VOLUME 1

STANDARD HANDBOOK OF PETROLEUM & NATURAL GAS *Engineering*

WILLIAM C. LYONS
EDITOR

STANDARD HANDBOOK OF PETROLEUM AND NATURAL GAS ENGINEERING

VOLUME 1

Petroleum engineering now has its own true classic handbook that reflects the profession's status as a mature major engineering discipline.

Formerly titled the *Practical Petroleum Engineer's Handbook,* by Joseph Zaba and W. T. Doherty (editors), this new, completely updated two-volume set is expanded and revised to give petroleum engineers a comprehensive source of industry standards and engineering practices. It is packed with the key, practical information and data that petroleum engineers rely upon daily.

The result of a fifteen-year effort, this handbook covers the gamut of oil and gas engineering topics to provide a reliable source of engineering and reference information for analyzing and solving problems. It also reflects the growing role of natural gas in industrial development by integrating natural gas topics throughout both volumes.

More than a dozen leading industry experts—academia and industry—contributed to this two-volume set to provide the best, most comprehensive source of petroleum engineering information available.

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Gulf Publishing Company
Houston, Texas

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Contents

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Preface

This petroleum and natural gas engineering two-volume handbook is written in the spirit of the classic handbooks of other engineering disciplines. The two volumes reflect the importance of the industry its engineers serve (i.e., *Standard and Poor's* shows that the fuels sector is the largest single entity in the gross domestic product) and the profession's status as a mature engineering discipline.

The project to write these volumes began with an attempt to revise the old *Practical Petroleum Engineer's Handbook* that Gulf Publishing had published since the 1940's. Once the project was initiated, it became clear that any revision of the old handbook would be inadequate. Thus, the decision was made to write an entirely new handbook and to write this handbook in the classic style of the handbooks of the other major engineering disciplines. This meant giving the handbook initial chapters on mathematics and computer applications, the sciences, general engineering, and auxiliary equipment. These initial chapters set the tone of the handbook by using engineering language and notation common to all engineering disciplines. This common language and notation is used throughout the handbook (language and notation in nearly all cases is consistent with Society of Petroleum Engineers publication practices). The authors, of which there are 27, have tried (and we hope succeeded) in avoiding the jargon that had crept into petroleum engineering literature over the past few decades. Our objective was to create a handbook for the petroleum engineering discipline that could be read and understood by any up-to-date engineer.

The specific petroleum engineering discipline chapters cover drilling and well completions, reservoir engineering, production, and economics and valuation. These chapters contain information, data, and example calculations related to practical situations that petroleum engineers often encounter. Also, these chapters reflect the growing role of natural gas in industrial operations by integrating natural gas topics and related subjects throughout both volumes.

This has been a very long and often frustrating project. Throughout the entire project the authors have been steadfastly cooperative and supportive of their editor. In the preparation of the handbook the authors have used published information from both the American

Petroleum Institute and the Society of Petroleum Engineers. The authors thank these two institutions for their cooperation in the preparation of the final manuscript. The authors would also like to thank the many petroleum production and service companies that have assisted in this project.

In the detailed preparation of this work, the authors would like to thank Jerry Hayes, Danette DeCristofaro, and the staff of ExecuStaff Composition Services for their very competent preparation of the final pages. In addition, the authors would like to thank Bill Lowe of Gulf Publishing Company for his vision and perseverance regarding this project; all those many individuals that assisted in the typing and other duties that are so necessary for the preparation of original manuscripts; and all the families of the authors that had to put up with weekends and weeknights of writing. The editor would especially like to thank the group of individuals that assisted through the years in the overall organization and preparation of the original written manuscripts and the accompanying graphics, namely; Ann Gardner, Britta Larrson, Linda Sperling, Ann Irby, Anne Cate, Rita Case, and Georgia Eaton.

All the authors and their editor know that this work is not perfect. But we also know that this handbook had to be written. Our greatest hope is that we have given those that will follow us, in future editions of this handbook, sound basic material to work with.

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STANDARD HANDBOOK OF PETROLEUM & NATURAL GAS *Engineering*

VOLUME 1

1

Mathematics

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1

Mathematics

GEOMETRY

See References 1–3 for additional information.

Sets and Functions

A *set* is a clearly defined collection of distinct objects or *elements*. The intersection of two sets S and T is the set of elements which belong to S *and* which also belong to T. The *union* of S and T is the set of all elements which belong to S *or* to T (or to both, i.e., inclusive or).

A *function* is a set of ordered elements such that no two ordered pairs have the same first element, denoted as (x,y) where x is the independent variable and y is the dependent variable. A function is established when a condition exists that determines y for each x, the condition usually being defined by an equation such as $y = f(x)$ [2].

Angles

An angle A may be *acute*, $0° < A < 90°$, *right*, $A = 90°$, or *obtuse*, $90° < A < 180°$. Directed angles, $A \leq 0°$ or $\geq 180°$, are discussed in the section "Trigonometry." Two angles are *complementary* if their sum is 90° or are *supplementary* if their sum is 180°. Angles are *congruent* if they have the same measurement in degrees and line segments are congruent if they have the same length. A *dihedral angle* is formed by two half-planes having the same edge, but not lying in the same plane. A *plane angle* is the intersection of a perpendicular plane with a dihedral angle.

Polygons

A *polygon* is the union of a finite number of triangular regions in a plane, such that if two regions intersect, their intersection is either a point or a line segment. Two polygons are *similar* if corresponding angles are congruent and corresponding sides are proportional with some constant k of proportionality. A segment whose end points are two nonconsecutive vertices of a polygon is a *diagonal*. The *perimeter* is the sum of the lengths of the sides.

Triangles

A *median* of a triangle is a line segment whose end points are a vertex and the midpoint of the opposite side. An *angle bisector* of a triangle is a median that lies on the ray bisecting an angle of the triangle. The *altitude* of a triangle is a perpendicular segment from a vertex to the opposite side. The sum of the angles of a triangle equals 180°. An *isosceles* triangle has two congruent sides and the angles opposite them are also congruent. If a triangle has three congruent sides (and, therefore, angles), it is *equilateral* and equiangular. A *scalene* triangle has no congruent sides. A set of congruent triangles can be drawn if one set of the following is given (where S = side length and A = angle measurement): SSS, SAS, AAS or ASA.

Quadrilaterals

A *quadrilateral* is a four-sided polygon determined by four coplanar points (three of which are noncollinear), if the line segments thus formed intersect each other only at their end points, forming four angles.

A *trapezoid* has one pair of opposite parallel sides and therefore the other pair of opposite sides is congruent. A *parallelogram* has both pairs of opposite sides congruent and parallel. The opposite angles are then congruent and adjacent angles are supplementary. The diagonals bisect each other and are congruent. A *rhombus* is a parallelogram whose four sides are congruent and whose diagonals are perpendicular to each other.

A *rectangle* is a parallelogram having four right angles, therefore both pairs of opposite sides are congruent. A rectangle whose sides are all congruent is a *square.*

Circles and Spheres

If P is a point on a given plane and r is a positive number, the *circle* with center P and radius r is the set of all points of the plane whose distance from P is equal to r. The *sphere* with center P and radius r is the set of all points in space whose distance from P is equal to r. Two or more circles (or spheres) with the same P, but different values of r are *concentric.*

A *chord* of a circle (or sphere) is a line segment whose end points lie on the circle (or sphere). A line which intersects the circle (or sphere) in two points is a *secant* of the circle (or sphere). A *diameter* of a circle (or sphere) is a chord containing the center and a radius is a line segment from the center to a point on the circle (or sphere).

The intersection of a sphere with a plane through its center is called a *great circle.*

A line which intersects a circle at only one point is a *tangent* to the circle at that point. Every tangent is perpendicular to the radius drawn to the point of intersection. Spheres may have tangent lines or tangent planes.

Pi (π) is the universal ratio of the circumference of any circle to its diameter and is equivalent to 3.1415927.... Therefore the circumference of a circle is πd or 2πr.

Arcs of Circles

A *central angle* of a circle is an angle whose vertex is the center of the circle. If P is the center and A and B are points, not on the same diameter, which lie on C (the circle), the *minor arc* AB is the union of A, B, and all points on C in the interior of <APB. The *major arc* is the union of A, B, and all points on C on the exterior of <APB. A and B are the end points of the arc and P is the center. If A and B are the end points of a diameter, the arc is a semicircle. A *sector* of a circle is a region bounded by two radii and an arc of the circle.

The *degree measure* (m) of a minor arc is the measure of the corresponding central angle (m of a semicircle is 180°) and of a major arc 360° minus the m of the corresponding minor arc. If an arc has a measure q and a radius r, then its *length* is

$L = q/180 \cdot \pi r$

Some of the properties of arcs are defined by the following theorems:

1. In congruent circles, if two chords are congruent, so are the corresponding minor arcs.
2. *Tangent-Secant Theorem*—If given an angle with its vertex on a circle, formed by a secant ray and a tangent ray, then the measure of the angle is half the measure of the intercepted arc.
3. *Two-Tangent Power Theorem*—The two tangent segments to a circle from an exterior point are congruent and determine congruent angles with the segment from the exterior point to the center of the circle.
4. *Two-Secant Power Theorem*—If given a circle C and an exterior point Q, let L_1 be a secant line through Q, intersecting C at points R and S, and let L_2 be another secant line through Q, intersecting C at U and T, then

   $$QR \cdot QS = QU \cdot QT$$

5. *Tangent-Secant Power Theorem*—If given a tangent segment $\overline{QT}$ to a circle and a secant line through Q, intersecting the circle at R and S, then

   $$QR \cdot QS = QT^2$$

6. *Two-Chord Power Theorem*—If $\overline{RS}$ and $\overline{TU}$ are chords of the same circle, intersecting at Q, then

   $$QR \cdot QS = QU \cdot QT$$

Concurrency

Two or more lines are *concurrent* if there is a single point which lies on all of them. The three altitudes of a triangle (if taken as lines, not segments) are always concurrent, and their point of concurrency is called the *orthocenter.* The angle bisectors of a triangle are concurrent at a point equidistant from their sides, and the medians are concurrent two thirds of the way along each median from the vertex to the opposite side. The point of concurrency of the medians is the *centroid.*

Similarity

Two figures with straight sides are *similar* if corresponding angles are congruent and the lengths of corresponding sides are in the same ratio. A line parallel to one side of a triangle divides the other two sides in proportion, producing a second triangle similar to the original one.

Prisms and Pyramids

A *prism* is a three dimensional figure whose bases are any congruent and parallel polygons and whose sides are parallelograms. A *pyramid* is a solid with one base consisting of any polygon and with triangular sides meeting at a point in a plane parallel to the base.

Prisms and pyramids are described by their bases: a *triangular prism* has a triangular base, a *parallelpiped* is a prism whose base is a parallelogram and a *rectangular parallelpiped* is a right rectangular prism. A cube is a rectangular parallelpiped all of whose edges are congruent. A *triangular pyramid* has a triangular base, etc. A *circular cylinder* is a prism whose base is a circle and a *circular cone* is a pyramid whose base is a circle.

Coordinate Systems

Each point on a plane may be defined by a pair of numbers. The coordinate system is represented by a line X in the plane (the *x-axis*) and by a line Y (the *y-axis*) perpendicular to line X in the plane, constructed so that their intersection, the *origin,* is denoted by zero. Any point P on the plane can now be described by its two coordinates which form an ordered pair, so that $P(x_1,y_1)$ is a point whose location corresponds to the real numbers x and y on the x-axis and the y-axis.

If the coordinate system is extended into space, a third axis, the z-axis, perpendicular to the plane of the x_1 and y_1 axes, is needed to represent the third dimension coordinate defining a point $P(x_1,y_1,z_1)$. The z-axis intersects the x and y axes at their origin, zero. More than three dimensions are frequently dealt with mathematically, but are difficult to visualize.

The *slope* m of a line segment in a plane with end points $P_1(x_1,y_1)$ and $P_2(x_2,y_2)$ is determined by the ratio of the change in the vertical (y) coordinates to the change in the horizontal (x) coordinates or

$$m = (y_2 - y_1)/(x_2 - x_1)$$

except that a vertical line segment (the change in x coordinates equal to zero) has no slope, i.e., m is undefined. A horizontal segment has a slope of zero. Two lines with the same slope are parallel and two lines whose slopes are negative reciprocals are perpendicular to each other.

Since the distance between two points $P_1(x_1,y_1)$ and $P_2(x_2,y_2)$ is the hypotenuse of a right triangle, the length of the line segment P_1P_2 is equal to

$$L = \sqrt{(x_2 - x_1)^2 + (y_2 - y_1)^2}$$

Graphs

A *graph* is a figure, i.e., a set of points, lying in a coordinate system and a graph of a condition (such as $x = y + 2$) is the set of all points that satisfy the condition. The graph of the *slope-intercept equation,* $y = mx + b$, is the line which passes through the point (0,b), where b is the y-intercept ($x = 0$) and m is the slope. The graph of the equation

$$(x - a)^2 + (y - b)^2 = r^2$$

is a circle with center (a,b) and radius r.

Vectors

A *vector* is described on a coordinate plane by a *directed segment* from its initial point to its terminal point. The directed segment represents the fact that every vector determines not only a magnitude, but also a direction. A vector **v** is not changed when moved around the plane, if its magnitude and angular orientation with respect to the x-axis is kept constant. The initial point of **v** may therefore be placed at the origin of the coordinate system and $\vec{v}$ may be denoted by

$$\vec{v} = \langle a,b \rangle$$

where a is the x-component and b is the y-component of the terminal point. The magnitude may then be determined by the Pythagorean theorem

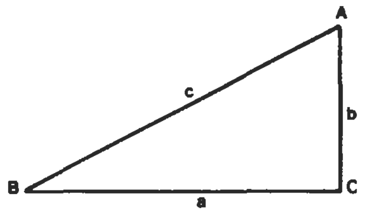
$$\mathbf{v} = \sqrt{a^2 + b^2}$$

For every pair of vectors (x_1,y_1) and (x_2,y_2), the *vector sum* is given by $(x_1 + x_2, y_1 + y_2)$. The *scalar product* of a vector $P = (x,y)$ and a real number (a *scalar*) r is $rP = (rx,ry)$. Also see the discussion of polar coordinates in the section "Trigonometry" and Chapter 2, "Basic Mechanics."

Lengths and Areas of Plane Figures [1]

(For definitions of trigonometric functions, see "Trigonometry.")

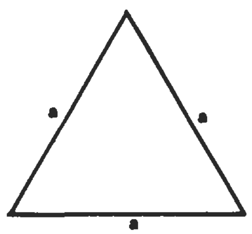
- *Right triangle* (Figure 1-1)



$c^2 = a^2 + b^2$

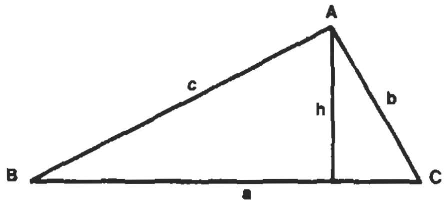
area $= 1/2 \bullet ab = 1/2 \bullet a^2 \cot A = 1/2 \bullet b^2 \tan A = 1/4 \bullet c^2 \sin 2A$

- *Equilateral triangle* (Figure 1-2)



area $= 1/4 \bullet a^2\sqrt{3} = 0.43301a^2$

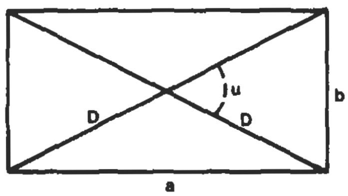
- *Any triangle* (Figure 1-3)



area = 1/2 base • altitude = 1/2 • ah = 1/2 • ab sin C
$= \pm 1/2 \bullet \{(x_1y_2 - x_2y_1) + (x_2y_3 - x_3y_2) + (x_3y_1 - x_1y_3)\}$

where (x_1,y_1), (x_2,y_2), (x_3,y_3) are coordinates of vertices.

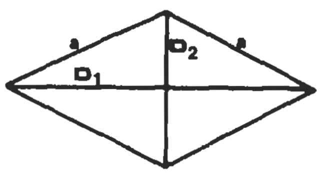
- *Rectangle* (Figure 1-4)



area = ab = $1/2 \bullet D^2 \sin u$

where u = angle between diagonals D, D

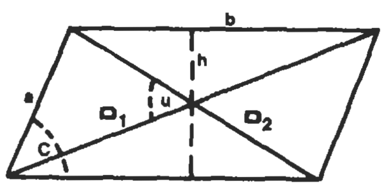
- *Rhombus* (Figure 1-5)



Area = $a^2 \sin C = 1/2 \bullet D_1D_2$

where C = angle between two adjacent sides
D_1, D_2 = diagonals

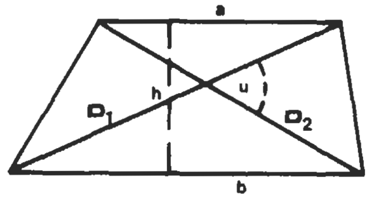
- *Parallelogram* (Figure 1-6)



area = bh = ab sin c = 1/2 • D_1D_2 sin u

where u = angle between diagonals D_1 and D_2

- *Trapezoid* (Figure 1-7)

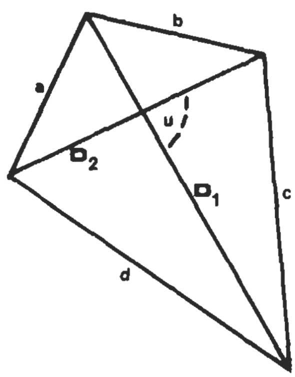


area = 1/2 • (a + b)h = 1/2 • D_1D_2 sin u

where u = angle between diagonals D_1 and D_2

and where bases a and b are parallel.

- *Any quadrilateral* (Figure 1-8)



area = 1/2 • D_1D_2 sin u

Note: $a^2 + b^2 + c^2 + d^2 = D_1^2 + D_2^2 + 4m^2$

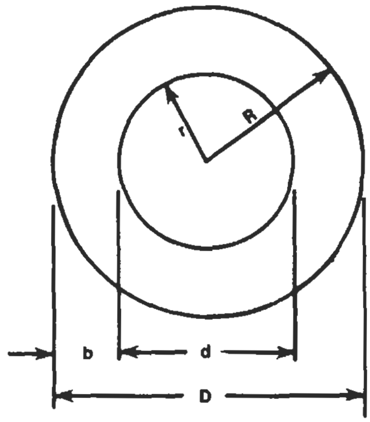
where m = distance between midpoints of D_1 and D_2

- *Circles*

area = πr^2 = 1/2 • Cr = 1/4 • Cd = 1/4 • πd^2 = $0.785398d^2$

where r = radius
d = diameter
C = circumference = $2\pi r$ = πd

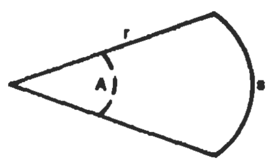
- *Annulus* (Figure 1-9)



area $= \pi(R^2 - r^2) = \pi(D^2 - d^2)/4 = 2\pi R'b$

where R' = mean radius $= 1/2 \bullet (R + r)$
$b = R - r$

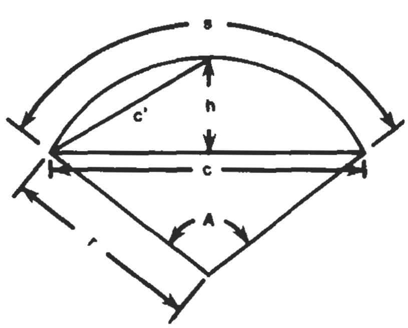
- *Sector* (Figure 1-10)



area $= 1/2 \bullet rs = \pi r^2 A/360° = 1/2 \bullet r^2$ rad A

where rad A = radian measure of angle A
s = length of arc = r rad A

- *Segment* (Figure 1-11)



area $= 1/2 \bullet r^2$ (rad A − sin A) $= 1/2[r(s - c) + ch]$

where rad A = radian measure of angle A

For small arcs,

$s = 1/3 \bullet (8c' - c)$

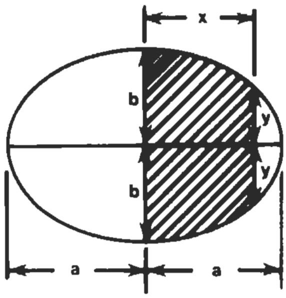
where c' = chord of half of the arc (Huygen's approximation)

Note: $c = 2\sqrt{h(d-h)}$

$$c' = \sqrt{dh} \text{ or } d = c'^2/h$$

where d = diameter of circle
$h = r[1 - \cos(1/2 \bullet A)]$
$s = 2r \text{ rad } (1/2 \bullet A)$

- *Ellipse* (Figure 1-12)



area of ellipse = πab

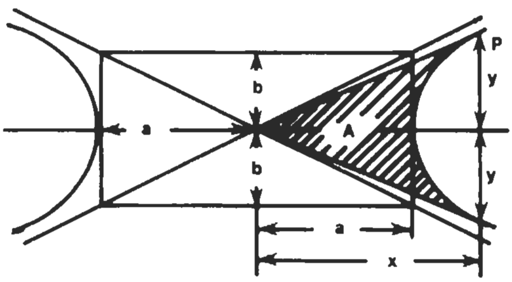
area of shaded segment = $xy + ab \sin^{-1}(x/a)$

length of perimeter of ellipse = $\pi(a + b)K$,

where $K = (1 + 1/4 \bullet m^2 + 1/64 \bullet m^4 + 1/256 \bullet m^6 + \ldots)$
$m = (a - b)/(a + b)$

For m = 0.1	0.2	0.3	0.4	0.5
K = 1.002	1.010	1.023	1.040	1.064
For m = 0.6	0.7	0.8	0.9	1.0
K = 1.092	1.127	1.168	1.216	1.273

- *Hyperbola* (Figure 1-13)



In any hyperbola,

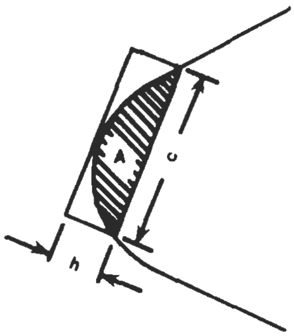
shaded area $A = ab \bullet \ln [(x/a) + (y/b)]$

In an equilateral hyperbola ($a = b$),

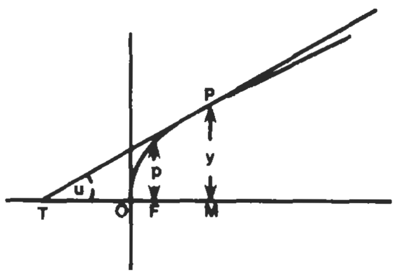
area $A = a^2 \sinh^{-1} (y/a) = a^2 \cosh^{-1} (x/a)$

Here x and y are coordinates of point P.

- *Parabola* (Figure 1-14)



shaded area $A = 2/3 \bullet ch$



In Figure 1-15,

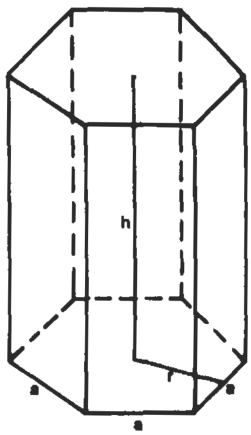
length of arc $OP = s = 1/2 \bullet PT + 1/2 \bullet p \bullet \ln [\cot(1/2 \bullet u)]$

Here c = any chord
p = semilatus rectum
PT = tangent at P

Note: OT = OM = x

Surfaces and Volumes of Solids [1]

- *Regular prism* (Figure 1-16)

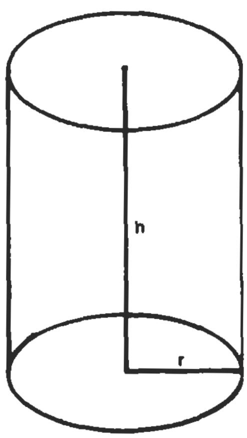


volume = 1/2 • nrah = Bh

lateral area = nah = Ph

where n = number of sides
B = area of base
P = perimeter of base

- *Right circular cylinder* (Figure 1-17)

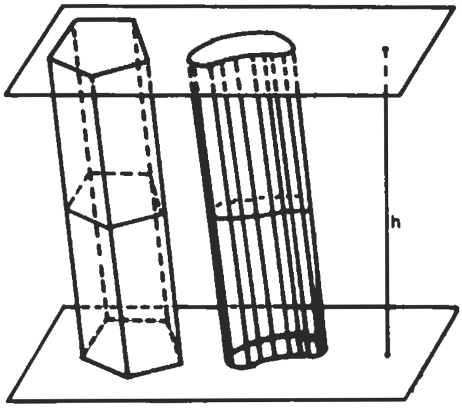


volume = $\pi r^2 h$ = Bh

lateral area = $2\pi rh$ = Ph

where B = area of base
P = perimeter of base

- *Any prism or cylinder* (Figure 1-18)



volume = Bh = Nl

lateral area = Ql

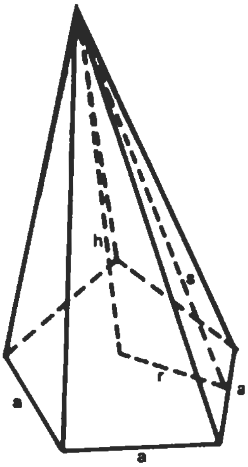
where l = length of an element or lateral edge
B = area of base
N = area of normal section
Q = perimeter of normal section

- *Hollow cylinder* (right and circular)

volume = $\pi h(R^2 - r^2) = \pi hb(D - b) = \pi hb(d + b) = \pi hbD' = \pi hb(R + r)$

where h = altitude
r,R, (d,D) = inner and outer radii (diameters)
b = thickness = R − r
D′ = mean diam = 1/2 • (d + D) = D − b = d + b

- *Regular pyramid* (Figure 1-19)



volume = 1/3 altitude • area of base = 1/6 • hran

lateral area = 1/2 slant height • perimeter of base = 1/2 • san

where r = radius of inscribed circle
a = side (of regular polygon)
n = number of sides
$s = \sqrt{r^2 + h^2}$

(vertex of pyramid directly above center of base)

- *Right circular cone*

volume = 1/3 • $\pi r^2 h$

lateral area = πrs

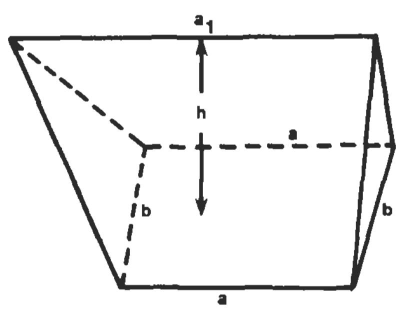
where r = radius of base
h = altitude
s = slant height = $\sqrt{r^2 + h^2}$

- *Any pyramid or cone*

volume = 1/3 • Bh

where B = area of base
h = perpendicular distance from vertex to plane in which base lies

- *Wedge* (Figure 1-20)



(Rectangular base; a_1 parallel to a,a and at distance h above base)

volume = 1/6 • $hb(2a + a_1)$

- *Sphere*

volume = $V = 4/3 \cdot \pi r^3 = 4.188790r^3 = 1/6 \cdot \pi d^3 = 0.523599d^3$

area = $A = 4\pi r^2 = \pi d^2$

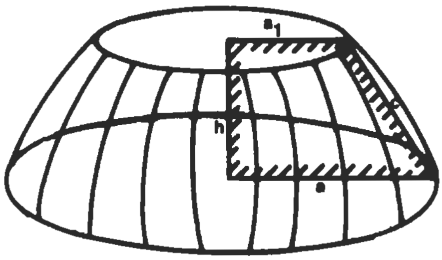
where r = radius
$d = 2r = \text{diameter} = \sqrt[3]{6V/\pi} = 1.24070\sqrt[3]{V} = \sqrt{A/\pi} = 0.56419\sqrt{A}$

- *Hollow sphere,* or spherical shell

  volume = $4/3 \cdot \pi(R^3 - r^3) = 1/6 \cdot \pi(D^3 - d^3) = 4\pi R_1^2 t + 1/3 \cdot \pi t^3$

  where R,r = outer and inner radii
  D,d = outer and inner diameters
  t = thickness = R − r
  R_1 = mean radius = $1/2 \cdot (R + r)$

- *Any spherical segment (Zone)* (Figure 1-21)

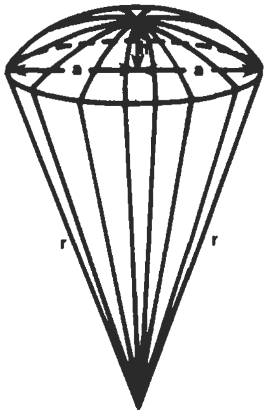


volume = $1/6 \cdot \pi h(3a^2 + 3a_1^2 + h^2)$

lateral area (zone) = $2\pi rh$

where r = radius of sphere

- *Spherical sector* (Figure 1-22)

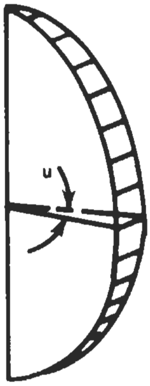


volume = $1/3 \cdot r \cdot$ area of cap = $2/3 \cdot \pi r^2 h$

total area = area of cap + area of cone = $2\pi rh + \pi ra$

Note: $a^2 = h(2r - h)$

- *Spherical wedge* bounded by two plane semicircles and a *lune* (Figure 1-23)



volume of wedge/volume of sphere = u/360°

area of lune/area of sphere = u/360°

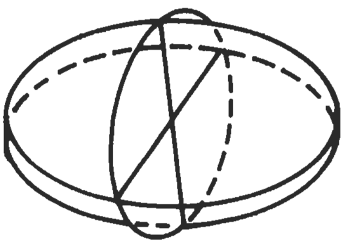
where u = dihedral angle of the wedge

- *Regular polyhedra*

Name of Solid	Bounded By	A/a^2	V/a^3
Tetrahedron	4 triangles	1.7321	0.1179
Cube	6 squares	6.0000	1.0000
Octahedron	8 triangles	3.4641	0.4714
Dodecahedron	12 pentagons	20.6457	7.6631
Icosahedron	20 triangles	8.6603	2.1817

where A = area of surface
V = volume
a = edge

- *Ellipsoid* (Figure 1-24)



volume = 4/3 • πabc

where a, b, c = semiaxes

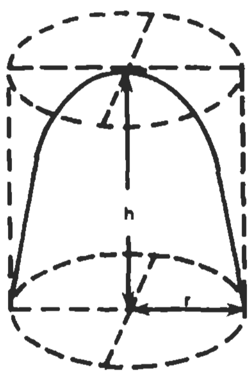
- *Spheroid* (or ellipsoid of revolution)

  By the prismoidal formula:

  volume = 1/6 • h(A + B + 4M)

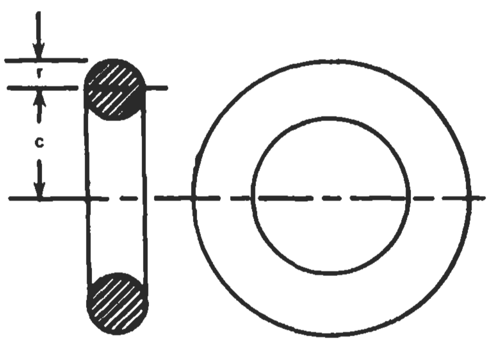
  where h = altitude
  A and B = areas of bases
  M = area of a plane section midway between the bases

- *Paraboloid of revolution* (Figure 1-25)



  volume = $1/2 \cdot \pi r^2 h$ = 1/2 volume of circumscribed cylinder

- *Torus*, or *anchor ring* (Figure 1-26)



  volume = $2\pi^2 cr^2$

  area = $4\pi r^2 cr$ (proof by theorems of Pappus)

ALGEBRA

See References 1 and 4 for additional information.

Operator Precedence and Notation

Operations in a given equation are performed in decreasing order of precedence as follows:

1. Exponentiation
2. Multiplication or division
3. Addition or subtraction

from left to right unless this order is changed by the insertion of parentheses. For example:

$$a + b \bullet c - d^3/e$$

will be operated upon (calculated) as if it were written

$$a + (b \bullet c) - [(d^3)/e]$$

The symbol $|a|$ means "the absolute value of a," or the numerical value of a regardless of sign, so that

$$|-2| = |2| = 2$$

n! means "n factorial" (where n is a whole number) and is the product of the whole numbers 1 to n inclusive, so that

$$4! = 1 \bullet 2 \bullet 3 \bullet 4 = 24$$

The notation for the sum of any real numbers $a_1, a_2, \ldots, a_n$ is

$$\sum_{i=1}^{n} a_i$$

and for their product

$$\prod_{i=1}^{n} a_i$$

The notation "$x \propto y$" is read "x varies directly with y" or "x is directly proportional to y," meaning $x = ky$ where k is some constant. If $x \propto 1/y$, then x is inversely proportional to y and $x = k/y$.

Rules of Addition

$$a + b = b + a$$

$$(a + b) + c = a + (b + c)$$

$$a - (-b) = a + b \text{ and}$$

$$a - (x - y + z) = a - x + y - z$$

(i.e., a minus sign preceding a pair of parentheses operates to reverse the signs of each term within, if the parentheses are removed)

Rules of Multiplication and Simple Factoring

$a \bullet b = b \bullet a$

$(ab)c = a(bc)$

$a(b + c) = ab + ac$

$a(-b) = -ab$ and $-a(-b) = ab$

$(a + b)(a - b) = a^2 - b^2$

$(a + b)^2 = a^2 + 2ab + b^2$

and

$(a - b)^2 = a^2 - 2ab + b^2$

$(a + b)^3 = a^3 + 3a^2 + 3ab^2 + b^3$

and

$(a - b)^3 = a^3 - 3a^2 + 3ab^2 - b^3$

(For higher order polynomials see the section "Binomial Theorem")
$a^n + b^n$ is factorable by $(a + b)$ if n is odd, thus

$a^3 + b^3 = (a + b)(a^2 - ab + b^2)$

and $a^n - b^n$ is factorable by $(a - b)$, thus

$a^n - b^n = (a - b)(a^{n-1} + a^{n-2}b + . . . + ab^{n-2} + b^{n-1})$

Fractions

The numerator and denominator of a fraction may be multiplied or divided by any quantity (other than zero) without altering the value of the fraction, so that, if $m \neq 0$,

$$\frac{ma + mb + mc}{mx + my} = \frac{a + b + c}{x + y}$$

To add fractions, transform each to a common denominator and add the numerators ($b,y \neq 0$):

$$\frac{a}{b} + \frac{x}{y} = \frac{ay}{by} + \frac{bx}{by} = \frac{ay + bx}{by}$$

To multiply fractions (denominators $\neq 0$):

$$\frac{a}{b} \bullet \frac{x}{y} = \frac{ax}{by}$$

$$\frac{a}{b} \bullet x = \frac{ax}{b}$$

$$\frac{a}{b} \bullet \frac{x}{y} \bullet \frac{c}{z} = \frac{axc}{byz}$$

To divide one fraction by another, invert the divisor and multiply:

$$\frac{a}{b} \div \frac{x}{y} = \frac{a}{b} \bullet \frac{y}{x} = \frac{ay}{bx}$$

Exponents

$a^m \bullet a^n = a^{m+n}$ and $a^m \div a^n = a^{m-n}$

$a^0 = 1 \ (a \neq 0)$ and $a^1 = a$

$a^{-m} = 1/a^m$

$(a^m)^n = a^{mn}$

$a^{1/n} = \sqrt[n]{a}$ and $a^{m/n} = \sqrt[n]{a^m}$

$(ab)^n = a^n b^n$

$(a/b)^n = a^n/b^n$

Except in simple cases (square and cube roots) radical signs are replaced by fractional exponents. If n is odd,

$$\sqrt[n]{-a} = -\sqrt[n]{a}$$

but if n is even, the nth root of $-a$ is imaginary.

Logarithms

The logarithm of a positive number N is the power to which the base (10 or e) must be raised to produce N. So, $x = \log_e N$ means that $e^x = N$, and $x = \log_{10} N$ means that $10^x = N$. Logarithms to the base 10, frequently used in numerical computation, are called *common* or *denary logarithms,* and those to base e, used in theoretical work, are called *natural logarithms* and frequently notated as *ln*. In either case,

$$\log(ab) = \log a + \log b$$

$\log(a/b) = \log a - \log b$

$\log(1/n) = -\log n$

$\log(a^n) = n \log a$

$\log_b(b) = 1$, where b is either 10 or e

$\log 0 = -\infty$

$\log 1 = 0$

$\log_{10}e = M = 0.4342944819\ldots$, so for conversion

$\log_{10}x = 0.4343 \log_e x$

and since $1/M = 2.302585$, for conversion ($\ln = \log_e$)

$\ln x = 2.3026 \log_{10}x$

Binomial Theorem

Let

$n_1 = n$

$$n_2 = \frac{n(n-1)}{2!}$$

$$n_3 = \frac{n(n-1)(n-2)}{3!}$$

etc. Then for any n, $|x| < 1$,

$$(1 + x)^n = 1 + n_1x + n_2x^2 + n_3x^3 + \ldots$$

If n is a positive integer, the system is valid without restriction on x and completes with the term n_nx^n.

Some of the more useful special cases are the following [1]:

$$\sqrt{1+x} = (1+x)^{1/2} = 1 + \frac{1}{2}x - \frac{1}{8}x^2 + \frac{1}{16}x^3 - \frac{5}{128}x^4 + \ldots \quad (|x|< 1)$$

$$\sqrt[3]{1+x} = (1+x)^{1/3} = 1 + \frac{1}{3}x - \frac{1}{9}x^2 - \frac{5}{81}x^3 - \frac{10}{243}x^4 + \ldots \quad (|x|< 1)$$

$$\frac{1}{1+x} = (1+x)^{-1} = 1 - x + x^2 - x^3 + x^4 - \ldots \quad (|x|< 1)$$

$$\frac{1}{\sqrt{1+x}} = (1+x)^{-1/2} = 1 - \frac{1}{2}x + \frac{3}{8}x^2 - \frac{5}{16}x^3 + \frac{35}{128}x^4 - \ldots \quad (|x|< 1)$$

$$\frac{1}{\sqrt[3]{1+x}} = (1+x)^{-1/3} = 1 - \frac{1}{3}x + \frac{2}{9}x^2 - \frac{14}{81}x^3 + \frac{35}{243}x^4 - \ . \ . \ . \qquad (|x| < 1)$$

$$\sqrt{(1+x)^3} = (1+x)^{3/2} = 1 - \frac{3}{2}x + \frac{3}{8}x^2 - \frac{1}{16}x^3 + \frac{3}{128}x^4 - \ . \ . \ . \qquad (|x| < 1)$$

$$\frac{1}{\sqrt{(1+x)^3}} = (1+x)^{-3/2} = 1 - \frac{3}{2}x + \frac{15}{8}x^2 - \frac{35}{16}x^3 + \frac{315}{128}x^4 - \ . \ . \ . \qquad (|x| < 1)$$

with corresponding formulas for $(1 - x)^{1/2}$, etc., obtained by reversing the signs of the odd powers of x. Also, provided $|b| < |a|$:

$$(a+b)^n = a^n\left(1 + \frac{b}{a}\right)^n = a^n + n_1a^{n-1}b + n_2a^{n-2}b^2 + n_3a^{n-3}b^3 + \ . \ . \ .$$

where n_1, n_2, etc., have th values given above.

Progressions

In an *arithmetic progression,* (a, a + d, a + 2d, a + 3d, . . .), each term is obtained from the preceding term by adding a constant difference, d. If n is the number of terms, the last term is $p = a + (n - 1)d$, the "average" term is $1/2(a + p)$ and the sum of the terms is n times the average term or $s = n/2(a + p)$. The *arithmetic mean* between a and b is $(a + b)/2$.

In a *geometric progression,* (a, ar, ar^2, ar^3, . . .), each term is obtained from the preceding term by multiplying by a constant ratio, r. The nth term is ar^{n-1}, and the sum of the first n terms is $s = a(r^n - 1)/(r - 1) = a(1 - r^n)/(1 - r)$. If r is a fraction, r^n will approach zero as n increases and the sum of n terms will approach $a/(1 - r)$ as a limit. The *geometric mean,* also called the "mean proportional," between a and b is $\sqrt{ab}$. The *harmonic mean* between a and b is $2ab/(a + b)$.

Summation of Series by Difference Formulas

a_1, a_2, . . ., a_n is a series of n numbers, and D′ (first difference), D″ (second difference), . . . are found by subtraction in each column as follows:

a	D′	D″	D‴	D⁗
-26				
	28			
2		-16		
	12		7	
14		-9		0
	3		7	
17		-2		0
	1		7	
18		5		0
	6		7	
24		12		
	18			
42				

If the kth differences are equal, so that subsequent differences would be zero, the series is an arithmetical series of the kth order. The nth term of the series is a_n, and the sum of the first n terms is S_n, where

$$a_n = a_1 + (n - 1)D' + (n - 1)(n - 2)D''/2! + (n - 1)(n - 2)(n - 3)D'''/3! + \ldots$$

$$S_n = na_1 + n(n - 1)D'/2! + n(n - 1)(n - 2)D''/3! + \ldots$$

In this third-order series just given, the formulas will stop with the term in D‴.

Sums of the First n Natural Numbers

- To the first power:

$$1 + 2 + 3 + \ldots + (n - 1) + n = n(n + 1)/2$$

- To the second power (squared):

$$1^2 + 2^2 + \ldots + (n - 1)^2 + n^2 = n(n + 1)(2n + 1)/6$$

- To the third power (cubed):

$$1^3 + 2^3 + \ldots + (n - 1)^3 + n^3 = [n(n + 1)/2]^2$$

Solution of Equations in One Unknown

Legitimate operations on equations include addition of any quantity to both sides, multiplication by any quantity of both sides (unless this would result in division by zero), raising both sides to any positive power (if ± is used for even roots) and taking the logarithm or the trigonometric functions of both sides.

Any *algebraic equation* may be written as a polynomial of nth degree in x of the form

$$a_0x^n + a_1x^{n-1} + a_2x^{n-2} + \ldots + a_{n-1}x + a_n = 0$$

with, in general, n roots, some of which may be imaginary and some equal. If the polynomial can be factored in the form

$$(x - p)(x - q)(x - r) \ldots = 0$$

then p, q, r, . . . are the roots of the equation. If $|x|$ is very large, the terms containing the lower powers of x are least important, while if $|x|$ is very small, the higher-order terms are least significant.

First degree equations (*linear equations*) have the form

$$x + a = b$$

with the solution $x = b - a$ and the root $b - a$.

Second-degree equations (*quadratic equations*) have the form

$$ax^2 + bx + c = 0$$

with the solution

$$x = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

and the roots

$$1.\ \frac{-b + \sqrt{b^2 - 4ac}}{2a}$$

and

$$2.\ \frac{-b - \sqrt{b^2 - 4ac}}{2a}$$

The sum of the roots is $-b/a$ and their product is c/a.

Third-degree equations (*cubic equations*), in the general case, have the form, after division by the coefficient of the highest-order term,

$$x^3 + ax^2 + bx + c = 0$$

with the solution

$$x_1^3 = Ax_1 + B$$

where $x_1 = x - a/3$
$A = 3(a/3)^2 - b$
$B = -2(a/3)^3 + b(a/3) - c$

Exponential equations are of the form

$$a^x = b$$

with the solution $x = (\log b)/(\log a)$ and the root $(\log b)/(\log a)$. The complete logarithm must be taken, not only the mantissa.

Trigonometric equations are of the form

$$a \cos x \pm b \sin x$$

If an acute angle u is found, where

$$\tan u = b/a$$

and an angle v ($0° < v < 180°$) is found, where

$$\cos^2 v = c^2/(a^2 + b^2)$$

the solution is $x = \pm(u \pm v)$ and the roots are $\pm(u + v)$ and $\pm(u - v)$, depending on the sign of b.

Solution of Systems of Simultaneous Equations

A set of *simultaneous equations* is a system of n equations in n unknowns. The solutions (if any) are the sets of values for the unknowns which will satisfy all the equations in the system.

First-degree equations in 2 unknowns are of the form

$$a_1x_1 + b_1x_2 = c_1 \tag{a}$$

$$a_2x_1 + b_2x_2 = c_2 \tag{b}$$

The solution is found by multiplication of Equations a and b by some factors that will produce one term in each that will, upon addition of Equations a and b, become zero. The resulting equation may then be rearranged to solve for the remaining unknown. For example, by multiplying Equation a by a_2 and Equation b by $-a_1$, adding Equation a and Equation b and rearranging their sum

$$x_2 = \frac{a_2c_1 - a_1c_2}{a_2b_1 - a_1b_2}$$

and by substitution in Equation a:

$$x_1 = \frac{b_1c_2 - b_2c_1}{a_2b_1 - a_1b_2}$$

A set of *n first-degree equations in n unknowns* is solved in a similar fashion by multiplication and addition to eliminate n - 1 unknowns and then back substitution. *Second-degree equations in 2 unknowns* may be solved in the same way when two of the following are given: the product of the unknowns, their sum or difference, the sum of their squares. For further solutions, see "Numerical Methods."

Determinants

Determinants of the second order are of the following form and are evaluated as

$$\begin{vmatrix} a_1 b_1 \\ a_2 b_2 \end{vmatrix} = a_1b_2 - a_2b_1$$

and of the third order as

$$\begin{vmatrix} a_1 b_1 c_1 \\ a_2 b_2 c_2 \\ a_3 b_3 c_3 \end{vmatrix} = a_1 \begin{vmatrix} b_2 c_2 \\ b_3 c_3 \end{vmatrix} - a_2 \begin{vmatrix} b_1 c_1 \\ b_3 c_3 \end{vmatrix} + a_3 \begin{vmatrix} b_1 c_1 \\ b_2 c_2 \end{vmatrix}$$

and of higher orders, by the general rule as follows. To evaluate a determinant of the nth order, take the elements of the first column with alternate plus and minus signs and form the sum of the products obtained by multiplying each of these elements by its corresponding *minor*. The minor corresponding to any element e_n is the determinant (of the next lowest order) obtained by striking out from the given determinant the row and column containing e_n.

Some of the general properties of determinants are:

1. Columns may be changed to rows and rows to columns.
2. Interchanging two adjacent columns changes the sign of the result.

3. If two columns are equal or if one is a multiple of the other, the determinant is zero.
4. To multiply a determinant by any number m, multiply all elements of any one column by m.

Systems of simultaneous equations may be solved by the use of determinants in the following manner. Although the example is a third-order system, larger systems may be solved by this method. If

$$a_1x + b_1y + c_1z = p_1$$

$$a_2x + b_2y + c_2z = p_2$$

$$a_3x + b_3y + c_3z = p_3$$

and if

$$D = \begin{vmatrix} a_1 b_1 c_1 \\ a_2 b_2 c_2 \\ a_3 b_3 c_3 \end{vmatrix} \neq 0$$

then

$$x = D_1/D$$

$$y = D_2/D$$

$$z = D_3/D$$

where

$$D_1 = \begin{vmatrix} p_1 b_1 c_1 \\ p_2 b_2 c_2 \\ p_3 b_3 c_3 \end{vmatrix}$$

$$D_2 = \begin{vmatrix} a_1 p_1 c_1 \\ a_2 p_2 c_2 \\ a_3 p_3 c_3 \end{vmatrix}$$

$$D_3 = \begin{vmatrix} a_1 b_1 p_1 \\ a_2 b_2 p_2 \\ a_3 b_3 p_3 \end{vmatrix}$$

TRIGONOMETRY

Directed Angles

If AB and AB′ are any two rays with the same end point A, the directed angle <BAB′ is the ordered pair $(\overrightarrow{AB}, \overrightarrow{AB'})$. $\overrightarrow{AB}$ is the initial side of <BAB′ and $\overrightarrow{AB'}$ the terminal side. <BAB′ ≠ <B′AB and any directed angle may be ≤ 0° or ≥ 180°.

A directed angle may be thought of as an amount of rotation rather than a figure. If $\overrightarrow{AB}$ is considered the initial position of the ray, which is then rotated about its end point A to form <BAB′, $\overrightarrow{AB}'$ is its terminal position.

Basic Trigonometric Functions

A trigonometric function of a real number, i.e., a, b, . . ., is the same function as that of the corresponding directed angle, <A, etc., that is,

a = degree measure of <A

The basic trigonometric functions are the sine, cosine, and tangent.
Given the triangle in Figure 1-27

sine <A = sin a° = a/r

cosine <A = cos a° = b/r

tangent <A = tan a° = a/b

Radian Measure

From the properties of arcs of circles, if an arc has degree measure m and radius r, its length L is

$L = m/180 \bullet \pi r$

therefore

$L/r = m\pi/180$

The value L/r is the *radian measure* of the arc and therefore of its central angle θ, so that

radian measure = degree measure • $\pi/180$

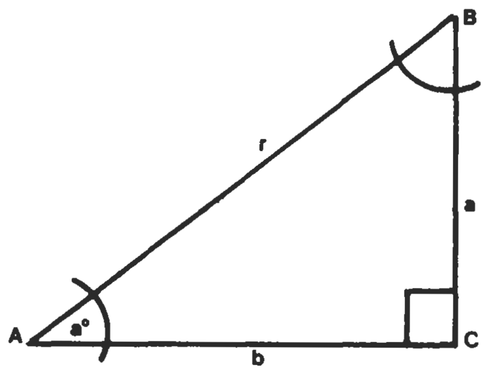


Figure 1-27. Trigonometric functions of real numbers.

Trigonometric Properties

For any angle θ as in Figure 1-28

$\sin\theta$ = opposite side/hypotenuse = s_1/h

$\cos\theta$ = adjacent side/hypotenuse = s_2/h

$\tan\theta$ = opposite side/adjacent side = $s_1/s_2 = \sin\theta/\cos\theta$

and the reciprocals of the basic functions (where the function $\neq 0$)

$$\text{cotangent } \theta = \cot\theta = 1/\tan\theta = s_2/s_1$$

$$\text{secant } \theta = \sec\theta = 1/\cos\theta = h/s_2$$

$$\text{cosecant } \theta = \csc\theta = 1/\sin\theta = h/s_1$$

The functions versed sine, coversed sine, and exterior secant are defined as

$$\text{vers } \theta = 1 - \cos\theta$$

$$\text{covers } \theta = 1 - \sin\theta$$

$$\text{exsec } \theta = \sec\theta - 1$$

To reduce an angle to the first quadrant of the unit circle, that is, to a degree measure between 0° and 90°, see Table 1-1. For function values at major angle values, see Tables 1-2 and 1-3. Relations between functions and the sum/difference of two functions are given in Table 1-4. Generally, there will be two angles between 0° and 360° that correspond to the value of a function.

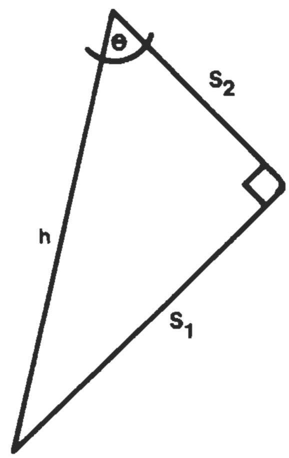


Figure 1-28. Trigonometric functions of angles.

(text continued on page 32)

Table 1-1
Angle Reduction to First Quadrant

If	$90° < x < 180°$	$180° < x < 270°$	$270° < x < 360°$
sin x =	+cos(x − 90°)	−sin(x − 180°)	−cos(x − 270°)
cos x =	−sin(x − 90°)	−cos(x − 180°)	+sin(x − 270°)
tan x =	−cot(x − 90°)	+tan(x − 180°)	−cot(x − 270°)
csc x =	+sec(x − 90°)	−csc(x − 180°)	−sec(x − 270°)
sec x =	−csc(x − 90°)	−sec(x − 180°)	+csc(x − 270°)
cot x =	−tan(x − 90°)	+cot(x − 180°)	−tan(x − 270°)

Table 1-2
Trigonometric Function Values by Quadrant

If	$0° < x < 90°$	$90° < x < 180°$	$180° < x < 270°$	$270° < x < 360°$
sin x	+0 to +1	+1 to +0	−0 to −1	−1 to −0
cos x	+1 to +0	−0 to −1	−1 to −0	+0 to +1
tan x	+0 to +∞	−∞ to −0	+0 to +∞	−∞ to −0
csc x	+∞ to +1	+1 to +∞	−∞ to −1	−1 to −∞
sec x	+1 to +∞	−∞ to −1	−1 to −∞	+∞ to +1
cot x	+∞ to +0	−0 to −∞	+∞ to +0	−0 to −∞

Table 1-3
Trigonometric Function Values at Major Angle Values

Values at	30°	45°	60°
sin x	1/2	$1/2\sqrt{2}$	$1/2\sqrt{3}$
cos x	$1/2\sqrt{3}$	$1/2\sqrt{2}$	1/2
tan x	$1/3\sqrt{3}$	1	$\sqrt{3}$
csc x	2	$\sqrt{2}$	$2/3\sqrt{3}$
sec x	$2/3\sqrt{3}$	$\sqrt{2}$	2
cot x	$\sqrt{3}$	1	$1/3\sqrt{3}$

Table 1-4
Relations Between Trigonometric Functions of Angles

Single Angle

$\sin^2 x + \cos^2 x = 1$
$\tan x = (\sin x)/(\cos x)$
$\cot x = 1/(\tan x)$

$1 + \tan^2 x = \sec^2 x$
$1 + \cot^2 x = \csc^2 x$
$\sin(-x) = -\sin x,\ \cos(-x) = \cos x,\ \tan(-x) = -\tan x$

Two Angles

$\sin(x + y) = \sin x \cos y + \cos x \sin y$
$\sin(x - y) = \sin x \cos y - \cos x \sin y$
$\cos(x + y) = \cos x \cos y - \sin x \sin y$
$\cos(x - y) = \cos x \cos y + \sin x \sin y$
$\tan(x + y) = (\tan x + \tan y)/(1 - \tan x \tan y)$
$\tan(x - y) = (\tan x - \tan y)/(1 + \tan x \tan y)$
$\cot(x + y) = (\cot x \cot y - 1)/(\cot y + \cot x)$
$\cot(x - y) = (\cot x \cot y + 1)/(\cot y - \cot x)$
$\sin x + \sin y = 2 \sin[1/2(x + y)] \cos[1/2(x - y)]$
$\sin x - \sin y = 2 \cos[1/2(x + y)] \sin[1/2(x - y)]$
$\cos x + \cos y = 2 \cos[1/2(x + y)] \cos[1/2(x - y)]$
$\cos x - \cos y = - 2 \sin[1/2(x + y)] \sin[1/2(x - y)]$
$\tan x + \tan y = [\sin(x + y)]/[\cos x \cos y]$
$\tan x - \tan y = [\sin(x - y)]/[\cos x \cos y]$
$\cot x + \cot y = [\sin(x + y)]/[\sin x \sin y]$
$\cot x - \cot y = [\sin(y - x)]/[\sin x \sin y]$
$\sin^2 x - \sin^2 y = \cos^2 y - \cos^2 x$
$= \sin(x + y) \sin(x - y)$
$\cos^2 x - \sin^2 y = \cos^2 y - \sin^2 x$
$= \cos(x + y) \cos(x - y)$
$\sin(45° + x) = \cos(45° - x),\quad \tan(45° + x) = \cot(45° - x)$
$\sin(45° - x) = \cos(45° + x),\quad \tan(45° - x) = \cot(45° + x)$

Multiple and Half Angles

$\tan 2x = (2 \tan x)/(1 - \tan^2 x)$
$\cot 2x = (\cot^2 x - 1)/(2 \cot x)$
$\sin(nx) = n \sin x \cos^{n-1} x - (n)_3 \sin^3 x \cos^{n-3} x + (n)_5 \sin^5 x \cos^{n-5} x - \ldots$
$\cos(nx) = \cos^n x - (n)_2 \sin^2 x \cos^{n-2} x + (n)_4 \sin^4 x \cos^{n-4} x - \ldots$
(Note: $(n)_2, \ldots$ are the binomial coefficients)
$\sin(x/2) = \pm \sqrt{1/2\,(1 - \cos x)}$
$\cos(x/2) = \pm \sqrt{1/2\,(1 + \cos x)}$
$\tan(x/2) = (\sin x)/(1 + \cos x) = \pm \sqrt{(1 - \cos x)/(1 + \cos x)}$

Three Angles Whose Sum = 180°

$\sin A + \sin B + \sin C = 4 \cos(A/2) \cos(B/2) \cos(C/2)$
$\cos A + \cos B + \cos C = 4 \sin(A/2) \sin(B/2) \sin(C/2) + 1$
$\sin A + \sin B - \sin C = 4 \sin(A/2) \sin(B/2) \cos(C/2)$
$\cos A + \cos B - \cos C = 4 \cos(A/2) \cos(B/2) \sin(C/2) - 1$
$\sin^2 A + \sin^2 B + \sin^2 C = 2 \cos A \cos B \cos C + 2$
$\sin^2 A + \sin^2 B - \sin^2 C = 2 \sin A \sin B \cos C$
$\tan A + \tan B + \tan C = \tan A \tan B \tan C$
$\cot(A/2) + \cot(B/2) + \cot(C/2) = \cot(A/2) \cot(B/2) \cot(C/2)$
$\sin 2A + \sin 2B + \sin 2C = 4 \sin A \sin B \sin C$
$\sin 2A + \sin 2B - \sin 2C = 4 \cos A \cos B \sin C$

(text continued from page 29)

Graphs of Trigonometric Functions

Graphs of the sine and cosine functions are identical in shape and periodic with a period of 360°. The sine function graph translated ±90° along the x-axis produces the graph of the cosine function. The graph of the tangent function is discontinuous when the value of tan θ is undefined, that is, at odd multiples of 90° (. . ., 90°, 270°, . . .). For abbreviated graphs of the sine, cosine, and tangent functions, see Figure 1-29.

Inverse Trigonometric Functions

The inverse sine of x (also referred to as the arc sine of x), denoted by $\sin^{-1}x$, is the principal angle whose sine is x, that is,

$$y = \sin^{-1}x \quad \text{means} \quad \sin y = x$$

Inverse functions $\cos^{-1}x$ and $\tan^{-1}x$ also exist for the cosine of y and the tangent of y. The principal angle for $\sin^{-1}x$ and $\tan^{-1}x$ is an angle a, where $-90° < a < 90°$, and for $\cos^{-1}x$, $0° < a < 180°$.

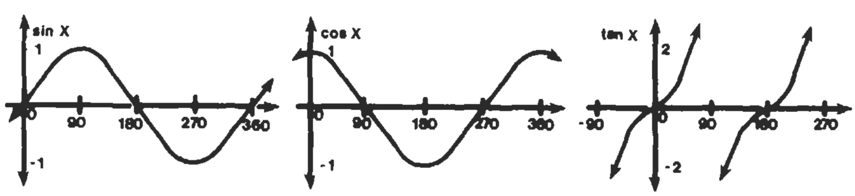
Solution of Plane Triangles

The solution of any part of a plane triangle is determined in general by any other three parts given by one of the following groups, where S is the length of a side and A is the degree measure of an angle:

- AAS
- SAS
- SSS

The fourth group, two sides and the angle opposite one of them, is ambiguous since it may give zero, one, or two solutions. Given an example triangle with sides a, b, and c and angles A, B, and C (A being opposite a, etc., and A + B + C = 180°), the fundamental laws relating to the solution of triangles are

1. Law of Sines: $a/(\sin A) = b/(\sin B) = c/(\sin C)$
2. Law of Cosines: $c^2 = a^2 + b^2 - 2ab \cos C$



The sine graph period 360° | The cosine graph period 360° | The tangent graph period 180°

Figure 1-29. Graphs of the trigonometric functions.

Hyperbolic Functions

The *hyperbolic sine, hyperbolic cosine,* etc. of any number x are functions related to the exponential function e^x. Their definitions and properties are very similar to the trigonometric functions and are given in Table 1-5.

The *inverse hyperbolic functions,* $\sinh^{-1}x$, etc., are related to the logarithmic functions and are particularly useful in integral calculus. These relationships may be defined for real numbers x and y as

$$\sinh^{-1}(x/y) = \ln(x + \sqrt{x^2 + y^2}) - \ln y$$

$$\cosh^{-1}(x/y) = \ln(x + \sqrt{x^2 - y^2}) - \ln y$$

$$\tanh^{-1}(x/y) = 1/2 \bullet \ln[(y + x)/(y - x)]$$

$$\coth^{-1}(x/y) = 1/2 \bullet \ln[(x + y)/(x - y)]$$

Table 1-5
Hyperbolic Functions

$\sinh x = 1/2(e^x - e^{-x})$
$\cosh x = 1/2(e^x + e^{-x})$
$\tanh x = \sinh x/\cosh x$
$\operatorname{csch} x = 1/\sinh x$
$\operatorname{sech} x = 1/\cosh x$
$\coth x = 1/\tanh x$

$\sinh(-x) = -\sinh x$
$\cosh(-x) = \cosh x$
$\tanh(-x) = -\tanh x$

$\cosh^2 x - \sinh^2 x = 1$
$1 - \tanh^2 x = \operatorname{sech}^2 x$
$1 - \coth^2 x = -\operatorname{csch}^2 x$

$\sinh(x \pm y) = \sinh x \cosh y \pm \cosh x \sinh y$
$\cosh(x \pm y) = \cosh x \cosh y \pm \sinh x \sinh y$
$\tanh(x \pm y) = (\tanh x \pm \tanh y)/(1 \pm \tanh x \tanh y)$
$\sinh 2x = 2 \sinh x \cosh x$
$\cosh 2x = \cosh^2 x + \sinh^2 x$
$\tanh 2x = (2 \tanh x)/(1 + \tanh^2 x)$
$\sinh(x/2) = \sqrt{1/2(\cosh x - 1)}$
$\cosh(x/2) = \sqrt{1/2(\cosh x + 1)}$
$\tanh(x/2) = (\cosh x - 1)/(\sinh x) = (\sinh x)/(\cosh x + 1)$

Polar Coordinate System

The *polar coordinate system* describes the location of a point (denoted as $[r,\theta]$) in a plane by specifying a distance r and an angle θ from the origin of the system. There are several relationships between polar and rectangular coordinates, diagrammed in Figure 1-30. From the Pythagorean Theorem

$$r = \pm\sqrt{x^2 + y^2}$$

Also

$$\sin\theta = y/r \quad \text{or} \quad y = r\sin\theta$$

$$\cos\theta = x/r \quad \text{or} \quad x = r\cos\theta$$

$$\tan\theta = y/x \quad \text{or} \quad \theta = \tan^{-1}(y/x)$$

To convert rectangular coordinates to polar coordinates, given the point (x,y), using the Pythagorean Theorem and the preceding equations.

$$[r,\theta] = [\sqrt{x^2 + y^2}, \tan^{-1}(y/x)]$$

To convert polar to rectangular coordinates, given the point $[r,\theta]$:

$$(x,y) = [r\cos\theta, r\sin\theta]$$

For graphic purposes, the polar plane is usually drawn as a series of concentric circles with the center at the origin and radii 1, 2, 3, ... Rays from the center are drawn at 0°, 15°, 30°, . . ., 360° or 0, $\pi/12$, $\pi/6$, $\pi/4$, . . ., 2π

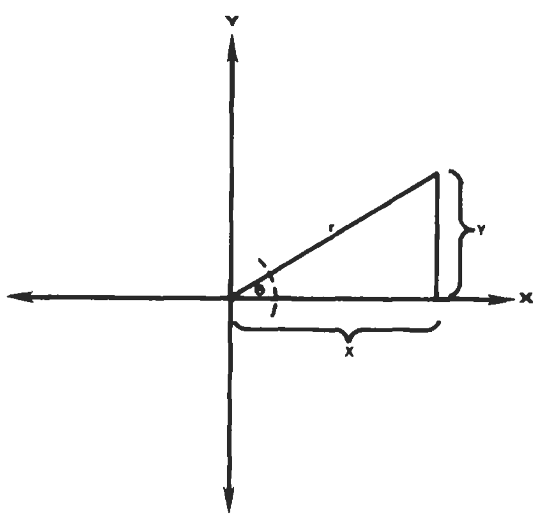


Figure 1-30. Polar coordinates.

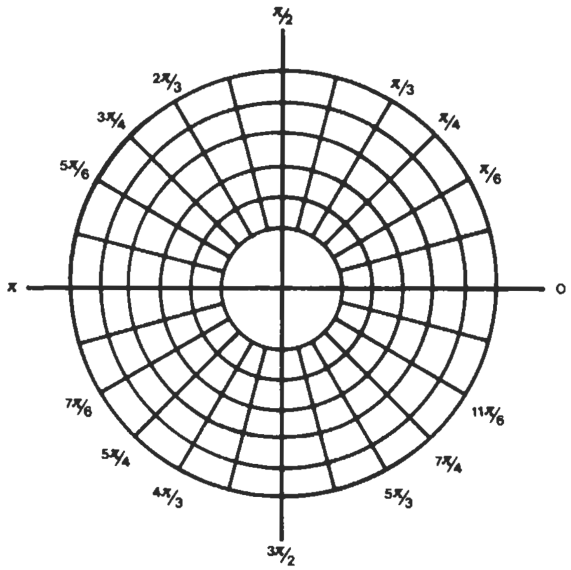


Figure 1-31. The polar plane.

radians. The origin is called the *pole,* and points $[r,\theta]$ are plotted by moving a positive or negative distance r horizontally from the pole, and through an angle θ from the horizontal. See Figure 1-31 with θ given in radians as used in calculus. Also note that

$$[r, \theta] = [-r, \theta + \pi]$$

DIFFERENTIAL AND INTEGRAL CALCULUS

See References 1 and 5–8 for additional information.

Derivatives

Geometrically, the derivative of $y = f(x)$ at any value x_n is the slope of a tangent line T intersecting the curve at the point $P(x,y)$. Two conditions applying to differentiation (the process of determining the derivatives of a function) are:

1. The primary (necessary and sufficient) condition is that

   $$\lim_{\Delta x \to 0} \frac{\Delta y}{\Delta x}$$

   exists and is independent of the way in which $\Delta x \to 0$
2. A secondary (necessary, not sufficient) condition is that

   $$\lim_{\Delta x \to 0} f(x + \Delta x) = f(x)$$

A short table of derivatives will be found in Table 1-6.

Table 1-6
Table of Derivatives*

$$\frac{d}{dx}(x) = 1$$

$$\frac{d}{dx}(a) = 0$$

$$\frac{d}{dx}(u \pm v \pm .....) = \frac{du}{dx} \pm \frac{dv}{dx} \pm .....$$

$$\frac{d}{dx}(au) = a\frac{du}{dx}$$

$$\frac{d}{dx}(uv) = u\frac{dv}{dx} + v\frac{du}{dx}$$

$$\frac{d}{dx}\frac{u}{v} = \frac{v\frac{du}{dx} - u\frac{dv}{dx}}{v^2}$$

$$\frac{d}{dx}(u^a) = nu^{a-1}\frac{du}{dx}$$

$$\frac{d}{dx}\log_a u = \frac{\log_a e}{u}\frac{du}{dx}$$

$$\frac{d}{dx}\log u = \frac{1}{u}\frac{du}{dx}$$

$$\frac{d}{dx}a^u = a^u \bullet \log_e a \bullet \frac{du}{dx}$$

$$\frac{d}{dx}e^u = e^u\frac{du}{dx}$$

$$\frac{d}{dx}u^v = vu^{v-1}\frac{du}{dx} + u^v \log_e u\frac{dv}{dx}$$

$$\frac{d}{dx}\sin u = \cos u\frac{du}{dx}$$

$$\frac{d}{dx}\cos u = -\sin u\frac{du}{dx}$$

$$\frac{d}{dx}\tan u = \sec^2 u\frac{du}{dx}$$

$$\frac{d}{dx}\cot u = -\csc^2 u\frac{du}{dx}$$

$$\frac{d}{dx}\sec u = \sec u \tan u\frac{du}{dx}$$

$$\frac{d}{dx}\csc u = -\csc u \cot u\frac{du}{dx}$$

$$\frac{d}{dx}\operatorname{vers} u = \sin u\frac{du}{dx}$$

$$\frac{d}{dx}\sin^{-1} u = \frac{1}{\sqrt{1-u^2}}\frac{du}{dx}$$

$$\frac{d}{dx}\cos^{-1} u = -\frac{1}{\sqrt{1-u^2}}\frac{du}{dx}$$

$$\frac{d}{dx}\tan^{-1} u = \frac{1}{1+u^2}\frac{du}{dx}$$

$$\frac{d}{dx}\cot^{-1} u = -\frac{1}{1+u^2}\frac{du}{dx}$$

$$\frac{d}{dx}\sec^{-1} u = \frac{1}{u\sqrt{u^2-1}}\frac{du}{dx}$$

$$\frac{d}{dx}\csc^{-1} u = -\frac{1}{u\sqrt{u^2-1}}\frac{du}{dx}$$

$$\frac{d}{dx}\operatorname{vers}^{-1} u = \frac{1}{\sqrt{2u-u^2}}\frac{du}{dx}$$

$$\frac{d}{dx}\sinh u = \cosh u\frac{du}{dx}$$

$$\frac{d}{dx}\cosh u = \sinh u\frac{du}{dx}$$

$$\frac{d}{dx}\tanh u = \operatorname{sech}^2 u\frac{du}{dx}$$

$$\frac{d}{dx}\coth u = -\operatorname{csch}^2 u\frac{du}{dx}$$

$$\frac{d}{dx}\operatorname{sech} u = -\operatorname{sech} u \tanh u\frac{du}{dx}$$

$$\frac{d}{dx}\operatorname{csch} u = -\operatorname{csch} u \coth u\frac{du}{dx}$$

$$\frac{d}{dx}\sinh^{-1} u = \frac{1}{\sqrt{u^2-1}}\frac{du}{dx}$$

$$\frac{d}{dx}\cosh^{-1} u = \frac{1}{\sqrt{u^2-1}}\frac{du}{dx}$$

$$\frac{d}{dx}\tanh^{-1} u = \frac{1}{1-u^2}\frac{du}{dx}$$

$$\frac{d}{dx}\coth^{-1} u = -\frac{1}{u^2-1}\frac{du}{dx}$$

$$\frac{d}{dx}\operatorname{sech}^{-1} u = -\frac{1}{u\sqrt{1-u^2}}\frac{du}{dx}$$

$$\frac{d}{dx}\operatorname{csch}^{-1} u = -\frac{1}{u\sqrt{u^2-1}}\frac{du}{dx}$$

*Note: u and v represent funcitons of x. All angles are in radians.

Higher-Order Derivatives

The *second derivative* of a function $y = f(x)$, denoted $f''(x)$ or d^2y/dx^2 is the derivative of $f'(x)$ and the *third derivative,* $f'''(x)$ is the derivative of $f''(x)$. Geometrically, in terms of $f(x)$: if $f''(x) > 0$ then $f(x)$ is concave upwardly, if $f''(x) < 0$ then $f(x)$ is concave downwardly.

Partial Derivatives

If $u = f(x,y, \ldots)$ is a function of two or more variables, the *partial derivative* of u with respect to x, $f_x(x,y, \ldots)$ or $\partial u/\partial x$, may be formed by assuming x to be the independent variable and holding $(y, \ldots)$ as constants. In a similar manner, $f_y(x,y, \ldots)$ or $\partial u/\partial y$ may be formed by holding $(x, \ldots)$ as constants. Second-order partial derivatives of $f(x,y)$ are denoted by the manner of their formation as f_{xx}, f_{xy} (equal to f_{yx}), f_{yy} or as $\partial^2u/\partial x^2$, $\partial^2u/\partial x\partial y$, $\partial^2u/\partial y^2$, and the higher-order partial derivatives are likewise formed.

Implicit functions, i.e., $f(x,y) = 0$, may be solved by the formula

$$\frac{dy}{dx} = -\frac{f_x}{f_y}$$

at the point in question.

Maxima and Minima

A *critical point* on a curve $y = f(x)$ is a point where $y' = 0$, that is, where the tangent to the curve is horizontal. A critical value of x, therefore, is a value such that $f'(x) = 0$. All roots of the equation $f'(x) = 0$ are critical values of x, and the corresponding values of y are the critical values of the function.

A function $f(x)$ has a *relative maximum* at $x = a$ if $f(x) < f(a)$ for all values of x (except a) in some open interval containing a and a *relative minimum* at $x = b$ if $f(x) > f(b)$ for all x (except b) in the interval containing b. At the relative maximum a of $f(x)$, $f'(a) = 0$, i.e., slope = 0, and $f''(a) < 0$, i.e., the curve is downwardly concave at this point, and at the relative minimum b, $f'(b) = 0$ and $f''(b) > 0$ (upward concavity). In Figure 1-32, A, B, C, and D are critical points

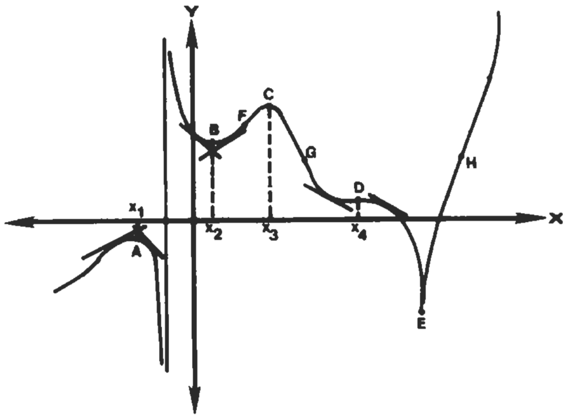


Figure 1-32. Maxima and minima.

and x_1, x_2, x_3, and x_4 are critical values of x. A and C are maxima, B is a minimum, and D is neither. D, F, G, and H are *points of inflection* where the slope is minimum or maximum. In special cases, such as E, maxima or minima may occur where f'(x) is undefined or infinite.

The partial derivatives $f_x = 0$, $f_y = 0$, $f_{xx} < 0$ (or > 0), $f_{yy} < 0$ (or > 0) determine the minima (or maxima) for a function of two variables f(x,y).

The *absolute maximum* (or minimum) of f(x) at x = a exists if $f(x) \le f(a)$ (or $f(x) \ge f(a)$) for all x in the domain of the function and need not be a relative maximum or minimum. If a function is defined and continuous on a closed interval, it will always have an absolute minimum and an absolute maximum, and they will be found either at a relative minimum and a relative maximum or at the endpoints of the interval.

Differentials

If y = f(x) and Δx and Δy are the increments of x and y, respectively, since $y + \Delta y = f(x + \Delta x)$, then

$$\Delta y = f(x + \Delta x) - f(x)$$

As Δx approaches its limit 0 and (since x is the independent variable) $dx = \Delta x$

$$\frac{dy}{dx} \cong \frac{f(x+\Delta x) - f(x)}{\Delta x}$$

and

$$dy \cong \Delta y$$

By defining dy and dx separately, it is now possible to write

$$\frac{dy}{dx} = f'(x) \quad \text{as}$$

$$dy = f' = (x)dx$$

Differentials of higher orders are of little significance unless dx is a constant, in which case the first, second, third, etc. differentials approximate the first, second, third, etc. differences and may be used in constructing difference tables (see "Algebra").

In functions of two or more variables, where f(x, y, . . .) = 0, if dx, dy, . . . are assigned to the independent variables x, y, . . ., the differential du is given by differentiating term by term or by taking

$$du = f_x \bullet dx + f_y \bullet dy + \ldots$$

If x, y, . . . are functions of t, then

$$\frac{du}{dt} = (f_x)\frac{dx}{dt} + (f_y)\frac{dy}{dt} + \ldots$$

expresses the rate of change of u with respect to t, in terms of the separate rates of change of x, y, . . . with respect to t.

Radius of Curvature

The *radius of curvature* R of a plane curve at any point P is the distance along the normal (the perpendicular to the tangent to the curve at point P) on the concave side of the curve to the *center of curvature* (Figure 1-33). If the equation of the curve is y = f(x)

$$R = \frac{ds}{du} = \frac{[1 + f'(x)^2]^{3/2}}{f''(x)}$$

where the rate of change (ds/dx) and the differential of the arc (ds), s being the length of the arc, are defined as

$$\frac{ds}{dx} = \sqrt{1 + \left(\frac{dy}{dx}\right)^2}$$

and

$$ds = \sqrt{dx^2 + dy^2}$$

and $dx = ds \cos u$
$dy = ds \sin u$
$u = \tan^{-1}[f'(x)]$

u being the angle of the tangent at P with respect to the x-axis. (Essentially, ds, dx, and y correspond to the sides of a right triangle.) The curvature K is the rate at which <u is changing with respect to s, and

$$K = \frac{1}{R} = \frac{du}{ds}$$

If f'(x) is small, $K \cong f''(x)$.

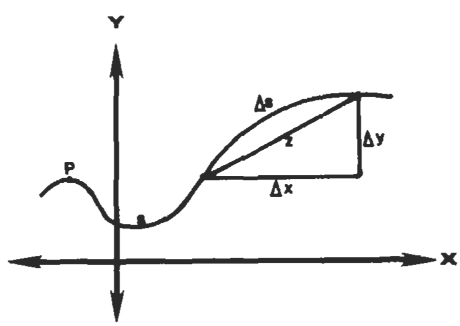


Figure 1-33. Radius of curvature in rectangular coordinates.

In polar coordinates (Figure 1-34), $r = f(\theta)$, where r is the radius vector and θ is the polar angle, and

$$ds = \sqrt{d\rho^2 + \rho^2 d\theta^2}$$

so that by $x = \rho \cos \theta$, $y = \rho \sin \theta$ and $K = 1/R = d\theta/ds$, then

$$R = \frac{ds}{d\theta} = \frac{[r^2 + (r')^2]^{3/2}}{r^2 - rr'' + 2(r')^2}$$

If the equation of the circle is

$$R^2 = (x - \alpha) + (y - \beta)^2$$

by differentiation and simplification

$$\alpha = x - \frac{y'[1 + (y')^2]}{y''}$$

and

$$\beta = y + \frac{1 + (y')^2}{y''}$$

The *evolute* is the locus of the centers of curvature, with variables α and β, and the parameter x (y, y', and y″ all being functions of x). If f(x) is the evolute of g(x), g(x) is the *involute* of f(x).

Indefinite Integrals

Integration by Parts makes use of the differential of a product

$$d(uv) = u\,dv + v\,du$$

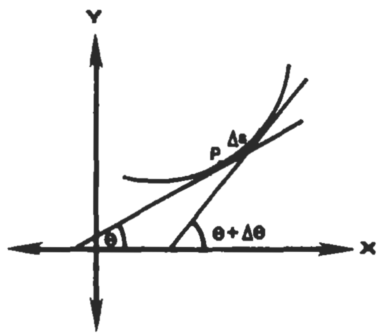


Figure 1-34. Radius of curvature in polar coordinates.

or

$$u\,dv = d(uv) - v\,du$$

and by integrating

$$\int u\,dv = uv - \int v\,du$$

where $\int v\,du$ may be recognizable as a standard form or may be more easily handled than $\int u\,dv$.

Integration by Transformation may be useful when, in certain cases, particular transformations of a given integral to one of a recognizable form suggest themselves.

For example, a given integral involving such quantities as

$$\sqrt{u^2 - a^2}, \quad \sqrt{u^2 + a^2}, \quad \text{or} \quad \sqrt{a^2 - u^2}$$

may suggest appropriate trigonometric transformations such as, respectively,

$$u = a \csc \theta,$$

$$u = a \tan \theta,$$

or

$$u = a \sin \theta$$

Integration by Partial Fractions is of assistance in the integration of rational fractions. If

$$\frac{ax + b}{x^2 + px + q} = \frac{ax + b}{(x - \alpha)(x - \beta)} = \frac{A}{x - \alpha} + \frac{B}{x - \beta}$$

where $A + B = \alpha$
$A\beta + B\alpha = -b$

and A and B are found by use of determinants (see "Algebra"), then

$$\int \frac{(ax + b)dx}{(x - \alpha)(x - \beta)} = \int \frac{Adx}{x - \alpha} + \int \frac{Bdx}{x - \beta} = A\log(x - \alpha) + B\log(x - \beta) + C$$

Integration by Tables is possible if an integral may be put into a form that can be found in a table of integrals, such as the one given in Table 1-7. More complete tables may be found in Bois, "Tables of Indefinite Integrals," Dover, and in others.

Definite Integrals

The *Fundamental Theorem of Calculus* states that if f(x) is the derivative of F(x) and if f(x) is continuous in the interval [a,b], then

$$\int_a^b f(x)dx = F(b) - F(a)$$

(text continued on page 44)

Table 1-7
Table of Integrals*

1. $\int df(x) = f(x) + C$
2. $d\int f(x)dx = f(x)dx$
3. $\int 0 \cdot dx = C$
4. $\int af(x)dx = a\int f(x)dx$
5. $\int (u \pm v)dx = \int udx \pm \int vdx$
6. $\int udv = uv - \int vdu$
7. $\int \frac{udv}{dx}dx = uv - \int v\frac{du}{dx}dx$
8. $\int f(y)dx = \int \frac{f(y)dy}{\frac{dy}{dx}}$
9. $\int u^n du = \frac{u^{n+1}}{n+1} + C, n \neq -1$
10. $\int \frac{du}{u} = \log_e u + C$
11. $\int e^u du = e^u + C$
12. $\int b^u du = \frac{b^u}{\log_e b} + C$
13. $\int \sin u\, du = -\cos u + C$
14. $\int \cos u\, du = \sin u + C$
15. $\int \tan u\, du = \log_e \sec u + C = -\log_e \cos u + C$
16. $\int \cot u\, du = \log_e \sin u + C = -\log_e \csc u + C$
17. $\int \sec u\, du = \log_e(\sec u + \tan u) + C = \log_e \tan\left(\frac{u}{2} + \frac{\pi}{4}\right) + C$
18. $\int \csc u\, du = \log_e(\csc u - \cot u) + C = \log_e \tan\frac{u}{2} + C$
19. $\int \sin^2 u\, du = \frac{1}{2}u - \frac{1}{2}\sin u \cos u + C$
20. $\int \cos^2 u\, du = \frac{1}{2}u + \frac{1}{2}\sin u \cos u + C$
21. $\int \sec^2 u\, du = \tan u + C$
22. $\int \csc^2 u\, du = -\cot u + C$
23. $\int \tan^2 u\, du = \tan u - u + C$
24. $\int \mathrm{ctn}^2 u\, du = -\cot u - u + C$
25. $\int \frac{du}{u^2 + a^2} = \frac{1}{a}\tan^{-1}\frac{u}{a} + C$

26. $\int \dfrac{du}{u^2 - a^2} = \dfrac{1}{2a}\log_e\left(\dfrac{u-a}{u+a}\right) + C = -\dfrac{1}{a}\coth^{-1}\left(\dfrac{u}{a}\right) + C,\ \text{if } u^2 > a^2$

$= \dfrac{1}{2a}\log_e\left(\dfrac{a-u}{a+u}\right) + C = -\dfrac{1}{a}\tanh^{-1}\left(\dfrac{u}{a}\right) + C,\ \text{if } u^2 < a^2$

27. $\int \dfrac{du}{\sqrt{a^2 - u^2}} = \sin^{-1}\left(\dfrac{u}{a}\right) + C$

28. $\int \dfrac{du}{\sqrt{u^2 \pm a^2}} \log_e(u + \sqrt{u^2 \pm a^2})^{\dagger} + C$

29. $\int \dfrac{du}{\sqrt{au - u^2}} = \cos^{-1}\left(\dfrac{a-u}{a}\right) + C$

30. $\int \dfrac{du}{u\sqrt{u^2 - a^2}} = \dfrac{1}{a}\sec^{-1}\left(\dfrac{u}{a}\right) + C = \dfrac{1}{a}\cos^{-1}\dfrac{a}{u} + C$

31. $\int \dfrac{du}{u\sqrt{a^2 \pm u^2}} = -\dfrac{1}{a}\log_e\left(\dfrac{a + \sqrt{a^2 - u^2}}{u}\right)^{\dagger} + C$

32. $\int \sqrt{a^2 - u^2} \bullet du = \dfrac{1}{2}\left(u\sqrt{a^2 - u^2} + a^2 \sin^{-1}\dfrac{u}{a}\right) + C$

33. $\int \sqrt{u^2 \pm a^2} \bullet du = \dfrac{1}{2}[u\sqrt{u^2 \pm a^2} \pm a^2 \log_e(u + \sqrt{u^2 \pm a^2})]^{\dagger} + C$

34. $\int \sinh u\ du = \cosh u + C$

35. $\int \cosh u\ du = \sinh u + C$

36. $\int \tanh u\ du = \log_e(\cosh u) + C$

37. $\int \coth u\ du = \log_e(\sinh u) + C$

38. $\int \operatorname{sech} u\ du = \sin^{-1}(\tanh u) + C$

39. $\int \operatorname{csch} u\ du = \log_e\left(\tanh\dfrac{u}{2}\right) + C$

40. $\int \operatorname{sech} u \bullet \tanh u \bullet du = -\operatorname{sech} u + C$

41. $\int \operatorname{csch} u \bullet \coth u \bullet du = -\operatorname{csch} u + C$

---

* Note: u and v represent functions of x.

† $\log_e\left(\dfrac{u + \sqrt{u^2 + a^2}}{a}\right) = \sinh^{-1}\left(\dfrac{u}{a}\right); \log_e\left(\dfrac{a + \sqrt{a^2 - u^2}}{u}\right) = \operatorname{sech}^{-1}\left(\dfrac{u}{a}\right);$

$\log_e\left(\dfrac{u + \sqrt{u^2 - a^2}}{a}\right) = \cosh^{-1}\left(\dfrac{u}{a}\right); \log_e\left(\dfrac{a + \sqrt{a^2 - u^2}}{u}\right) \operatorname{csch}^{-1}\left(\dfrac{u}{a}\right)$

(text continued from page 41)

Geometrically, the integral of f(x)dx over the interval [a,b] is the area bounded by the curve y = f(x) from f(a) to f(b) and the x-axis from x = a to x = b, or the "area under the curve from a to b."

Properties of Definite Integrals

$$\int_a^b = -\int_b^a$$

$$\int_a^c + \int_c^b = \int_a^b$$

The *mean value* of f(x), $\bar{f}$, between a and b is

$$\bar{f} = \frac{1}{b-a}\int_a^b f(x)dx$$

If the upper limit b is a variable, then $\int_a^b f(x)dx$ is a function of b and its derivative is

$$f(b) = \frac{d}{db}\int_a^b f(x)dx$$

To differentiate with respect to a parameter

$$\frac{\partial}{\partial c}\int_a^b f(x,c)dx = \int_a^b \frac{\partial f(x,c)}{\partial c}dx$$

Some methods of integration of definite integrals are covered in "Numerical Methods."

Improper Integrals

If one (or both) of the limits of integration is infinite, or if the integrand itself becomes infinite at or between the limits of integration, the integral is an *improper integral.* Depending on the function, the integral may be defined, may be equal to ∞, or may be undefined for all x or for certain values of x.

Multiple Integrals

If $\int f(x)dx = F(x)dx + C_1$, then

$$\int (F(x) + C_1)dx = G(x) + C_1 + C_2$$

or

$$\int [\int f(x)dx]dx = \iint f(x)dx\ dx = \iint f(x)dx^2$$

and the process may be repeated, as in

$$\int \{\int [\int f(x)dx]dx\}dx + \iiint f(x)dx^3$$

Definite multiple integrals are solved from the inner integral to the outer.

The double integral of a function with two independent variables is of the form

$$\iint f(x,y)dy\ dx = \int [\int f(x,y)dy]dx$$

where the limits of the inner integral are functions of x or constants as in the definite integral

$$\int_c^d \int_a^b f(x,y)dy\ dx$$

where a and b may be $f_1(x)$ and $f_2(x)$.

Second Fundamental Theorem

If A is an area bounded by a closed curve (see Figure 1-35) and $x_1 = a$ and $x_{n+1} = b$, then

$$A = \int_a^b \int_{f_1(x)}^{f_2(x)} dydx$$

where $x = \gamma_1(y)$ = equation of curve CBE
$x = \gamma_2(y)$ = equation of curve CDE
$y = f_1(x)$ = equation of curve BED
$y = f_2(x)$ = equation of curve BCD

Differential Equations

An *ordinary differential equation* contains a single independent variable and a single unknown function of that variable, with its derivatives. A *partial differential*

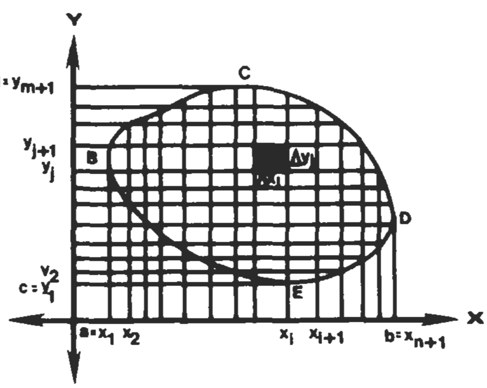


Figure 1-35. Area included in a closed curve.

equation involves an unknown function of two or more independent variables, and its partial derivatives. The general solution of a differential equation of order n is the set of all functions that possess at least n derivatives and satisfy the equation, as well as any auxiliary conditions.

Methods of Solving Ordinary Differential Equations

For *first-order equations,* if possible, separate the variables, integrate both sides, and add the constant of integration, C. If the equation is *homogeneous* in x and y, the value of dy/dx in terms of x and y is of the form $dy/dx = f(y/x)$ and the variables may be separated by introducing new independent variable $v = y/x$ and then

$$x\frac{dv}{dx} + v = f(v)$$

The expression $f(x,y)dx + F(x,y)dy$ is an *exact differential* if

$$\frac{\partial f(x,y)}{\partial y} = \frac{\partial F(x,y)}{\partial x} \quad (= P, \text{ for example})$$

Then, the solution of $f(x,y)dx + F(x,y)dy = 0$ is

$$\int f(x,y)dx + \int [F(x,y) - \int P\, dx]dy = C$$

or

$$\int F(x,y)dy + \int [f(x,y) - \int P\, dy]dx = C$$

A *linear differential equation* of the first order such as

$$dy/dx + f(x) \bullet y = F(x)$$

has the solution

$$y = e^{-P}[\int e^{P}F(x)dx + C] \quad \text{where } P = \int f(x)dx$$

In the class of nonlinear equations known as *Bernoulli's equations,* where

$$dy/dx + f(x) \bullet y = F(x) \bullet y^n$$

substituting $y^{1-n} = v$ gives

$$dv/dx + (1 - n)f(x) \bullet v = (1 - n)F(x) \quad [n \neq 0 \text{ or } 1]$$

which is linear in v and x. In *Clairaut's equations*

$$y = xp + f(p) \quad \text{where } p = dy/dx,$$

the solution consists of the set of lines given by $y = Cx + f(C)$, where C is any constant, and the curve obtained by eliminating p between the original equation and $x + f'(p) = 0$ [1].

Some differential equations of the *second order* and their solutions follow:

For $d^2y/dx^2 = -n^2y$
$$y = C_1\sin(nx + C_2)$$
$$= C_3\sin nx + C_4\cos nx$$

For $d^2y/dx^2 = + n^2y$
$$y = C_1\sinh(nx + C_2)$$
$$= C_3e^{nx} + C_4e^{-nx}$$

For $d^2y/dx^2 = f(y)$

$$x = \int \frac{dy}{\sqrt{C_1 + 2P}} + C_2$$

where $P = \int f(y)dy$

For $d^2y/dx^2 = f(x)$
$$y = \int Pdx + C_1x + C_2 \text{ where } P = \int f(x)dx$$
$$= xP - \int xf(x)dx + C_1x + C_2$$

For $d^2y/dx^2 = f(dy/dx)$, setting $dy/dx = z$ and $d^2/dx^2 = dz/dx$
$$x = \int dz/f(z) + C_1 \text{ and}$$
$$y = \int zdz/f(z) + C_2\text{, then eliminating } z$$

For $d^2y/dx^2 + 2b(dy/dx) + a^2y = 0$ (the equation for damped vibration)

- If $a^2 - b^2 > 0$,

  then $m = \sqrt{a^2 - b^2}$
  $$y = C_1e^{-bx}\sin(mx + C_2)$$
  $$= e^{-bx}[C_3\sin(mx) + C_4\cos(mx)]$$
- If $a^2 - b^2 = 0$,
  $$y = e^{-bx}(C_1 + C_2x)$$
- If $a^2 - b^2 < 0$,

  then $n = \sqrt{b^2 - a^2}$ and
  $$y = C_1e^{-bx}\sinh(nx + C_2)$$
  $$= C_3e^{-(b+n)x} + C_4e^{-(b-n)x}$$

For $d^2y/dx^2 + 2b(dy/dx) + a^2y = c$

$$y = c/a^2 + y_1$$

where y_1 is the solution of the previous equation with second term zero.

The preceding two equations are examples of linear differential equations with constant coefficients and their solutions are often found most simply by the use of Laplace transforms [1].

For the linear equation of the n^{th} order

$$A_n(x)d^ny/dx^n + A_{n-1}(x)d^{n-1}y/dx^{n-1} + \ldots + A_1(x)dy/dx + A_0(x)y = E(x)$$

the general solution is

$$y = u + c_1u_1 + c_2u_2 + \ldots + c_nu_n,$$

where u is any solution of the given equation and $u_1, u_2, \ldots, u_n$ form a *fundamental system* of solutions to the homogeneous equation [$E(x) \leftarrow$ zero]. A set of functions has linear independence if its Wronskian determinant, W(x), $\neq 0$, where

$$W(x) = \begin{vmatrix} u_1 & u_2 & \ldots u_n \\ u_1 & u_2 & \ldots u_n \\ \cdot & \cdot & \ldots \cdot \\ u_1^m & u_2^m & \ldots u_n^m \end{vmatrix}$$

and $m = n - 1^{th}$ derivative. (In certain cases, a set of functions may be linearly independent when $W(x) = 0$.)

The Laplace Transformation

The *Laplace transformation* is based upon the Laplace integral which transforms a differential equation expressed in terms of time to an equation expressed in terms of a complex variable $\sigma + j\omega$. The new equation may be manipulated algebraically to solve for the desired quantity as an explicit function of the complex variable.

Essentially three reasons exist for the use of the Laplace transformation:

1. The ability to use algebraic manipulation to solve high-order differential equations
2. Easy handling of boundary conditions
3. The method is suited to the complex-variable theory associated with the Nyquist stability criterion [1].

In Laplace-transformation mathematics, the following symbols and variables are used:

f(t) = a function of time
s = a complex variable of the form $(\sigma + j\omega)$
F(s) = the Laplace transform of f, expressed in s, resulting from operating on f(t) with the Laplace integral.
$\mathcal{L}$ = the Laplace operational symbol, i.e., $F(s) = \mathcal{L}[f(t)]$.

The Laplace integral is defined as

$$\mathcal{L} = \int_0^\infty e^{-st}dt \quad \text{and so}$$

$$\mathcal{L}[f(t)] = \int_0^\infty e^{-st}f(t)dt$$

Table 1-8 lists the transforms of some common time-variable expressions.

Table 1-8
Laplace Transforms

$f(t)$	$F(s) = \mathcal{L}[f(t)]$
A	A/s
$1 = u(t)$	$1/s$
$e^{-\alpha t}$	$\frac{1}{s+\alpha}$
$1/re^{-t/\tau}$	$\frac{1}{\tau s + 1}$
$Ae^{-\alpha t}$	$\frac{A}{s+\alpha}$
$\sin \beta t$	$\frac{\beta}{s^2+\beta^2}$
$\cos \beta t$	$\frac{s}{s^2+\beta^2}$
$\frac{1}{\beta} e^{-\alpha t} \sin \beta t$	$\frac{1}{s^2 + 2\alpha s + \alpha^2 + \beta^2}$
$\frac{e^{-\alpha t}}{\beta - \alpha} - \frac{e^{-\beta t}}{\beta - \alpha}$	$\frac{1}{(s+\alpha)(s+\beta)}$
$\frac{Ae^{-\alpha t} - Be^{-\beta t}}{C}$ where $A = a - \alpha$, $B = a - \beta$, $C = \beta - \alpha$	$\frac{s+a}{(s+\alpha)(s+\beta)}$
$\frac{e^{-\alpha t}}{A} + \frac{e^{-\beta t}}{B} + \frac{e^{-\delta t}}{C}$ where $A = (\beta - \alpha)(\delta - \alpha)$, $B = (\alpha - \beta)(\delta - \beta)$, $C = (\alpha - \delta)(\beta - \delta)$	$\frac{1}{(s+\alpha)(s+\beta)(s+\delta)}$
t^n	$\frac{n!}{s^{n+1}}$
$d/dt[f(t)]$	$sF(s) - f(0^+)$
$d^2/dt^2[f(t)]$	$s^2F(s) - sf(0^+) - \frac{df}{dt}(0^+)$
$d^3/dt^3[f(t)]$	$s^3F(s) - s^2f(0^+) - s\frac{df}{dt}(0^+) - \frac{d^2f}{dt^2}(0^+)$
$\int f(t)dt$	$\frac{1}{s}[F(s) + \int f(t)dt \mid 0^+]$
$\frac{1}{\alpha} \sinh \alpha t$	$\frac{1}{s^2 - \alpha^2}$
$\cosh \alpha t$	$\frac{s}{s^2 - \alpha^2}$

The transform of a first derivative of f(t) is

$$\mathcal{L}\left[\frac{d}{dt}f(t)\right] = sF(s) - f(0^+)$$

where $f(0^+)$ = initial value of f(t) as $t \to 0$ from positive values.
The transform of a second derivative of f(t) is

$$\mathcal{L}[f''(t)] = s^2F(s) - sf(0^+) - f'(0^+)$$

and of $\int f(t)dt$ is

$$\mathcal{L}\left[\int f(t)dt\right] = \frac{f^{-1}(0^+)}{s} + \frac{F(s)}{s}$$

Solutions derived by Laplace transformation are in terms of the complex variable s. In some cases, it is necessary to retransform the solution in terms of time, performing an *inverse transformation*

$$\mathcal{L}^{-1}F(s) = f(t)$$

Just as there is only one direct transform F(s) for any f(t), there is only one inverse transform f(t) for any F(s) and inverse transforms are generally determined through use of tables.

ANALYTIC GEOMETRY

Symmetry

Symmetry exists for the curve of a function about the y-axis if $F(x,y) = F(-x,y)$, about the x-axis if $F(x,y) = F(x,-y)$, about the origin if $F(x,y) = F(-x,-y)$, and about the 45° line if $F(x,y) = F(y,x)$.

Intercepts

Intercepts are points where the curve of a function crosses the axes. The x intercepts are found by setting $y = 0$ and the y intercepts by setting $x = 0$.

Asymptotes

As a point P(x,y) on a curve moves away from the region of the origin (Figure 1-36), the distance between P and some fixed line may tend to zero. If so, the line is called an asymptote of the curve. If N(x) and D(x) are polynomials with no common factor, and

$$y = N(x)/D(x)$$

where $x = c$ is a root of D(x), then the line $x = c$ is an asymptote of the graph of y.

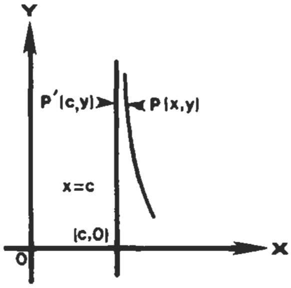


Figure 1-36. Asymptote of a curve.

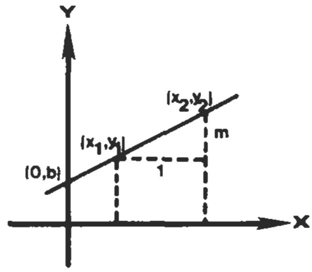


Figure 1-37. Slope of a straight line.

Equations of Slope

1. Two-point equation (Figure 1-37)

$$\frac{y - y_1}{x - x_1} = \frac{y_2 - y_1}{x_2 - x_1}$$

2. Point slope equation (Figure 1-37)

$$y - y_1 = m(x - x_1)$$

3. Slope intercept equation (Figure 1-37)

$$y = mx + b$$

Tangents

If the slope m of the curve of f(x) at (x_1,y_1) is given by (Figure 1-38)

$$m = \frac{dy}{dx(x_1,y_1)} = f'(x)$$

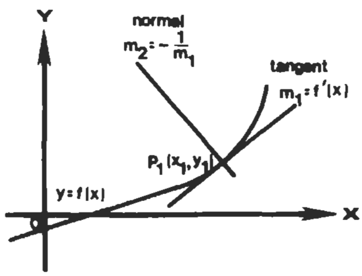


Figure 1-38. Tangent and normal to a curve.

then the equation of the line tangent to the curve at this point is

$$y - y_1 = f'(x_1)\,(x - x_1)$$

and the normal to the curve is the line perpendicular to the tangent with slope m_2 where

$$m_2 = -1/m_1 = -1/f'(x)$$

or

$$y - y_1 = -\,(x - x_1)/f'(x_1)$$

Equations of a Straight Line

- General equation

  $ax + by + c = 0$

- Intercept equation

  $x/a + y/b = 1$

- Normal form (Figure 1-39)

  $x \cos\theta + y \sin\theta - p = 0$

- Distance d from a straight line ($ax + by + c = 0$) to a point $P(x_1,y_1)$

$$d = \frac{ax_1 + by_1 + c}{\pm\sqrt{a^2 + b^2}}$$

- If u is the angle between $ax + by + c = 0$ and $a'x + b'y + c' = 0$ then

$$\cos u = \frac{aa' + bb'}{\pm\sqrt{(a^2 + b^2)(a'^2 + b'^2)}}$$

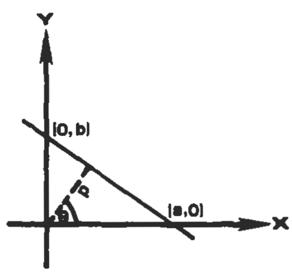


Figure 1-39. Equation of a straight line (normal form).

Equations of a Circle (Center (h,k))

- $(x - h)^2 + (y - k)^2 = r^2$
- Origin at center

  $x^2 + y^2 = r^2$
- General equation

  $x^2 + y^2 + Dx + Ey + F = 0$

  where center = $(-D/2, -E/2)$

  $$\text{radius} = \sqrt{(D/2)^2 + (E/2)^2 - F}$$
- Tangent to circle at (x_1,y_1)

  $$x_1x + y_1y + \frac{1}{2}D(x + x_1) + \frac{1}{2}E(y + y_1) + F = 0$$
- Parametric form, replacing x and y by

  $x = a \cos u$

  and

  $y = a \sin u$

Equations of a Parabola (Figure 1-40)

A parabola is the set of points that are equidistant from a given fixed point (the focus) and from a given fixed line (the directrix) in the plane. The key feature of a parabola is that it is quadrilateral in one of its coordinates and linear in the other.

- $(y - k)^2 = 4p(x - h)$
- Coordinates of the vertex V(h,k) and of the focus F(h + p,k)

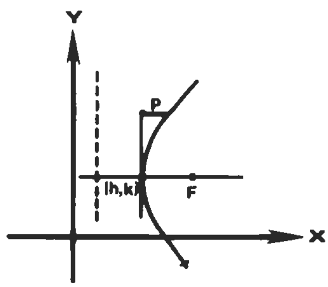


Figure 1-40. Equation of a parabola.

- Origin at vertex

  $y^2 = 4px$
- Equation of the directrix

  $x = h - p$
- Length of latus rectum

  $LL' = 4p$
- Polar equation (focus as origin)

  $r = p/(1 - \cos\theta)$
- Equation of the tangent to $y^2 = 2\,px$ at (x_1,y_1)

  $y_1y = p(x + x_1)$

Equations of an Ellipse of Eccentricity e (Figure 1-41)

- $\dfrac{(x-h)^2}{a^2} + \dfrac{(y-k)^2}{b^2} = 1$
- Coordinates of center C(h,k), of vertices V(h + a,k) and V'(h - a,k), and of foci F(h + ae,k) and F'(h - ae,k)
- Center at origin

  $x^2/a^2 + y^2/b^2 = 1$
- Equation of the directrices

  $x = h \pm a/e$
- Equation of the eccentricity

  $$e = \frac{\sqrt{a^2 - b^2}}{a} < 1$$
- Length of the latus rectum

  $LL' = 2b^2/a$
- Parametric form, replacing x and y by

  $x = a \cos u$ and $y = b \sin u$

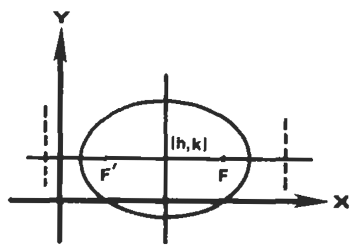


Figure 1-41. Ellipse of eccentricity e.

- Polar equation (focus as origin)

  $r = p/(1 - e \cos \theta)$

- Equation of the tangent at (x_1,y_1)

  $b^2x_1x + a^2y_1y = a^2b^2$

Equations of a Hyperbola (Figure 1-42)

- $\dfrac{(x-h)^2}{a^2} - \dfrac{(y-k)^2}{b^2} = 1$

- Coordinates of the center $C(h,k)$, of vertices $V(h + a,k)$ and $V'(h - a,k)$, and of the foci $F(h + ae,k)$ and $F'(h - ae,k)$

- Center at origin

  $x^2/a^2 - y^2/b^2 = 1$

- Equation of the directrices

  $x = h \pm a/e$

- Equation of the asymptotes

  $y - k = \pm\, b/a \bullet (x - k)$

- Equation of the eccentricity

  $$e = \frac{\sqrt{a^2 + b^2}}{a} > 1$$

- Length of the latus rectum

  $LL' = 2b^2/a$

- Parametric form, replacing x and y

  $x = a \cosh u$ and $y = b \sinh u$

- Polar equation (focus as origin)

  $r = p/(1 - e \cos \theta)$

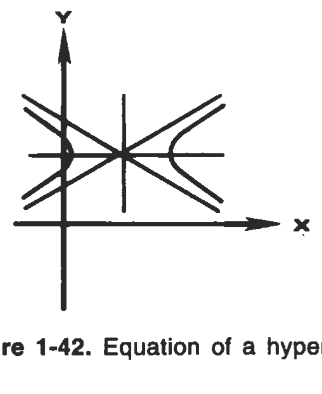


Figure 1-42. Equation of a hyperbola.

- Equation of the tangent at (x,y)

  $b^2x_1x - a^2y_1y = a^2b^2$

Equations of Three-Dimensional Coordinate Systems (Figure 1-43)

- Distance d between two points

  $$d = \sqrt{(x_2 - x_1)^2 + (y_2 - y_1)^2 + (z_2 - z_1)^2}$$

- Direction cosines of a line

  $\lambda = \cos \alpha,\ \mu = \cos \beta,\ v = \cos \gamma$

- Direction numbers, proportional to the direction cosines with k

  $a = k\lambda,\ b = k\mu,\ c = kv$

Equations of a Plane

- $ax + by + cz + d = 0$
- Intercept

  $x/a + y/b + z/c = 1$

- Normal form

  $\lambda x + \mu y + vz - p = 0$

- Distance from $ax + by + cz + d = 0$ to a point $P(x_1,y_1,z_1)$

  $$D = \frac{ax_1 + by_1 + c_1 + d}{\pm\sqrt{a^2 + b^2 + c^2}}$$

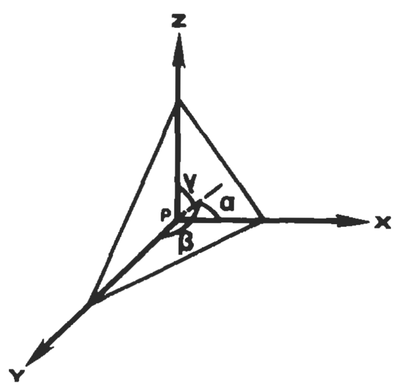


Figure 1-43. Three-dimensional coordinate systems.

Equations of a Line

- Intersection of two planes

$$\begin{cases} a_1x + b_1y + c_1z + d_1 = 0 \\ a_2x + b_2y + c_2z + x_2 = 0 \end{cases}$$

For this line

$$\lambda:\mu:v = \begin{vmatrix} b_1 & c_1 \\ b_2 & c_2 \end{vmatrix} : -\begin{vmatrix} a_1 & c_1 \\ a_2 & c_2 \end{vmatrix} : \begin{vmatrix} a_1 & b_1 \\ a_2 & b_2 \end{vmatrix}$$

- Symmetric form, i.e., through (x_1,y_1,z_1) with direction numbers a, b, and c

$(x - x_1)/a = (y - y_1)/b = (z - z_1)/c$

Through two points

$$\frac{x - x_1}{x_2 - x_1} = \frac{y - y_1}{y_2 - y_1} = \frac{z - z_1}{z_2 - z_1}$$

where $\lambda:\mu:v = (x_2 - x_1):(y_2 - y_1):(z_2 - z_1)$

Equations of Angles

- Between two lines

$\cos\theta = \lambda_1\lambda_2 + \mu_1\mu_2 + v_1v_2$

and the lines are parallel if $\cos\theta = 1$ or perpendicular if $\cos\theta = 0$

- Between two planes, given by the angle between the normals to the planes.

Equation (standard form) of a Sphere (Figure 1-44)

$x^2 + y^2 + z^2 = r$

Equation (standard form) of an Ellipsoid (Figure 1-45)

$x^2/a^2 + y^2/b^2 + z^2/c^2 = 1$

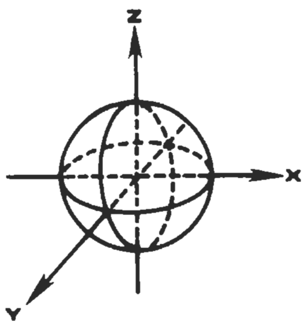


Figure 1-44. Sphere.

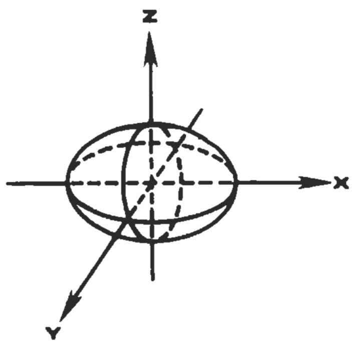


Figure 1-45. Equation of an ellipsoid.

Equations (standard form) of Hyperboloids

- Of one sheet (Figure 1-46)

  $x^2/a^2 + y^2/b^2 - z^2/c^2 = 1$
- Of two sheets (Figure 1-47)

  $x^2/a^2 - y^2/b^2 - z^2/c^2 = 1$

Equations (standard form) of Paraboloids

- Of elliptic paraboloid (Figure 1-48)

  $x^2/a^2 + y^2/b^2 = cz$
- Of hyperbolic paraboloid (Figure 1-49)

  $x^2/a^2 - y^2/b^2 = cz$

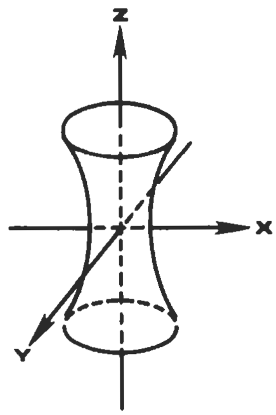


Figure 1-46. Hyperboloid of one sheet.

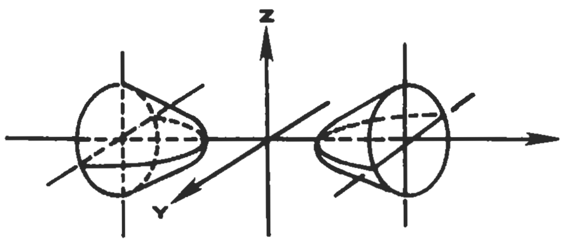


Figure 1-47. Hyperboloid of two sheets.

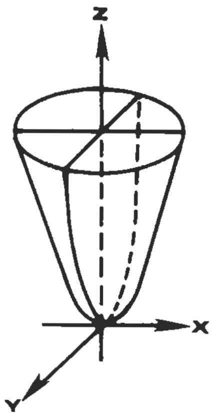


Figure 1-48. Elliptic paraboloid.

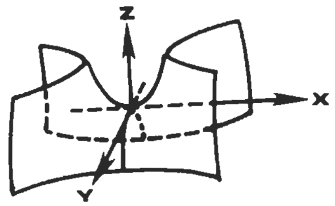


Figure 1-49. Hyperbolic paraboloid.

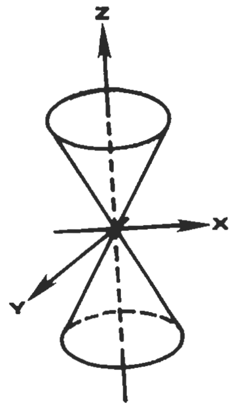


Figure 1-50. Elliptic cone.

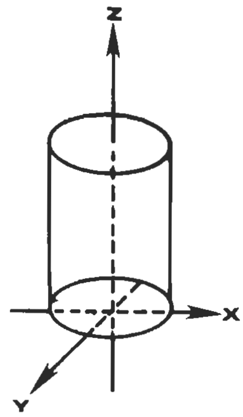


Figure 1-51. Elliptic cylinder.

Equation (standard form) of an Elliptic Cone (Figure 1-50)

$x^2/a^2 + y^2/b^2 - z^2/c^2 = 0$

Equation (standard form) of an Elliptic Cylinder (Figure 1-51)

$x^2/a^2 + y^2/b^2 = 1$

NUMERICAL METHODS

See References 1 and 9–22 for additional information.

Expansion in Series

If the value of a function f(x) can be expressed in the region close to x = a, and if all derivatives of f(x) near a exist and are finite, then by the infinite power series

$$f(x) = f(a) + (x-a)f'(a) + \frac{(x-a)^2}{2!}f''(a) + \ldots + \frac{(x-a)^n}{n!}f^n(a) + \ldots$$

and f(x) is *analytic* near x = a. The preceding power series is called the *Taylor series expansion* of f(x) near x = a. If for some value of x as [x − a] is increased, then the series is no longer convergent, then that value of x is outside the radius of convergence of the series.

The error due to truncation of the series is partially due to [x − a] and partially due to the number of terms (n) to which the series is taken. The quantities [x − a] and n can be controlled and the truncation error is said to be of the order of $(x-a)^{n+1}$ or $\mathcal{O}(x-a)^{n+1}$.

Finite Difference Calculus

In the finite difference calculus, the fundamental rules of ordinary calculus are employed, but Δx is treated as a small quantity, rather than infinitesimal.

Given a function f(x) which is analytic (i.e., can be expanded in a Taylor series) in the region of a point x, where h = Δx, if f(x + h) is expanded about x, f′(x) can be defined at $x = x_i$ as

$$f'(x_i) = f_i' = (f_{i+1} - f_i)/h + \mathcal{O}(h)$$

The first *forward difference* of f at x_i may be written as

$$\Delta f_i = f_{i+1} - f_i$$

and then

$$f'(x) = (\Delta f_i)/h + \mathcal{O}(h)$$

The first *backward difference* of f at x_i is

$$\nabla f_i = f_i - f_{i-1}$$

and f′(x) may also be written as

$$f'(x) = (\nabla f_i)/h + \mathcal{O}(h)$$

The second forward difference of f(x) at x_i is

$$\Delta^2 f_i = f_{i+2} - 2f_{i+1} + f_i$$

and the second derivative of f(x) is then given by

$$f''(x) = (\Delta^2 f_i)/h^2 + \mathcal{O}(h)$$

The second backward difference of f at x_i is

$$\nabla^2 f_i = f_i - 2f_{i-1} + f_{i-2}$$

and f''(x) may also be defined as

$$f''(x) = (\nabla^2 f_i)/h^2 + \mathcal{O}(h)$$

Approximate expressions for derivatives of any order are given in terms of forward and backward difference expressions as

$$f_i^{(n)} = (\Delta^n f_i)/h^n + \mathcal{O}(h) = (\nabla^n f_i)/h^n + \mathcal{O}(h)$$

Coefficients of forward difference expressions for derivatives of up to the fourth order are given in Figure 1-52 and of backward difference expressions in Figure 1-53.

More accurate difference expressions may be found by expanding the Taylor series. For example, f'(x) to ∇(h) is given by forward difference by

$$f'(x) = (-f_{i+2} + 4f_{i+1} - 3f_i)/(2h) + \mathcal{O}(h^2)$$

and a similar backward difference representation can also be easily obtained. These expressions are exact for a parabola. Forward and backward difference expressions of $\mathcal{O}(h^2)$ are contained in Figures 1-54 and 1-55.

A *central difference* expression may be derived by combining the equations for forward and backward differences.

	f_i	f_{i+1}	f_{i+2}	f_{i+3}	f_{i+4}	
$hf'(x_i)$ =	-1	1				
$h^2f''(x_i)$ =	1	-2	1			+ $\mathcal{O}(h)$
$h^3f'''(x_i)$ =	-1	3	-3	1		
$h^4f''''(x_i)$ =	1	-4	6	-4	1	

Figure 1-52. Forward difference coefficients of σ(h).

	f_{i-4}	f_{i-3}	f_{i-2}	f_{i-1}	f_i	
$hf'(x_i)$ =				−1	1	
$h^2f''(x_i)$ =			1	−2	1	$+\ \sigma(h)$
$h^3f'''(x_i)$ =		−1	3	−3	1	
$h^4f''''(x_i)$ =	1	−4	6	−4	1	

Figure 1-53. Backward difference coefficients of σ(h).

	f_i	f_{i+1}	f_{i+2}	f_{i+3}	f_{i+4}	f_{i+5}	
$2hf'(x_i)$ =	−3	4	−1				
$h^2f''(x_i)$ =	2	−5	4	−1			$+\ \sigma(h)^2$
$2h^3f'''(x_i)$ =	−5	18	−24	14	−3		
$h^4f''''(x_i)$ =	3	−14	26	−24	11	−2	

Figure 1-54. Forward difference coefficients of σ(h)².

	f_{i-5}	f_{i-4}	f_{i-3}	f_{i-2}	f_{i-1}	f_i	
$2hf'(x_i)$ =				1	−4	3	
$h^2f''(x_i)$ =			−1	4	−5	2	$+\ \sigma(h)^2$
$2h^3f'''(x_i)$ =		3	−14	24	−18	5	
$h^4f''''(x_i)$ =	−2	11	−24	26	−14	3	

Figure 1-55. Backward difference coefficients of σ(h)².

$$\delta f_i = 1/2 \bullet (\Delta f_i + \nabla f_i) = 1/2 \bullet (f_{i+1} - f_{i-1})$$

The first derivative of f at x_i may then be given in terms of the central difference expression as

$$f'_i = (\delta f_i)/h + \mathcal{O}(h^2)$$

and is accurate to a greater degree than the forward or backward expressions of f′. Central difference expressions for derivatives of any order in terms of forward and backward differences are given by

$$f_i^{(n)} = [\nabla^n f_{i+n/2} + \Delta^n f_{i-n/2}]/(2h^n) + \mathcal{O}(h^2), \text{ n even}$$

and

$$f_i^{(n)} = [\nabla^n f_{i+(n-1)/2} + \Delta^n f_{i-(n-1)/2}]/(2h^n) + \mathcal{O}(h^2), \text{ n odd}$$

Coefficients of central difference expressions for derivatives up to order four of $\mathcal{O}(h^2)$ are given in Figure 1-56 and of $\mathcal{O}(h^4)$ in Figure 1-57.

	f_{i-2}	f_{i-1}	f_i	f_{i+1}	f_{i+2}	
$2hf'(x_i)$ =		-1	0	1		
$h^2f''(x_i)$ =		1	-2	1		$+ \mathcal{O}(h)^2$
$2h^3f'''(x_i)$ =	-1	2	0	-2	1	
$h^4f''''(x_i)$ =	1	-4	6	-4	1	

Figure 1-56. Central difference coefficients of $\sigma(h)^2$.

	f_{i-3}	f_{i-2}	f_{i-1}	f_i	f_{i+1}	f_{i-2}	f_{i+3}	
$12hf'(x_i)$ =		1	-8	0	8	-1		
$12h^2f''(x_i)$ =		-1	16	-30	16	-1		$+ \mathcal{O}(h)^4$
$8h^3f'''(x_i)$ =	1	-8	13	0	-13	8	-1	
$6h^4f''''(x_i)$ =	-1	12	-39	56	-39	12	-1	

Figure 1-57. Central difference coefficients of $\sigma(h)^4$.

Interpolation

A *forward difference table* may be generated (see also "Algebra") using notation consistent with numerical methods as given in Table 1-9. In a similar manner, a *backward difference table* can be calculated as in Table 1-10. A *central difference table* is constructed in the same general manner, leaving a space between each line of original data, then taking the differences and entering them on alternate full lines and half lines (see Table 1-11). The definition of the central difference δ is

$$\delta f_{i+1/2} = f_{i+1} - f_i$$

The quarter lines in the table are filled with the arithmetic mean of the values above and below (Table 1-12).

Given a data table with evenly spaced values of x, and rescaling x so that h = one unit, forward differences are usually used to find f(x) at x near the top of the table and backward differences at x near the bottom. Interpolation near the center of the set is best accomplished with central differences.

The *Gregory-Newton forward formula* is given as

$$f(x) = f(0) + x(\Delta f_0) + \frac{x(x-1)}{2!}\Delta^2 f_0 + \frac{x(x-1)(x-2)}{3!}\Delta^3 f_0 + \ . \ . \ .$$

Table 1-9
Forward Difference Table

x	f(x)	Δf	$\Delta^2 f$	$\Delta^3 f$	$\Delta^4 f$	$\Delta^5 f$
0	0	2	−2	4	2	1
1	2	0	2	6	3	
2	2	2	8	9		
3	4	10	17			
4	14	27				
5	41					

Table 1-10
Backward Difference Table

x	f(x)	Δf	$\Delta^2 f$	$\Delta^3 f$	$\Delta^4 f$	$\Delta^5 f$
0	0					
1	2	2				
2	2	0	−2			
3	4	2	2	4		
4	14	10	8	6	2	
5	41	27	17	9	3	1

Table 1-11
Central Difference Table
(Original Data)

x	f(x)	Δf	Δ^2f	Δ^3f	Δ^4f	Δ^5f
0	0					
		2				
1	2		−2			
		0		4		
2	2		2		2	
		2		6		1
3	4		8		3	
		10		9		
4	14		17			
		27				
5	41					

Table 1-12
Central Difference Table
(Filled)

x	f(x)	Δf	Δ^2f	Δ^3f	Δ^4f	Δ^5f
0	0					
0.5	1	2				
1	2	1	−2			
1.5	2	0	0	4		
2	2	1	2	5	2	
2.5	3	2	5	6	2.5	1
3	4	6	8	7.5	3	
3.5	9	10	12.5	9		
4	14	18.5	17			
4.5	27.5	27				
5	41					

and the *Gregory-Newton backward formula* as

$$f(x) = f(0) + x(\nabla f_0) + \frac{x(x+1)}{2!}\nabla^2 f_0 + \frac{x(x+1)(x+2)}{3!}\nabla^3 f_0 + \ldots$$

To use central differences, the origin of x must be shifted to a base line (shaded area in Table 1-13) and x rescaled so one full (two half) line spacing = 1 unit. *Sterling's formula* (full lines as base) is defined as

$$f(x) = f(0) + x(\delta y_0) + \frac{x^2}{2!}(\delta^2 y_0) + \frac{x(x^2-1)}{3!}(\delta^3 y_0) + \frac{x^2(x^2-1)}{4!}(\delta^4 y_0)$$
$$+ \frac{x(x^2-1)(x^4-4)}{5!}(\delta^5 y_0) + \ldots$$

Table 1-13
Central Difference Table with Base Line

Old x	New x	f(x)	Δf	Δ^2f	Δ^3f	Δ^4f	Δ^5f
0	−2.5	0					
0.5	−2.0	1	2				
1	−1.5	2	1	−2			
1.5	−1.0	2	0	0	4		
2	−0.5	2	1	2	5	2	
2.5	0.0	3	2	5	6	2.5	1
3	+0.5	4	6	8	7.5	3	
3.5	+1.0	9	10	12.5	9		
4	+1.5	14	18.5	17			
4.5	+2.0	27.5	27				
5	+2.5	41					

and *Bessel's formula* (half line as base) as

$$f(x) = f(0) + x(\delta y_0) = \frac{\left(x^2 - \frac{1}{4}\right)}{2!}(\delta^2 y_0) + \frac{x\left(x^2 - \frac{1}{4}\right)}{3!}(\delta^3 y_0)$$

$$+ \frac{\left(x^2 - \frac{1}{4}\right)\left(x^2 - \frac{9}{4}\right)}{4!}(\delta^4 y_0) + \frac{x\left(x^2 - \frac{1}{4}\right)\left(x^2 - \frac{9}{4}\right)}{5!}(\delta^5 y_0) + \ldots$$

Interpolation with nonequally spaced data may be accomplished by the use of *Lagrange Polynomials,* defined as a set of n^{th} degree polynomials such that each one, $P_j(x)$ ($j = 0, 1, \ldots, n$), passes through zero at each of the data points except one, x_k, where $k = j$. For each polynomial in the set

$$P_j(x) = A_j \prod_{\substack{i=0 \\ i \neq j}}^{n} (x - x_i)$$

where if

$$A_j = \frac{1}{\prod_{\substack{i=0 \\ i \neq j}}^{n} (x_j - x_i)}$$

then

$$P_j(x) = \begin{cases} 0, & k \neq j \\ 1, & k = j \end{cases}$$

and the linear combination of $P_j(x)$ may be formed

$$p_n(x) = \sum_{j=0}^{n} f(x_j)P_j(x)$$

It can be seen that for any x_i, $p_n(x_i) = f(x_i)$.

Interpolation of this type may be extremely unreliable toward the center of the region where the independent variable is widely spaced. If it is possible to select the values of x for which values of f(x) will be obtained, the maximum error can be minimized by the proper choices. In this particular case Chebyshev polynomials can be computed and interpolated [11].

Neville's algorithm constructs the same unique interpolating polynomial and improves the straightforward Lagrange implementation by the addition of an error estimate.

If $P_i(i = 1, \ldots, n)$ is defined as the value at x of the unique polynomial of degree zero passing through the point (x_i,y_i) and P_{ij} $(i = 1, \ldots, n - 1, j = 2, \ldots, n)$ the polynomial of degree one passing through both (x_i,y_i) and (x_j,y_j), then the higher-order polynomials may likewise be defined up to $P_{123\ldots n}$, which is the value of the unique interpolating polynomial passing through all n points. A table may be constructed, e.g., if n = 3

$$\begin{array}{lll} x_1\colon\ y_1 = P_1 & & \\ & P_{12} & \\ x_2\colon\ y_2 = P_2 & & P_{123} \\ & P_{23} & \\ x_3\colon\ y_3 = P_3 & & \end{array}$$

Neville's algorithm recursively calculates the preceding columns from left to right as

$$P_{i(i+1)\ldots(i+m)} = \frac{(x - x_{i+m})P_{i(i+1)\ldots(i+m-1)} + (x_i - x)P_{(i+1)\ldots(i+m)}}{x_i - x_{i+m}}$$

In addition the differences between the columns may be calculated as

$$D_{m+1,i} = \frac{(x_{i+m+1} - x)(C_{m,i+1} + D_{m,i})}{x_i - x_{i+m+1}}$$

$$C_{m+1,i} = \frac{(x_i - x)(C_{m,i+1} - D_{m,i})}{x_i - x_{i+m+1}}$$

and $P_{12\ldots n}$ is equal to the sum of any y_i plus a set of C's and/or D's that lead to the rightmost member of the table [22].

Functions with localized strong inflections or poles may be approximated by *rational functions* of the general form

$$R(x) = \frac{\sum_{i=0}^{n} a_i x^i}{\sum_{i=0}^{m} b_i x^i}$$

as long as there are sufficient powers of x in the denominator to cancel any nearby poles. Bulirsch and Stoer [16] give a Neville-type algorithm that performs rational function extrapolation on tabulated data

$$R_{i(i+1)\ldots(i+m)} = R_{(i+1)\ldots(i+m)} + \frac{R_{(i+1)\ldots(i+m)} - R_{i\ldots(i+m-1)}}{\left[\frac{x - x_i}{x - x_{i+m}}\right]\left[1 - \frac{R_{(i+1)\ldots(i+m)} - R_{i\ldots(i+m-1)}}{R_{(i+1)\ldots(i+m)} - R_{(i+1)\ldots(i+m-1)}}\right] - 1}$$

starting with $R_i = y_i$ and returning an estimate of error, calculated by C and D in a manner analogous with Neville's algorithm for polynomial approximation.

In a high-order polynomial, the highly inflected character of the function can more accurately be reproduced by the *cubic spline function*. Given a series of $x_i(i = 0, 1, \ldots, n)$ and corresponding $f(x_i)$, consider that for two arbitrary and adjacent points x_i and x_{i+j}, the cubic fitting these points is

$$F_i(x) = a_0 + a_i x + a_2 x^2 + a_3 x^3$$
$$(x_i \leq x \leq x_{i-1})$$

The approximating cubic spline function g(x) for the region $(x_0 \leq x \leq x_n)$ is constructed by matching the first and second derivatives (slope and curvature) of $F_i(x)$ to those of $F_{i-1}(x)$, with special treatment (outlined below) at the end points, so that g(x) is the set of cubics $F_i(x)$, $i = 0, 1, 2, \ldots, n - 1$, and the second derivative $g''(x)$ is continuous over the region. The second derivative varies linearly over $[x_0, x_n]$ and at any x $(x_i \leq x \leq x_{i+1})$

$$g''(x) = g''(x_i) + \frac{x - x_i}{x_{i+1} - x_i}[g''(x_{i+1}) - g''(x_i)]$$

Integrating twice and setting $g(x_i) = f(x_i)$ and $g(x_{i+1}) = f(x_{i+1})$, then using the derivative matching conditions

$$F_i'(x_i) = F_{i+1}'(x_i) \text{ and } F_i''(x_i) = F_{i-1}''(x_i)$$

and applying the condition for $i = [1, n - 1]$ finally yields a set of linear simultaneous equations of the form

$$[\Delta x_{i-1}]g''(x_{i-1}) + [2(x_{i+1} - x_{i-1})]g''(x_i) + [\Delta x_i]g''(x_{i+1})$$
$$= 6\left[\frac{f(x_{i+1}) - f(x_i)}{\Delta x_i} - \frac{f(x_i) - f(x_{i-1})}{\Delta x_{i-1}}\right]$$

where $i = 1, 2, \ldots, n - 1$
$\Delta x_i = x_{i+1} - x_i$

If the x_i are equally spaced by Δx, then the preceding equation becomes

$$[1]g''(x_{i-1}) + [4]g''(x_i) + [1]g''(x_{i+1}) = 6\left[\frac{f(x_{i+1}) - 2f(x_i) + f(x_{i-1})}{(\Delta x_i)^2}\right]$$

There are $n - 1$ equations in $n + 1$ unknowns and the two necessary additional equations are usually obtained by setting

$$g''(x_0) = 0 \text{ and } g''(x_n) = 0$$

and g(x) is now referred to as a *natural cubic spline.* $g''(x_0)$ or $g''(x_n)$ may alternatively be set to values calculated so as to make g' have a specified value on either or both boundaries. The cubic appropriate for the interval in which the x value lies may now be calculated (see "Solutions of Simultaneous Linear Equations").

Extrapolation is required if f(x) is known on the interval [a,b], but values of f(x) are needed for x values not in the interval. In addition to the uncertainties of interpolation, extrapolation is further complicated since the function is fixed only on one side. Gregory-Newton and Lagrange formulas may be used for extrapolation (depending on the spacing of the data points), but all results should be viewed with extreme skepticism.

Roots of Equations

Finding the root of an equation in x is the problem of determining the values of x for which $f(x) = 0$. *Bisection,* although rarely used now, is the basis of several more efficient methods. If a function f(x) has one and only one root in [a,b], then the interval may be bisected at $x_m = (a + b)/2$. If $f(x_m) \bullet f(b) < 0$, the root is in $[x_m,b]$, while if $f(x_m) \bullet f(b) > 0$, the root is in $[a,x_m]$. Bisection of the appropriate intervals, where $x'_m = (a' + b')/2$, is repeated until the root is located $\pm \varepsilon$, ε being the maximum acceptable error and $\varepsilon \leq 1/2 \bullet$ size of interval.

The *Regula Falsa method* is a refinement of the bisection method, in which the new end point of a new interval is calculated from the old end points by

$$x_m = a - (b - a)\frac{f(a)}{f(b) - (a)}$$

Whether x_m replaces a or replaces b depends again on the sign of a product, thus

if $f(a) \bullet f(x_m) < 0$, then the new interval is $[a,x_m]$

or

if $f(x_m) \bullet f(b) < 0$, then the new interval is $[x_m,b]$

Because of round off errors, the Regula Falsa method should include a check for excessive iterations. A *modified Regula Falsa method* is based on the use of a *relaxation factor,* i.e., a number used to alter the results of one iteration before inserting into the next. (See the section on relaxation methods and "Solution of Sets of Simultaneous Linear Equations.")

By iteration, the general expression for the *Newton-Raphson method* may be written (if f′ can be evaluated and is continuous near the root):

$$x^{(n+1)} - x^{(n)} = \delta^{(n+1)} = -\frac{f(x^{(n)})}{f'(x^{(n)})}$$

where (n) denotes values obtained on the n^{th} iteration and (n + 1) those obtained on the $(n + 1)^{th}$ iteration. The iterations are terminated when the magnitude of $|\delta^{(n+1)} - \delta^{(n)}| < \varepsilon$, being the predetermined error factor and $\varepsilon \cong 0.1$ of the permissible error in the root.

The *modified Newton method* [12] offers one way of dealing with multiple roots. If a new function is defined

$$u(x) = \frac{f(x)}{f'(x)}$$

since u(x) = 0 when f(x) = 0 and if f(x) has a multiple root at x = c of multiplicity r, then Newton's method can be applied and

$$x^{(n+1)} - x^{(n)} = \delta^{(n+1)} = -\frac{u(x^{(n)})}{u'(x^{(n)})}$$

where

$$u'(x) = 1 - \frac{f(x)f''(x)}{[f'(x)]^2}$$

If multiple or closely spaced roots exist, both f and f′ may vanish near a root and therefore methods that depend on tangents will not work. Deflation of the polynomial P(x) produces, by factoring,

$$P(x) = (x - r)Q(x)$$

where Q(x) is a polynomial of one degree lower than P(x) and the roots of Q are the remaining roots of P after factorization by synthetic division. Deflation avoids convergence to the same root more than one time. Although the calculated roots become progressively more inaccurate, errors may be minimized by using the results as initial guesses to iterate for the actual roots in P.

Methods such as Graeffe's root-squaring method, Muller's method, Laguerre's method, and others exist for finding all roots of polynomials with real coefficients. [12 and others]

Solution of Sets of Simultaneous Linear Equations

A matrix is a rectangular array of numbers, its size being determined by the number of rows and columns in the array. In this context, the primary concern is with square matrices, and matrices of column dimension 1 (column vectors) and row dimension 1 (row vectors).

Certain configurations of square matrices are of particular interest. If

$$C = \begin{vmatrix} c_{11} & . & . & c_{14} \\ . & . & . & . \\ . & . & . & . \\ c_{41} & . & . & c_{44} \end{vmatrix}$$

the diagonal consisting of c_{11}, c_{22}, c_{33} and c_{44} is the *main diagonal.* The matrix is *symmetric* if $c_{ij} = c_{ji}$. If all elements below the main diagonal are zero (blank), it is an *upper triangular matrix,* while if all elements above the main diagonal are zero, it is a *lower triangular matrix.* If all elements are zero except those on the main diagonal, the matrix is a *diagonal matrix* and if a diagonal matrix has all ones on the diagonal, it is the *unit, or identity, matrix.*

Matrix addition (or subtraction) is denoted as S = A + B and defined as

$$s_{ij} = a_{ij} + b_{ij}$$

where A, B, and S have identical row and column dimensions. Also,

$$A + B = B + A$$

$$A - B = -B + A$$

Matrix multiplication, represented as P = AB, is defined as

$$P_{ij} = \sum_{k=1}^{n} a_{ik} b_{kj}$$

where n is the column dimension of A and the row dimension of B. P will have row dimension of A and column dimension of B. Also

$$AI = A$$

and

$$IA = A$$

while, in general,

$$AB \neq BA$$

Matrix division is not defined, although if C is a square matrix, C^{-1} (the *inverse* of C) can usually be defined so that

$$CC^{-1} = I$$

and

$$(C^{-1})^{-1} = C$$

The *transpose* of A if

$$A = \begin{vmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \\ a_{31} & a_{32} \end{vmatrix}$$

is

$$A^T = \begin{vmatrix} a_{11} & a_{21} & a_{31} \\ a_{12} & a_{22} & a_{32} \end{vmatrix}$$

A square matrix C is *orthogonal* if

$$C^T = C^{-1}$$

The *determinant* of a square matrix C (det C) is defined as the sum of all possible products found by taking one element from each row in order from the top and one element from each column, the sign of each product multiplied by $(-1)^r$, where r is the number of times the column index decreases in the product.

For a 2 × 2 matrix

$$C = \begin{vmatrix} c_{11} & c_{12} \\ c_{21} & c_{22} \end{vmatrix}$$

$$\det C = c_{11}c_{22} - c_{12}c_{21}$$

(Also see discussion of determinants in "Algebra.")

Given a set of simultaneous equations, for example, four equations in four unknowns:

$$\begin{aligned}
c_{11}x_1 + c_{12}x_2 + c_{13}x_3 + c_{14}x_4 &= r_1 \\
c_{21}x_1 + c_{22}x_2 + c_{23}x_3 + c_{24}x_4 &= r_2 \\
c_{31}x_1 + c_{32}x_2 + c_{33}x_3 + c_{34}x_4 &= r_3 \\
c_{41}x_1 + c_{42}x_2 + c_{43}x_3 + c_{44}x_4 &= r_4
\end{aligned}$$

and in matrix form

$$\begin{vmatrix} c_{11} & c_{12} & c_{13} & c_{14} \\ c_{21} & c_{22} & c_{23} & c_{24} \\ c_{31} & c_{32} & c_{33} & c_{34} \\ c_{41} & c_{42} & c_{43} & c_{44} \end{vmatrix} \begin{vmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{vmatrix} = \begin{vmatrix} r_1 \\ r_2 \\ r_3 \\ r_4 \end{vmatrix}$$

or

$$CX = R$$

The solution for x_k in a system of equations such as given in the matrix above is

$$x_k = (\det C_k)/(\det C)$$

where C_k is the matrix C, with its k^{th} column replaced by R (Cramer's Rule). If det C = 0, C and its equations are singular and there is no solution.

Sets of simultaneous linear equations are frequently defined as [12]:

- *Sparse* (many zero elements) and large
- *Dense* (few zero elements) and small. A *banded matrix* has all zero elements except for a band centered on the main diagonal, e.g.,

$$C = \begin{vmatrix} c_{11} & c_{12} & & \\ c_{21} & c_{22} & c_{23} & \\ & c_{32} & c_{33} & c_{34} \\ & & c_{43} & c_{44} \end{vmatrix}$$

then C is a banded matrix of bandwidth 3, also called a *tridiagonal matrix.*

Equation-solving techniques may be defined as *direct,* expected to yield results in a predictable number of operations, or *iterative,* yielding results of increasing accuracy with increasing numbers of iterations. Iterative techniques are in general preferable for very large sets and for large, sparse (not banded) sets. Direct methods are usually more suitable for small, dense sets and also for sets having banded coefficient matrices.

Gauss elimination is the sequential application of the two operations:

1. Multiplication, or division, of any equation by a constant.
2. Replacement of an equation by the sum, or difference, of that equation and any other equation in the set, so that a set of equations

$$\begin{vmatrix} c_{11} & c_{12} & c_{13} & c_{14} \\ c_{21} & c_{22} & c_{23} & c_{24} \\ c_{31} & c_{32} & c_{33} & c_{34} \\ c_{41} & c_{42} & c_{43} & c_{44} \end{vmatrix} \begin{vmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{vmatrix} = \begin{vmatrix} r_1 \\ r_2 \\ r_3 \\ r_4 \end{vmatrix}$$

becomes, by division of the first equation by c_{11},

$$\begin{vmatrix} 1 & c'_{12} & c'_{13} & c'_{14} \\ c_{21} & c_{22} & c_{23} & c_{24} \\ c_{31} & c_{32} & c_{33} & c_{34} \\ c_{41} & c_{42} & c_{43} & c_{44} \end{vmatrix} \begin{vmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{vmatrix} = \begin{vmatrix} r'_1 \\ r_2 \\ r_3 \\ r_4 \end{vmatrix}$$

then, by replacement of the next three equations,

$$\begin{vmatrix} 1 & c'_{12} & c'_{13} & c'_{14} \\ 0 & c'_{22} & c'_{23} & c'_{24} \\ 0 & c'_{32} & c'_{33} & c'_{34} \\ 0 & c'_{42} & c'_{43} & c'_{44} \end{vmatrix} \begin{vmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{vmatrix} = \begin{vmatrix} r'_1 \\ r'_2 \\ r'_3 \\ r'_4 \end{vmatrix}$$

and finally

$$\begin{vmatrix} 1 & c'_{12} & c'_{13} & c'_{14} \\ & 1 & c'_{23} & c'_{24} \\ & & 1 & c'_{34} \\ & & & 1 \end{vmatrix} \begin{vmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{vmatrix} = \begin{vmatrix} r'_1 \\ r'_2 \\ r'_3 \\ r'_4 \end{vmatrix}$$

Gauss-Jordan elimination is a variation of the preceding method, which by continuation of the same procedures yields

$$\begin{vmatrix} 1 & & & \\ & 1 & & \\ & & 1 & \\ & & & 1 \end{vmatrix} \begin{vmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{vmatrix} = \begin{vmatrix} r''_1 \\ r''_2 \\ r''_3 \\ r''_4 \end{vmatrix}$$

Therefore, $x_1 = r''_1$, etc., i.e., the r vector is the solution vector. If the element in the current pivot position is zero or very small, switch the position of the entire pivot row with any row below it, including the x vector element, but not the r vector element.

If det $C \neq 0$, C^{-1} exists and can be found by *matrix inversion* (a modification of the Gauss-Jordan method), by writing C and I (the identity matrix) and then performing the same operations on each to transform C into I and, therefore, I into C^{-1}.

If a matrix is ill-conditioned, its inverse may be inaccurate or the solution vector for its set of equations may be inaccurate. Two of the many ways to recognize possible ill-conditioning are

1. If there are elements of the inverse of the matrix that are larger than elements of the original matrix.
2. If the magnitude of the determinant is small, i.e., if

$$\frac{\det C}{\sqrt{\sum_{i=1}^{n}\sum_{j=1}^{n} c_{ij}^2}} \leq 1$$

Gauss-Siedel method is an iterative technique for the solution of sets of equations. Given, for example, a set of three linear equations

$$c_{11}x_1 + c_{12}x_2 + c_{13}x_3 = r_1$$
$$c_{21}x_1 + c_{22}x_2 + c_{23}x_3 = r_2$$
$$c_{31}x_1 + c_{32}x_2 + c_{33}x_3 = r_3$$

solving for the unknowns yields

$$x_1 = \frac{r_1 - c_{12}x_2 - c_{13}x_3}{c_{11}}$$

$$x_2 = \frac{r_2 - c_{21}x_1 - c_{23}x_3}{c_{22}}$$

$$x_3 = \frac{r_3 - c_{31}x_1 - c_{32}x_2}{c_{33}}$$

By making an initial guess for x_1, x_2, and x_3, denoted as x_1^0, x_2^0, and x_3^0, the value of x_1 on the first iteration is

$$x_1^{(1)} = \frac{r_1 - c_{12}x_2^{(0)} - c_{13}x_3^{(0)}}{c_{11}}$$

Using the most recently obtained values for each unknown (as opposed to the fixed point or Jacobi method), then

$$x_2^{(1)} = \frac{r_2 - c_{21}x_1^{(1)} - c_{23}x_3^{(0)}}{c_{22}}$$

$$x_3^{(1)} = \frac{r_3 - c_{31}x_1^{(1)} - c_{32}x_2^{(1)}}{c_{33}}$$

If the equations have the proper characteristics, the iterative process will eventually converge. Commonly used convergence criteria are of two types:

1. *Absolute convergence* criteria of the form

$$\left|x_i^{(n+1)} - x_i^{(n)}\right| \leq \varepsilon$$

are most useful when approximate magnitudes of x_i are known beforehand so that ε may be chosen to be proportional to x_i.

2. *Relative convergence* criteria of the form

$$\left|(x_i^{(n+1)} - x_i^{(n)})/x_i^{(n+1)}\right| \leq \varepsilon$$

is the choice if the magnitudes of x_i are uncertain.

Relaxation methods may also be used to modify the value of an unknown before it is used in the next calculation. The effect of the relaxation factor λ may be seen in the following equation, where $x_i^{(n+1)*}$ is the value obtained at the present iteration.

$$x_i^{(n+1)} = \lambda x_i^{(n+1)*} + (1 - \lambda)x_i^{(n)}$$

and $0 < \lambda < 2$. If $0 < \lambda < 1$, the effect is termed underrelaxation, which is frequently employed to produce convergence in a nonconvergent process. If $1 < \lambda < 2$, the effect, overrelaxation, will be to accelerate an already convergent process.

Least Squares Curve Fitting

For a function f(x) given only as discrete points, the measure of accuracy of the fit is a function $d(x) = | f(x) - g(x) |$ where g(x) is the approximating function to f(x). If this is interpreted as minimizing d(x) over all x in the interval, one point in error can cause a major shift in the approximating function towards that point. The better method is the least squares curve fit, where d(x) is minimized if

$$E = \sum_{i=1}^{n} [g(x_i) - f(x_i)]^2$$

is minimized, and for a polynomial of order m

$$E = \sum_{i=1}^{n} [a_0 + a_1 x_i + a_2 x_i^2 + \ . \ . \ . \ + a_m x_i^m - f(x_i)]^2$$

Setting the partial derivatives of E with respect to each of the coefficients of g(x) equal to zero, differentiating and summing over 1, . . . , n forms a set of m + 1 equations [9] so that

$$\begin{vmatrix} n & \sum x_i & \sum x_i^2 \ldots \\ \sum x_i & \sum x_i^2 & \sum x_i^3 \ldots \\ \ldots & & \end{vmatrix} \begin{vmatrix} a_0 \\ a_1 \\ \ldots \end{vmatrix} = \begin{vmatrix} \sum f(x_i) \\ \sum x_i f(x_i) \\ \ldots \end{vmatrix}$$

If the preceding solution is reduced to a linear approximation (n = 1), the matrix will be (n = 1)

$$\begin{vmatrix} n & \sum x_i \\ \sum x_i & \sum x_i^2 \end{vmatrix} \begin{vmatrix} a_0 \\ a_1 \end{vmatrix} = \begin{vmatrix} \sum f(x)_i \\ \sum x_i f(x_i) \end{vmatrix}$$

and for a parabola (n = 2), the first three rows and columns.

Another possible form is the exponential function

$$F(t) = \alpha e^{\beta t}$$

and although partial differentiation will produce two equations in two unknowns, they will be nonlinear and cannot be written in matrix form. However, a change in variable form may produce a model that is linear, for example, for the preceding equation

$$\ln (F) = \ln (\alpha) + \beta t$$

and if X is defined to be t, Y to be ln (F), $a_0 = \ln (\alpha)$, and $a_1 = \beta$, the equation becomes

$$Y(x) = a_0 + a_1X$$

and linear least squares analysis may be applied.

In order to determine the quality (or the validity) of fit of a particular function to the data points given, a comparison of the deviation of the curve from the data to the size of the experimental error can be made. The deviations (i.e., the scatter off the curve) should be of the same order of magnitude as the experimental error, so that the quantity "chi-squared" is defined as

$$X^2 = \sum_{i=1}^{n} \frac{[y_i' - y_i]^2}{(\Delta y_i)^2}$$

where y_i' = is the fitted function and y_i is the measured value of y at x_i, so that Δy_i is the magnitude of the error of y_i'. The sum is over n points and if the number of parameters in the model function is g_2, then if $\mathcal{O}(X^2) > \mathcal{O}(n - g)$, the approximating function is a poor fit, while if $\mathcal{O}(X^2) < \mathcal{O}(n - g)$, the function may be overfit, representing noise [10].

The simplest form of approximation to a *continuous* function is some polynomial. Continuous functions may be approximated in order to provide a "simpler form" than the original function. Truncated power series representations (such as the Taylor series) are one class of polynomial approximations.

The Chebyshev polynomials, $T_n(x)$ or T_n, which exist on the interval $-1 \leq x \leq 1$, form the series

$$T_0 = 1$$
$$T_1 = x$$
$$\cdot$$
$$\cdot$$
$$T_{n+1} = 2xT_n - T_{n-1}$$

(see Table 1-14 for a more complete list). If these polynomials are inverted, powers of x are given in terms of $T_n(x)$ (Table 1-15). Any finite interval $a \leq y \leq b$ can be mapped onto the interval $-1 \leq x \leq 1$ by the formula

$$x = (2y - b - a)/(b - a)$$

Table 1-14
Chebyshev Polynomials

$T_0(x) = 1$
$T_1(x) = x$
$T_2(x) = 2x^2 - 1$
$T_3(x) = 4x^3 - 3x$
$T_4(x) = 8x^4 - 8x^2 + 1$
$T_5(x) = 16x^5 - 20x^3 + 5x$
$T_6(x) = 32x^6 - 48x^4 + 18x^2 - 1$
$T_7(x) = 64x^7 - 112x^5 + 56x^3 - 7x$
$T_8(x) = 128x^8 - 256x^6 + 160x^4 - 32x^2 + 1$

Table 1-15
Inverted Chebyshev Polynomials

$$1 = T_0$$

$$x = T_1$$

$$x^2 = \frac{1}{2}(T_0 + T_2)$$

$$x^3 = \frac{1}{4}(3T_1 + T_3)$$

$$x^4 = \frac{1}{8}(3T_0 + 4T_2 + T_4)$$

$$x^5 = \frac{1}{16}(10T_1 + 5T_3 + T_5)$$

$$x^6 = \frac{1}{32}(10T_0 + 15T_2 + 6T_4 + T_6)$$

$$x^7 = \frac{1}{64}(35T_1 + 21T_3 + 7T_5 + T_7)$$

$$x^8 = \frac{1}{128}(35T_0 + 56T_2 + 28T_4 + 8T_8 + T_8)$$

and the inverted Chebyshev polynomials can be substituted for powers of x in a power series representing any function f(x). Since the maximum magnitude for $T_n = 1$ because of the interval, the sum of the magnitudes of lower-order terms is relatively small. Therefore, even with truncation of the series after comparatively few terms, the altered series can provide sufficient accuracy.

See also the discussion on cubic splines in "Interpolation."

Numerical Integration

By assuming that a function can be replaced over a limited range by a simpler function and by first considering the simplest function, a straight line, the areas under a complicated curve may be approximated by the *trapezoidal rule*. The area is subdivided into n panels and

$$I = T_n = \frac{1}{2}\Delta x_n\left(f_a + 2\sum_{i=1}^{n-1} f_i + f_b\right)$$

where $\Delta x_n = (b - a)/n$ and f_i is the value of the function at each x_i. If the number of panels $n = 2^k$, an alternate form of the trapezoidal can be given, where

$$I = T_k = \frac{1}{2}T_{k-1} + \Delta x_k \sum_{\substack{i=1 \\ i\ \mathrm{odd}}}^{n-1} f(a + i\,\Delta x_k)$$

where $\Delta x_k = (b - a)/2^k$, $T_0 = 1/2(f_a + f_b)(b - a)$, and the equation for T_k is repeatedly applied for $k = 1,2, \ldots$ until sufficient accuracy has been obtained.

If the function f(x) is approximated by parabolas, *Simpson's Rule* is obtained, by which (the number of panels n being even)

$$I = S_n = \frac{1}{3}\Delta x\left[f_0 + 4\sum_{\substack{i=1 \\ i\ \mathrm{odd}}}^{n-1} f_i + 2\sum_{\substack{i=2 \\ i\ \mathrm{even}}}^{n-2} f_i + f_n\right] + E$$

where E is the dominant error term involving the fourth derivative of f, so that it is impractical to attempt to provide error correction by approximating this term. Instead Simpson's rule *with end correction* (sixth order rather than fourth order) may be applied where

$$I = S_n = \frac{1}{15}\Delta x\left[14\left\{\frac{1}{2}(f_0 + f_n) + \sum_{\substack{i=2 \\ i\ \mathrm{even}}}^{n-2} f_i\right\} + 16\sum_{\substack{i=1 \\ i\ \mathrm{odd}}}^{n-1} f_i + \Delta x[f'(a) - f'(b)]\right]$$

The original Simpson's formula without end correction may be generalized in a similar way as the trapezoidal formula for $n = 2^k$ panels, using $\Delta x_k = (b - a)/2^k$ and increasing k until sufficient accuracy is achieved, where

$$S_k = \frac{1}{3}\Delta x_k\left[f_a + 4\sum_{\substack{i=1 \\ i\ \mathrm{odd}}}^{n-1} f(a + i\Delta x_k) + 2\sum_{\substack{i=2 \\ i\ \mathrm{even}}}^{n-2} f(a + i\Delta x_k) + f_b\right]$$

For the next higher level of integration algorithm, f(x) over segments of [a,b] can be approximated by a cubic and if this k^{th} order result is C_k, then *Cote's rule* can be given as

$$C_k = S_k + (S_k - S_{k-1})/15$$

and the next higher degree approximation as

$$D_k = C_k + (C_k - C_{k-1})/63$$

The limit suggested by the sequence

$$T_k \rightarrow S_k \rightarrow C_k \rightarrow D_k \rightarrow \dots$$

is known as *Romberg Integration.*

If a new notation $T_k^{(m)}$ is defined, where k is the order of the approximation ($n = 2^k$) and m is the level of the integration algorithm, then

m = 0 (trapezoidal rule)

$$T_k^{(0)} = T_k$$

m = 1 (Simpson's rule)

$$T_k^{(1)} = S_k$$

m = 2 (Cote's rule)

$$T_k^{(2)} = C_k$$

m = 3

$$T_k^{(3)} = D_k$$

etc.

The generalization of the preceding definitions leads to the Romberg equation

$$T_k^{(m+1)} = T_k^{(m)} + \frac{T_k^{(m)} - T_{k-1}^{(m)}}{4^{(m+1)} - 1}$$

The procedure is to start with the one-panel trapezoidal rule

$$T_0 \rightarrow T_0^{(0)} = \frac{1}{2}(b-a)(f_a + f_b)$$

and then increase the order (k) of the calculation by

$$T_k = \frac{1}{2} T_{k-1} + \Delta x_k \bullet \sum_{\substack{i=1 \\ i \text{ odd}}}^{n-1} f(a + i \Delta k_k)$$

and next increase the level of the algorithm m by the equation for $T_k^{(m+1)}$ just shown. In terms of $T_k^{(m)}$ for the first few k and m [10].

m	0	1	2	3	4
k					
0	$T_0^{(0)}$				
1	$T_1^{(0)}$	$T_1^{(1)}$			
2	$T_2^{(0)}$	$T_2^{(1)}$	$T_2^{(2)}$		
3	$T_3^{(0)}$	$T_3^{(1)}$	$T_3^{(2)}$	$T_3^{(3)}$	
4	$T_4^{(0)}$	$T_4^{(1)}$	$T_4^{(2)}$	$T_4^{(3)}$	$T_4^{(4)}$

Increasing accuracy may be obtained by stepping down or across the table, while the most accurate approximation will be found on the lower vertex of the diagonal. The Romberg procedure is terminated when the values along the diagonal no longer change significantly, i.e., when the relative convergence criterion is less than some predetermined ε. In higher-level approximations, subtraction of like numbers occurs and the potential for round-off error increases. In order to provide a means of detecting this problem, a value is defined

$$R_k^{(m)} = \frac{1}{4^{(m+1)}} \frac{T_{k-1}^{(m)} - T_{k-2}^{(m)}}{T_k^{(m)} - T_{k-1}^{(m)}}$$

and since $R_k^{(m)}$ should approach 1 as a limit, a satisfactory criterion of error is if $R_k^{(m)}$ begins to differ significantly from 1.

An improper integral has one or more of the following qualities [38]:

1. Its integrand goes to finite limiting values at finite upper and lower limits, but cannot be integrated right on one or both of these limits.
2. Its upper limit equals ∞, or its lower limit equals −∞.
3. It has an integrable singularity at (a) either limit, (b) a known place between its limits, or (c) an unknown place between its limits.

In the case of 3b, Gaussian quadrature can be used, choosing the weighting function to remove the singularities from the desired integral. A variable step size differential equation integration routine [38, Chapter 15] produces the only practicable solution to 3c.

Improper integrals of the other types whose problems involve both limits are handled by open formulas that do not require the integrand to be evaluated at its endpoints. One such formula, the extended midpoint rule, is accurate to the same order as the extended trapezoidal rule and is used when the limits of integration are located halfway between tabulated abscissas:

$$I = M_n = \Delta x(f_{3/2} + f_{5/2} + \ldots + f_{n-3/2} + f_{n-1/2})$$

Semi-open formulas are used when the problem exists at only one limit. At the closed end of the integration, the weights from the standard closed-type formulas are used and at the open end, the weights from open formulas are used. (Weights for closed and open formulas of various orders of error may be found in standard numerical methods texts.) Given a closed extended trapezoidal rule of one order higher than the preceding formula, i.e.,

$$I = T_{2_c} = \Delta x\left[\frac{5}{12}(f_1 + f_n) + \frac{13}{12}(f_2 + f_{n-1}) + \sum_{i=2}^{n-2} f_i\right]$$

and the open extended formula of the same order of accuracy

$$I = T_{2_o} = \Delta x\left[\frac{23}{12}(f_1 + f_{n-1}) + \frac{7}{12}(f_3 + f_{n-2}) + \sum_{i=4}^{n-3} f_i\right]$$

a semi-open formula can be constructed that, in this example, is closed on the right and open on the left:

$$I = T_{2s} = \Delta x\left[\frac{23}{12}f_2 + \frac{7}{12}f_3 + \left(\sum_{i=4}^{n-2} f_i\right) + \frac{13}{12}f_{n-1} + \frac{5}{12}f_n\right]$$

In order to eliminate the restriction of evenly spaced points, *Gauss Quadrature* algorithms may be constructed. In these algorithms not only the function values are weighted, but the position of the function evaluations as well as the set of weight factors are left as parameters to be determined by optimizing the overall accuracy. If the function is evaluated at points $x_0, x_1, \ldots, x_n$, the procedure has $2n + 2$ parameters to be determined (the x_i, and the w_i for each x_i) and is required to be accurate for any polynomial of degree $N = 2n + 1$ or less.

These algorithms are frequently stated in terms of integrals over $[-1,1]$, termed Gauss-Legendre quadrature, and the general formula then is

$$\int_{-1}^{1} f(x)dx \cong w_0 f_0 + w_1 f_1 + \ldots + w_n f_n$$

For example, for $n = 1$,

$$\int_{-1}^{1} f(x)dx \cong f\left(x = \frac{-1}{\sqrt{3}}\right) + f\left(x = \frac{1}{\sqrt{3}}\right)$$

For each choice of n (the number of points), the w_k and the n zeros (ξ_k) of the nth degree Legendre polynomial must be determined by requiring that the approximation be exact for polynomials of degree less than $2n + 1$. These have been determined for $n = 2$ through 95 and an abbreviated table for some n is given in Table 1-16. The interval $-1 \le \xi \le 1$ is transformed onto the interval $a \le x \le b$ by calculating for each x_k $(k = 1, \ldots, n)$

$$x_k = \frac{b+a}{2} + \frac{b-a}{2}\xi_k$$

and an approximation to the integral is then

$$I \cong \frac{b-a}{2}\sum_{k=1}^{m} w_k f(x_k)$$

Some other typical Gaussian quadrature formulas are:

(a,b)	W(x)	Gauss-
$(-1,1)$	$\frac{1}{\sqrt{1-x^2}}$	Chebyshev
$(0,\infty)$	$x^c e^{-x}$	Laguerre $(c = 0,1, \ldots)$
$(-\infty,\infty)$	e^{-x^2}	Hermite

Table 1-16
Sampling Points and Weight Factors for Gauss Quadratures

n	i	x_i	ω_i
2	0	−0.5773502692	1.0000000000
	1	−0.5773502692	1.0000000000
3	0	−0.7745966692	0.5555555556
	1	0.0	0.8888888889
	2	0.7745966692	0.5555555556
5	0	−0.9061798459	0.2369268850
	1	−0.5384693101	0.4786286705
	2	0.0	0.5688888889
	3–4	see Note below	
10	0	−0.9739065285	0.0666713443
	1	−0.8650633667	0.1494513492
	2	−0.6794095683	0.2190863625
	3	−0.4333953941	0.2692667193
	4	−0.1488743390	0.2955242247
	5–9	see Note below	
20	0	−0.9931285992	0.0176140071
	1	−0.9639719273	0.0406014298
	2	−0.9122344283	0.0626720483
	3	−0.8391169718	0.0832767416
	4	−0.7463319065	0.1019301198
	5	−0.6360536807	0.1181945320
	6	−0.5108670020	0.1316886384
	7	−0.3737060887	0.1420961093
	8	−0.2277858511	0.1491729865
	9	−0.0765265211	0.1527533871
	10–19	see Note below	

Note: Points and weight factors are symmetric with respect to zero.

Weights and zeros for the above formulas (and for other Gaussian formulas) may be found in references such as Stroud (*Gaussian Quadrature Formulas,* Prentice-Hall, 1966).

Since the dominant error term in Gauss Quadrature involves very high-order derivatives, the best method for determining the accuracy of an integration is to compare the results for several different n. However, in certain cases, a comparison may result in a set of significantly different answers, due to the presence of one or more singularities in f(x) or to a highly oscillatory function. Also if very large values of n are employed, round-off error can cause a major deterioration in accuracy (see previous discussion of Romberg integration).

Numerical Solution of Differential Equations

The two major categories of *ordinary differential equations* are

1. *Initial value problems* where conditions are specified at some starting value of the independent variable.

2. *Boundary value problems* where conditions are specified at two (or, rarely, more) values of the independent variable.

(The solution of boundary value problems depends to a great degree on the ability to solve initial value problems.) Any n^{th}-order initial value problem can be represented as a system of n coupled first-order ordinary differential equations, each with an initial condition. In general

$$\frac{dy_1}{dt} = f_1(y_1, y_2, \ldots, y_n, t)$$

$$\frac{dy_2}{dt} = f_2(y_1, y_2, \ldots, y_n, t)$$

$$\frac{dy_n}{dt} = f_n(y_1, y_2, \ldots, y_n, t)$$

and

$$y_1(0) = y_{10},\ y_2(0) = y_{20}, \ldots, y_n(0) = y_{n0}$$

The *Euler method,* while extremely inaccurate, is also extremely simple. This method is based on the definition of the derivative

$$\frac{dy}{dx} \cong \frac{y(x_i + \Delta x) - y(x_i)}{\Delta x}$$

or

$$y_{i+1} = y_i + f_i \Delta x$$

where $f_i = f(x_i, y_i)$ and $y(x = a) = y_0$ (initial condition).

Discretization error depends on the step size, i.e., if $\Delta x_i \to 0$, the algorithm would theoretically be exact. The error for Euler method at step N is $\mathcal{O}\ N(\Delta x)^2$ and total accumulated error is $\mathcal{O}\ (\Delta x)$, that is, it is a first-order method.

The *modified Euler method* needs two initial values y_0 and y_1 and is given by

$$y_n = y_{n-2} + f_{n-1}(2\Delta x) + \mathcal{O}\ (\Delta x)^2$$

If y_0 is given as the initial value, y_1 can be computed by Euler's method, or more accurately as

$$\Delta y_a = f(x_0, y_0)\Delta x$$

$$y_1 = y_0 + \Delta y_a$$

$$f_1 = f(x_1, y_1)$$

and

$$\Delta y_b = f_1 \Delta x$$

therefore

$$\Delta y = \frac{1}{2}(\Delta y_a + \Delta y_b)$$

and

$$y_1 = y_0 + \Delta y$$

Another improvement on the basic Euler method is to approximate the slope in the middle of the interval by the average of the slopes at the end points, or

$$y_{i+1} = y_i + \frac{1}{2}(f_i + f_{i+1})\Delta x$$

This form is a *closed-type* formula since it does not allow direct steps from x_i to x_{i+1}, but uses the basic Euler's method to estimate y_{i+1}, thus

$$y_{i+1} = y_i + f_i\Delta x$$

$$f_{i+1} = f(x_{i+1}, y_{i+1})$$

$$y_{i+1} = y_i + \frac{1}{2}(f_i + f_{i+1})\Delta x$$

The *Runge-Kutta method* takes the weighted average of the slope at the left end point of the interval and at some intermediate point. This method can be extended to a fourth-order procedure with error $\mathcal{O}(\Delta x)^4$ and is given by

$$y_{i+1} = y_i + \frac{1}{6}[\Delta y_0 + 2\Delta y_1 + 2\Delta y_2 + \Delta y_3]$$

where $\Delta y_0 = f(x_i, y_i)\Delta x$

$$\Delta y_1 = f\left(x_i + \frac{1}{2}\Delta x, y_i + \frac{1}{2}\Delta y_0\right)\Delta x$$

$$\Delta y_2 = f\left(x_i + \frac{1}{2}\Delta x, y_i + \frac{1}{2}\Delta y_1\right)\Delta x$$

$$\Delta y_3 = f(x_{i+1}, y_i + \Delta y_2)\Delta x$$

Runge-Kutta formulas of the sixth and eighth orders are also available, but less commonly used.

If two values of y_{i+1} are calculated, y_{i+1} by using one step between x_i and x_{i+1} with Δx, and $\hat{y}_{i+1}$ by taking two steps with $\Delta x/2$, the estimate of the truncation error is

$$E_{i+1} \sim \frac{\hat{y}_{i+1} - y_{i+1}}{2^{-k} - 1}$$

where k is the order of the expression (e.g., k = 4 for the foregoing Runge-Kutta formula). The step size can be adjusted to keep the error E below some predetermined value.

The *Adams Open Formulas* are a class of multistep formulas such that the first-order formula reproduces the Euler formula. The second-order Adams open formula is given by

$$y_{i+1} = y_i + \Delta x\left[\frac{3}{2}f_i - \frac{1}{2}f_{i-1}\right] + \mathcal{O}(\Delta x)^2$$

This formula and the higher-order formulas are not self starting since they require f_{i-1}, f_{i-2}, etc. The common practice is to employ a Runge-Kutta formula of the same order to compute the first term(s) of y_i. The general Adams open formula may be written as

$$y_{i+1} = y_i + \Delta x\sum_{k=0}^{n} \beta_{nk} f_{i-k} + \mathcal{O}(\Delta x)^{n+2}$$

and the coefficients β are given in Table 1-17 for n = 0,1, . . .,5.

Adams Closed Formulas require an iterative method to solve for y_{i+1}, since the right side of the expression requires a value for f_{i+1}. The iteration of estimating y, evaluating f, and obtaining a new estimate of y is repeated until it converges to the desired accuracy. The general formula is

$$y_{i+1} = y_i + \Delta x\sum_{k=0}^{n} \beta^*_{nk} f_{i+1-k} + \mathcal{O}(\Delta x)^{n+2}$$

and the coefficients β* are given in Table 1-18.

Table 1-17
Coefficients β_{nk} of the Open Adams Formulas

n \ k	0	1	2	3	4	5	$\mathcal{O}$*
0	1						1
1	3/2	−1/2					2
2	23/12	−16/12	5/12				3
3	55/24	−59/24	37/24	−9/24			4
4	1901/720	−2774/720	2616/720	−1274/720	251/720		5
5	4277/1440	−7923/1440	9982/1440	−7298/1440	2877/1440	−475/1440	6

$\mathcal{O}$ is the order of the method

Table 1-18
Coefficients β^*_{nk} of the Closed Adams Formulas

n \ k	0	1	2	3	4	5	$\mathcal{O}$*
0	1						1
1	1/2	1/2					2
2	5/12	8/12	−1/12				3
3	9/24	19/24	−5/24	1/24			4
4	251/720	646/720	−264/720	106/720	−19/720		5
5	475/1440	1427/1440	−798/1440	482/1440	−173/1440	27/1440	6

$\mathcal{O}$ is the order of the method

A combination of open- and closed-type formulas is referred to as the *predictor-corrector method.* First the open equation (the predictor) is used to estimate a value for y_{i+1}, this value is then inserted into the right side of the corrector equation (the closed formula) and iterated to improve the accuracy of y. The predictor-corrector sets may be the low-order modified (open) and improved (closed) Euler equations, the Adams open and closed formulas, or the *Milne method,* which gives the following system

1. Predictor

$$y_{i+1} = y_{i-3} + \frac{4}{3}\Delta x(2f_i - f_{i-1} + 2f_{i-2})$$

2. Corrector

$$y_{i+1} = \frac{1}{3}\Delta x(f_{i+1} + 4f_i + f_{i-1})$$

although the Milne method, like the Adams formulas, is not self starting.

The *Hamming method* [12] applies a predictor y^0, then a modifier $\hat{y}^0$ which provides a correction for the estimate of error in the predictor and corrector, and then iterates the corrector y^n as desired. The procedure is

1. Predictor

$$y^{(0)}_{i+1} = y_{i-3} + \frac{4}{3}\Delta x(2f_i - f_{i-1} + 2f_{i-2})$$

2. Modifier

$$\hat{y}^{(0)}_{i+1} = y^{(0)}_{i+1} + \frac{112}{121}(y_i - y^{(0)}_i)$$

3. Corrector

$$y_{i+1}^{(n+1)} = \frac{1}{8}(9y_i - y_{i-2}) + \frac{3}{8}\Delta x(f_{i+1}^{(n)} + 2f_i - f_{i-1})$$

Truncation error estimates can be made to determine if the step size should be reduced or increased. For example, for the Hamming method,

$$E_{i+1} \sim \frac{9}{121}(y_{i+1} - y_{i+1}^{(0)})$$

The Gear Algorithm [15], based on the Adams formulas, adjusts both the order and mesh size to produce the desired local truncation error. *Bulirsch and Stoer method* [16, 22] is capable of producing accurate solutions using step sizes that are much smaller than conventional methods. Packaged Fortran subroutines for both methods are available.

One approach to second-order *boundary value problems* is a matrix formulation. Given

$$\frac{d^2y}{dx^2} + Ay = B, \quad y(0) = 0, \quad y(L) = 0$$

the function can be represented at i by

$$\frac{y_{i+1} - 2y_i + y_{i-1}}{(\Delta x)^2} + Ay_i = B$$

Since there are n equations of this form and n values of y, the set can be written in matrix form as

$$\begin{vmatrix} \alpha & 1 & & & & & \\ 1 & \alpha & 1 & & & & \\ & 1 & \alpha & 1 & & & \\ & & - & - & - & & \\ & & & - & - & - & \\ & & & & 1 & \alpha & 1 \\ & & & & & 1 & \alpha & 1 \end{vmatrix} \begin{vmatrix} y_1 \\ y_2 \\ y_3 \\ - \\ - \\ y_{n-1} \\ y_n \end{vmatrix} = \begin{vmatrix} B(\Delta x)^2 \\ B(\Delta x)^2 \\ B(\Delta x)^2 \\ - \\ - \\ B(\Delta x)^2 \\ B(\Delta x)^2 \end{vmatrix}$$

where $\alpha = -2 + A(\Delta x)^2$ and the error is essentially second order. Row manipulation may be necessary if there are boundary conditions on the derivatives. Equations of higher order and sets of coupled ordinary differential equations may be solved this way if central difference representations of $\mathcal{O}\,(\Delta x)^2$ are used for the derivatives.

Shooting methods attempt to convert a boundary value problem into an initial value problem. For example, given the preceding example restated as an initial value problem for which

$$y(0) = 0 \text{ and } \frac{dy}{dx}(0) = U$$

U is unknown and must be chosen so that y(L) = 0. The equation may be solved as an initial value problem with predetermined step sizes so that x_n will equal L at the end point. Since y(L) is a function of U, it will be denoted as $y_L(U)$ and an appropriate value of U sought so that

$$y_L(U) = y(L) = 0$$

Any standard root-seeking method that does not utilize explicitly the derivative of the function may be employed.

Given two estimates of the root U_{00} and U_0, two solutions of the initial value problem are calculated, $y_L(U_{00})$ and $y_L(U_0)$, a new estimate of U is obtained where

$$U_1 = U_0 - \frac{y_L(U_0)}{[y_L(U_0) - y_L(U_{00})]/(U_0 - U_{00})}$$

and the process is continued to convergence.

There are three basic classes of second-order *partial differential equations* involving two independent variables:

1. Parabolic

$$\frac{\partial^2 u}{\partial x^2} = \phi$$

2. Elliptic

$$\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} = \phi$$

3. Hyperbolic

$$\frac{\partial^2 u}{\partial x^2} - \frac{\partial^2 u}{\partial y^2} = \phi$$

where $\phi = \phi(x,y,u,\partial u/\partial x,\partial u/\partial y)$. Each class requires a different numerical approach. (For higher-order equations and equations in three or more variables, the extensions are usually straightforward.)

Given a parabolic equation of the form

$$\alpha \frac{\partial^2 u}{\partial x^2} = \frac{\partial u}{\partial y}$$

with boundary conditions

$$u(a,y) = u_a$$

$$u(b,y) = u_b$$

$$u(x,0) = u_0$$

the equation can be written in a finite difference form, considered over the grid as shown in Figure 1-58. Using a central difference form for the derivative with respect to x and a forward difference form for the derivative with respect to y gives

$$\frac{u_{j,k+1} - 2u_{j,k} + u_{j,k-1}}{(\Delta x)^2} = \frac{1}{\alpha}\left[\frac{u_{j+1,k} - u_{j,k}}{\Delta y}\right]$$

and

$$u_{j+1,k} = \left[\frac{\alpha(\Delta y)}{(\Delta x)^2}\right]u_{j,k-1} + \left[1 - 2\frac{\alpha(\Delta y)}{(\Delta x)^2}\right]u_{j,k} + \left[\frac{\alpha(\Delta y)}{(\Delta x)^2}\right]u_{j,k+1}$$

If j is set to zero, the procedure can be used to take the first step after the initial conditions are set.

If $\partial^2 u/\partial x^2$ is represented by a central difference expression and $\partial u/\partial y$ by a backward difference expression an *implicit* solution may be obtained where

$$\frac{u_{j+1,k+1} - 2u_{j+1,k} + u_{j+1,k-1}}{(\Delta x)^2} = \frac{1}{\alpha}\left[\frac{u_{j+1,k} - u_{j,k}}{\Delta y}\right]$$

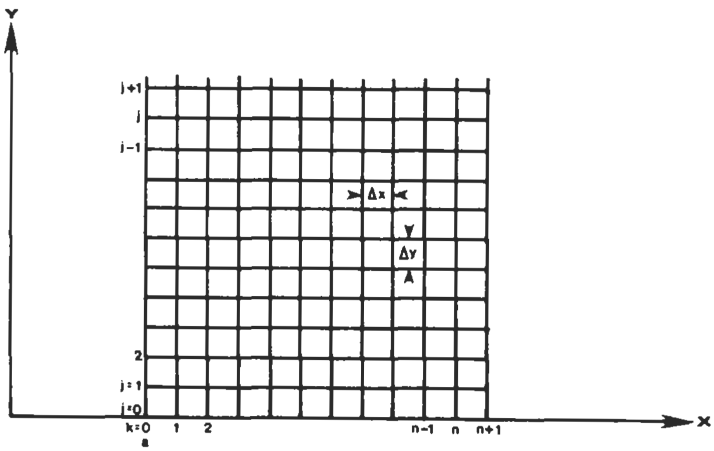


Figure 1-58. Finite difference grid.

is written for all k = 1,2, . . ., n. A set of n linear algebraic equations in n unknowns is now defined, expressed in matrix form as

$$\begin{vmatrix} \beta & 1 & & & & & \\ 1 & \beta & 1 & & & & \\ & 1 & \beta & 1 & & & \\ & & - & - & - & & \\ & & & - & - & - & \\ & & & & 1 & \beta & 1 \\ & & & & & 1 & \beta & 1 \end{vmatrix} \begin{vmatrix} u_{j+1,1} \\ u_{j+1,2} \\ u_{j+1,3} \\ - \\ - \\ u_{j+1,n-1} \\ u_{j+1,n} \end{vmatrix} = \begin{vmatrix} \Omega u_{j,1} - u_a \\ \Omega u_{j,2} \\ \Omega u_{j,3} \\ - \\ - \\ \Omega u_{j,n-1} \\ \Omega u_{j,n} - u_b \end{vmatrix}$$

where $\beta = -2 - [(\Delta x)^2]/[\alpha(\Delta y)]$
$\Omega = -[(\Delta x)^2]/[\alpha(\Delta y)]$

The *Crank-Nicholson method* is a special case of the formula

$$\alpha\left[\theta\frac{u_{j+1,k+1} - 2u_{j+1,k} + u_{j+1,k-1}}{(\Delta x)^2} + (1-\theta)\frac{u_{j,k+1} - 2u_{j,k} + u_{j,k-1}}{(\Delta x)^2}\right] = \frac{u_{j+1,k} - u_{j,k}}{\Delta y}$$

where θ is the degree of implicitness, $\theta = 1$ yields implicit representation, $\theta = 1/2$ gives Crank-Nicholson method, and $\theta = 0$, the explicit representation. $\theta \geq 1/2$ is universally stable, while $\theta < 1/2$ is only conditionally stable.

Given a partial differential equation of the elliptic form

$$\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} = 0$$

and a grid as shown in Figure 1-58, then the equation may be written in central difference form at (j,k) as

$$\frac{u_{j,k+1} - 2u_{j,k} + u_{j,k-1}}{(\Delta x)^2} + \frac{u_{j+1,k} - 2u_{j,k} + u_{j-1,k}}{(\Delta y)^2} = 0$$

and there are mn simultaneous equations in mn unknowns $u_{j,k}$.

The most effective techniques for hyperbolic partial differential equations are based on the *method of characteristics* [19] and an extensive treatment of this method may be found in the literature of compressible fluid flow and plasticity fields.

Finite element methods [20,21] have replaced finite difference methods in many fields, especially in the area of partial differential equations. With the finite element approach, the continuum is divided into a number of "finite elements" that are assumed to be joined by a discrete number of points along their boundaries. A function is chosen to represent the variation of the quantity over each element in terms of the value of the quantity at the boundary points. Therefore a set of simultaneous equations can be obtained that will produce a large, banded matrix.

The three primary advantages of the finite element approach over finite difference methods are [9]:

1. Easy handling of irregularly shaped regions.
2. Variation in size of elements over a region, allowing smaller elements where strong variations occur.
3. Larger elements can produce comparable accuracy to smaller mesh elements of a finite difference grid, which is especially useful in handling elliptic partial differential matrices.

Other methods for solving PDE's include Monte Carlo, spectral, and variational. Spectral methods in particular converge more rapidly than finite difference methods, but do not handle problems involving irregular geometries or discontinuities well.

APPLIED STATISTICS

See References 23–28 for additional information.

A *discrete random variable* is one that may take on only distinct, usually integer, values. A *continuous random variable* is one that may take on any value within a continuum of values.

Moments

The moments describe the characteristics of a sample or distribution function. The *mean,* which locates the average value on the measurement axis, is the first moment of values measured about the origin. The mean is denoted by μ for the population and $\overline{X}$ for the sample and is given for a continuous random variable by

$$\overline{X} \text{ or } \mu = \int_{-\infty}^{\infty} Xf(X)dX$$

For a discrete random variable, the mean is given by

$$\overline{X} \text{ or } \mu = \sum_{i=1}^{n} X_i f(X_i)$$

and if each observation is given equal weight, f(X) = 1/n and

$$\overline{X} \text{ or } \mu = \frac{1}{n}\sum_{i=1}^{n} X_i$$

The *variance* is the second moment about the mean and indicates the closeness of values to the mean. It is denoted by σ^2 (population) or S^2 (sample) and is given for a continuous random variable by

$$\sigma^2 \text{ or } S^2 = \int_{-\infty}^{\infty} (X - \mu)^2 f(X) dX$$

For a discrete random variable, the variance is

$$\sigma^2 \text{ or } S^2 = \sum_{i=1}^{n} (X_i - \mu)^2 f(X_i)$$

and if f(X) = 1/n

$$\sigma^2 \text{ or } S^2 = \frac{1}{n} \sum_{i=1}^{n} (X_i - \mu)^2 (\text{biased})$$

or

$$\sigma^2 \text{ or } S^2 = \frac{1}{n-1} \sum_{i=1}^{n} (X_i - \mu)^2 (\text{unbiased})$$

The *standard deviation* is the square root of the variance and is denoted by σ (population) or S (sample).

The *skew*, the third moment about the mean, is a measure of symmetry of distribution and can be denoted by γ (population) or g (sample). It is given for a continuous random variable by

$$\gamma \text{ or } g = \int_{-\infty}^{\infty} (X - \mu)^3 f(X) dX$$

and for a discrete random variable by

$$\gamma \text{ or } g = \sum_{i=1}^{n} (X_i - \mu)^3 f(X_i)$$

A completely symmetrical distribution will have a skew = 0, and in a non-symmetric distribution the sign of the skew will depend upon the location of the tail of the distribution.

The *kurtosis*, the fourth moment, is a measure of peakedness or flatness of a distribution. It has no common notation (k is used here) and is given for a continuous random variable by

$$k = \int_{-\infty}^{\infty} (X - \mu)^4 f(X) dX$$

and for a discrete random variable by

$$k = \sum_{i=1}^{n} (X_i - \mu)^4 f(X_i)$$

Moment Ratios

The *moment ratios* are dimensionless coefficients used to compare characteristics of distributions measured on different scales.

The *coefficient of variation* is a measure of relative dispersion of a set of values and is given for the population by

$$C_v = \sigma/\mu$$

and for the sample by

$$C_v = S/\overline{X}$$

The *coefficient of skewness* is a measure of relative symmetry of a distribution and is given for the population by

$$\beta_1 = \gamma^2/\sigma^6$$

and for the sample by

$$\beta_1 = g^2/S^6$$

The *coefficient of kurtosis* is a measure of relative peakedness and is given by

$$\beta_2 = k/S^4$$

Common Probability Distributions for Continuous Random Variables

The *parameters* of a distribution control its geometric characteristics [23]:

1. A *location parameter* is the abscissa of a location point and may be a measure of central tendency, such as a mean.
2. A *scale parameter* determines the location of fractiles of the distribution relative to some specified point, often the value of the location parameter.
3. *Shape parameters* control the geometric configuration of a distribution. There may be zero, one, or multiple shape parameters.

A bounded continuous random variable with *uniform distribution* has the probability function

$$f(X) = \left| \begin{array}{ll} 1/(\beta - \alpha) & \alpha \le X \le \beta \\ 0 & \text{otherwise} \end{array} \right.$$

where α = location parameter, representing lower limit of the distribution
β = scale parameter, representing upper bound of the distribution

Probabilities are determined by integration over the necessary range, i.e.,

$$P(X_1 < X < X_2) = \int_{X_1}^{X_2} 1/(\beta - \alpha)dX$$

The *normal (Gaussian) distribution* is the most frequently used probability function and is given by

$$f(X) = \frac{1}{\sqrt{2\pi}\,\sigma} \exp\left[-\frac{1}{2}\left(\frac{X-\mu}{\sigma}\right)^2\right] \quad \text{for } -\infty < X < \infty$$

where μ = location parameter
σ = scale parameter

The cumulative function for this distribution is $\int f(X)$.

The *standard normal distribution* is determined by calculating a random variable z where

$z = (X - \mu)/\sigma$ for the population

$z = (X - \overline{X})/S$ for the sample

The probability function for the standard normal distribution is then

$$f(z) = \frac{1}{\sqrt{2\pi}} e^{-.05z^2}$$

where z has a mean of zero and a standard deviation of one. Probability estimates are evaluated by integrating f(z)

$$P(z_1 \le z \le z_2) = \int_{z_1}^{z_2} \frac{1}{\sqrt{2\pi}} e^{-.05z^2} dz$$

The *t (Student's t) distribution* is an unbounded distribution where the mean is zero and the variance is $v/(v - 2)$, v being the scale parameter (also called "degrees of freedom"). As $v \rightarrow \infty$, the variance $\rightarrow 1$ (standard normal distribution). A t table such as Table 1-19 is used to find values of the t statistic where v is located along the vertical margin and the probability is given on the horizontal margin. (For a one-tailed test, given the probability for the left tail, the t value must be preceded by a negative sign.)

The *chi-square distribution* gives the probability for a continuous random variable bounded on the left tail. The probability function has a shape parameter v (degrees of freedom), a mean of v, and a variance of 2v. Values of the X^2 characteristic are obtained from a table such as Table 1-20, which is of similar construction as the t table (Table 1-19).

The *F distribution* has two shape parameters, v_1 and v_2. Table 1-21 shows F values for 1% and 5% probabilities.

Note: $F(v_1,v_2) \neq F(v_2,v_1)$

Table 1-19
Critical Values for the t Distribution

	Level of Significance for One-Tailed Test							
	.250	.100	.050	.025	.010	.005	.0025	.0005
	Level of Significance for Two-Tailed Test							
	.500	.200	.100	.050	.020	.010	.005	.001
v (degrees of freedom)								
1.	1.000	3.078	6.314	12.706	31.821	63.657	127.321	536.627
2.	.816	1.886	2.920	4.303	6.965	9.925	14.089	31.599
3.	.765	1.638	2.353	3.182	4.541	5.841	7.453	12.924
4.	.741	1.533	2.132	2.776	3.747	4.604	5.598	8.610
5.	.727	1.476	2.015	2.571	3.365	4.032	4.773	6.869
6.	.718	1.440	1.943	2.447	3.143	3.707	4.317	5.959
7.	.711	1.415	1.895	2.365	2.998	3.499	4.029	5.408
8.	.706	1.397	1.860	2.306	2.896	3.355	3.833	5.041
9.	.703	1.383	1.833	2.262	2.821	3.250	3.690	4.781
10.	.700	1.372	1.812	2.228	2.764	3.169	3.581	4.587
11.	.697	1.363	1.796	2.201	2.718	3.106	3.497	4.437
12.	.695	1.356	1.782	2.179	2.681	3.055	3.428	4.318
13.	.694	1.350	1.771	2.160	2.650	3.012	3.372	4.221
14.	.692	1.345	1.761	2.145	2.624	2.977	3.326	4.140
15.	.691	1.341	1.753	2.131	2.602	2.947	3.286	4.073
16.	.690	1.337	1.746	2.120	2.583	2.921	3.252	4.015
17.	.689	1.333	1.740	2.110	2.567	2.898	3.222	3.965
18.	.688	1.330	1.734	2.101	2.552	2.878	3.197	3.922
19.	.688	1.328	1.729	2.093	2.539	2.861	3.174	3.883
20.	.687	1.325	1.725	2.086	2.528	2.845	3.153	3.850
21.	.686	1.323	1.721	2.080	2.518	2.831	3.135	3.819
22.	.686	1.321	1.717	2.074	2.508	2.819	3.119	3.792
23.	.685	1.319	1.714	2.069	2.500	2.807	3.104	3.768
24.	.685	1.318	1.711	2.064	2.492	2.797	3.091	3.745
25.	.684	1.316	1.708	2.060	2.485	2.787	3.078	3.725
26.	.684	1.315	1.706	2.056	2.479	2.779	3.067	3.707
27.	.684	1.314	1.703	2.052	2.473	2.771	3.057	3.690
28.	.683	1.313	1.701	2.048	2.467	2.763	3.047	3.674
29.	.683	1.311	1.699	2.045	2.462	2.756	3.038	3.659
30.	.683	1.310	1.697	2.042	2.457	2.750	3.030	3.646
40.	.681	1.303	1.684	2.021	2.423	2.704	2.971	3.551
50.	.679	1.299	1.676	2.009	2.403	2.678	2.937	3.496
60.	.679	1.296	1.671	2.000	2.390	2.660	2.915	3.460
70.	.678	1.294	1.667	1.994	2.381	2.648	2.899	3.435
80.	.678	1.292	1.664	1.990	2.374	2.639	2.887	3.416
90.	.677	1.291	1.662	1.987	2.368	2.632	2.878	3.402
100.	.677	1.290	1.660	1.984	2.364	2.626	2.871	3.390
150.	.676	1.287	1.655	1.976	2.351	2.609	2.849	3.357
200.	.676	1.286	1.653	1.972	2.345	2.601	2.839	3.340
∞	.6745	1.2816	1.6448	1.9600	2.3267	2.5758	2.8070	3.2905

Probability Distributions for Discrete Random Variables

The *binomial distribution* applies to random variables where there are only two possible outcomes (A or B) for each trial and where the outcome probability is constant over all n trials. If the probability of A occurring on any one trial is denoted as p and the number of occurrences of A is denoted as x, then the *binomial coefficient* is given by

$$\binom{n}{x} = \frac{n!}{x!(n-x)!}$$

and the probability of getting x occurrences of A in n trials is

$$b(x;n,p) = \binom{n}{x} p^x (1-p)^{n-x} \quad \text{for } x = 0,1,2,\ldots,n$$

The cumulative probability of the binomial distribution is given by

$$B(x;n,p) = \sum_{i=0}^{x} b(i;n,p)$$

For the binomial distribution

$$\mu = np$$

$$\sigma = \sqrt{np(1-p)}$$

For $np \geq 5$ and $n(1 - p) \geq 5$, an approximation of binomial probabilities is given by the standard normal distribution where z is a standard normal deviate and

$$z = \frac{x - np}{\sqrt{np(1-p)}}$$

The *negative binomial distribution* defines the probability of the k^{th} occurrence of an outcome occurring on the x^{th} trial as

$$b^-(x;k,p) = \binom{x-1}{k-1} p^k (1-p)^{x-k} \quad \text{for } x = k, k+1, k+2, \ldots$$

and

$$\mu = k(1 - p)/p$$

$$\sigma^2 = k(1 - p)/p^2$$

(text continued on page 102)

Table 1-20
Critical Values for the Chi-Square Distribution

Degrees of Freedom v / α	.999	.995	.990	.975	.950	.900	.700	.500	.200	.100	.050	.020	.010	.005	.001
1.	.000	.000	.000	.001	.004	.016	.148	.455	1.642	2.706	3.842	5.405	6.637	7.905	10.809
2.	.002	.010	.020	.051	.102	.211	.713	1.386	3.219	4.604	5.995	7.822	9.221	10.589	13.691
3.	.024	.071	.115	.216	.352	.584	1.424	2.366	4.642	6.252	7.817	9.841	11.325	12.819	16.292
4.	.090	.205	.297	.484	.711	1.064	2,195	3.357	5.989	7.782	9.492	11.660	13.280	14.824	18.432
5.	.209	.411	.553	.831	1.145	1.610	3.000	4.352	7.291	9.237	11.073	13.385	15.088	16.762	20.751
6.	.377	.673	.871	1.236	1.635	2.204	3.828	5.349	8.559	10.646	12.596	15.033	16.810	18.550	22.677
7.	.597	.988	1.237	1.688	2.167	2.833	4.671	6.346	9.804	12.020	14.070	16.624	18.471	20.270	24.527
8.	.850	1.341	1.642	2.179	2.732	3.489	5.527	7.344	11.031	13.363	15.512	18.171	20.082	21.938	26.318
9.	1.135	1.728	2.086	2.699	3.324	4.168	6.393	8.343	12.243	14.686	16.925	19.683	21.654	23.563	28.061
10.	1.446	2.152	2.555	3.244	3.938	4.864	7.266	9.342	13.443	15.990	18.311	21.165	23.194	25.154	29.763
11.	1.819	2.597	3.047	3.815	4.574	5.576	8.148	10.341	14.633	17.278	19.681	22.623	24.755	26.714	31.431
12.	2.188	3.064	3.568	4.402	5.225	6.303	9.034	11.340	15.813	18.551	21.030	24.059	26.246	28.249	33.070
13.	2.577	3.560	4.102	5.006	5.890	7.041	9.926	12.340	16.986	19.814	22.367	25.477	27.717	29.878	34.683
14.	3.018	4.066	4.653	5.624	6.568	7.789	10.821	13.339	18.152	21.067	23.691	26.879	29.169	31.376	36.272
15.	3.449	4.588	5.226	6.260	7.260	8.546	11.721	14.339	19.312	22.310	25.000	28.266	30.605	32.857	37.842
16.	3.894	5.135	5.807	6.905	7.960	9.311	12.624	15.339	20.466	23.546	26.301	29.640	32.027	34.321	39.392

17.	4.350	5.687	6.400	7.560	8.670	10.083	13.530	16.338	21.616	24.771	27.593	31.002	33.435	35.771	40.926
18.	4.864	6.251	7.004	8.225	9.388	10.864	14.440	17.337	22.761	25.992	28.877	32.353	34.831	37.208	42.444
19.	5.351	6.825	7.627	8.904	10.114	11.650	15.351	18.338	23.902	27.206	30.148	33.694	36.216	38.633	43.949
20.	5.848	7.422	8.252	9.587	10.849	12.442	16.265	19.337	25.039	28.415	31.416	35.026	37.591	40.046	45.440
21.	6.398	8.018	8.886	10.278	11.590	13.238	17.182	20.337	26.173	29.619	32.678	36.350	38.957	41.449	46.919
22.	6.919	8.622	9.528	10.976	12.336	14.040	18.100	21.337	27.304	30.817	33.933	37.666	40.314	42.843	48.387
23.	7.447	9.247	10.187	11.685	13.088	14.846	19.020	22.337	28.431	32.012	35.178	38.975	41.662	44.228	49.845
24.	8.027	9.869	10.846	12.397	13.845	15.657	19.943	23.337	29.556	33.199	36.421	40.277	43.004	45.604	51.293
25.	8.576	10.498	11.510	13.115	14.607	16.471	20.866	24.337	30.678	34.384	37.660	41.573	44.338	46.973	52.732
26.	9.130	11.132	12.190	13.837	15.377	17.291	21.792	25.337	31.796	35.566	38.894	42.863	45.665	48.334	54.162
27.	9.735	11.789	12.868	14.565	16.149	18.113	22.718	26.336	32.913	36.745	40.119	44.147	46.986	49.688	55.584
28.	10.306	12.438	13.551	15.304	16.925	18.938	23.646	27.336	34.028	37.920	41.344	45.426	48.301	51.036	56.998
29.	10.882	13.092	14.240	16.042	17.705	19.766	24.576	28.336	35.140	39.092	42.565	46.699	49.610	52.378	58.405
30.	11.509	13.767	14.943	16.784	18.488	20.598	25.507	29.336	36.251	40.261	43.782	47.968	50.914	53.713	59.805
40.	17.846	20.669	22.139	24.423	26.508	29.055	34.879	39.337	47.261	51.796	55.753	60.443	63.710	66.802	73.490
50.	24.609	27.957	29.685	32.349	34.763	37.693	44.319	49.336	58.157	63.159	67.501	72.619	76.172	79.523	86.740
60.	31.678	35.503	37.465	40.474	43.187	46.463	53.814	59.336	68.966	74.390	79.078	84.586	88.396	91.982	99.679
70.	38.980	43.246	45.423	48.750	51.739	55.333	63.351	69.335	79.709	85.521	90.528	96.393	100.441	104.243	112.383
80.	46.466	51.145	53.523	57.147	60.391	64.282	72.920	79.335	90.400	96.572	101.876	108.075	112.344	116.348	124.901
90.	54.104	59.171	61.738	65.641	69.126	73.295	82.515	89.335	101.048	107.559	113.143	119.654	124.130	128.324	137.267
100.	61.869	67.303	70.049	74.216	77.929	82.362	92.133	99.335	111.662	118.493	124.340	131.147	135.820	140.193	149.505
150.	102.073	109.122	112.655	117.980	122.692	128.278	140.460	149.334	164.345	172.577	179.579	187.683	193.219	198.380	209.310
200.	143.807	152.224	156.421	162.724	168.279	174.838	189.051	199.334	216.605	226.017	233.993	243.191	249.455	255.281	267.579

Table 1-21
Critical Values for the Cumulative F Distribution

Degrees of Freedom v_2 \ v_1	1	2	3	4	5	6	7	8	9	10	15	20	30	40	50	100	500	1000	10000
1.	161.	200.	216.	225.	230.	234.	237.	239.	241.	242.	246.	248.	250.	251.	252.	253.	254.	254.	254.
	4052.	5000.	5403.	5625.	5764.	5859.	5928.	5961.	6022.	6056.	6157.	6209.	6261.	6287.	6303.	6334.	6360.	6363.	6364.
2.	18.51	19.00	19.16	19.25	19.30	19.33	19.35	19.37	19.38	19.40	19.43	19.45	19.46	19.47	19.48	19.49	19.49	19.49	19.49
	98.50	99.00	99.17	99.25	99.30	99.33	99.36	99.37	99.39	99.40	99.43	99.45	99.47	99.47	99.46	99.49	99.50	99.50	99.49
3.	10.13	9.55	9.28	9.12	9.01	8.94	8.89	8.85	8.81	8.79	8.70	8.66	8.62	8.59	6.58	8.55	8.53	8.53	8.52
	34.12	30.82	29.46	28.71	28.24	27.91	27.67	27.49	27.34	27.23	26.87	26.69	26.50	26.41	26.35	26.24	26.15	26.14	26.12
4.	7.71	6.94	6.59	6.39	6.26	6.16	6.09	6.04	6.00	5.96	5.86	5.80	5.75	5.72	5.70	5.66	5.64	5.63	5.63
	21.20	18.00	16.69	15.98	15.52	15.21	14.98	14.80	14.66	14.55	14.20	14.02	13.84	13.75	13.69	13.58	13.49	13.47	13.46
5.	6.61	5.79	5.41	5.19	5.05	4.95	4.88	4.82	4.77	4.74	4.62	4.56	4.50	4.46	4.44	4.41	4.37	4.37	4.37
	16.26	13.27	12.06	11.39	10.97	10.67	10.46	10.29	10.16	10.05	9.72	9.55	9.38	9.29	9.24	9.13	9.04	9.03	9.02
6.	5.99	5.14	4.76	4.53	4.39	4.28	4.21	4.15	4.10	4.06	3.94	3.87	3.81	3.77	3.75	3.71	3.68	3.67	3.67
	13.75	10.92	9.78	9.15	8.75	8.47	8.26	8.10	7.98	7.87	7.56	7.40	7.23	7.14	7.09	6.99	6.90	6.89	6.88
7.	5.59	4.74	4.35	4.12	3.97	3.87	3.79	3.73	3.68	3.64	3.51	3.44	3.38	3.34	3.32	3.27	3.24	3.23	3.23
	12.25	9.55	8.45	7.85	7.46	7.19	6.99	6.84	6.72	6.62	6.31	6.16	5.99	5.91	5.86	5.75	5.67	5.66	5.65
8.	5.32	4.46	4.07	3.84	3.69	3.58	3.50	3.44	3.39	3.35	3.22	3.15	3.08	3.04	3.02	2.97	2.94	2.93	2.93
	11.26	8.65	7.59	7.01	6.63	6.37	6.18	6.03	5.91	5.81	5.52	5.36	5.20	5.12	5.07	4.96	4.88	4.87	4.86

9.	5.12	4.26	3.86	3.63	3.48	3.37	3.29	3.23	3.18	3.14	3.01	2.94	2.86	2.83	2.80	2.76	2.72	2.71	2.71
	10.56	8.02	6.99	6.42	6.06	5.80	5.61	5.47	5.35	5.26	4.96	4.81	4.65	4.57	4.52	4.41	4.33	4.32	4.31
10.	4.96	4.10	3.71	3.48	3.33	3.22	3.14	3.07	3.02	2.98	2.85	2.77	2.70	2.66	2.64	2.59	2.55	2.54	2.54
	10.04	7.56	6.55	5.99	5.64	5.39	5.20	5.06	4.94	4.85	4.56	4.41	4.25	4.17	4.12	4.01	3.93	3.92	3.91
15.	4.54	3.68	3.29	3.06	2.90	2.79	2.71	2.64	2.59	2.54	2.40	2.33	2.25	2.20	2.18	2.12	2.08	2.07	2.07
	8.68	6.36	5.42	4.89	4.56	4.32	4.14	4.00	3.89	3.80	3.52	3.37	3.21	3.13	3.08	2.98	2.89	2.88	2.87
20.	4.35	3.49	3.10	2.87	2.71	2.60	2.51	2.45	2.39	2.35	2.20	2.12	2.04	1.99	1.97	1.91	1.86	1.85	1.85
	8.10	5.85	4.94	4.43	4.10	3.87	3.70	3.56	3.46	3.37	3.09	2.94	2.78	2.69	2.64	2.54	2.44	2.43	2.42
30.	4.17	3.32	2.92	2.69	2.53	2.42	2.33	2.27	2.21	2.16	2.01	1.93	1.84	1.79	1.76	1.70	1.64	1.63	1.62
	7.56	5.39	4.51	4.02	3.70	3.47	3.30	3.17	3.07	2.98	2.70	2.55	2.39	2.30	2.25	2.13	2.03	2.02	2.01
40.	4.08	3.23	2.84	2.61	2.45	2.34	2.25	2.18	2.12	2.08	1.92	1.84	1.74	1.69	1.66	1.59	1.53	1.52	1.51
	7.31	5.18	4.31	3.83	3.51	3.29	3.12	2.99	2.89	2.80	2.52	2.37	2.20	2.11	2.06	1.94	1.83	1.82	1.82
50.	4.03	3.18	2.79	2.56	2.40	2.29	2.20	2.13	2.07	2.03	1.87	1.78	1.69	1.63	1.60	1.52	1.46	1.45	1.44
	7.17	5.06	4.20	3.72	3.41	3.19	3.02	2.89	2.78	2.70	2.42	2.27	2.10	2.01	1.95	1.82	1.71	1.70	1.68
100.	3.94	3.09	2.70	2.46	2.31	2.19	2.10	2.03	1.97	1.93	1.77	1.68	1.57	1.52	1.48	1.39	1.31	1.30	1.28
	6.90	4.82	3.98	3.51	3.21	2.99	2.82	2.69	2.59	2.50	2.22	2.07	1.89	1.80	1.74	1.60	1.47	1.45	1.43
500.	3.86	3.01	2.62	2.39	2.23	2.12	2.03	1.96	1.90	1.85	1.69	1.59	1.48	1.42	1.38	1.28	1.16	1.14	1.12
	6.69	4.65	3.82	3.36	3.05	2.84	2.68	2.55	2.44	2.36	2.07	1.92	1.74	1.63	1.57	1.41	1.23	1.20	1.17
1000.	3.85	3.00	2.61	2.38	2.22	2.11	2.02	1.95	1.89	1.84	1.68	1.58	1.47	1.41	1.36	1.26	1.13	1.11	1.08
	6.66	4.63	3.80	3.34	3.04	2.82	2.66	2.53	2.43	2.34	2.06	1.90	1.72	1.61	1.54	1.38	1.19	1.16	1.12
10000.	3.84	3.00	2.61	2.37	2.21	2.10	2.01	1.94	1.88	1.83	1.67	1.57	1.46	1.40	1.35	1.25	1.11	1.08	1.03
	6.64	4.61	3.78	3.32	3.02	2.80	2.64	2.51	2.41	2.32	2.04	1.88	1.70	1.59	1.53	1.36	1.16	1.11	1.05

Note: The upper and lower values in the table are for F_{05} and F_{01}.

(text continued from page 97)

If the probabilities do not remain constant over the trials and if there are k (rather than two) possible outcomes of each trial, the *hypergeometric distribution* applies. For a sample of size N of a population of size T, where

$t_1 + t_2 + \ldots + t_k = T$, and

$n_1 + n_2 + \ldots + n_k = N$

the probability is

$$h(n_i; N, t_i, T) = \frac{\binom{t_1}{n_1}\binom{t_2}{n_2} \cdots \binom{t_k}{n_k}}{\binom{T}{N}}$$

The *Poisson distribution* can be used to determine probabilities for discrete random variables where the random variable is the number of times that an event occurs in a single trial (unit of time, space, etc.). The probability function for a Poisson random variable is

$$P(x;\mu) = \frac{e^{-\mu}\mu^x}{x!} \quad \text{for } x = 0,1,2, \ldots$$

where μ = mean of the probability function (and also the variance)

The cumulative probability function is

$$F(X;\mu) = \sum_{i=0}^{x} P(i;\mu)$$

Univariate Analysis

For Multivariate Analysis, see McCuen, Reference 23, or other statistical texts.

The first step in data analysis is the selection of the best fitting probability function, often beginning with a *graphical analysis* of the frequency histogram. Moment ratios and *moment-ratio diagrams* (with β_1 as abscissa and β_2 as ordinate) are useful since probability functions of known distributions have characteristic values of β_1 and β_2.

Frequency analysis is an alternative to moment-ratio analysis in selecting a representative function. Probability paper (see Figure 1-59 for an example) is available for each distribution, and the function is presented as a cumulative probability function. If the data sample has the same distribution function as the function used to scale the paper, the data will plot as a straight line.

The procedure is to fit the population frequency curve as a straight line using the sample moments and parameters of the proposed probability function. The data are then plotted by ordering the data from the largest event to the smallest and using the rank (i) of the event to obtain a probability plotting position. Two of the more common formulas are Weibull

$pp_w = i/(n + 1)$

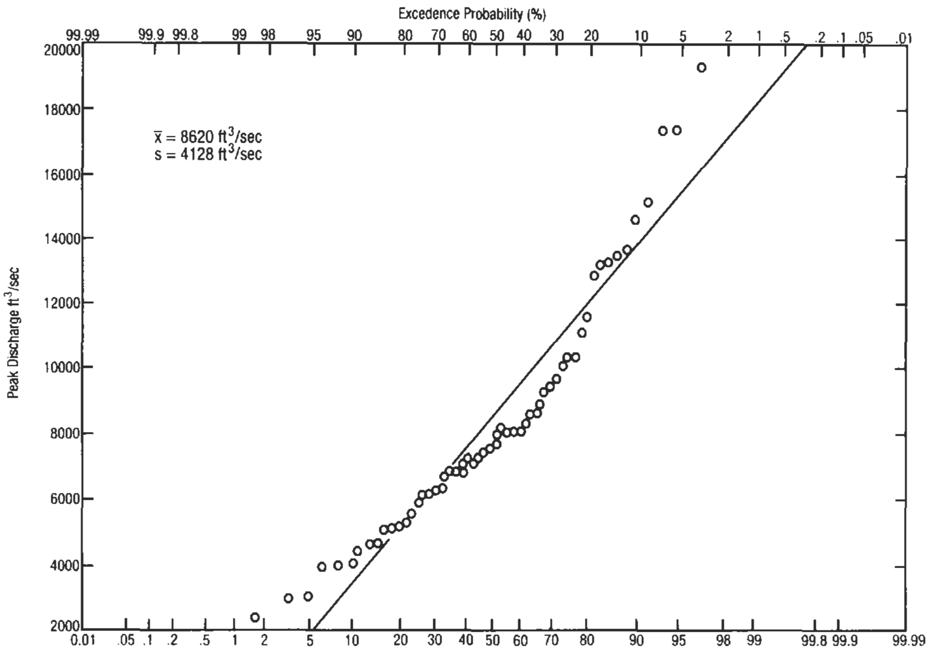


Figure 1-59. Probability paper for frequency analysis.

and Hazen

$$pp_h = (2i - 1)/(2n)$$

where n is the sample size. If the data do not show a reasonable fit to the population curve, a different function should be investigated.

Estimation of model parameters is frequently accomplished by the *method of moments.* For example, for the uniform distribution, the mean is

$$\mu = \int_\alpha^\beta X\left(\frac{1}{\beta - \alpha}\right)dX = \frac{\beta + \alpha}{2} = \overline{X}$$

and the variance is

$$\sigma^2 = (\beta - \alpha)^2/12 = S^2$$

Solving for α and β gives

$$\alpha = \overline{X} - S\sqrt{3}$$

$$\beta = \overline{X} + S\sqrt{3}$$

Confidence Intervals

Confidence intervals provide a method of calculating the range of values that contains the true value of an estimate. A general equation for a two-sided confidence interval is

$K_{est} = \pm FD$

where K_{est} = estimated value of the K statistic
F = distribution factor
D = measure of dispersion

(For one-sided confidence intervals, the ± is replaced by + or by −.)
If σ is known, the *confidence interval on the mean* is

1. Two sided

$$\overline{X} - z_{\alpha/2} \frac{\sigma}{\sqrt{n}} \leq \mu \leq \overline{X} + z_{\alpha/2} \frac{\sigma}{\sqrt{n}}$$

2. One sided, lower limit

$$\overline{X} - z_{\alpha} \frac{\sigma}{\sqrt{n}} \leq \mu \leq \infty$$

3. One sided, upper limit

$$-\infty \leq \mu \leq \overline{X} + z_{\alpha} \frac{\sigma}{\sqrt{n}}$$

where $\overline{X}$ = sample mean = K_{est}
n = sample size
$z_\alpha, z_{\alpha/2}$ = values of random variables, with standard normal distribution, cutting off $(1 - \gamma)$ and $(1 - \gamma/2)$ respectively in the tail of the distribution, and $\alpha = 1 - \gamma$ (the level of significance) = F
$\sigma/\sqrt{n}$ = measure of dispersion

If σ is unknown, the equations are

1. Two sided

$$\overline{X} - t_{\alpha/2} \frac{s}{\sqrt{n}} \leq \mu \leq \overline{X} + t_{\alpha/2} \frac{s}{\sqrt{n}}$$

2. One sided, lower limit

$$\overline{X} - t_{\alpha} \frac{s}{\sqrt{n}} \leq \mu \leq \infty$$

3. One sided, upper limit

$$-\infty \leq \mu \leq \overline{X} + t_{\alpha} \frac{s}{\sqrt{n}}$$

where s = standard deviation of sample
$t_\alpha, t_{\alpha/2}$ = values of variables having a t distribution with $v = n - 1$, and α % of distribution cut off in one tail and $\alpha/2$ % in both tails = F
$s/\sqrt{n}$ = measure of dispersion

The *confidence interval on the variance* is computed by

1. $$\frac{(n-1)S^2}{X^2_{\alpha/2}} \leq \sigma^2 \leq \frac{(n-1)S^2}{X^2_{(1-\alpha/2)}}$$

2. $$\frac{(n-1)S^2}{X^2_{\alpha}} \leq \sigma^2 \leq \infty$$

3. $$-\infty \leq \sigma^2 \leq \frac{(n-1)S^2}{X^2_{(1-\alpha)}}$$

where $X^2_{\alpha/2}, X^2_\alpha$ = values of a random variable with a chi-square distribution cutting off $\alpha/2$ % and α %, respectively, of the right tail
$X^2_{(1-\alpha/2)}, X^2_{(1-\alpha)}$ = values of a random variable with a chi-square distribution cutting off $(1 - \alpha/2)$ % and $(1 - \alpha)$ %, respectively, of the left tail
$S^2 = K_{est}$
X^2 values = distribution factors

Correlation

Correlation analysis quantifies the degree to which the value of one variable can be used to predict the value of another. The most frequently used method is the *Pearson product-moment correlation coefficient.*

The *coefficient of determination* is the fraction of the variation that is explained by a linear relationship between two variables and is given by

$$R^2 = \frac{\sum_{i=1}^{n}(\hat{Y}_i - \overline{Y})^2}{\sum_{i=1}^{n}(Y_i - \overline{Y})^2}$$

where Y = observation on the random variable
$\hat{Y}$ = value of Y estimated from the best linear relationship with the second variable X
$\overline{Y}$ = mean of the observations on Y

and R is the *correlation coefficient.* A perfect association is indicated by $R = 1$ for a direct relationship and $R = -1$ for an inverse relationship. $R = 0$ indicates no linear association between X and Y.

A second definition of R is

$$R = \frac{1}{nS_xS_y}\sum_{i=1}^{n}(X_i - \overline{X})(Y_i - \overline{Y})$$

where n = number of observations on Y
S_x,S_y = biased (n degrees of freedom) estimates of the standard deviations of X and Y

Note: For small n, even high correlation may not indicate a significant relationship between the variables.

Regression

The relationship between a criterion variable and two or more predictor variables is given by a *linear multivariate model:*

$$\hat{Y} = b_o + b_1x_1 + b_2x_2 + \ldots + b_px_p$$

where p = number of predictor variables
x_i = i^{th} predictor variable
b_i = i^{th} slope coefficient
b_o = intercept coefficient
i = 1,2, . . .,p

The coefficients b_i are the *partial regression coefficients.*

The *principle of least squares* is used to correlate Y with the X_i values. The error e (or residual) is defined as

$$e_i = \hat{Y}_i - Y_i$$

where $\hat{Y}_i$ = i^{th} predicted value of the criterion variable
Y_i = i^{th} measured value of the criterion variable
e_i = i^{th} error

The purpose of the principle of least squares is to minimize the sum of the squares of the errors so that

$$E = \min\sum_{i=1}^{n}(\hat{Y}_i - Y)^2$$

where n = number of observations of the criterion variable (i.e., sample size)

$$E = \sum_{i=1}^{n}(b_o + b_1X_i - Y_i)^2$$

By differentiating with respect to b_o and b_1 and setting the equations equal to zero, two equations in two unknowns are obtained (the summations are for i = 1, . . .,n)

$$b_1 = \frac{\sum X_i Y_i - \left(\sum X_i \sum Y_i\right)/n}{\sum X_i^2 - \left(\sum X_i\right)^2/n}$$

and

$$b_o = \frac{\sum Y_i}{n} - \frac{b_1 \sum X_i}{n}$$

and by solution of the set, b_0 and b_1 can be obtained.

The standard deviation S_y gives the accuracy of prediction. If Y is related to one or more predictor variables, the error of prediction is reduced to the *standard error of estimate* S_e (the standard deviation of the errors), where

$$S_e = \left[\frac{1}{v}\sum_{i=1}^{n}(\hat{Y}_i - Y_i)^2\right]^{0.5}$$

where v = degrees of freedom, or sample size − number of unknowns. For the general linear model with an intercept, there are (p + 1) unknowns and v = n − (p + 1)

If $S_e = 0$, then R = 1, and if $S_e = S_y$, then R = 0.

The *standardized partial regression coefficient* t is a measure of the relative importance of the corresponding predictor and is given by

$$t = (b_1 S_x)/S_y$$

where $-1 < t < 1$ for rational models

The two-sided confidence intervals for the coefficients b_0 and b_1, when β_o and β_1 are random variables having t distributions with (n − 2) degrees of freedom and error variances of

$$S_e^2(b_0) = \frac{S_e^2 \sum X_i^2}{n\sum (X_i - \overline{X})^2}$$

and

$$S_e^2(b_1) = \frac{S_e^2}{\sum (X_i - \overline{X})^2}$$

are (if α is the level of significance)

$$b_0 \pm t_{(\alpha/2,n-2)} S_e(b_0)$$

$$b_1 \pm t_{(\alpha/2,n-2)} S_e(b_1)$$

The confidence interval for a line of m points may be plotted by computing the confidence limits $\hat{Y}_{ci}$ at each point, $(X_{xi}, \hat{Y}_{ai}, i = 1,2, \ldots, m)$ of the regression line when

$$\hat{Y}_{ci} = \hat{Y}_{ai} \pm S_e \sqrt{2F}\left[\frac{1}{n} + \frac{(X_{ai} - \overline{X})^2}{\sum (X - \overline{X})^2}\right]^{0.5}$$

where $\hat{Y}_{ai} = \overline{Y} + b_1(X_{ai} - \overline{X})$

F = F statistic obtained for $(2, n - 2)$ degrees of freedom and a level of significance $\alpha = 1 - \gamma$, γ being the level of confidence.

The confidence interval for a single point, say X_o, can be computed using the interval

$$\hat{Y} \pm t_{\alpha/2} S_e \left[\frac{1}{n} + \frac{(X_o - \overline{X})^2}{\sum (X - \overline{X})^2}\right]^{0.5}$$

where $t_{\alpha/2}$ = value of random variable having a t distribution with $(n - 2)$ degrees of freedom and a level of confidence $\alpha = 1 - \gamma$

$\hat{Y} = \overline{Y} + b_1(X_o - \overline{X})$

The confidence interval for a future value X_f is given by

$$\hat{Y} \pm t_{\alpha/2} S_e \left[1 + \frac{1}{n} + \frac{(X_f - \overline{X})^2}{\sum (X - \overline{X})^2}\right]^{0.5}$$

where $\hat{Y} = \overline{Y} + b_1(X_f - \overline{X})$

COMPUTER APPLICATIONS

See References 29–40 for additional information.

Some areas of digital computer use that are highly applicable to the engineering field include

- Numerical computations for design and modeling
- Information storage and retrieval
- Data sorting and reduction
- Computer-aided graphics for illustration, as well as for design
- Word processing
- Communication networks and database access

The rapid increase in computer applications is partly attributable to both the decreasing costs of hardware and software and to the increasing costs of human labor. This shift has given rise to a productivity factor assigned to various tasks performed by computers versus people. One figure recently quoted was a minimum factor of 4 to 5 for CAD (computer aided design), that is, one draftsman with a CAD system can replace 4 to 5 manual draftsmen.

Problem Solving

The initial outline of a solution to a problem is the *algorithm,* i.e., a list of English-language instructions with the following properties:

1. The execution of the proposed algorithm must be completed after a finite number of operations, the number depending on the complexity of the problem and the degree of detail of the algorithm.
2. The representation of the solution must have a *unique interpretation,* computers having less tolerance for ambiguity than humans, so when executing the steps with the same input data, the same outputs are obtained.
3. The algorithm must present the computer with sufficient information and instructions to carry out the solution.
4. The scope of the algorithm may be predefined by the range of the inputs.

Programming Languages

A wide variety of programming languages are available ranging from *machine code,* composed of sequences of 0's and 1's representing either data or instructions, which is completely processor dependent and not transferable, through *assembly languages* consisting of mnemonics for instructions and usually hexadecimal representation of storage location addresses and of data to *high-level programming languages* such as BASIC and FORTRAN. High-level languages may be divided into procedure-oriented languages and problem-oriented languages.

The term problem-oriented languages should be read as "special-purpose languages," or "applications-oriented languages," since in a more general context all high-level languages may be used to solve problems. Some of these languages have been designed for special applications such as electronic circuit analysis, while others are more general purpose, such as those written for simulation or statistical packages.

Procedure-oriented languages, so called because they allow the programmer to concentrate on the process rather than on the machine architecture, include familiar languages, such as FORTRAN, as well as many more recently developed ones. Three of the high-level languages of common interest to engineers are:

1. BASIC: Beginners All Purpose Symbolic Instruction Code, most often an interpreted (i.e., each line is translated into (object) machine code immediately before execution and no object code is retained for future executions), highly interactive language, with a minimum of imposed structure and the ability to allow the user to access the memory and input/output hardware of the machine. Its disadvantages are slow execution speed (although compiled BASICs are frequently faster) and the great degree of programming freedom, allowing users to create terrible messes if they so desire, with little chance of error detection. Because of the great degree of variation in BASIC implementations on different systems, it is necessary to refer to a specific system manual for its BASIC syntax, as well as for the DOS (Disk Operating System) instructions for that system, or to the BASIC texts in the reference section for some general forms of the language.
2. FORTRAN: Formula Translating Language, the original, still most commonly used language for engineering computations. It is a compiled language (the entire program is translated to object code and saved before execution) that links to many libraries of subroutines and has a number of special purpose extensions. FORTRAN has been much improved by the

addition of control structures which eliminate the necessity for many unstructured leaps of logic through the program. (See the FORTRAN language section for details on structure and syntax of FORTRAN 77.)

3. Pascal: A general purpose language, designed by Niklaus Wirth specifically to teach structured, modular programming and to provide reliable and efficient programs on available computers. It provides a high degree of error checking during compilation and an extensive set of data types. It also allows the use of recursion, i.e., a procedure is allowed to call itself, which may produce elegant solutions to certain types of problems. (See the Pascal language section for details on structure and syntax of Wirth "standard" Pascal.)

Other familiar languages may be C, ADA, LISP, ALGOL, PL/I, Prolog, and APL which are now available on many microcomputer systems.

Packaged programs are available in many areas of general interest to engineers, including mathematics, statistics, and structural design. A number of vendors also offer specialized petroleum engineering packages relating to such areas as EOR, drilling fluids, corrosion control, cementing, and well production histories. Some private vendors also maintain databases on specific subjects such as well production histories.

Common Data Types

Although the number of data types available varies with the programming language and a particular vendor's restrictions and extensions of the standard, the following types are particularly useful in scientific programming. (Some languages permit user definition of nonstandard data types, usually for the purpose of limiting the range of values accepted by a variable of that type.)

1. *Integer*—A signed number with no fractional part.
2. *Real*—A signed number with an integer part and a fractional part.
3. *Double precision*—Value stored as two words, rather than one, representing a real number, but allowing for approximately double the number of significant digits.
4. *Complex*—Value stored as two words, one representing the real part of the number and the other representing the imaginary part.
5. *Character*—Alphanumerical item (2,m,!,etc.) represented in memory as a binary code (see Table 1-22 for ASCII and Table 1-23 for EBCDIC).
6. *Logical*—Data type with only two possible values: True (represented as 1) and False (represented as 0), also referred to as Boolean.
7. *Pointer*—Identifies addresses of other data items; used to create linked data structures.

Common Data Structures

The following data structures are available, or can be constructed, in most high-level languages.

1. *Variable*—Named data item of a specific type; may be assigned one or more values during the course of a program run (in some languages, a constant may be defined with a specified initial value which may not be changed).
2. *Array*—A collection of data items of the same type, referred to collectively by a single name. The individual items, the *array elements*, are ordered by

Table 1-22
ASCII (American Standard Code for Information Exchange)

Left Digit(s)	Right Digit(s)	0	1	2	3	4	5	6	7	8	9
3					!	"	#	$	%	&	'
4		(	)	*	+	,	–	.	/	0	1
5		2	3	4	5	6	7	8	9	:	;
6		<	=	>	?	@	A	B	C	D	E
7		F	G	H	I	J	K	L	M	N	O
8		P	Q	R	S	T	U	V	W	X	Y
9		Z	[	\	]	∧	_	‘	a	b	c
10		d	e	f	g	h	i	j	k	l	m
11		n	o	p	q	r	s	t	u	v	w
12		x	y	z	{	|	}	~			

Note: Decimal codes 00 to 31 and 127 and higher represent nonprintable control characters and special character codes.

Table 1-23
EBCDIC (Extended Binary Coded Decimal Interchange Code)

Left Digit(s)	Right Digit(s)	0	1	2	3	4	5	6	7	8	9
6											
7						¢	.	<	(	+	|
8		&									
9		!	$	*	)	;	¬	–	/		
10								∧	,	%	_
11		>	?								
12				:	#	@	'	"			a
13		b	c	d	e	f	g	h	i		
14							j	k	l	m	n
15		o	p	q	r						
16				s	t	u	v	w	x	y	z
17									\	{	}
18		[	]								
19					A	B	C	D	E	F	G
20		H	I								J
21		K	L	M	N	O	P	Q	R		
22								S	T	U	V
23		W	X	Y	Z						
24		0	1	2	3	4	5	6	7	8	9

Note: Decimal codes 00 to 63 and 250 to 255 represent nonprintable control characters.

their subscripts, the number of subscripts being determined by the dimensionality of the array. An element is referred to by the array name followed by its parenthesized subscripts, e.g., TEMP(I,J) might refer to the temperature at the I^{th} time increment and at the J^{th} pipe node.
3. *Record*—A collection of data items (fields) of various types, which may also be records themselves. If EMP1 is a record in the master file of employees, EMP1.NAME may be a character field, EMP1.ZIP an integer field, and EMP1.SAL a real field.
4. *File*—Collection of records that normally consist of matching types of fields. Records in a file may be accessed sequentially (the entire file must be read until the needed record is reached) or randomly (the record contains a key field, which determines its physical location in storage).
5. *Linked lists*—Data items linked by pointers. In the general form, each item, except the first, has one predecessor, and each item, except the last, has one successor, with pointers linking items to their successors. *Doubly linked lists* have pointers to both the predecessor and the successor of an item and a *circular list* has a pointer from the final item to the initial item (producing a predecessor to the initial item and a successor to the final item). Restricted lists also exist, such as *stacks,* where items may only be added (pushed) or deleted (popped) at one end (the top), and *queues,* where items must be inserted at one end and deleted from the other. *Trees* are linked lists in which each item (node) except the root node has one predecessor, but all nodes may have any finite number, or zero, successors; *graphs* contain both nodes and edges, which connect the nodes and define their relationships.

Program Statements

The statements of which the program consists may be either executable or nonexecutable. *Nonexecutable statements* consist of comments, which explain the data and logic of the program, and declarations, which are orders to the translator or to other system programs and which usually serve to allocate memory space for data.

Executable statements, which are translated into machine code, are instructions by which operations are performed on data or by which the sequence of execution is changed. Statements producing operations on data are:

1. *Assignment*—Assign a value, either a constant or a computed value, to a variable, an array element, a node, or a field.
2. *Input*—Transfer data from external devices, such as a keyboard or disk file, to the program.
3. *Output*—Transfer data from the program to an external device, such as a printer, screen, or a disk file.

Executable statements affecting the order in which the program instructions are executed include conditional (branching) statements, iterative (looping) statements, and statements which call subprogram units.

1. *Conditional statements*—Change the sequence in which instructions are executed depending upon the logical relationship(s) between variables and/or between variable(s) and set value(s).
2. *Iterative statements*—Force the repetition of instructions depending on preset conditions.
3. *Calling statements*—Transfer control to a subprogram unit.

Subprograms

The division of a program into a main program unit and one or more subprogram units allows logical organization of the program into sections of related operations and facilitates the coding, debugging, and replacement of units of the program. Data are passed from one unit (or module) to another through parameters (arguments) and/or through shared memory locations. There are two types of subprograms usually available:

1. *Function subprograms*—Return a single value as the value of the function name; these may be either extrinsic (user-defined) or intrinsic (provided as part of the system library).
2. *Subroutine or procedure subprograms*—Return values through parameters or global variables (see scope below).

The *scope* of a data item in a program determines which program units may access (and change) the value of that data item. *Global* data may be accessed by all program units, whereas *local* data are visible only to the unit(s) in which they are defined. The method by which scope is determined depends on the type of language being used. In FORTRAN, a COMMON block defined by a COMMON statement in the main program allows data in that block to be accessed by any module in which the block is defined. Parameters allow two units to share data values; some types of parameters allow the passing of only the value and not the location of the item, so that the subunit can read, but not change, the value. In block structured languages, such as Pascal, the structure determines the scope of the variables; the scope of a variable is the block in which it was defined and all blocks contained therein. A variable in this case is global to a sub-unit if it was declared not in that sub-unit but in a higher-level unit which contains the sub-unit, e.g., all variables declared in the main program are global to all program units.

Recursion is available in languages having dynamic memory allocation such as ALGOL and Pascal. (Many versions, especially older ones, of FORTRAN and BASIC have static memory allocation and do not permit recursion.) Direct recursion occurs when a program unit calls itself; indirect recursion occurs when a chain of subprogram calls results in the original calling unit being called again without returning to a higher-level unit, e.g., MAIN → SUBA → SUBB → SUBC → SUBA, where

```
 MAIN
-------
| | | |
A B C D
```

General Programming Principles

Two characteristics of well-designed programs are

1. *Generality*—To as great an extent as possible for a particular problem, a program should be able to operate on a wide variety of data sets with a minimum of program revision, and the necessary changes should be as simple to make as possible.
2. *Portability*—A program should adhere as closely as possible to a standard version of a language and avoid highly machine-dependent constants and constructions. Unidentified machine-dependent information should be localized and identified for simplification of transport.

With little extra effort, code can be written so as to minimize the difficulty a reader will encounter in comprehending the program logic. Several considerations in improving readability of a program are

1. Names should be as descriptive as possible, e.g., DEPTH rather than I2.
2. Comments should be liberally used to describe the data and logic of the program; they should be brief, when placed in the body of a program unit, but may be longer and more descriptive at the beginning of a program unit.
3. Indentation, when possible, clarifies conditional and iterative constructions, and spacing improves the general readability of a program.
4. The use of subprogram units allows separation of the various operations of a program into modules, thereby clearly delineating the program logic. Specific types of calculations, input, and output may be done in distinct modules; in many cases, the main unit will consist primarily of calling statements to a few modules.

The proper handling of certain common errors can improve the run-time behavior of programs. In most cases, awareness of inherent problems in machine handling of data and attention to program details can avoid program crashes due to error. The following should be considered when the program code is being designed:

1. Input validation statements should be used to automatically check input data, to produce a clear message when an error is found, and to allow reentry of erroneous data. Data input should be echoed for user verification and an opportunity allowed for alteration of specific data items.
2. A method of exiting the program in case of a run-time error, which produces a message to the user as to the type and location of the error, should be provided. The possibility of certain errors occurring can be anticipated, and the use of flags and conditional constructions may provide a path to exit the program gracefully.
3. It is necessary to avoid predicting the exact value of a real variable, since, after several operations, it may have been rounded off one or more times.
4. Side effects in subprogram units are unintentional changes in data values defined in other units. These frequently occur when the scope of a variable is mistakenly considered, because of insufficient cross-checking, to be local, when it is, in fact, global.

FORTRAN Language

FORTRAN names (unit, variable, array, etc.) consist of an initial letter (see defaults for real and integer types in the following) followed by letters or digits, the maximum length of which is 6 characters.

Data Types. See the section "Statements" for forms of declaration of type.

Integer—Variable names starting with I-N, unless otherwise declared.
Real—Variable names starting with A-H and O-Z, unless otherwise declared.
Double precision—Must be declared.
Complex—Must be declared.
Logical—Must be declared.
Character—Must be declared with length of string; default length is 1.

Data Structures.

Variable—May be assigned value by a numerical or character constant, by input, or by an expression.
Array—May have up to seven dimensions; number and size (upper and lower boundaries) of dimensions are declared in DIMENSION or TYPE statements.
File—External (physical) only; may be sequential or random access.

Statements. Most statements, except where noted otherwise, begin in the seventh column of a page considered to be 80 columns wide; *continuation lines* are indicated by a "+" (symbol may vary with version of language) in the sixth column. The first five columns are reserved for *labels* (line numbers), which are only required if the line is referenced by another statement, and for *comment lines*, which are determined by a character in the first column. Columns 7 to 72 are reserved for statements; 73–80 are not read. (See Table 1-24 for required order of statements in FORTRAN.)

Nonexecutable Statements.

Program unit heading—Program name, function name, or subroutine name.
Type declaration—Specifies data type to be represented by a variable name (overrides defaults), e.g.,

```
REAL MSR
INTEGER COUNT, PNUM, AP
LOGICAL TEST1
CHARACTER*10 LNAME
```

Implicit declaration—Allows type specification for all names beginning with the given first letter(s)

```
IMPLICIT DOUBLE PRECISION (A-Z)
IMPLICIT COMPLEX (C)
```

Table 1-24
Order of Statements in FORTRAN

<table>
<tr><td rowspan="6">COMMENT Lines</td><td colspan="3">PROGRAM, FUNCTION, SUBROUTINE or BLOCK DATA Statements</td></tr>
<tr><td rowspan="5">FORMAT and ENTRY Statements</td><td colspan="2">IMPLICIT Statements</td></tr>
<tr><td colspan="2">PARAMETER Statements</td></tr>
<tr><td colspan="2">DIMENSION, COMMON TYPE, and EQUIVALENCE Statements</td></tr>
<tr><td rowspan="2">DATA Statements</td><td>Statement Function Definitions</td></tr>
<tr><td>Executable Statements</td></tr>
<tr><td colspan="4">END Statement</td></tr>
</table>

Dimension statement–Specifies number and size of dimensions for each array (may be included in Type statement); lower bound default is 1

```
DIMENSION A1(5:10),A2(15)
DIMENSION A3(0:5,0:10,10:100),A4(10,10,10,10)
```

or

```
INTEGER A1(5:10),A2(15)
```

Common statement–Defines a common block of global variables (one common block may have no specified name)

```
COMMON/blockname/varnames/blockname/varnames
COMMON X,Y,Z/C2BLK/A,B,C
```

Data may be entered into variables declared in a labeled COMMON block by an assignment or an input statement or through a BLOCK DATA subprogram as defined in subprogram statements.

Equivalence statement–Assigns two or more variable names to the same memory location

```
EQUIVALENCE(A,B,C)
```

Parameter statement–Declares the name of a constant whose value cannot be changed in the program

```
PARAMETER name
```

End statement–Compiler signal for end of unit

```
END
```

Executable Statements.

Assignment statements may be DATA statements and are used mainly to assign initial values to variables

```
DATA A,B,C,D,E/1.,2.,3.,4.,5./X,Y,Z/3*10.
```

or values assigned by numerical or character constants or by expressions (see Table 1-25 for arithmetic operators and precedence)

```
PI = 3.1415927
C1 = 'This is a test'
X(I,J) = X(I - 1,J) + Y(J)*Z**3
```

Input/Output statements may be either list-directed (stream) or formatted. List-directed I/O statements may be

```
READ(device#,*)var1,var2, . . .
WRITE(device#,*)var1,var2, . . .
PRINT * var1,var2, . . .
```

Table 1-25
Precedence of FORTRAN Operators

Class	Level	Symbol or Mnemonic
EXPONENTIAL	First	**
ARITHMETIC	Second	– (negation) and + (identity)
	Third	*, /
	Fourth	+, –
RELATIONAL	Fifth	.GT.,.GE.,.LT.,.LE.,.EQ.,.NE.
LOGICAL	Sixth	.NOT.
	Seventh	.AND.
	Eighth	.OR.
	Ninth	.EQV. ,.NEQV.

where device# refers to either a device such as a screen or printer or to a disk file. Usually the default input device (referenced by an asterisk rather than a device#) is the keyboard and the default output device (also referenced by an asterisk) is the screen. PRINT connects only to the printer. Other devices and files may be assigned device numbers through file handling statements (see later). A statement requesting list-directed input from the default device (the keyboard) might be, for example,

```
READ(*,*)TEMP,PRSSR,LENGTH
```

Formatted I/O statements require edit specifiers (see Table 1-26 for list) for each variable to be handled and may also include strings of characters enclosed in single quote marks. The format may be given in a separate format statement (referenced by a line number and labeled) or, in many systems, may be enclosed in quotes and parentheses in the I/O statement itself. The general form for formatted I/O statements is

```
      READ(device#,label)var1,var2, . . .
      WRITE(device#,label)var1,var2, . . .
      PRINT label var1,var2, . . .
label FORMAT(list of specifications)
```

For example,

```
    READ(*,100)X(1),Y,I
100 FORMAT(1X,2F12.4,I5)

    WRITE(6,110)(TEMP(I),I = 1,5)
110 FORMAT(1X,5(F7.3,2X))

    WRITE(*,'("TEMP AT SURFACE IS ",F8.2)')TSURF
```

File handling statements allow the manipulation of sequential and random access files. (Since there is considerable variation from one system to the next, the following information is given in general terms only.) Devices are treated

Table 1-26
FORTRAN Edit Specifiers

A	Character data fields
D	Double precision data fields
E	Real data fields-exponential (E) notation
F	Real data fields-decimal notation
G	General form
H	Character constants
I	Integer data fields
L	Logical data fields
P	Scaie factors, used with D, E, F. and G specifiers to shift the decimal point or exponent size for output
S	Restores the optional + convention to the compiler
SP	Prints + with all subsequent positive data
SS	Suppresses + for all subsequent positive data
TL	Next character will be inputloutput the specified number of spaces left of the current position
TR	Next character will be input/output the specified number of spaces right of the current position
X	Skip the specified number of spaces before next character is input/output
:	Ends format control if no more data items in list
/	Skips a record
	Specifiers for input operations only
BN	Specifies that blank characters are to be ignored
BZ	Specifies blank characters are to be read as zeros
	Specifications for time four most common data types
Integer	Iw
Real	Fw.d
Character	Aw
Exponential	Ew.d

as sequential files, while disk files may be sequential or random access. The following statements are generally accepted forms:

OPEN(list of specifiers)-Connects an existing file to an I/O device or generates a new file, and specifies a device (unit) number. The specifiers (required and optional) are:

UNIT =	file unit (device) number
IOSTAT	integer variable for I/O status
FILE	name of file
ERR	label for error transfer
STATUS	file status descriptor, may be OLD, NEW, SCRATCH, or UNKNOWN
ACCESS	may be SEQUENTIAL or DIRECT
FORM	may be FORMATTED or UNFORMATTED
RECL	record length, if file access is DIRECT
BLANK	specifies blank handling, either NULL or ZERO

CLOSE(list of specifiers)—Disconnects a file. The specifiers may be:

UNIT =	file unit (device) number
IOSTAT	as in OPEN
ERR	as in OPEN
STATUS	may be KEEP or DELETE

INQUIRE(list of specifiers)-Returns information about the attributes of a file. Besides the UNIT = , IOSTAT and ERR specifiers, the following specifiers may be included:

EXIST	returns .TRUE. if file exists, else .FALSE.
OPENED	returns .TRUE. if open, else .FALSE.
NUMBER	returns number of connected device
NAMED	returns .TRUE. or .FALSE.
NAME	returns name of file
FORM	returns FORMATTED or UNFORMATTED
RECL	returns record length in direct access file
NEXT REC	returns number of next record in direct access file
BLANK	returns whether blanks or zeros specified

The following statements must include the UNIT = and may include the IOSTAT and/or the ERR specifiers:

REWIND(list of specifiers)	— Causes a sequential file to be rewound to first record
BACKSPACE(list of specifiers)	— Causes a sequential file to rewind one record
ENDFILE(list of specifiers)	— Places an end-of-file mark on a sequential file

A few examples of file-handling statements are:

```
OPEN(UNIT = 6,IOSTAT = FSTAT,FILE = 'PRINTER',STATUS = 'NEW')
OPEN(4,FILE = 'DATA1',STATUS = 'OLD')
CLOSE(6)
BACKSPACE(4,ERR = 500)
```

Control statements affect the flow of instructions within a program unit. (For control between units, see subprogram statements later.) These may be general, conditional, or iterative statements. *General control statements* are:

PAUSE n—Interrupts program run, resumption upon pressing "Enter"; n is an optional character constant or integer of less than 5 digits, e.g.,

PAUSE 'VALUE INVALID'

STOP n—Halts program run (n as above)

STOP 10050

GOTO i—Transfers control to statement labeled i, where i is an integer constant or variable with value of label, e.g.,

GOTO 700

Conditional statements are as follows:

IF(e)s1,s2,s3—Arithmetic if, where e is an arithmetic expression and s1,s2,s3 are statement labels; transfers control to a labeled statement depending on whether e evaluates to a negative, zero, or positive value, respectively. For example,

```
IF(I - 5) 30,40,50
```

will transfer control to the statement labeled 40 if I = 5.

IF(e)st—Logical if, where e is a logical expression (see Table 1-26 for relational and logical operators) and st is any executable statement except DO, IF, ELSEIF, ENDIF, or END. For example,

```
IF(I.EQ.1) WRITE(*,*)'YES'
```

IF(e) THEN—Block if, where e is a logical expression, followed by a sequence of statements and completed by an ENDIF statement. The block may include sub-blocks introduced by one or more ELSEIF statements and/or one ELSE statement and all sub-blocks may contain nested IF-THEN-ELSE blocks within them. For example,

```
IF(I.EQ.J) THEN
  X = 4
  Y = 5
ENDIF
IF(I.EQ.J) THEN
  X = 4
ELSE
  X = 5
ENDIF
IF(I.EQ.J) THEN
  X = 4
ELSEIF(I.EQ.K) THEN
  X = 5
ELSE
  X = Y
ENDIF
```

The **iterative statement** in FORTRAN is the DO statement (although others may be constructed using conditional statements and GOTOs), where

```
DO st i = init,term,incr
```

introduces the repetitive section, and st is the label of the executable statement marking the end of the loop (usually, but not necessarily, a CONTINUE statement), i is the index or control integer variable and init, term, and incr (optional) are the initial value for i, the terminal value for i, and the increment of i to be used, respectively. These values must be integer constants, variables, or expressions in standard FORTRAN 77, but many extensions to the language allow real values to be used. DO loops may be nested to a level determined by a specific compiler. For example,

```
      DO 100 I = 1,10,2
        PR(I) = PPE(I - 1)
  100 CONTINUE

      DO 100 I = 10,1, - 1
        DO 100 J = 1,10
  100 ATR(I,J) = P(J,I)*FRIC

      DO 100 I = - 5,25,5
        PF(I) = TFP(I) - FFG
        IF(PF(I).GE.LMT) GOTO 110
  100 CONTINUE
  110 MAXC = I
```

Statement functions are defined before any other executable statements in the program and are called in the same way that subprogram or intrinsic functions are called (see subprogram statements later). They are one-line expressions that receive one or more parameters from the calling statement and return a single calculated value to the function name in the calling statement. For example, a statement function defined as

```
FDPT(X(I),Y(I),Z,I) = X(I)*Y(I) + Z**I
```

will calculate a value depending on the values of X(I),Y(I),Z, and I at the time of calling and return the calculated value to the calling statement through FDPT.

Subprogram statements are those used to transfer control between program units—the main program, functions, and subroutines. A *function call* is performed by invoking the name of the function module in an assignment statement, such as

```
X = FDPR(Z,Y(I))*PRF
```

which will transfer control to the function FDPR and pass the values Z, and Y(I) to that unit. An intrinsic function or a statement function may be called in the same way. (See Table 1-27 for a list of FORTRAN 77 intrinsic functions.) A *subroutine call* is performed by a statement such as

```
CALL CALCSUB(MATFOR,I,J,PVAL)
```

which will transfer control to the subroutine CALCSUB and pass (and/or return) the values MATFOR,I,J, and PVAL. A subroutine may have an ENTRY name (parameter list) statement imbedded within it, which when called in the same manner as the main subroutine call, will receive control transfer at that point. Control passes from the called unit back to the calling unit when a RETURN statement is encountered. Given a subroutine

```
Subroutine CALCSUB(MAT,M,N,P1)
  REAL MAT(100)
  P1 = MAT(M) + MAT(N)
  RETURN
  ENTRY NEWCALC(MAT,M,N,P2)
  P2 = MAT(M) + MAT(N)
  RETURN
END
```

Table 1-27
FORTRAN Intrinsic Functions

	Integer
IABS	Returns the absolute value of an argument
IDIM	Returns the positive difference between two arguments
IDINT	Converts a double-precision argument to integer by truncation
IFIX	Converts a real argument to integer by truncation
INT	Truncates the decimal part of an argument
ISIGN	Transfers the sign from one integer argument to the other
MAX0	Selects the largest value of several arguments
MAX1	Selects the largest value of several arguments, but converts any real result to integer
MIN0	Selects the smallest value of several arguments
MIN1	Selects the smallest value of several arguments, but converts any real result to integer
MOD	Returns the remainder from division of two arguments
	Real
ABS	Returns the absolute value of an argument
ACOS	Returns the arc cosine of an argument
AIMAG	Returns the imaginary part of a complex number
AINT	Truncates the decimal part of an argument
ALOG	Returns the natural logarithm of an argument
ALOG10	Returns the common logarithm of an argument
AMAX1	Selects the largest value of several arguments
AMIN1	Selects the smallest value of several arguments
AMOD	Returns the remainder from division of two arguments
ANINT	Returns the whole number nearest in value to the argument
ASIN	Returns the arc sine of an argument
ATAN	Returns the arc tangent of an argument
ATAN2	Returns the arc tangent of two arguments [$\arctan(a_1/a_2)$]
COS	Returns the cosine of an argument
COSH	Returns the hyperbolic cosine of an argument
DIM	Returns the positive difference between two arguments
EXP	Returns the exponential e raised to the power of the argument
FLOAT	Converts an argument to a real number
NINT	Returns the nearest integer value
REAL	Converts a complex argument to a real value
SIGN	Transfers the sign from one argument to the other
SIN	Returns the sine of an argument
SINH	Returns the hyperbolic sine of an argument
SQRT	Returns the square root of an argument
SNGL	Converts a double precision argument to single precision
TAN	Returns the tangent of an argument
TANH	Returns the hyperbolic tangent of an argument
	Double Precision
DABS	Returns the absolute value of an argument
DACOS	Returns the arc cosine of an argument

DASIN	Returns the arc sine of an argument
DATAN	Returns the arc tangent of one argument
DATAN2	Returns the arc tangent of two arguments [arctan(a_1/a_2)]
DBLE	Converts an argument to double precision
DCOS	Returns the cosine of an argument
DCOSH	Returns the hyperbolic cosine of an argument
DDIM	Returns the positive difference between two arguments
DEXP	Returns the exponential e raised to the power of the argument
DINT	Truncates the decimal part of an argument
DLOG	Returns the natural logarithm of an argument
DLOG10	Returns the common logarithm of an argument
DMAX1	Selects the largest value of several arguments
DMIN1	Selects the smallest value of several arguments
DMOD	Returns the remainder from division of two arguments
DNINT	Returns the whole number ciosest in value to the argument
DPROD	Converts the product of two real arguments to double precision
DSIGN	Transfers the sign from one argument to the other
DSIN	Returns the sine of an argument
DSINH	Returns the hyperbolic sine of an argument
DSQRT	Returns the square root of an argument
DTAN	Returns the tangent of an argument
DTANH	Returns the hyperbolic tangent of an argument
IDINT	Converts the argument to the nearest integer value
	Complex
CABS	Returns the absolute value of an argument
CCOS	Returns the cosine of an angle
CEXP	Returns the exponential e raised to the power of the argument
CLOG	Returns the natural logarithm of the argument
CMPLX	Converts the argument to a complex number
CONJ	Returns the conjugate of a complex function
CSQRT	Returns the square root of an argument
CSIN	Returns the sine of an argument

a call to CALCSUB as before will return a value through P1 to PVAL and a call

```
CALL NEWCALC(MAT,K,L,PVAL)
```

will transfer control in at the ENTRY statement and return a value through P2 to PVAL. A BLOCK DATA subprogram enters data into the variables declared in a labeled COMMON block and has the form

```
BLOCK DATA
  (DATA, DIMENSION, IMPLICIT, TYPE, EQUIVALENCE,
  COMMON and PARAMETER statements)
END
```

Pascal Language

Pascal names (of units, variables, array elements, etc.), consisting of an initial letter followed by letters or digits, may be as long as needed in standard Pascal; however, in practice, most compilers require that a name be unique within a certain number of characters. (See Table 1-28 for reserved words which may not be used as names.)

Data Types (Data structures of all types, predefined and user-defined, must be declared; see "Statements" for type declaration form.)

Integer—Predefined, scalar (ordered) type with a machine-dependent limit, predeclared as a constant MAXINT.
Real—Predefined scalar type, decimal, or scientific notation may be used.
Boolean—Predefined logical type, ordered so that false < true.
Char—Predefined character type, ordered by code. (See Table 1-22 for two common code sets.)
User-defined—See "Statements."

Data Structures

Constant—Type and value assigned at declaration, and once defined, neither type nor value may be changed.
Variable—Type assigned at declaration and may not be changed; value assigned by a numerical or character constant, by input or by an expression.
Array—May have more than one dimension; number of dimensions and type are assigned at declaration and may not be changed; values are assigned to array elements (which must be of same type) as they are to variables.
Packed array—Produces more efficient use of memory, but slower program execution than a regular array.
Record—The elements (fields) may be of different types and may be accessed at random; fields and their types are assigned at declaration and may not be changed; field values are assigned as are variable values.
File—Composed of elements (which may be records or other data structures) of the same type; sequential files only in standard Wirth Pascal; external (physical) files (for input and output) must be declared with the program heading (see "Statements") and internal (temporary) files may be added for use within the program; all files except INPUT and OUTPUT must be declared in TYPE and/or VAR sections.
Set—Type must be ordinal; must be processed as whole (cannot be broken down into elements).

Table 1-28
Pascal Reserved Words

and	array	begin	case	const
div	do	downto	else	end
file	for	function	goto	if
in	label	mod	nil	not
of	or	packed	procedure	program
record	repeat	set	then	to
type	until	var	while	with

Pointer—Each value of a variable declared to be of a pointer type gives the memory location of the element to which it points.

Statements. A Pascal statement may be shorter than or may continue over more than one line, since a semi-colon separates statements. Reserved words are not separated from statements; therefore the statement preceding the word END (see later sections) requires no punctuation. END of the main program unit is followed by a period. Comment lines are included in the program by placing them between the symbols (* and *). Statements may be simple (single) or compound, i.e., a series of simple statements joined together in a block which is introduced by BEGIN and completed by END.

Nonexecutable Statements

Program unit heading—Program, procedure, or function name; the forms are:

```
PROGRAM name (input file name(s), output file name(s));
```

where one input and one output file name, designated as INPUT and OUTPUT, are assigned to default devices. For example,

```
PROGRAM TEST(INPUT,OUTPUT);
PROGRAM EDITOR(EDFILE, INPUT,OUTPUT);
```

and

```
FUNCTION name(parameter list:type, . . .):function type;
PROCEDURE name(parameter list:type, . . .);
```

The declaration VAR preceding a variable name(s) in a parameter list allows the sub-program to modify the value of the variable stored in memory; if there is no VAR preceding a variable name, the sub-program is passed only the value of the parameter at the time of calling and cannot modify the stored value of that variable in the calling unit, e.g.,

```
PROCEDURE SAMPLE(X:REAL; Y:CHAR);
PROCEDURE LOOP(VAR A,B,C:INTEGER; Y:CHAR);
FUNCTION CHECK(VAR NUM:REAL; VAR PNR:INTEGER):REAL;
FUNCTION BOUND(TEMP:REAL):BOOLEAN;
```

In standard Pascal, although only in a few implementations of the language, a function or procedure name may also be a parameter to a sub-program. The constants, types, and variables used in a sub-program, as well as those in the main program, must be declared (see following declarations).

Label declaration—Labels identify an executable statement which may then be referenced by GOTO; they may be 1-4 digits and must be declared by

```
LABEL unsigned integer(s);
```

For example,

```
LABEL 50,999;
```

Constant declaration—Must precede variable declarations and be of the form

```
CONST
  constant name = value;
  constant name = value;
  etc.
```

For example,

```
CONST
            PI = 3.1415927;
          FACT = TRUE;
       PIPENUM = 27;
       MINDPTH = 1500;
       MAXDPTH = 7500.0;
       CLRRADS = .34;
```

Type declaration—Must be placed after CONST sequence and before variable declarations; used to define new (not predefined) data types either as a range of values or as specific values within parentheses and to define arrays, records, and files. User-defined scalar types are ordered in increasing left to right sequence. For example,

```
TYPE
     UNIT = (PIPE,COLLAR,MOTOR);
     TEMP = ARRAY[1..34 OF REAL;
      ATP = ARRAY[1..50,1..10] OF CHAR;
     PREC = RECORD
                  TIME:  REAL;
                   RUN:  INTEGER;
                CRUNIT:  UNIT;
                 PRSSR:  REAL;
            END;
STACKPTR = ↑STACKELEMENT;
  STRING = PACKED ARRAY[1..MAXLENGTH OF CHAR;
             (* may be predefined on some systems *)
   TSURF = 0..80;
    CSET = SET OF 'A'..'Z';
    NODE = 1..PIPENUM;
```

Variable declaration—Placed after TYPE statements and defines the type of each variable, array, and record used in the program unit, for example

```
  NXTUNIT:  NODE;
      LOG:  PREC;
    COUNT:  INTEGER;
    PTEMP:  TEMP;
 PREVTEMP:  TEMP;
   LETTER:  CSET;
     NAME:  ARRAY[1..20] OF CHAR;
     RUNS:  FILE OF PREC;
```

BEGIN—Introduces the executable statements in each block of statements.

END—Completes the executable statements in each block; followed by a period at the end of the main program.

Executable Statements

Assignment statements—Assign values by numerical or character constants, by expressions (see Table 1-29 for arithmetic and set operators), or by input to variables, array elements, and fields

```
PIPENUM:    = 1;
MAXTEMP:    = PTEMP;
LETTER:     = [ ];     (* the empty set *)
NAME:       = 'WELL #1';
```

Input/Output statements—Two boolean functions that are useful in input processing are EOF, which is TRUE if the pointer is currently at the end of the input file and FALSE otherwise, and EOLN, which is TRUE if the pointer is at the end of the current input line and FALSE otherwise.

Table 1-29
Pascal Operators

Operator	Description	Operand(s) Type	Result Type
:=	Assignment	Any, except file	—
+	Addition	Integer or real	Integer or real
	Set union	Any set type	Same as operand
–	Subtraction	Integer or real	Integer or real
	Set difference	Any set type	Same as operand
*	Multiplication	Integer or real	Integer or real
	Set intersection	Any set type	Same as operand
div	Integer division	Integer	Integer
/	Real division	Integer or real	Real
mod	Modulus	Integer	Integer
not	Logical negation	Boolean	Boolean
or	Disjunction	Boolean	Boolean
and	Conjunction	Boolean	Boolean
<=	Set inclusion	Any set type	Boolean
	Less than or equal	Any scalar type	Boolean
=	Equivalence	Boolean	Boolean
	Equality	Scalar, set, pointer	Boolean
<>	Exclusive or	Boolean	Boolean
	Inequality	Scalar, set, pointer	Boolean
>=	Set inclusion	Any set type	Boolean
	Greater than or equal	Any scalar type	Boolean
<	Less than	Any scalar type	Boolean
>	Greater than	Any scalar type	Boolean
in	Set membership	Left operand scalar Right operand: set with base type same as left operand	Boolean

READ(variable list);–Stores each item of data in a location indicated by the variable associated with it

READLN(variable list);–Causes input control to shift to the next line when the end of the variable list or line of input is reached

WRITE(variable list);–Can also output a character string by placing it, enclosed by single quotes, inside the parentheses; a field (a format for output) may be assigned to a data item as

```
WRITE(SUM:5);
```

or, in the case of a real number, as

```
WRITE(X:10:2);
```

specifying the total width and also the number of decimal places. If no field is specified, a default width is assumed.

WRITELN(variable list);–Output shifts to the next line when the output list has been completed

Some examples of Input/Output statements are:

```
WRITE(I,X);
WRITELN('J = ', ABS(J):6);
WRITE(X + Y:5:2);
READLN(X,Y);
READ(A,B,C,D);
```

If no other files are specified for input or output in the program heading, the default files or devices (usually the keyboard and the printer or the screen) are used. If other (declared) files are to be used, they must be prepared (see file handling statements) and then specified in the input/output statements, as follows

```
WRITE(filename, variable);
READ(filename, variable);
```

For example,

```
WRITE(PFILE,PRSSR(I));
READ(TEMP,TFCT);
```

For details on file pointers, buffer variables, and GET and PUT operations, see one of the advanced Pascal texts listed in the references.

File-handling statements–Every file, except the INPUT and OUTPUT files, must be defined in a VAR statement that gives its name and type, e.g.,

```
VAR
 TEMP: FILE OF CHAR;
PRSSR: FILE OF REAL;
```

and must be prepared before data can be written to or read from that file.

REWRITE(filename);–Produces an empty file, to which data can be written, and

WRITE(filename, a,b,c,d);

can then write four items to the file.

RESET(filename);—Prepares a file for reading, beginning with the first item in the file, and

READ(filename,X);

can then read the first item from the file.

Control statements—Affect the flow of statement execution within a program unit (see subprogram statements below for passing of control between units). The only general control statement in Pascal (which is rarely needed) is:

GOTO line number;—Shifts control to the statement beginning at the specified line number.

Conditional statements may be nested with other conditional statements and/ or with iterative statements. They are:

IF boolean statement THEN statement-Where statement may be a simple or a compound statement (for boolean and relational operators, see Table I-29) and

```
IF boolean statement
   THEN statement 1
   ELSE statement 2—where the statements are as defined above
```

For example,

```
IF (PRSSR < MINP) OR (PRSSR > MAXP) THEN
   BEGIN
      X(PIPENUM: = PREVX;
      CORRPR(PRSSR,X(PIPENUM),AVT)
   END
ELSE
      PREVX: = X(IP);
```

and

```
CASE  selector expression OF
         label: statement;
         label: statement;
               . . .
         label: statement
END
```

where, when the selector expression is evaluated, producing an ordinal value, the statement following the label value which matches the selector expression is executed, e.g.,

```
CASE PIPENUM OF
        1: PREV: = FALSE;
   MAXP: NEXT: = FALSE
END;
```

Iterative statements control how many times a simple statement or a compound statement is repeated, and also may be nested with conditional and/or other iterative statements. They may be

```
WHILE boolean expression DO
  statement
```

or

```
REPEAT
  statement
UNTIL boolean expression
```

or

```
FOR variable: = expression TO expression DO
  statement
```

where the variable and expressions must be of ordinal (not real) type and TO may be replaced by DOWNTO. In all types of iterative constructions, statement may refer to a simple statement or to a compound statement introduced by BEGIN and completed with END, for example,

```
WHILE NOT EOF DO
  BEGIN
    I: = 0;
    REPEAT
      I: = I + 1;
      READ(TEMP(I))
    UNTIL I > MAXI
  END;
FOR I: = MAXI DOWNTO MINI DO
  BEGIN
    X(I): = FNC(X(I - 1) * ABS(Z);
    IF X(I) > XFT THEN AGSTP(X(I),I,PTP(I))
  END;
```

Subprogram statements—Used to transfer control between program units-the main program, functions, and procedures. A *function call* is the invocation of a function name in an assignment statement, such as

```
X: = DEF(Y)*Z;
```

which will transfer control to DEF and pass the variable Y to that function. Control returns to the calling unit when the END statement of the function is encountered. Intrinsic functions, supplied in the system library, are called in the same manner. (See Table 1-30 for a list of standard Pascal intrinsic functions.) A *procedure call* is made by stating the name of the procedure and the variables to be passed, e.g.,

```
CALCPRSSR(PIPENUM,TEMP,FFACTR);
```

will transfer control to procedure CALCPRSSR. Control returns to the calling unit when the procedure END statement is found.

Table 1-30
Pascal Intrinsic Functions

ABS	Returns the absolute value of an argument
ARCTAN	Returns the arc tangent of an argument
CHR	Returns a character in the position in the collating sequence given by the argument
COS	Returns the cosine of an argument
EOF	Returns TRUE if the end of file has been reached, otherwise returns FALSE
EXP	Returns the exponential e raised to the power of the argument
LN	Returns the natural logarithm of an argument
ODD	Returns TRUE if an integer argument is odd, otherwise returns FALSE
ORD	Returns the ordinal number of an argument
PRED	Returns the predecessor to an ordinal argument
RESET	Initializes an input file to accept values
REWRITE	Initializes an output file to accept values
ROUND	Converts a real argument to an integer value by rounding
SIN	Returns the sine of an argument
SQR	Returns the square of an argument
SQRT	Returns the square root of an argument
SUCC	Returns the successor of an ordinal argument
TRUNC	Converts a real argument to an integer value by truncating

System Software

System software is the connection between the user and the machine. It provides management of the system resources and utilities which simplify development of applications programs. Essential system software includes

1. **Translators**—Assemblers, interpreters, and/or compilers that translate symbolic language into machine code.
2. **Linkers and loaders**—Linkers resolve references between program units and allow access to system libraries; loaders place code into the main memory locations from which it will be executed.
3. **Operating systems**—Manage hardware resources of the computer system. Utilization may be of the *batch* method, in which program units, libraries, and data are submitted to the system along with the job control language statements needed to run the program. The operating system allocates the central processing unit to one batch job at a time, according to a hierarchical system. *Time-sharing* systems provide interactive sharing of resources by many users. The system must interweave allocation of resources to users and manage memory locations.
4. **Utility programs**—Simplify use by performing particular tasks for the programmer, such as editing, debugging, etc.
5. **File manager systems**—Maintain files and handle data input to and output from the files. *Database management systems* (DBMS) contain integrated sets of files related by their use and provide uniform software interfaces for accessing data. The essential relationships between records in the files may be of several types, including sequential, associative, or hierarchical.
6. **Telecommunications monitors**—Supervise communications between remote terminals and the central computer.

System Hardware

System hardware consists of the central processor, the input devices (usually a keyboard), the output devices (probably both a video display terminal and a hardcopy printer), long-term storage devices, and perhaps communications components. In smaller systems, more than one of these components may be "built in" to one unit, while in larger systems there may be many units each of several components associated with the system.

The *central processing unit* (CPU) consists of the arithmetic-logic unit (ALU), the control unit, and the central storage (short-term memory) unit. The CPU is normally classified by size of the word (number of bits in one piece of information or address), size of memory (which is partly dependent upon the size of the word available and is expressed in KB, i.e., kilobytes) and by speed of operations (given in megaHertz, mH). There are many combinations of these factors, depending upon the processor chip used and upon the architecture of the machine. Speed of operation and of data transfer is of major importance in large number-crunching programs. Memory size affects the size of the program and the amount of data that may be held at one time, while word size primarily affects the size of memory available. A useful addition to the system, if a need for large-scale number crunching is anticipated, is the arithmetic coprocessor chip, which performs high-speed numerical operations.

Keyboards are the most widely used *input devices,* but optical scanners and digital pads (for computer-aided design) are some additional input devices. Input may also be from files stored on a disk or tape.

Video display terminals for output are available in several sizes (measured diagonally in inches), various resolutions (number of pixels, an important factor in graphics) and a choice of monochrome or color. The type of screen chosen will depend upon the anticipated use. Hardcopy *output devices* usually include a printer, either a dot-matrix printer, in which the characters are formed by dot combinations, or a laser printer. The number of dots per character unit is one consideration in the quality of print in a dot-matrix printer, although options such as double-strike may increase the print quality at the cost of speed. Speeds of laser printers (as opposed to the larger line printers) are measured in pages per minute (ppm), but the actual throughput on printers is not always a direct correlation of ppm. Laser printers have become available at affordable prices and have the major advantage of producing a much lower level of noise, as well as of high-speed operation. For engineering and design use, plotters, either black-white or multicolored are frequently added to the system. Output may also be sent to a disk or tape for long-term storage.

Long-term memory storage devices may be cassettes, for small amounts of storage on personal computers, several sizes of floppy diskettes, hard (fixed) disks or magnetic tapes (frequently used for duplicate "back-up" storage). This choice can be based on the amount of available storage needed without changing cassettes or floppy diskettes and on the speed of access to the device required, with cassettes being the slowest and hard disks the fastest accessed devices.

Systems may also include *modems,* which connect small computers or terminals to other computers or workstations either in-house or over telephone system lines and whose speed of transmission is rated by *baud* (for binary information units, the number of bits transmitted per second). Common baud rates for small systems are 2400 or 9600 with higher rates possible. Computers may also be networked together to share data or peripheral components such as high-speed printers.

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2

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2

General Engineering and Science

BASIC MECHANICS (STATICS AND DYNAMICS)

Mechanics is the physical science that deals with the effects of forces on the state of motion or rest of solid, liquid, or gaseous bodies. The field may be divided into the mechanics of rigid bodies, the mechanics of deformable bodies, and the mechanics of fluids.

A rigid body is one that does not deform. True rigid bodies do not exist in nature; however, the assumption of rigid body behavior is usually an acceptable accurate simplification for examining the state of motion or rest of structures and elements of structures. The rigid body assumption is not useful in the study of structural failure. Rigid body mechanics is further subdivided into the study of bodies at rest, *statics*, and the study of bodies in motion, *dynamics*.

Definitions, Laws, and Units

Fundamental Quantities

All of Newtonian mechanics is developed from the independent and absolute concepts of *space, time*, and *mass*. These quantities cannot be exactly defined, but they may be functionally defined as follows:

Space. Some fixed reference system in which the position of a body can be uniquely defined. The concept of space is generally handled by imposition of a coordinate system, such as the Cartesian system, in which the position of a body can be stated mathematically.

Time. Physical events generally occur in some causal sequence. Time is a measure of this sequence and is required in addition to position in space in order to fully specify an event.

Mass. A measure of the resistance of a body to changes in its state of motion.

Derived Quantities

The concepts of space, time, and mass may be combined to produce additional useful measures and concepts.

Particle. An entity which has mass, but can be considered to occupy a point in space. Rigid bodies that are not subject to the action of an unbalanced couple often may be treated as particles.

Body. A collection of particles. A rigid body is a rigidly connected collection of particles.

Force. The action of one body on another. This action will cause a change in the motion of the first body unless counteracted by an additional force or forces. A force may be produced either by actual contact or remotely (gravitation, electrostatics, magnetism, etc.). Force is a vector quantity.

Couple. If two forces of equal magnitude, opposite direction, and different lines of action act on a body, they produce a tendency for rotation, but no tendency for translation. Such a pair of forces is called a couple. The magnitude of the moment produced by a couple is calculated by multiplying the magnitude of one of the two forces times the perpendicular distance between them. Moment is a vector quantity, and its sense of direction is considered to be outwardly perpendicular to the plane of counterclockwise rotation of the couple. The moment of a single force about some point A is the magnitude of the force times the perpendicular distance between A and the line of action of the force.

Velocity. A measure of the instantaneous rate of change of position in space with respect to time. Velocity is a vector quantity.

Acceleration. A measure of the instantaneous rate of change in velocity with respect to time. Acceleration is a vector quantity.

Gravitational acceleration. Every body falling in a vacuum at a given position above and near the surface of the earth will have the same acceleration, g. While this acceleration varies slightly over the earth's surface due to local variations in its shape and density, it is sufficiently accurate for most engineering calculations to assume that $g = 32.2\ ft/s^2$ or $9.81\ m/s^2$ at the surface of the earth.

Weight. A measure of the force exerted on a body of mass M by the gravitational attraction of the earth. The magnitude of this force is

$$\mathbf{W} = \mathbf{Mg} \tag{2-1}$$

where **W** is the weight of the body. Strictly speaking, weight is a vector quantity since it is a force acting in the direction of the gravitational acceleration.

General Laws

The foregoing defined quantities interact according to the following fundamental laws, which are based upon empirical evidence.

Conservation of mass. The mass of a system of particles remains unchanged during the course of ordinary physical events.

Parallelogram law for the addition of forces. Two forces, $\mathbf{F}_1$ and $\mathbf{F}_2$, acting on a particle may be replaced by a single force, **R**, called their resultant. If the two forces are represented as the adjacent sides of a parallelogram, the diagonal of the parallelogram will represent the resultant (Figure 2-1).

Principle of transmissibility. A force acting at a point on a body can be replaced by a second force acting at a different point on the body without changing the state of equilibrium or motion of the body as long as the second force has the same magnitude and line of action as the first.

Newton's Laws of Motion

1. A particle at rest will remain at rest, and a particle in motion will remain in motion along a straight line with no acceleration unless acted upon by an unbalanced system of forces.

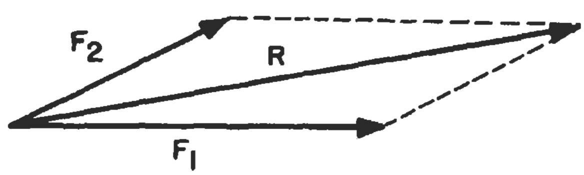


Figure 2-1. Parallelogram law for addition of forces.

2. If an unbalanced system of forces acts upon a particle, it will accelerate in the direction of the resultant force at a rate proportional to the magnitude of the resultant force. This law expresses the relationship between force, mass, and acceleration and may be written as

$$\mathbf{F} = M\mathbf{a} \tag{2-2}$$

where **F** is the resultant force, M is the mass of the particle, and **a** is the acceleration of the particle.
3. Contact forces between two bodies have the same magnitude, the same line of action, and opposite direction.

Gravitation. Two particles in space are attracted toward each other by a force that is proportional to the product of their masses and inversely proportional to the square of the distance between them. Mathematically this may be stated as

$$|\mathbf{F}| = \frac{Gm_1m_2}{r^2} \tag{2-3}$$

where $|\mathbf{F}|$ is the magnitude of the force of gravitational attraction, G is the universal gravitational constant ($6.673 \times 10^{-11} m^3/kg - s^2$ or $3.44 \times 10^{-8}\ ft^4/lb - s^4$), m_1 and m_2 are the masses of particles 1 and 2, and r is the distance between the two particles.

Systems of Units

Two systems of units are in common usage in mechanics. The first, the SI system, is an absolute system based on the fundamental quantities of space, time, and mass. All other quantities, including force, are derived. In the SI system the basic unit of mass is the kilogram (kg), the basic unit of length (space) is the meter (m), and the basic unit of time is the second (s). The derived unit of force is the Newton (N), which is defined as the force required to accelerate a mass of 1 kg at a rate of 1 m/s^2.

The U.S. customary or English system of units is a gravitational system based upon the quantities of space, time, and force (weight). All other quantities including mass are derived. The basic unit of length (space) is the foot (ft), the basic unit of time is the second (s), and the basic unit of force is the pound (lb). The derived unit of mass is the slug, which is the unit of mass that will be accelerated by a force of one pound at a rate of 1 ft/s^2. To apply the slug in practice, as in Equation 2-2, the weight in pounds mass must first be divided by $g = 32.2\ ft/s^2$, thus generating a working mass in units of $lb - s^2/ft$, or slugs.

Statics

If there are no unbalanced forces acting on a *particle*, the particle is said to be in static equilibrium, and Newton's second law reduces to

$$\sum \mathbf{F} = 0 \tag{2-4}$$

Thus, solving a problem in particle statics reduces to finding the unknown force or forces such that the resultant force will be zero. To facilitate this process it is useful to draw a diagram showing the particle of interest and all the forces acting upon it. This is called a *free-body diagram.* Next a coordinate system (usually Cartesian) is superimposed on the free-body diagram, and the forces are decomposed into their components along the coordinate axes. For the particle to be in equilibrium, the sum of the force components along each of the axes must be zero. This yields an algebraic equation to be solved for the forces in each coordinate direction.

Example 2-1

Block W, weighing 100 lb (see Figure 2-2) is attached at point A to a cable, which is, in turn, attached to vertical walls at points B and C. What are the tensions in segments AB and BC?

Breaking down the diagram into the various forces (Figure 2-2b):

- Force balance in the y direction:

$$\sum F_y = -100 + T_{AC} \sin 45° + T_{AB} \sin 15° = 0$$

$$0.707T_{AC} + 0.259T_{AB} = 100 \qquad \text{(a)}$$

- Force balance in the x direction:

$$\sum F_x = T_{AC} \cos 45° - T_{AB} \cos 15° = 0$$

$$0.707T_{AC} - 0.966T_{AB} = 0 \qquad \text{(b)}$$

and solving Example Equations a and b simultaneously yields

$T_{AB} = 81.6$ lb

$T_{AC} = 111.5$ lb

If there are no unbalanced forces and no unbalanced moments acting on a *rigid body,* the rigid body is said to be in static equilibrium. That is, Equation 2-4 must be satisfied just as for particles, and furthermore:

$$\sum \mathbf{M}_A = 0 \qquad (2\text{-}5)$$

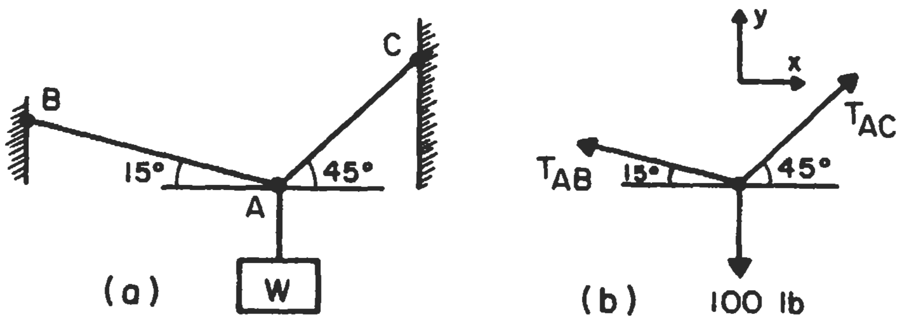


Figure 2-2. Diagram for Example 2-1.

where $\Sigma \mathbf{M}_A$ is the sum of the vector moments of all the forces acting on the body about any arbitrarily selected point A. In two dimensions this constitutes an algebraic equation because all moments must act about an axis perpendicular to the plane of the forces. In three dimensions the moments must be decomposed into components parallel to the principal axes, and the components along each axis must sum algebraically to zero.

Example 2-2

A weightless beam 10 ft in length (see Figure 2-3a) supports a 10-lb weight, W, suspended by a cable at point C. The beam is inclined at an angle of 30° and rests against a step at point A and a frictionless fulcrum at point B, a distance of $L_1 = 6$ ft from point A. What are the reactions at points A and B?

Breaking the diagram down into the various forces (Figure 2-3b):

- Force balance in the x direction:

$$\sum F_x = R_{Ax} - 10 \cos 60° = 0$$

$$R_{Ax} = 5 \text{ lb}$$

- Moment balance about point A:

$$\overset{+}{\sum} M_A = (10)(10)\sin 60° - 6R_{By} = 0$$

$$R_{By} = 14.43 \text{ lb}$$

- Force balance in the y direction:

$$\sum Fy = R_{Ay} + R_{By} - 10 \sin 60° = 0$$

$$R_{Ay} - 10 \sin 60° = -R_{By}$$

$$R_{Ay} = -4.43 \text{ lb}$$

Note that although the direction assumed for R_{Ay} was incorrect, the sign of the result indicates the correct direction.

Whenever the weight of a body is significant in comparison to the external forces, the weight, or body force, must be considered in both the force and moment balances.

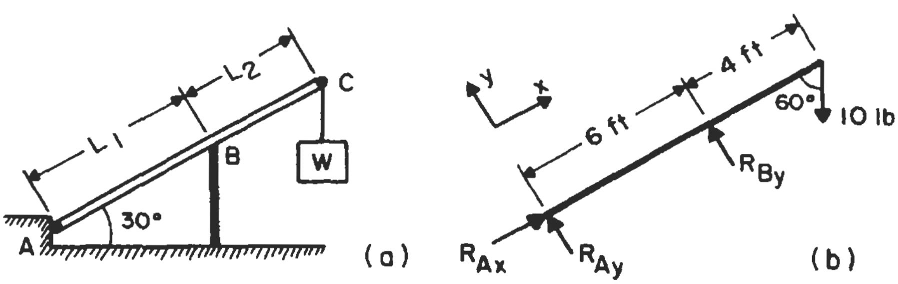


Figure 2-3. Diagram for Example 2-2.

The weight W of the body acts at the *center of gravity,* the Cartesian coordinates of which are found by:

$$\bar{x} = \frac{1}{W}\int_v x\,dw \tag{2-6}$$

$$\bar{y} = \frac{1}{W}\int_v y\,dw \tag{2-7}$$

$$\bar{z} = \frac{1}{W}\int_v z\,dw \tag{2-8}$$

The foregoing are volume integrals evaluated over the entire volume of the rigid body and dw is an infinitesimal element of weight. If the body is of uniform density, then the center of gravity is also called the *centroid.* Centroids of common lines, areas, and volumes are shown in Tables 2-1, 2-2, and 2-3. For a composite body made up of elementary shapes with known centroids and known weights the center of gravity can be found from

$$\bar{x} = \frac{\Sigma_i\, \bar{x}_i W_i}{\Sigma_i\, W_i} \tag{2-9}$$

$$\bar{y} = \frac{\Sigma_i\, \bar{y}_i W_i}{\Sigma_i\, W_i} \tag{2-10}$$

$$\bar{z} = \frac{\Sigma_i\, \bar{z}_i W_i}{\Sigma_i\, W_i} \tag{2-11}$$

Table 2-1
Centroids of Common Lines [2]

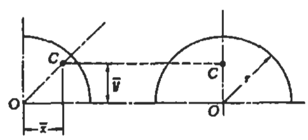
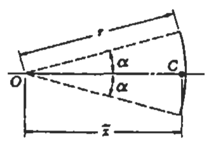
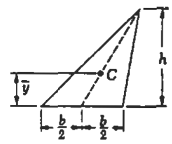
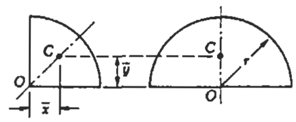
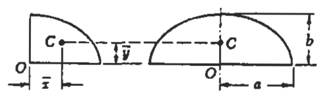
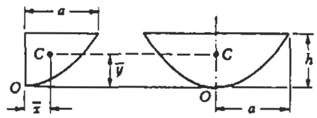
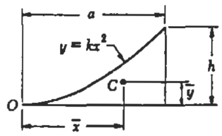
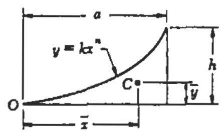
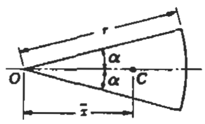
Shape		$\bar{x}$	$\bar{y}$	Length
Quarter-circular arc		$\frac{2r}{\pi}$	$\frac{2r}{\pi}$	$\frac{\pi r}{2}$
Semicircular arc		0	$\frac{2r}{\pi}$	πr
Arc of circle		$\frac{r \sin \alpha}{\alpha}$	0	$2\alpha r$

Table 2-2
Centroids of Common Areas [2]

Shape		$\bar{x}$	$\bar{y}$	Area
Triangular area			$\frac{h}{3}$	$\frac{bh}{2}$
Quarter-circular area		$\frac{4r}{3\pi}$	$\frac{4r}{3\pi}$	$\frac{\pi r^2}{4}$
Semicircular area		0	$\frac{4r}{3\pi}$	$\frac{\pi r^2}{2}$
Quarter-elliptical area		$\frac{4a}{3\pi}$	$\frac{4b}{3\pi}$	$\frac{\pi ab}{4}$
Semielliptical area		0	$\frac{4b}{3\pi}$	$\frac{\pi ab}{2}$
Semiparabolic area		$\frac{3a}{8}$	$\frac{3h}{5}$	$\frac{2ah}{3}$
Parabolic area		0	$\frac{3h}{5}$	$\frac{4ah}{3}$
Parabolic spandrel		$\frac{3a}{4}$	$\frac{3h}{10}$	$\frac{ah}{3}$
General spandrel		$\frac{n+1}{n+2}a$	$\frac{n+1}{4n+2}h$	$\frac{ah}{n+1}$
Circular sector		$\frac{2r\sin\alpha}{3\alpha}$	0	αr^2

Example 2-3

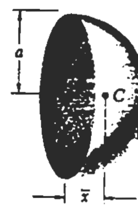
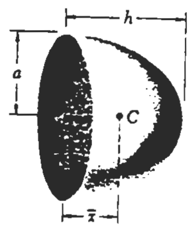
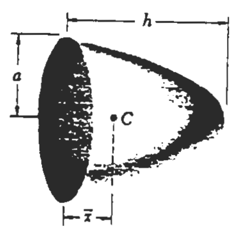
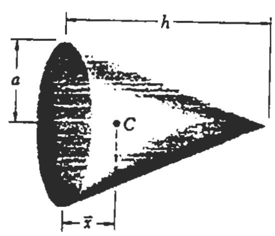
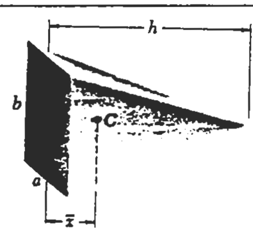
A mallet is composed of a section of a right circular cylinder welded to a cylindrical shaft, as shown in Figure 2-4a and b. Both components are steel, and the density is uniform throughout. Find the centroid of the mallet.

$r = 2$ in. $\quad d = 1$ in.

$L_1 = 6$ in. $\quad L_2 = 5$ in.

$L_3 = 1.5$ in. $\quad \gamma = 0.283$ lb/in.^{3}

Table 2-3
Centroids of Common Volumes [2]

Shape		$\bar{x}$	Volume
Hemisphere		$\frac{3a}{8}$	$\frac{2}{3}\pi a^3$
Semiellipsoid of revolution		$\frac{3h}{8}$	$\frac{2}{3}\pi a^2 h$
Paraboloid of revolution		$\frac{h}{3}$	$\frac{1}{2}\pi a^2 h$
Cone		$\frac{h}{4}$	$\frac{1}{3}\pi a^2 h$
Pyramid		$\frac{h}{4}$	$\frac{1}{3}abh$

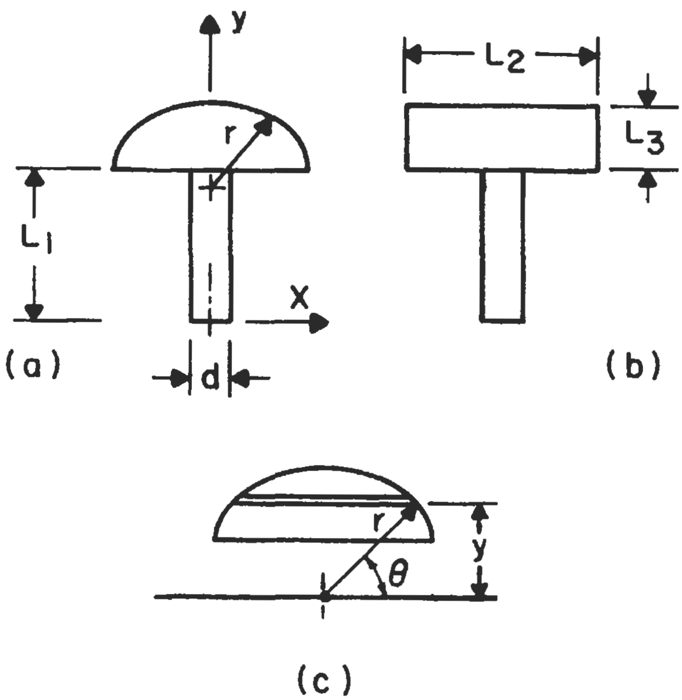


Figure 2-4. Diagram for Example 2-3.

Letting the center of the bottom of the handle be the origin, the centroid of Section 1, the handle, can be found by inspection as

$$\bar{x}_1 = 0$$

$$\bar{y}_1 = \frac{L_1}{2} = 3 \text{ in.}$$

$$w_1 = \frac{\pi}{4} d^2 L_1 \gamma = 1.33 \text{ lb}$$

In Section 2, the integral formula, 2-7, is applied

$$\bar{x}_2 = 0$$

$$\bar{y}_2 = a + \frac{1}{W_2}\int_a^b y dW_2$$

where $a = L_1 - (r - L_3)$
$b = L_1 + L_3$

Transforming the integral to polar coordinates as shown in Figure 2-4:

$$dW_2 = L_2\gamma 2r \cos\theta dy$$

$$y = r \sin\theta$$

$$dy = r \cos\theta d\theta$$

$$\theta_1 = \sin^{-1}\left(\frac{r - L_3}{r}\right) = 14.5° = 0.253 \text{ rad}$$

$$\theta_2 = 90° = \frac{\pi}{2} \text{ rad}$$

$$\cos\theta_1 = 0.968 \text{ rad}$$

$$\bar{y}_2 = L_1 - (r - L_3) + \frac{2L_2\gamma r^3}{W_2}\int_{\theta_1}^{\theta_2} \sin\theta \cos\theta \, d\theta$$

$$\bar{y}_2 = 5.5 + \frac{6.85}{W_2}$$

$$W_2 = \int_{L_1}^{L_1+L_3} dW = 2r^2L_2\gamma\int_{\theta_1}^{\theta_2} \cos^2\theta \, d\theta$$

$$W_2 = 6.09 \text{ lb}$$

Substituting W_2 into the equation for $\bar{y}_2$

$$\bar{y}_2 = 6.62 \text{ in.}$$

For the entire body

$$\bar{x} = 0$$

$$\bar{y} = \frac{\bar{y}_1W_1 + \bar{y}_2W_2}{W_1 + W_2} = \frac{3 \times 1.33 + 6.62 \times 6.09}{1.33 + 6.09}$$

$$\bar{y} = 5.97 \text{ in.}$$

When two bodies are in contact and there is a tendency for them to slide with respect to each other, a tangential *friction* force is developed that opposes the motion. For dry surfaces this is called *dry friction* or *coulomb friction*. For lubricated surfaces the friction force is called *fluid friction*, and it is treated in the study of fluid mechanics. Consider a block of weight **W** resting on a flat surface as shown in Figure 2-5. The weight of the block is balanced by a normal force **N** that is equal and opposite to the body force. Now, if some sufficiently small sidewise force **P** is applied (Figure 2-5b) it will be opposed by a friction force **F** that is equal and opposite to **P** and the block will remain fixed. If **P** is increased, **F** will simultaneously increase at the same rate until

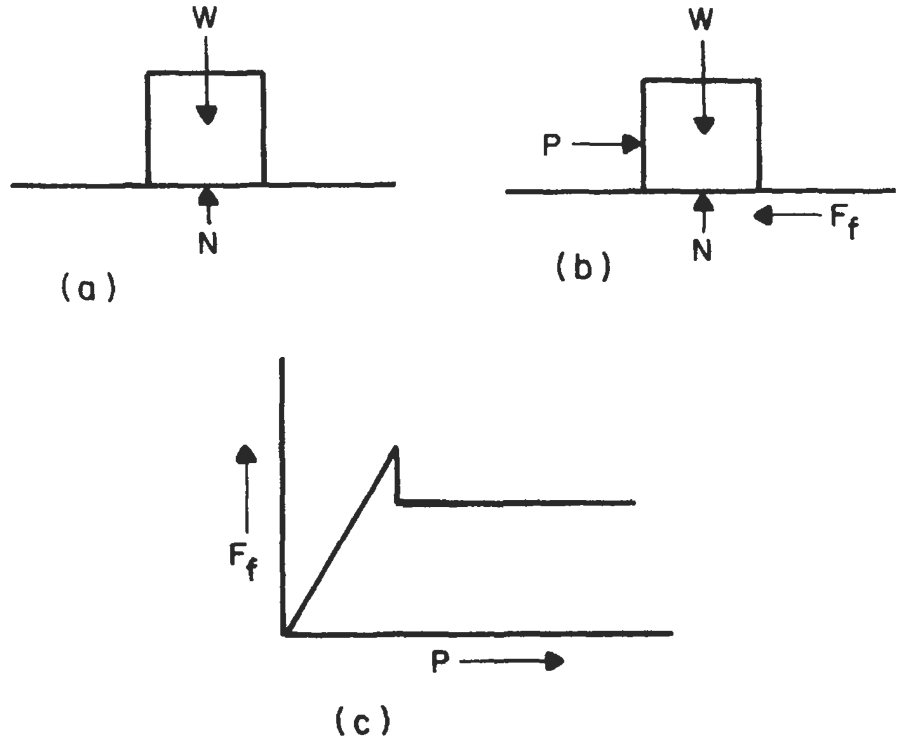


Figure 2-5. Dry friction force.

the maximum value of the static friction force is reached, at which point the block will begin to slide.

The maximum value of the static friction force is proportional to the normal force as

$$F_f = \mu_S N \tag{2-12}$$

where μ_S is called the *coefficient of static friction*. Once the block begins to slide, the friction force decreases slightly and remains at a constant value defined by

$$F_f = \mu_K N \tag{2-13}$$

where μ_K is the *coefficient of kinetic friction*. The magnitude of the friction force as a function of the applied force P is illustrated in Figure 2-5c, and typical values for μ_S and μ_K are given for both dry and lubricated surfaces in Table 2-4.

It is often necessary to compute the forces in *structures* made up of connected rigid bodies. A free-body diagram of the entire structure is used to develop an equation or equations of equilibrium based on the body weight of the structure and the external forces. Then the structure is decomposed into its elements and equilibrium equations are written for each element, taking advantage of the fact that by Newton's third law the forces between two members at a common frictionless joint are equal and opposite.

Table 2-4
Typical Values for μ_s and μ_K for Dry and Lubricated Surfaces [1]

(Reference letters indicate the lubricant used: numbers in parentheses give the sources. See footnote)

	Static		Sliding	
Materials	**Dry**	**Greasy**	**Dry**	**Greasy**
Hard steel on hard steel	0.78 (1)	0.11 (1, *a*)	0.42 (2)	0.029 (5, *h*)
		0.23 (1, *b*)		0.081 (5, *c*)
		0.15 (1, *c*)		0.080 (5, *i*)
		0.11 (1, *d*)		0.058 (5, *j*)
		0.0075 (18, p)		0.084 (5, *d*)
		0.0052 (18, h)		0.105 (5, *k*)
				0.108 (5, *m*)
				0.12 (5, *a*)
Mild steel on mild steel	0.74 (19)		0.57 (3)	0.09 (3, *a*)
				0.19 (3, *u*)
Hard steel on graphite	0.21 (1)	0.09 (1, a)		
Hard steel on babitt (ASTM No. 1)	0.70 (11)	0.23 (1, b)	0.33 (6)	0.16 (1, b)
		0.15 (1, c)		0.06 (1, c)
		0.08 (1, d)		0.11 (1, d)
		0.085 (1, e)		
Hard steel on babbitt (ASTM No. 8)	0.42 (11)	0.17 (1, b)	0.35 (11)	0.14 (1, b)
		0.11 (1, c)		0.065 (1, d)
		0.09 (1, d)		0.07 (1, d)
		0.08 (1, e)		0.08 (11, h)
Hard steel on babbitt (ASTM No. 10)		0.25 (1, b)		0.13 (1, b)
		0.12 (1, c)		0.06 (1, c)
		0.10 1, d)		0.055 (1, d)
		0.11 (1, e)		
Mild steel on cadmium silver				0.173 (2, f)
Mild steel on phosphor bronze			0.34 (3)	0.173 (2, f)
Mild steel on copper lead				0.145 (2, f)
Mild steel on cast iron		0.183 (15, c)	0.23 (6)	0.133 (2, f)
Mild steel on lead	0.95 (11)	0.5 (1, f)	0.95 (11)	0.3 (11, f)
Nickel on mild steel			0.64 (3)	0.178 (3, x)
Aluminum on mild steel	0.61 (8)		0.47 (3)	
Magnesium on mild steel			0.42 (3)	
Magnesium on magnesium	0.6 (22)	0.08 (22, y)		
Teflon on Teflon	0.04 (22)			0.04 (22, f)
Teflon on steel	0.04 (22)			0.04 (22, f)
Tungsten carbide on tungsten carbide	0.2 (22)	0.12 (22, a)		
Tungsten carbide on steel	0.5 (22)	0.08 (22, a)		
Tungsten carbide on copper	0.35 (23)			
Tungsten on carhide on iron	0.8 (23)			
Bonded carbide on copper	0.35 (23)			
Bonded carbide on iron	0.8 (23)			
Cadmium on mild steel			0.46 (3)	
Copper on mild steel	0.53 (8)		0.36 (3)	0.18 (17, a)
Nickel on nickel	1.10 (16)		0.53 (3)	0.12 (3, w)
Brass on mild steel	0.51 (8)		0.44 (6)	
Brass on cast iron			0.30 (6)	
Zinc on cast iron	0.85 (16)		0.21 (7)	
Magnesium on cast iron			0.25 (7)	
Copper on cast iron	1.05 (16)		0.29 (7)	
Tin on cast iron			0.32 (7)	
Lead on cast iron			0.43 (7)	
Aluminum on aluminum	1.05 (16)		1.4 (3)	
Glass on glass	0.94 (8)	0.01 (10, p)	0.40 (3)	0.09 (3, a)
		0.005 (10, q)		0.116 (3, v)
Carbon on glass			0.18 (3)	
Garnet on mild steel			0.39 (3)	
Glass on nickel	0.78 (8)		0.56 (3)	
Copper on glass	0.68 (8)		0.53 (3)	
Cast iron on cast iron	1.10 (16)		0.15 (9)	0.070 (9, d)
				0.064 (9, n)
Bronze on cast iron			0.22 (9)	0.077 (9, n)
Oak on oak (parallel to grain)	0.62 (9)		0.48 (9)	0.164 (9, r)
				0.067 (9, s)
Oak on oak (perpendicular)	0.54 (9)		0.32 (9)	0.072 (9, s)
Leather on oak (parallel)	0.61 (9)		0.52 (9)	
Cast iron on oak			0.49 (9)	0.075 (9, n)
Leather on cast iron			0.56 (9)	0.36 (9, t)
				0.13 (9, n)
Laminated plastic on steel			0.35 (12)	0.05 (12, t)
Fluted rubber bearing on steel				0.05 (13, t)

(1) Campbell, Trans. *ASME*, 1939; (2) Clarke, Lincoln, and Sterrett, *Proc. API*, 1935; (3) Bears and Bowden, *Phil. Trans. Roy. Soc.*, 1935; (4) Dokos, Trans, ASME, 1946; (5) Boyd ani Robertson, Trans. *ASME*, 1945; (6) Sacha, Zed. *J. angeu. Math, und Mech.*, 1924; (7) Honda and Yania la. *Jour. I of M.* 1925; (8) Tomlinson, *Phil, Mag.*, 1929; (9) Morin, *Acad. Roy. des Sciences*, 1838; (10) Claypoole, *Trans. ASME*, 1943; (11) Tabor, Jour. Applied Phys., 1945; (12) Eyssen, General Discussion on Lubrication, *ASME*, 1937; (13) Brazier and Holland-Bowyer, General Discussion on Lubrication, ASME, 1937; (14) Burwell, *Jour. SAE*, 1942; (15) Stanton, "Friction," Longinans; (16) Ernst and Merchant, Conference on Friction and Surface Finish, M.I.T., 1940; (17) Gongwer, Conference on Friction and Surface Finish, M.I.T., 1940; (18) Hardy and Bircumshaw, *Proc. Rey. Soc.*, 1925; (19) Hardy and Hardy, *Phil. Mag.*, 1919; M.I.T., 1940; (20) Bowden and Young, *Proc. Roy. Soc.*, 1951; (21) Hardy and Doubleday, *Proc. Roy. Soc.*, 1923; (22) Bowqden and Tabor, "The Friction and Lubrication of Solids," Oxford; (23) Shooter. *Research*, 4, 1951.

(a) Oleic acid; (b) Atlantic spindle oil (light mineral); (c) castor oil; (d) lard oil; (e) Atlantic spindle oil plus 2 percent oleic acid; (f) medium ineral oil; (g) medium mineral oil plus ½ percent oleic acid; (h) stearic acid; (i) grease (xinc oxide base); (j) graphite; (k) turbine oil plus 1 percent graphite; (l) turbine oil plus 1 percent stearic acid; (m) turbine oil (medium mineral); (n) olive oil; (p) palmitic acid; (q) ricinoleic acid; (r) dry soap; (s) lard; (t) water; (u rape oil; (v) 3-in-1 oil; (w) octyl alcohol; (x) triolein; (y) 1 percent lauric acid in paraffin oil.

One of the simplest structures is the *truss*. A truss consists of straight members connected at their end points only. All loads, including the weight of the members themselves, are considered to be supported at the joints. Due to its construction and the assumption of loading at joints only, the members of a truss support only loads of axial tension or axial compression. A *rigid truss* or a rigid structure will not collapse and can only deform if its members deform. A *simple truss* is one that can be constructed, starting with three members arranged in a triangle, by adding new members in pairs, first connecting one end of each together to form a new joint, and then connecting the other ends at separate existing joints of the truss.

A *frame* is a structure with at least one member that supports more than two forces. Members of a frame may support lateral as well as axial forces. Connections in a frame need not be located at the ends of the members. Frames, like trusses, are designed to support loads, and are usually motionless. A *machine* also has multiforce members. It is designed to modify and transmit forces and, though it may sometimes be stationary, it always includes parts that move during some phase of operation.

Not all structures can be fully analyzed by the methods of statics. If the number of discrete equilibrium equations is equal to the number of unknown loads, then the structure is said to be *statically determinate* and rigid. If there are more unknowns than equations, then the structure is *statically indeterminate*. If there are more equations than unknowns, then the structure is said to be *statically indeterminate and nonrigid*.

For further information on this subject, refer to References 1 and 2.

DYNAMICS

Dynamics is the study of the mechanics of rigid bodies in motion. It is usually subdivided into *kinematics*, the study of the motion of bodies without reference to the forces causing that motion or to the mass of bodies, and *kinetics*, the study of the relationship between the forces acting on a body, the mass and geometry of the body, and the resulting motion of the body.

Kinematics

Kinematics is based on one-dimensional differential equations of motion. Suppose a particle is moving along a straight line, and its distance from some reference point is S (see Figure 2-6a). Then its linear velocity and linear acceleration are defined by the differential equations given in the top half of Column 1, Table 2-5. The solutions

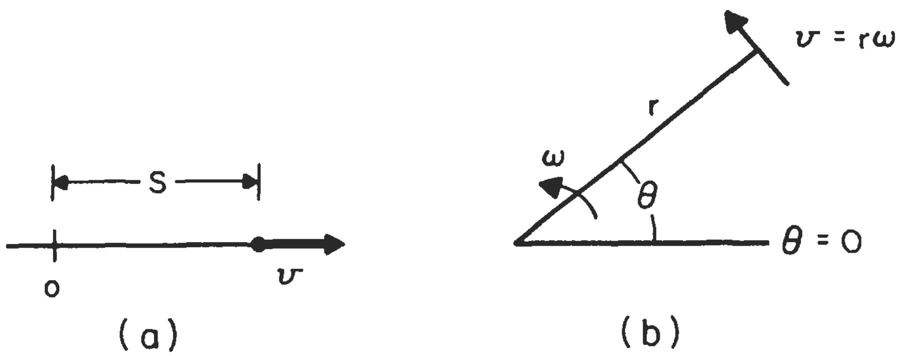


Figure 2-6. Diagrams of motion: (a) one-dimensional linear; (b) rotational.

Table 2-5
One-dimensional Differential Equations of Motion and Their Solutions

	Differential Equations	a/α = constant	$a = a(t);\ \alpha = \alpha(t)$	$a = a(v);\ \alpha = \alpha(\omega)$	$a = a(s);\ \alpha = \alpha(\theta)$
Linear	$v = \frac{ds}{dt}$	$v_f = v_i + at$	$s_f = s_i + \int_0^t v(t)dt$		$t = \int_{s_i}^{s} \frac{ds}{v(s)}$
	$a = \frac{dv}{dt}$	$s_f = s_i + v_i t + \frac{1}{2}at^2$	$v_f = v_i + \int_0^t a(t)dt$	$t = \int_{v_i}^{v_f} \frac{dv}{a(v)}$	
	$vdv = ads$	$v_f^2 = v_i^2 + 2a\Delta s$		$s_f = s_i + \int_{v_i}^{v_f} \frac{vdv}{a(v)}$	$v_f^2 = v_i^2 + 2\int_{s_i}^{s} a(s)ds$
Rotational	$\omega = \frac{d\theta}{dt}$	$\omega_f = \omega_i + \alpha t$	$\theta_f = \theta_i + \int_0^t w(t)\,dt$		$t = \int_{\theta_i}^{\theta} \frac{d\theta}{\omega(\theta)}$
	$\alpha = \frac{d\omega}{dt}$	$\theta_f = \theta_i + \omega_i t + \frac{1}{2}\alpha t^2$	$\omega_f = \omega_i + \int_0^t \alpha(t)\,dt$	$t = \int_{\omega_i}^{\omega_f} \frac{d\omega}{\alpha(\omega)}$	
	$\omega d\omega = \alpha d\theta$	$\omega_f^2 = \omega_i^2 + 2\alpha\Delta\theta$		$\theta_f = \theta_i + \int_{\omega_i}^{\omega_f} \frac{\omega d\omega}{\alpha(\omega)}$	$\omega_f^2 = \omega_i^2 + 2\int_{\theta_i}^{\theta} \alpha(\theta)\,d\theta$

to these equations are in Columns 2–5, for the cases of constant acceleration, acceleration as a function of time, acceleration as a function of velocity, and acceleration as a function of displacement s.

For rotational motion, as illustrated in Figure 2-6b, a completely analogous set of equations and solutions are given in the bottom half of Table 2-5. There ω is called the angular velocity and has units of radians/s, and α is called angular acceleration and has units of radians/s^2.

The equations of Table 2-5 are all scalar equations representing discrete components of motions along orthogonal axes. The axis along which the component ω or α acts is defined in the same fashion as for a couple. That is, the direction of ω is outwardly perpendicular to the plane of counterclockwise rotation (Figure 2-7).

The equations of Table 2-5 can be used to define orthogonal components of motion in space, and these components are then combined vectorally to give the complete motion of the particle or point in question.

The calculation and combination of the components of particle motion requires imposition of a coordinate system. Perhaps the most common is the Cartesian system illustrated in Figure 2-8. Defining unit vectors $\hat{i}$, $\hat{j}$, and $\hat{k}$ along the coordinate axes x, y, and z, the position of some point in space, P, can be defined by a position vector, $\mathbf{r}_p$:

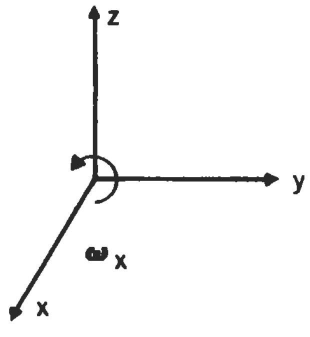


Figure 2-7. W_x is a vector of magnitude W_x acting along the x axis.

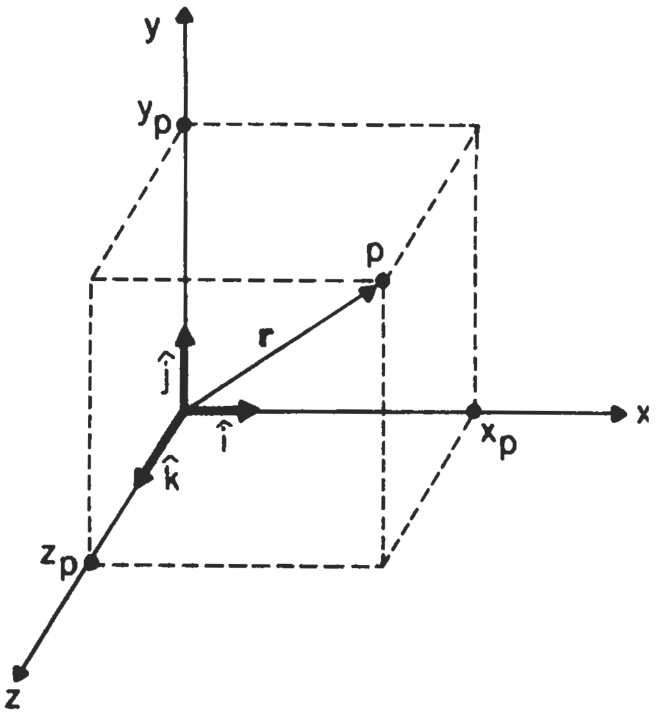


Figure 2-8. Equations of motion in a Cartesian coordinate system.

$$\mathbf{r}_p = x_p\hat{i} + y_p\hat{j} + z_p\hat{k} \tag{2-14}$$

In Equation 2-14, x, y, and z represent the coordinates of point P. The velocity of P is the vector sum of the component velocities:

$$v_x = \frac{dx_p}{dt}, \quad v_y = \frac{dy_p}{dt}, \quad \text{and} \quad v_z = \frac{dz_p}{dt}$$

$$v_p = v_x\hat{i} + v_y\hat{j} + v_z\hat{k} \tag{2-15}$$

Likewise, the acceleration of P is the vector sum of the components of the accelerations where

$$a_x = \frac{dv_x}{dt}, \quad a_y = \frac{dv_y}{dt}, \quad \text{and} \quad a_z = \frac{dv_z}{dt}$$

$$\mathbf{a}_p = a_x\hat{i} + a_y\hat{j} + a_z\hat{k} \tag{2-16}$$

For any vector, the magnitude is the square root of the sum of the squares of the components. Thus the magnitude of the velocity of point P would be

$$|v_p| = (v_x^2 + v_y^2 + v_z^2)^{0.5} \tag{2-17}$$

The angle between the total velocity (or any other vector) and any particular coordinate axis can be calculated from the scalar product of said vector and the unit vector along that axis. The scalar product is defined as

$$\mathbf{a} \bullet \mathbf{b} = a_x b_x + a_y b_y + a_z b_z = |a||b|\cos\theta$$

where θ is the angle between vectors **a** and **b**. Thus the angle between the velocity and the x axis is

$$\theta_x = \cos^{-1}\left(\frac{v \bullet \hat{i}}{|v|}\right)$$

$$= \cos^{-1}\left[-\frac{v_x}{(v_x^2 + v_y^2 + v_z^2)^{0.5}}\right] \tag{2-18}$$

If the magnitude and direction of a vector are known, its components are the products of the magnitude and the respective direction cosines. In the case of the velocity vector, for example, the components are

$$v_x = |v_p|\cos\theta_x$$

$$v_y = |v_p|\cos\theta_y \tag{2-19}$$

$$v_z = |v_p|\cos\theta_z$$

Example 2-4

A projectile is fired at an angle of 30° to the surface of the earth with an initial velocity of 1,000 ft/s (see Figure 2-9). What will be its velocity and the angle of its trajectory as a function of time?

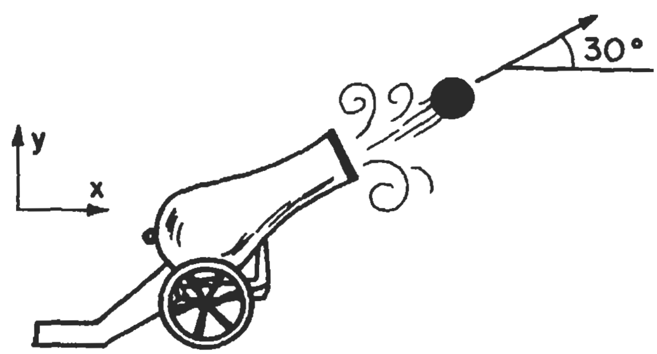


Figure 2-9. Diagram for Example 2-4.

x Component: The initial velocity in the x direction is

$$v_x = |v|\cos\theta_x = 1{,}000 \text{ ft/s} \cos 30°$$

$$v_x = 866 \text{ ft/s}$$

Assuming no air friction, this velocity is constant.

y Component: The initial velocity in the y direction is

$$v_{yo} = |v|\cos\theta_y = |v|\sin\theta_x = 1{,}000 \text{ ft/s} \sin 30°$$

$$v_{yo} = 500 \text{ ft/s}$$

In the y direction, the projectile has a constant acceleration of $-g = -32.2$ ft/sec^2. Thus by Equation 2-2, Column, 2, Table 2-5, its velocity as a function of time is

$$v_y(t) = v_{yo} + at = v_{yo} - gt$$

$$= 500 \text{ ft/s} - \frac{32.2 \text{ ft}}{s^2} t$$

and the total velocity vector is

$$v = 866\hat{i} + (500 - 32.2t)\hat{j}$$

The angle of the trajectory is found from Equation 2-18 as

$$\theta_x = \cos^{-1}\left\{\frac{866}{\left[866^2 + (500 - 32.2t)^2\right]^{0.5}}\right\}$$

It is often convenient to use some other coordinate system besides the Cartesian system. In the normal/tangential system (Figure 2-10), the point of reference is not fixed in space but is located on the particle and moves as the particle moves. There is no position vector and the velocity and acceleration vectors are written in terms of

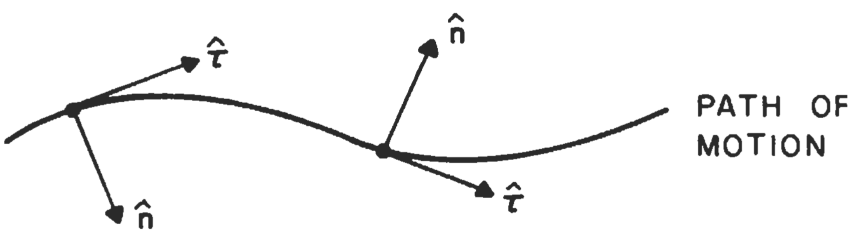


Figure 2-10. Normal and tangential unit vectors at different points on a path.

unit vectors $\hat{\tau}$, tangent to the path of motion, and $\hat{\eta}$, inwardly perpendicular to the path of motion.

The velocity is always tangent to the path of motion, and thus the velocity vector has only one component (Equation 2-20).

$$v = v\hat{\tau} \tag{2-20}$$

The acceleration vector has a component tangent to the path $a_t = d|v|/d_t$, which is the rate at which the magnitude of the velocity vector is changing, and a component perpendicular to the path $a_n = |v|^2/\rho$, which represents the rate at which the direction of motion is changing (Equation 2-21).

$$\mathbf{a} = a_t\hat{\tau} + a_n\tilde{\mathbf{n}} = \frac{d|v|}{dt}\hat{\tau} + \frac{|v|^2}{\rho}\tilde{\mathbf{n}} \tag{2-21}$$

In Equation 2-21 ρ is the local radius of curvature of the path. The normal component of acceleration can also be expressed as $a_n = \rho|\omega|^2$ or $a_n = |v||\omega|$ where ω is the angular velocity of the particle.

Example 2-5

A car is increasing in speed at a rate of 10 ft/s² when it enters a curve with a radius of 50 ft at a speed of 30 ft/s. What is the magnitude of its total acceleration?

$$a_t = \frac{d|v|}{dt} = 10 \text{ ft/s}^2$$

$$a_n = \frac{v^2}{\rho} = \frac{(30 \text{ ft/s})^2}{50 \text{ ft}} = 18 \text{ ft/s}^2$$

$$\mathbf{a} = 10\tilde{\tau} + 18\tilde{n}$$

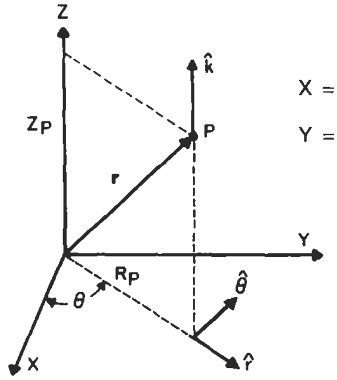
$$|\mathbf{a}| = (10^2 + 18^2)^{0.5} = 20.6 \text{ ft/s}^2$$

In addition to the Cartesian and normal/tangential coordinate systems, the cylindrical (Figure 2-11) and spherical (Figure 2-12) coordinate systems are often used.

When dealing with the motions of rigid bodies or systems of rigid bodies, it is sometimes quite difficult to directly write out the equations of motion of the point in question as was done in Examples 2-6 and 2-7. It is sometimes more practical to analyze such a problem by *relative motion*. That is, first find the motion with respect to a nonaccelerating reference frame of some point on the body, typically the center of mass or axis of rotation, and vectorally add to this the motion of the point in question with respect to the reference point.

Example 2-6

Consider an arm 2 ft in length rotating in the counterclockwise direction about a fixed axis at point A at a rate of 2 rpm (see Figure 2-13a). Attached to the arm at



$X = R_P \cos\theta$

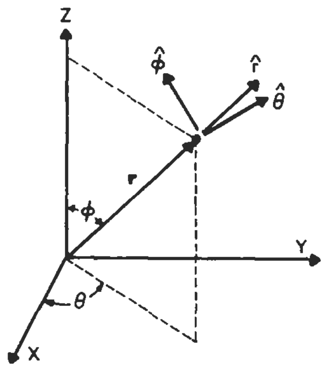
$Y = R_P \sin\theta$

$$\mathbf{r} = R_P\hat{r} + Z_P\hat{k}$$

$$\upsilon = \left(\frac{dR_P}{dt}\hat{r}\right) + \left(R_P\frac{d\theta}{dt}\right)\hat{\theta} + \left(\frac{dz}{dt}\right)\hat{k}$$

$$\mathbf{a} = \left[\frac{d^2R_P}{dt^2} - R_P\left(\frac{d\theta}{dt}\right)^2\right]\hat{r} + \left[R_P\frac{d^2\theta}{dt^2} + 2\frac{dR_P}{dt}\frac{d\theta}{dt}\right]\hat{\theta} + \left[\frac{d^2z}{dt^2}\right]\hat{k}$$

Figure 2-11. Equations of motion in a cylindrical coordinate system.



$X = |\mathbf{r}|\sin\phi\cos\theta$

$Y = |\mathbf{r}|\sin\phi\sin\theta$

$Z = |\mathbf{r}|\cos\phi$

$$\mathbf{r} = |\mathbf{r}|\hat{r}$$

$$\upsilon = \left(\frac{d|\mathbf{r}|}{dt}\right)\hat{r} + \left(|\mathbf{r}|\frac{d\theta}{dt}\cos\phi\right)\hat{\theta} + \left(|\mathbf{r}|\frac{d\phi}{dt}\right)\hat{\phi}$$

$$\mathbf{a} = \left[\frac{d^2|\mathbf{r}|}{dt^2} - |\mathbf{r}|\left(\frac{d\phi}{dt}\right)^2 - |\mathbf{r}|\left(\frac{d\theta}{dt}\right)^2\cos^2\phi\right]\hat{r} + \left[\frac{\cos\phi}{|\mathbf{r}|}\frac{d}{dt}\left(|\mathbf{r}|^2\frac{d\theta}{dt}\right) - 2|\mathbf{r}|\frac{d\theta}{dt}\frac{d\phi}{dt}\sin\phi\right]\hat{\theta}$$

$$+\left[\frac{1}{|\mathbf{r}|}\frac{d}{dt}\left(|\mathbf{r}|^2\frac{d\theta}{dt}\right) + |\mathbf{r}|\left(\frac{d\theta}{dt}\right)^2\sin\phi\cos\phi\right]\hat{\phi}$$

Figure 2-12. Equations of motion in a spherical coordinate system.

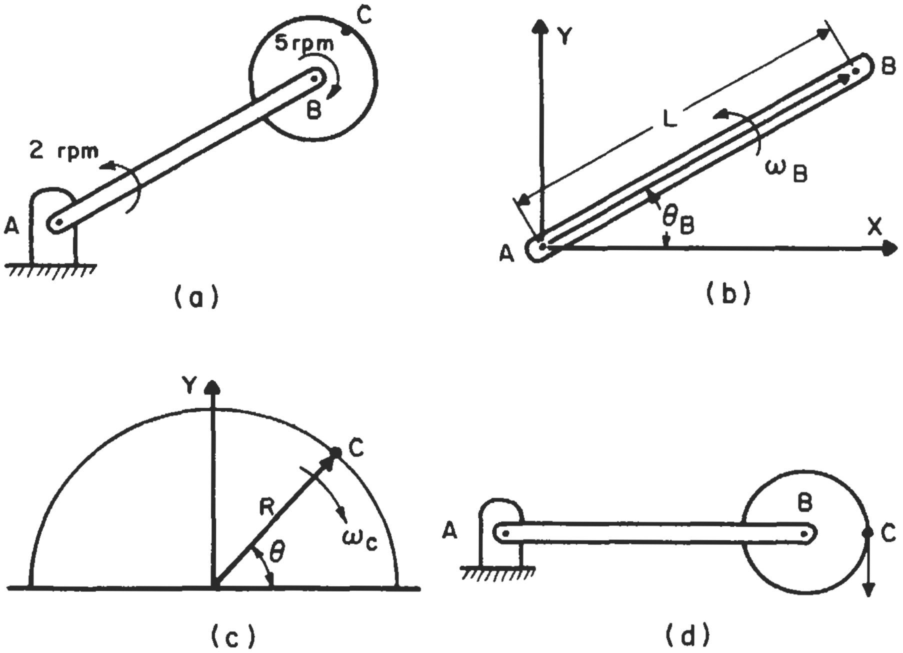


Figure 2-13. Diagram for Example 2-6.

point B is a disk with a radius of 1 ft, which is rotating in the clockwise direction about point B at a rate of 5 rpm. What is the velocity and acceleration of some arbitrary point C on the rim of the disk?

By relative motion the position vector of C is the sum of the position vector of B and the position vector of C with respect to B

$$\mathbf{r}_C = \mathbf{r}_B + \mathbf{r}_{C/B} \tag{2-22}$$

and likewise with the velocity and acceleration, which is

$$v_C = v_B + v_{C/B} \tag{2-23}$$

$$\mathbf{a}_C = \mathbf{a}_B + \mathbf{a}_{C/B} \tag{2-24}$$

Analyzing the motion of point B (see Figure 2-13b),

$$\mathbf{r}_B = L\cos\theta_B\hat{i} + L\sin\theta_B\hat{j}$$

Assuming rotation starts at $\theta_B = 0$,

$$\theta_B = \omega_B t$$

$$v_B = \frac{d}{dt}(\mathbf{R}_B) = -L\omega_B \sin\omega_B t\hat{i} + L\omega_B \cos\omega_B t\hat{j}$$

$$\mathbf{a}_B = \frac{d}{dt}(v_B) = -L\omega_B^2 \cos\omega_B t\hat{i} - L\omega_B^2 \sin\omega_B t\hat{j}$$

Assuming point B is fixed and analyzing the motion of point C with respect to B (see Figure 2-13c),

$$\mathbf{r}_{C/B} = R\cos\theta_C\hat{i} + R\sin\theta_C\hat{j}$$

$$v_{C/B} = -R\omega_C \sin\omega_C t\hat{i} + R\omega_C \cos\omega_C t\hat{j}$$

$$\mathbf{a}_{C/B} = -R\omega_C^2 \cos\omega_C t\hat{i} + R\omega_C^2 \sin\omega_C t\hat{j}$$

Thus the velocity of point C is

$$v_C = v_B + v_{C/B} = -(L\omega_B \sin\omega_B t + R\omega_C \sin\omega_C t)\hat{i}$$

$$+ (L\omega_B \cos\omega_B t + R\omega_C \cos\omega_C t)\hat{j}$$

$$\omega_B = \frac{2\text{ rev}}{\text{min}}\left|\frac{1\text{ min}}{60\text{ s}}\right|\frac{2\pi\text{ r}}{\text{revolution}} = 0.209\text{ rps}$$

$$L\omega_B = \frac{2\text{ ft}}{}\left|\frac{0.209\text{ r}}{\text{s}}\right. = 0.418\text{ ft/s}$$

$\omega_C = -0.524$ rps

$R\omega_C = -0.524$ ft/s

$$v_C = -[0.418\sin(0.209t) - 0.524\sin(-0.524t)]\hat{i}$$

$$+ [0.418\cos(0.209t) - 0.524\cos(-0.524t)]\hat{j}$$

At t = 0, for instance, the velocity is (see Figure 2-13d)

$$v_C = -0.106\hat{j}\text{ ft/s}$$

Likewise, the general expression for the acceleration of point C is

$$\mathbf{a}_C = \mathbf{a}_B + \mathbf{a}_{C/B} = -(L\omega_B^2 \cos\omega_B t + R\omega_C^2 \cos\omega_C t)\hat{i}$$

$$- (L\omega_B^2 \sin\omega_C t + R\omega_C^2 \sin\omega_C t)\hat{j}$$

At t = 0 this reduces to

$$\mathbf{a}_C = -(L\omega_B^2 + R\omega_C^2)\hat{i} = -[2\times 0.209^2 + 1\times(-0.524)^2]\hat{i}$$

$\mathbf{a}_C = -0.362\hat{i}\ \text{ft/s}^2$

When looking for the velocities of points on a rigid body, the method of *instantaneous centers* can often be used. If the velocity of two points on the body are known, those points and all other points on the body can be considered to be rotating with the same angular velocity about some motionless central point. This central point is called the instantaneous center of zero velocity. The instantaneous center generally moves through space as a function of time and has acceleration. It does *not* represent a point about which acceleration may be determined.

Example 2-7

Link AB of length 2 ft (see Figure 2-14a) is sliding down a wall with point A moving downward at 4 ft/s when θ is 30°. What is the angular velocity of the link and linear velocity of point B?

Because v_A (Figure 2-14b) is parallel to the vertical wall, it is rotating about a point on a line through A perpendicular to the wall. Likewise B is rotating about a

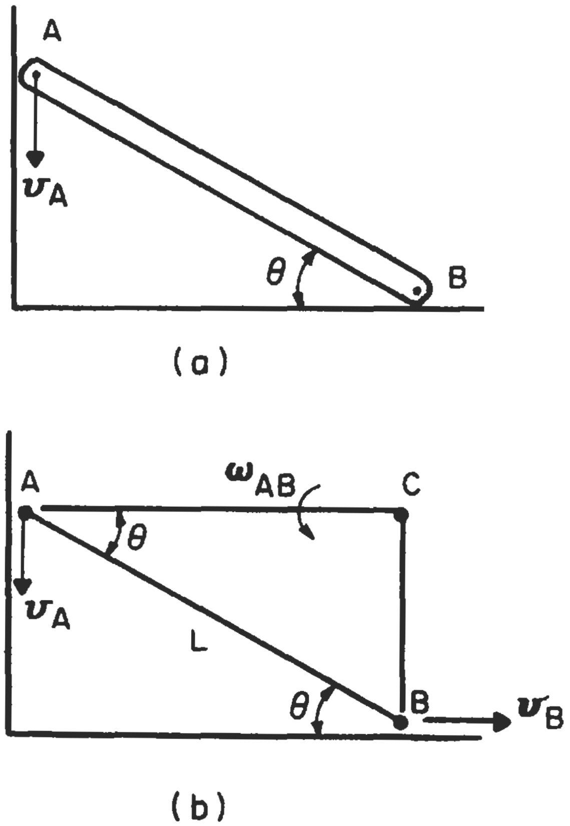


Figure 2-14. Diagram for Example 2-7.

point on a line through B perpendicular to v_B. These two lines intersect at C, the instantaneous center.

$$AC = L \cos \theta = Z \times 0.866 = 1.73 \text{ ft}$$

$$|v_A| = AC|\omega_{AB}|$$

$$\omega_{AB} = \frac{|v_A|}{AC} = \frac{4 \text{ ft}}{s} \bigg| \frac{}{1.73 \text{ ft}}$$

where $\omega_{AB} = 2.31$ rad/s = the angular velocity of the link and of any line on the link

$$|v_B| = BC|\omega_{AB}| = L \sin \theta |\omega_{AB}| = \frac{2 \text{ ft}}{} \bigg| \frac{0.5}{} \bigg| \frac{2.31 \text{ rev}}{s}$$

$$|v_B| = 2.31 \text{ ft/s}$$

Kinetics

In *kinetics,* Newton's second law, the principles of kinematics, conservation of momentum, and the laws of conservation of energy and mass are used to develop relationships between the forces acting on a body or system of bodies and the resulting motion.

Applications of Newton's Second Law. Problems involving no unbalanced couples can often be solved with the second law and the principles of kinematics. As in statics, it is appropriate to start with a free-body diagram showing all forces, decompose the forces into their components along a convenient set of orthogonal coordinate axes, and then solve a set of algebraic equations in each coordinate direction. If the accelerations are known, the solution will be for an unknown force or forces, and if the forces are known the solution will be for an unknown acceleration or accelerations.

Example 2-8

In Figure 2-15a, a 10-lb block slides down a ramp inclined at an angle of 30°. If the coefficient of kinetic friction between the block and the ramp is 0.1, what will be the acceleration of the block?

As shown in the free-body diagram of Figure 2-15b, all the motion of the block is parallel to the surface of the ramp; thus there is a static force balance in the y direction.

$$\sum Fy = N - W \cos 30° = 0$$

$$N = W \cos 30°$$

$$\rightarrow F_f = \mu N = \mu W \cos 30°$$

Also, by Newton's second law, the force in the x direction produces an acceleration a_x:

$$\sum F_x = W \sin 30° - F_f = ma_x = \frac{W}{g} a_x$$

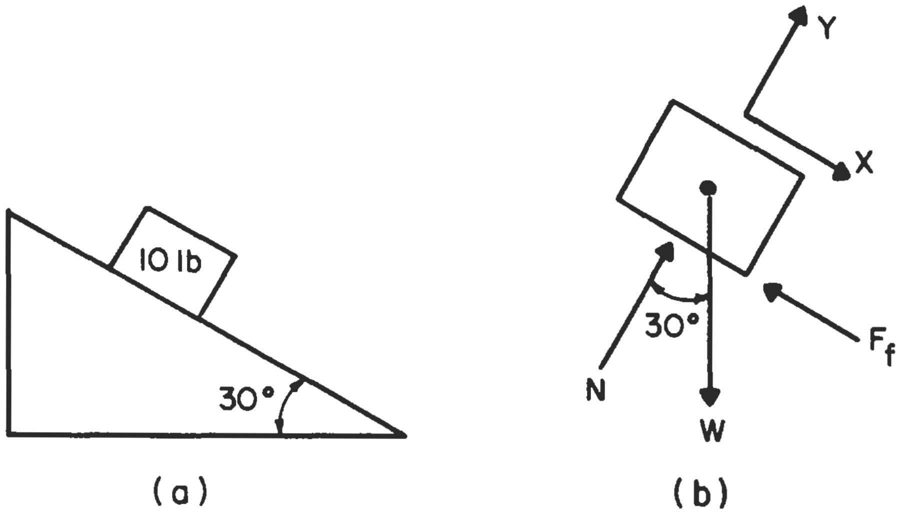


Figure 2-15. Diagram for Example 2-8.

$a_x = g(\sin 30° - \mu \cos 30°)$

$a_x = 13.31 \text{ ft/s}^2$

Where unbalanced couples are involved, a rotational analog to Newton's second law can be applied:

$$\sum \overline{\mathbf{M}} = \bar{I}\alpha \quad (2\text{-}25)$$

where $\sum \overline{\mathbf{M}}$ is the sum of all moments acting about the center of mass in the plane of rotation, $\bar{I}$ is the mass moment of inertia about the center of mass, and α is the angular acceleration of the body. The mass moment of inertia is defined by

$$I = \int r^2 dm = mk^2 \quad (2\text{-}26)$$

where r is the perpendicular distance from the axis of rotation to the differential element of mass, dm. I is sometimes expressed in terms of k, the radius of gyration, and m, the mass of the body. If the axis of rotation passes through the center of mass, then the mass moment of inertia is designated as $\bar{I}$. Mass moments of inertia of common shapes are compiled in Tables 2-6 and 2-7.

It is often convenient to sum the moments about some arbitrary point 0, other than the mass center. In this case, Equation 2-25 becomes

$$\sum \mathbf{M}_0 = \bar{I}\alpha + m\bar{a}d \quad (2\text{-}27)$$

where m is the mass of the body, $\bar{a}$ is the linear acceleration of the mass center, and d is the perpendicular distance between the vector $\bar{a}$ and point 0.

Table 2-6
Moments of Inertia of Common Areas and Volumes [3]

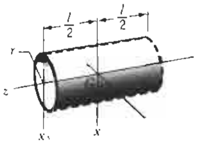
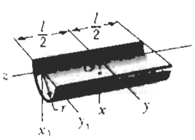
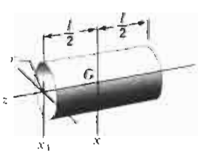
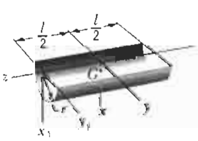
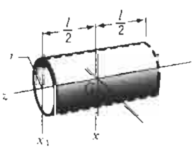
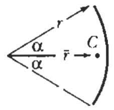
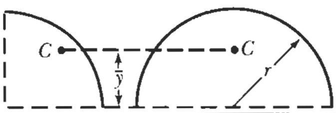
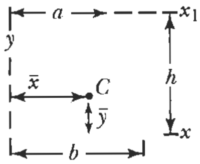
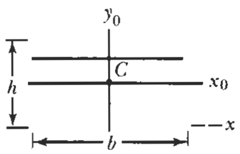
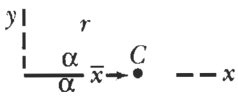
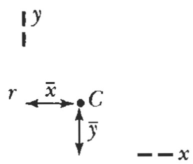
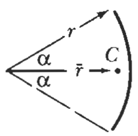
BODY	MASS CENTER	MOMENTS OF INERTIA
 Circular Cylindrical Shell	—	$I_{xx} = \frac{1}{2}mr^2 + \frac{1}{12}ml^2$ $I_{x_1x_1} = \frac{1}{2}mr^2 + \frac{1}{3}ml^2$ $I_{xx} = mr^2$
 Half Cylindrical Shell	$\bar{x} = \frac{2r}{\pi}$	$I_{xx} = I_{yy}$ $= \frac{1}{2}mr^2 + \frac{1}{12}ml^2$ $I_{x_1x_1} = I_{y_1y_1}$ $= \frac{1}{2}mr^2 + \frac{1}{3}ml^2$ $I_{xx} = mr^2$ $\bar{I}_{xx} = \left(1 - \frac{4}{\pi^2}\right)mr^2$
 Circular Cylinder	—	$I_{xx} = \frac{1}{4}mr^2 + \frac{1}{12}ml^2$ $I_{x_1x_1} = \frac{1}{4}mr^2 + \frac{1}{3}ml^2$ $I_{xx} = \frac{1}{2}mr^2$
 Semicylinder	$\bar{x} = \frac{4r}{3\pi}$	$I_{xx} = I_{yy}$ $= \frac{1}{4}mr^2 + \frac{1}{12}ml^2$ $I_{x_1x_1} = I_{v_1v_1}$ $= \frac{1}{4}mr^2 + \frac{1}{3}ml^2$ $I_{xx} = \frac{1}{2}mr^2$ $\bar{I}_{xx} = \left(\frac{1}{2} - \frac{16}{9\pi^2}\right)mr^2$
 Rectangular Parallelepiped	—	$I_{xx} = \frac{1}{12}m(a^2 + l^2)$ $I_{yy} = \frac{1}{12}m(b^2 + l^2)$ $I_{xx} = \frac{1}{12}m(a^2 + b^2)$ $I_{y_1y_1} = \frac{1}{12}mb^2 + \frac{1}{3}ml^2$

Table 2-7

FIGURE	CENTROID	AREA MOMENTS OF INERTIA
Arc Segment 	$\bar{r} = \frac{r \sin \alpha}{\alpha}$	—
Quarter and Semicircular Arcs 	$\bar{y} = \frac{2r}{\pi}$	—
Triangular Area 	$\bar{x} = \frac{a+b}{3}$ $\bar{y} = \frac{h}{3}$	$I_x = \frac{bh^3}{12}$ $\bar{I}_x = \frac{bh^3}{36}$ $I_{x_1} = \frac{bh^3}{4}$
Rectangular Area 	—	$I_x = \frac{bh^3}{3}$ $\bar{I}_x = \frac{bh^3}{12}$ $\bar{J} = \frac{bh}{12}(b^2 + h^2)$
Area of Circular Sector 	$\bar{x} = \frac{2}{3}\frac{r \sin \alpha}{\alpha}$	$I_x = \frac{r^4}{4}(\alpha - \frac{1}{2}\sin 2\alpha)$ $I_y = \frac{r^4}{4}(\alpha + \frac{1}{2}\sin 2\alpha)$ $J = \frac{1}{2}r^4\alpha$
Quarter Circular Area 	$\bar{x} = \bar{y} = \frac{4r}{3\pi}$	$I_x = I_y = \frac{\pi r^4}{16}$ $\bar{I}_x = \bar{I}_y = \left(\frac{\pi}{16} - \frac{4}{9\pi}\right)r^4$ $J = \frac{\pi r^4}{8}$
Area of Elliptical Quadrant Area $A = \frac{\pi ab}{4}$ 	$\bar{x} = \frac{4a}{3\pi}$ $\bar{y} = \frac{4b}{3\pi}$	$I_x = \frac{\pi ab^3}{16}, \bar{I}_x = \left(\frac{\pi}{16} - \frac{4}{9\pi}\right)ab^3$ $I_y = \frac{\pi a^3 b}{16}, \bar{I}_y = \left(\frac{\pi}{16} - \frac{4}{9\pi}\right)a^3 b$ $J = \frac{\pi ab}{16}(a^2 + b^2)$

SOURCE: Meriam pp. 498–499.

If 0 is a fixed axis or the instantaneous center of zero velocity, then Equation 2-27 reduces to

$$\sum \mathbf{M}_0 = I_0\alpha \tag{2-28}$$

where I_0 is the mass moment of inertia about point 0. I_0 may be found from Equation 2-26, or it may be calculated from the parallel axis theorem

$$I_0 = \bar{I} + m\bar{r}^2 \tag{2-29}$$

where $\bar{r}$ is the distance from 0 to the center of mass. The parallel axis theorem may be used to find I_0 regardless of whether I_0 is a fixed axis of instantaneous center of zero velocity.

Example 2-9

In Figure 2-16 a 10 lb cylinder with a 3-in. radius rolls down a 30° incline. What is its angular acceleration and the linear acceleration of its center of mass? In the free-body diagram of Figure 2-16, the point of contact between the wheel and the ramp is the instantaneous center of zero velocity. Thus,

$$\sum M_0 = rw \sin 30° = I_0\alpha$$

$$\alpha = \frac{rw \sin 30°}{I_0}$$

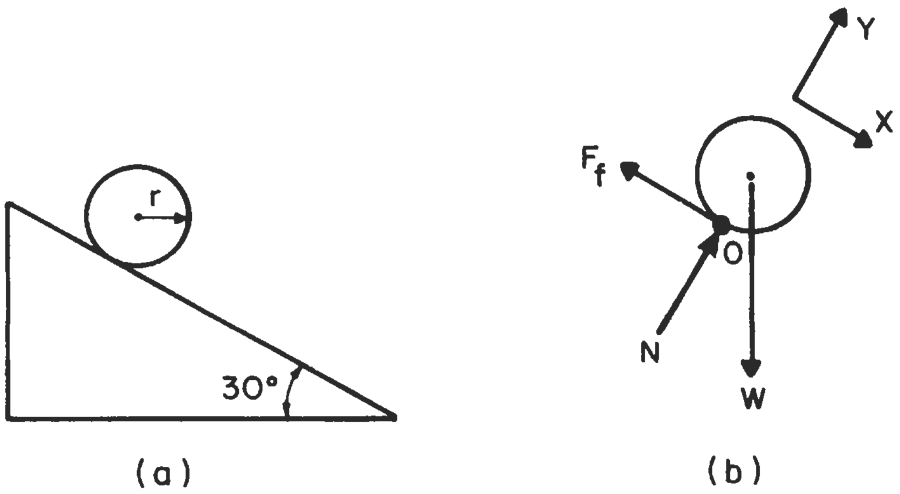


Figure 2-16. Diagram for Example 2-9.

From Tables 2-6 and 2-7 and the parallel axis theorem,

$$I_0 = \frac{W}{2g}r^2 + \frac{W}{g}r^2 = \frac{3}{2}\frac{W}{g}r^2$$

$$\alpha = \frac{2g \sin 30^\circ}{3r} = 3.57\,\text{rps}^2$$

$$a_x = r\alpha = 10.73\ \text{ft/s}^2$$

Conservation of Momentum. If the mass of a body or system of bodies remains constant, then Newton's second law can be interpreted as a balance between force and the time rate of change of momentum, *momentum* being a vector quantity defined as the product of the velocity of a body and its mass.

$$\mathbf{F} = m\mathbf{a} = \frac{d}{dt}m\mathbf{v} = \frac{d}{dt}\mathbf{G} \tag{2-30}$$

Integrating Equation 2-30 with respect to time yields the impulse/momentum equation

$$\int \mathbf{F}dt = \Delta\mathbf{G} \tag{2-31}$$

where **F**dt is called the impulse, and $\Delta\mathbf{G}$ is the change in momentum. Equation 2-31 can be applied explicitly and is particularly useful when the force is known as a function of time.

In collisions between two bodies the contact force and the duration of contact are usually unknown. However, the duration of contact is the same for both bodies, and the force on the first body is the negative of the force on the second body. Thus the net change in momentum is zero. This is called the principle of *conservation of momentum.*

If a collision is purely plastic, then the two colliding bodies will adhere to each other and move on as a single body. Knowing the initial velocities and masses thus allows calculation of the final velocity.

$$m_1v_1 + m_2v_2 = (m_1 + m_2)v \tag{2-32}$$

If the collision is purely elastic or elasto-plastic, then the two bodies will depart the collision with different velocities.

$$m_1v_{11} + m_2v_{21} = m_1v_{12} + m_2v_{22} \tag{2-33}$$

In this case, an additional equation is required before the final velocities may be found. Thus, the coefficient of restitution e is defined as the ratio of the velocity of separation to the velocity of approach:

$$e = \frac{v_{22x} - v_{12x}}{v_{11x} - v_{21x}} = \frac{v_{22y} - v_{12y}}{v_{11y} - v_{21y}} \tag{2-34}$$

Note that e is defined in terms of the components of the velocities, not the vector velocities, whereas the momentum balance is defined in terms of the vector velocities. To solve Equations 2-32 and 2-33 when all the velocities are not colinear, one writes the momentum balances along the principal axes and solves the resulting equations simultaneously.

For purely elastic impacts, e = 1, and for purely plastic impacts, e = 0. For elastoplastic impacts, e lies between zero and one and is a function of both the material properties and the velocity of impact.

Example 2-10

Sphere 1 weighs 1 lb and is traveling at 2 ft/s in the positive x direction when it strikes sphere 2, weighing 5 lb and traveling in the negative x direction at 1 ft/s. What will be the final velocity of the system if the collision is (a) plastic, or (b) Elastoplastic with e = 0.5?

(a) By Equation 2-32

$$v = \frac{m_1 v_{11} - m_2 v_{21}}{m_1 + m_2} = \frac{2-5}{6} = -0.5\ \text{ft/s}$$

(b) By Equation 2-34

$$e = \frac{v_{22} - v_{12}}{v_{11} - v_{21}}$$

$$v_{22} = v_{12} + 0.5[2-(-1)] = v_{12} + 1.5$$

By Equation 2-33

$$m_1 v_{11} + m_2 v_{21} = m_1 v_{12} + m_2 v_{22} = m_1 v_{12} + m_2(v_{12} + 1.5)$$

$$= (m_1 + m_2)v_{12} + 1.5 m_2$$

$$v_{12} = \frac{m_1 v_1 + m_2(v_2 - 1.5)}{m_1 + m_2} = \frac{1 \times 2 + 5(-1 - 1.5)}{1+5} = -1.75\ \text{ft/s}$$

$$v_{22} = -0.25\ \text{ft/s}$$

The foregoing discussion of impulse and momentum applies only when no change in rotational motion is involved. There is an analogous set of equations for angular impulse and impulse momentum. The angular momentum about an axis through the center of mass is defined as

$$\overline{\mathbf{H}} = \bar{I}\omega \tag{2-35}$$

and the angular momentum about any arbitrary point 0 is defined as

$$\mathbf{H}_o = \bar{I}\omega + m\bar{v}d \tag{2-36}$$

where $\bar{v}$ is the velocity of the center of mass and d is the perpendicular distance between the vector $\bar{v}$ and the point 0. And if 0 is a fixed axis or instantaneous center of zero velocity, then

$$H_0 = I_0\omega \tag{2-37}$$

Likewise, the angular impulse is defined as

$$\int \mathbf{M}_0 dt = \Delta \mathbf{H}_0 \tag{2-38}$$

In collisions, angular momentum, like linear momentum, is conserved.

Conservation of Energy

In a rigid-body system, energy is conserved in the sense that the net change in mechanical energy must be equal to the net work done on the system.

$$U = \Delta T + \Delta Vg + \Delta Ve \tag{2-39}$$

U is the net work done on the system and is defined as the sum of the work done by external forces and external moments.

$$U = \int \mathbf{F} \bullet ds + \int \mathbf{M} \bullet d\theta \tag{2-40}$$

The work of the force **F** is positive if it acts in the direction of the displacement **ds**, and the work of the moment **M** is positive if it acts in the direction of rotation dθ.

ΔT is the change in kinetic energy, made up of a change in linear kinetic energy and rotational kinetic energy.

$$\Delta T = \frac{1}{2} m(\bar{v}_f^2 - \bar{v}_i^2) + \frac{1}{2} \bar{I}(\omega_f^2 - \omega_i^2) \tag{2-41}$$

If the body in question has a fixed axis or an instantaneous center of zero velocity, then Equation 2-41 can be simplified to

$$\Delta T = \frac{1}{2} I_0(\omega_f^2 - \omega_i^2) \tag{2-42}$$

ΔV_g is the net change in gravitational potential energy. This term is path independent and depends only on the initial and final heights, h_i and h_f, above some arbitrary reference height with respect to the surface of the earth.

$$\Delta V_g = mg(h_f - h_i) = W(h_f - h_i) \tag{2-43}$$

ΔV_e is the net change in elastic energy stored in a massless spring, due to extension or compression (no spring is massless, but this assumption is reasonably accurate for most engineering calculations).

$$\Delta V_e = \frac{1}{2} k(X_f^2 - X_i^2) \tag{2-44}$$

The constant k, called the spring constant, represents the ratio of the force exerted by the spring to X, its net compression or extension from the rest length.

Example 2-11

A 1-lb sphere is dropped from a height of 20 ft to strike a 2-ft long relaxed vertical spring with a constant of 100 lb/ft (see Figure 2-17). What will be the velocity of the sphere at a height of 2 ft when it strikes the spring? What will be the maximum compression of the spring?

The sphere and the spring may be considered as a system in which no outside forces or moments are acting. Thus the work term in Equation 2-39 is zero. Before the collision with the spring, $\Delta V_e = 0$ also, and Equation 2-39 reduces to

$$\Delta T + \Delta V_g = 0$$

$$\Delta T = \frac{1}{2}\frac{W}{g}(v_f^2 - v_i^2) = -\frac{1}{2}\frac{W}{g}v_f^2$$

$$\Delta V_g = W(h_f - h_i)$$

which can be solved for the impact velocity.

$$v_f = \left[2g(h_i - h_f)\right]^{0.5} = \left[2 \times 32.2(20 - 2)\right]^{0.5}$$

$$v_f = 34 \text{ ft/s}$$

At full compression the velocity of the sphere is zero. Thus Equation 2-39 reduces to

$$\Delta V_e + \Delta V_g = 0$$

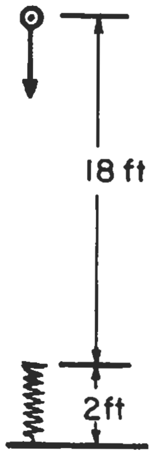


Figure 2-17. Diagram for Example 2-11.

$$\Delta V_c = \frac{1}{2}k(X_f^2 - X_i^2) = \frac{1}{2}kX_f^2$$

$$\Delta V_g = W(h_f - h_i)$$

$$h_f = L - X_f$$

(where L is the relaxed height of the spring) which can be solved for X_f, the maximum compression of the spring.

$$\frac{1}{2}kX_f^2 + W(L - X_f - h_i) = 0$$

$$X_f^2 \frac{2W}{k} X_f + \frac{2W}{k}(L - h_i) = 0$$

$$X_f = \frac{\frac{2W}{k} \pm \left[\left(\frac{2W}{k}\right)^2 - 4\left(\frac{2W}{k}(L - h_i)\right)\right]^{0.5}}{2}$$

$$= \frac{0.02 \pm \left[0.02^2 - 4\left[0.02(2 - 20)\right]\right]^{0.5}}{2}$$

$$X_f = 0.61 \text{ ft}$$

The negative root is ignored because it represents an extension of the spring rather than a compression.

For further information, refer to References 1–5.

FLUID MECHANICS

In fluid mechanics the principles of conservation of mass, conservation of momentum, the first and second laws of thermodynamics, and empirically developed correlations are used to predict the behavior of gases and liquids at rest or in motion. The field is generally divided into fluid statics and fluid dynamics and further subdivided on the basis of compressibility. Liquids can usually be considered as incompressible, while gases are usually assumed to be compressible.

Fluid Statics

Pressure is the force per unit area exerted by or on a fluid. In a static fluid the pressure increases with depth, but according to *Pascal's* principle it is the same in all directions at any given depth. Pressure may be specified as either *absolute,* or *gauge,* the relationship between the two being:

$$P_g = P_a - P_{atm} \tag{2-45}$$

where P_g is gauge pressure, P_a is absolute pressure, and P_{atm} is the *atmospheric pressure.* Fluid mechanics calculations are generally done in absolute pressure, and hereafter P will represent absolute pressure.

The governing equation for the pressure within a fluid at any depth h is

$$dP = \rho g dh \tag{2-46}$$

where ρ is the fluid density in mass per unit volume, and g is the acceleration due to gravity. In engineering calculations it is often convenient to replace the quantity ρg with γ, the *specific weight*, which is a measure of the weight of the fluid per unit volume.

If γ can be considered to be constant, the fluid is said to be incompressible and Equation 2-46 can be solved to yield

$$P = P_0 + \gamma(h - h_0) \tag{2-47}$$

where h_0 is some reference depth, h is depth increasing downward, and P_0 is the pressure at h_0. In a gas the specific weight of the fluid is a function of pressure and temperature. The concept of an *ideal* or *perfect gas* as one in which the molecules occupy no volume and the only intermolecular forces are due to intermolecular collisions leads to the *ideal gas law:*

$$\gamma = \frac{PS}{RT} \tag{2-48}$$

where P is the absolute pressure in pounds per square foot, T is the temperature in degrees Rankine, S is the specific gravity (the ratio of the density of the gas in question to the density of air at standard conditions), and R is Boltzman's constant (53.3 ft-lb/lb-°R). Under the assumption of an ideal gas at constant temperature, Equation 2-46 can be solved to yield

$$P = P_0 \exp\left[\frac{(h - h_0)S}{RT}\right] \tag{2-49}$$

If the gas behavior deviates markedly from ideal, the real gas law can be written as

$$\gamma = \frac{Ps}{ZRT} \tag{2-50}$$

where Z is an empirical compressibility factor that accounts for nonideal behavior (See Volume 2, Chapter 5).

Substituting the real gas law into Equation 2-47 yields

$$\frac{ZT}{P} dP = \frac{S}{R} dh \tag{2-51}$$

Equation 2-51 can be integrated under the assumption that Z and T are constant to yield Equation 2-52, or, if extreme accuracy is required, it is necessary to account for variations in Z and T and a numerical integration may be required.

$$P = P_0 \exp\left[\frac{(h - h_0)s}{ZRT}\right] \tag{2-52}$$

Example 2-12

Consider a 1,000-ft-deep hole. What will be the absolute pressure at the bottom if (a) it is filled with pure water or (b) it is filled with air at a constant temperature of 85°F?

(a)

$P = P_0 + \gamma(h-h_0)$

$h_0 = 0$

$h = 1,000$ ft

$$P_0 = \frac{14.7\ \text{lb}}{\text{in.}^2}\bigg|\frac{144\ \text{in.}^2}{\text{ft}^2} = 2,116.8\ \text{lb/ft}^2$$

$\gamma = 62.4\ \text{lb/ft}^3$

$P = 2,116.8 + 62.4(1,000 - 0) = 64,516.8\ \text{lb/ft}^2$

$P = 448$ psi

(b)

$$P = P_0 \exp\left[\frac{(h - h_0)s}{ZRT}\right]$$

$s = 1$

Assume

$Z = 1$

$T = 85 + 460 = 545°R$

$$P = 2,116.8 \exp\left[\frac{(1,000 - 0)1}{1(53.3)(545)}\right] = 2,190.94\ \text{lb/ft}^2$$

$P = 15.21$ psi

In a case where $Z \neq 1$, it is practical to assume $Z = 1$, perform Calculation (b), and then, based on the resultant, estimate for $P_{avg} = (P + P_0)/2$, find the value of Z, and repeat the calculation. Three iterations are generally sufficient. If T varies, it is usually sufficiently accurate to use an estimate of T_{avg} such as $T_{avg} = (T + T_0)/2$.

Fluid Dynamics

When fluids are in motion, the *pressure losses* may be determined through the principle of conservation of energy. For slightly compressible fluids this leads to

Bernoulli's equation (Equation 2-53), which accounts for static and *dynamic pressure losses* (due to changes in velocity), but does not account for frictional pressure losses, energy losses due to heat transfer, or work done in an engine.

$$\frac{P_1}{\gamma_1} + \frac{v_1^2}{2g} + h_1 = \frac{P_2}{\gamma_2} + \frac{v_2^2}{2g} + h_2 \tag{2-53}$$

where $v_1, v_2 \equiv$ velocity at points 1 and 2
$g \equiv$ the acceleration due to gravity
(See Figure 2-18.)

For flow in pipes and ducts, where frictional pressure losses are important, Equation 2-53 can be modified into

$$\frac{P_1}{\gamma_1} + \frac{v_1^2}{2g} + h_1 = \frac{P_2}{\gamma_2} + \frac{v_2^2}{2g} + h_2 + \frac{fLv^2}{2gD} \tag{2-54}$$

where $f \equiv$ an empirical friction factor
$v \equiv$ the average velocity along the flow path
$L \equiv$ the length of the flow path
$D \equiv$ the hydraulic diameter, 2(flow area)/(wetted perimeter)
(See Figure 2-18.)

If the fluid is highly compressible, Equation 2-53 must be further modified:

$$\frac{P_1}{\gamma_1}\left(\frac{k}{k-1}\right) + \frac{v_1^2}{2g} + h_1 = \frac{P_2}{\gamma_2}\left(\frac{k}{k-1}\right) + \frac{v_2^2}{2g} + h_2 \tag{2-55}$$

where $k \equiv$ ratio of specific heats, c_p/c_v; see Table 2-8.
(See Figure 2-18.)

The *friction factor* in Equations 2-54 and 2-55 is a function of the *surface roughness* of the pipe and the *Reynold's number.* Typical surface roughnesses of new commercial pipes are shown in Table 2-9. Old or corroded pipes may have a significantly higher roughness.

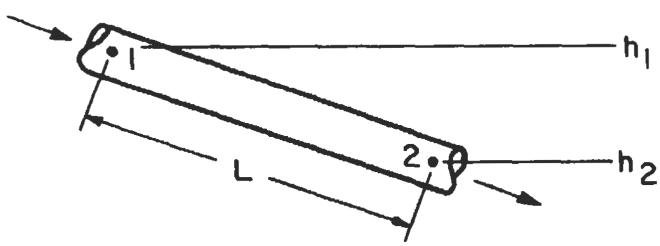


Figure 2-18. Flow in an inclined pipe.

Table 2-8
Critical Expansion Rates [7]

Gas	C_p Btu/(lb mole) (°F)	$k = C_p/C_v$ at 1 atm, 60°F	$\frac{P_c}{P_1}$	Acoustical velocity at 60°F, ft/sec
Air.........	7.00	1.410	0.528	1,031
Helium.....	4.968	1.66	0.486	2,840
Methane....	8.44	1.308	0.545	1,350
Ethane.....	12.30	1.193	0.565	967
Propane....	17.10	1.133	0.577	793
Isobutane...	22.4	1.097	0.585	681
n-Butane...	23.0	1.094	0.585	680
0.6 gravity..	8.84	1.299	0.546	1,309
0.7 gravity..	9.77	1.279	0.550	1,035

Table 2-9
Values of Absolute Roughness, New, Clean, Commercial Pipes [1]

Type of pipe or tubing	ϵ ft (0.3048 m) × 10^6 Range	Design	Probable max variation of f from design, %
Asphalted cast iron	400	400	−5 to +5
Brass and copper	5	5	−5 to +5
Concrete	1,000 10,000	4,000	−35 to 50
Cast iron	850	850	−10 to +15
Galvanized iron	500	500	0 to +10
Wrought iron	150	150	−5 to 10
Steel	150	150	−5 to 10
Riveted steel	3,000 30,000	6,000	−25 to 75
Wood stave	600 3,000	2,000	−35 to 20

Compiled from data given in "Pipe Friction Manual," Hydraulic Institute, 3d ed., 1961.

The Reynolds number is the ratio of the inertia forces acting on the fluid to the viscous forces acting on the fluid. It is dimensionless and may be calculated as

$$R = \frac{\gamma D v}{g \mu} \tag{2-56}$$

The term μ in Equation 2-56 is the *dynamic viscosity* of the fluid. The dynamic viscosity is the ratio of the shear stress to the shear rate. It has units of (force × time)/(area). The most common unit of viscosity is the centipoise (1 centipoise = 0.01 g/cm - s). Dynamic viscosity may be a function of temperature, pressure, and shear rate.

For *Newtonian fluids* the dynamic viscosity is constant (Equation 2-57), for *power-law fluids* the dynamic viscosity varies with shear rate (Equation 2-58), and for *Bingham plastic fluids* flow occurs only after some minimum shear stress, called the yield stress, is imposed (Equation 2-59).

$$\tau = \mu\left(\frac{dv}{dy}\right) \tag{2-57}$$

$$= \mu\left(\frac{dv}{dy}\right)^n \tag{2-58}$$

$$= \tau_o + \mu\left(\frac{dv}{dy}\right) \tag{2-59}$$

where τ is the shear stress in force per unit area, dv/dy is the shear rate (rate of change in velocity with respect to distance from measured perpendicular to the flow). The behavior of all three types of fluids is illustrated in Figure 2-19. Viscosities of common fluids that are normally Newtonian are given in Tables 2-10 and 2-11. Viscosities for hydrocarbon gases can be estimated from Figure 2-20.

If the calculated value of the Reynolds number is below 2,000, the flow will generally be laminar, that is, the fluid particles will follow parallel flow paths. For laminar flow the friction factor is

$$f = 64/R \tag{2-60}$$

If the Reynolds number is greater than 4,000, the flow will generally be turbulent and the friction factor can be calculated from the Colebrook equation:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left[\frac{\varepsilon/D}{3.7} + \frac{2.51}{R\sqrt{f}}\right] \tag{2-61}$$

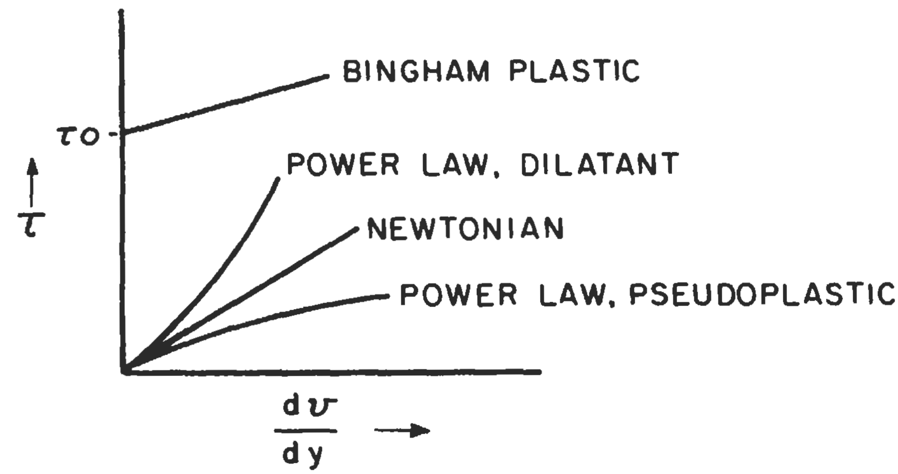


Figure 2-19. Viscous behavior of fluids.

Table 2-10
Dynamic Viscosity of Liquids at Atmospheric Pressure [1]

Temp: °C	0	20	40	60	80	100
°F	32	68	104	140	176	212
Liquid	μ, (lbf • s)/(ft^2) [47.88 (N • s)/(m^2)] × 10^6					
Alcohol, ethyl[a]	37.02	25.06	17.42	12.36	9.028	
Benzene[a]	19.05	13.62	10.51	8.187	6.871	
Carbon tetrachloride[e]	28.12	20.28	15.41	12.17	9.884	
Gasoline,[b] sp. gr. 0.68	7.28	5.98	4.93	4.28		
Glycerin[d]	252,000	29,500	5,931	1,695	666.2	309.1
Kerosene,[b] sp. gr. 0.81	61.8	38.1	26.8	20.3	16.3	
Mercury[a]	35.19	32.46	30.28	28.55	27.11	25.90
Oil, machine,[a] sp. gr. 0.907						
"Light"	7,380	1.810	647	299	164	102
"Heavy"	66,100	9,470	2,320	812	371	200
Water, fresh[c]	36.61	20.92	13.61	9.672	7.331	5.827
Water, salt[d]	39.40	22.61	18.20			

Computed from data given in:
[a]"Handbook of Chemistry and Physics," 52d ed., Chemical Rubber Company, 1971–1972.
[b]"Smithsonian Physical Tables," 9th rev. ed., 1954.
[c]"Steam Tables," ASME, 1967.
[d]"American Institute of Physics Handbook," 3d ed., McGraw-Hill, 1972.
[e]"International Critical Tables," McGraw-Hill.

Table 2-11
Viscosity of Gases at 1 Atm [1]

Temp: °C	0	20	60	100	200	400	600	800	1000
°F	32	68	140	212	392	752	1112	1472	1832
Gas	μ, (lbf • s)/(ft^2) [47.88 (N • s)/(m^2)] × 10^8								
Air*	35.67	39.16	41.79	45.95	53.15	70.42	80.72	91.75	100.8
Carbon dioxide*	29.03	30.91	35.00	38.99	47.77	62.92	74.96	87.56	97.71
Carbon monoxide†	34.60	36.97	41.57	45.96	52.39	66.92	79.68	91.49	102.2
Helium*	38.85	40.54	44.23	47.64	55.80	71.27	84.97	97.43	
Hydrogen*,†	17.43	18.27	20.95	21.57	25.29	32.02	38.17	43.92	49.20
Methane*	21.42	22.70	26.50	27.80	33.49	43.21			
Nitrogen*,†	34.67	36.51	40.14	43.55	51.47	65.02	76.47	86.38	95.40
Oxygen†	40.08	42.33	46.66	50.74	60.16	76.60	90.87	104.3	116.7
Steam‡		18.49	21.89	25.29	33.79	50.79	67.79	84.79	

Computed from data given in:
*"Handbook of Chemistry and Physics," 52d ed., Chemical Rubber Company, 1971–1972.
†"Tables of Thermal Properties of Gases," *NBS Circular* 564, 1955.
‡" Steam Tables," ASME, 1967.

where ε is the surface roughness. Equation 2-61 can be solved iteratively. If the Reynolds number falls between 2,000 and 4,000, the flow is said to be in the critical zone, and it may be either laminar or turbulent.

Equations 2-60 and 2-61 are illustrated graphically in Figure 2-21. This chart is called a Moody diagram, and it may be used to find the friction factor, given the Reynolds number and the surface roughness.

Example 2-13

Suppose 1,000 gal/min of light machine oil (see Table 2-10) flow through a 100-ft-long straight steel pipe with a square cross-section, 2 in. on a side. At the inlet of the

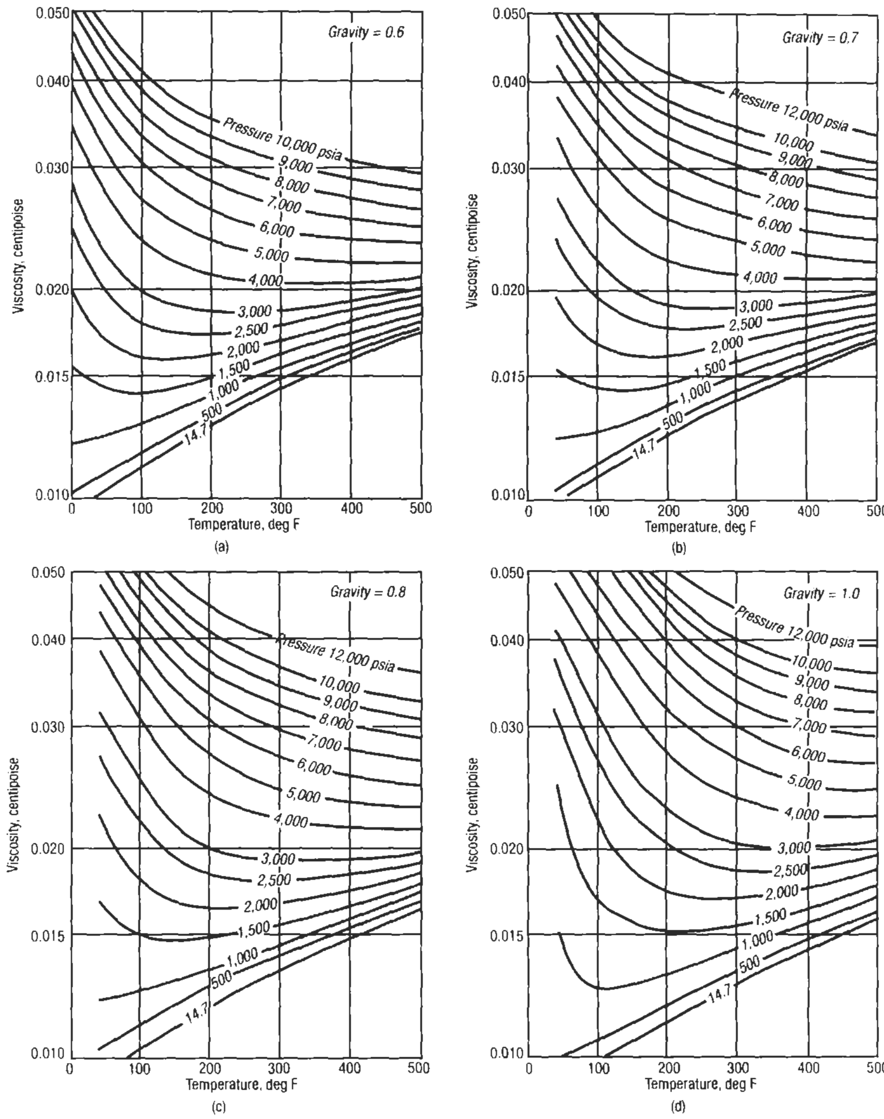


Figure 2-20. Viscosity of natural gases: (a) 0.6 gravity; (b) 0.7 gravity; (c) 0.8 gravity; (d) 1.0 gravity.

pipe the pressure is 2,000 psi and the elevation is 150 ft. At the outlet the elevation is 100 ft. What will be the pressure at the outlet when the temperature is 32°F? When the temperature is 104°F?

$$\gamma = (0.907)(62.4\ \text{lb/ft}^3) = 56.60\ \text{lb/ft}^3$$
$$A = [(2\ \text{in.})/(12\ \text{in./ft})]^2 = 0.0278\ \text{ft}^2$$
$$C = 4(2\ \text{in.})/(12\ \text{in./ft}) = 0.6667\ \text{ft}$$

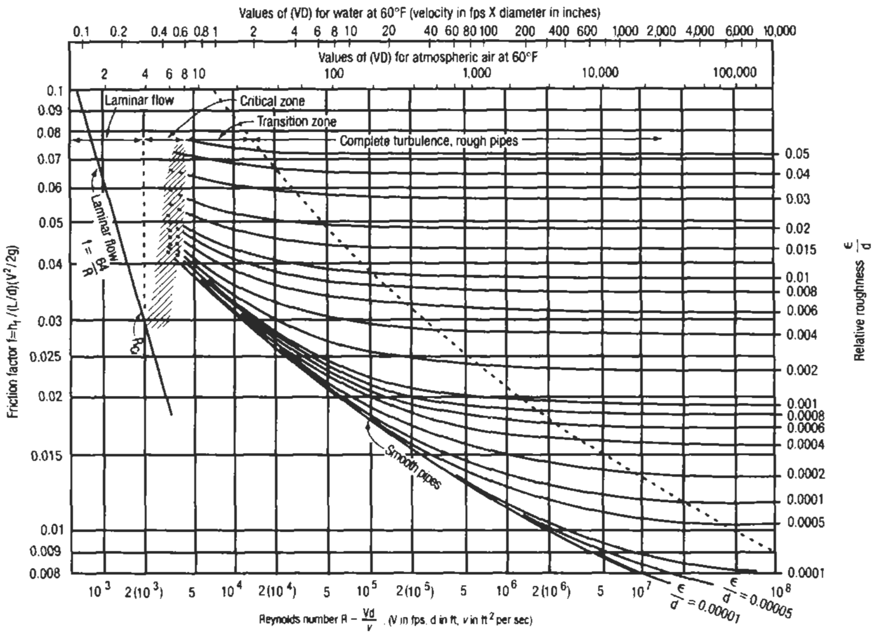


Figure 2-21. Friction factor for flow in pipes [1].

D = 2A/C = 0.0833 ft
v = Q/A = [(1,000 gal/min)(0.1337 ft³/gal)/(60 s/min)]
  = .0278 ft² = 80.21 ft/s
μ = 7,380 × 10^{-6} lb-s/ft² @ 32°F
  = 647 × 10^{-6} lb-s/ft² @ 104°F
g = 32.2 ft/s²

At 32°F

$$R = \frac{(56.60)(0.0833)(80.21)}{(32.2)(7,380\times10^{-6})} = 1,591$$

Because R < 2,000 the flow is laminar and f = 64/R = 0.04022. Assuming that the fluid is incompressible implies that $\gamma_1 = \gamma_2$ and $v_1 = v_2$, and Equation 2-54 can be rewritten as

$$P_2 = P_1 + \gamma\left[(h_1 - h_2) - \frac{fLv^2}{2gD}\right]$$

$$= 2,000 + 56.60\left[(150 - 100) - \frac{(0.04022)(100)(80.21)^2}{2(32.2)(0.0833)}\right]$$

/(144 in.²/ft²

= 124 psi

At 104°F

$$R = \frac{(56.6)(0.0833)(80.21)}{(32.2)(647 \times 10^{-6})} = 18{,}150$$

Because R > 4,000 the flow is turbulent. From Table 2-9, $\varepsilon = 150 \times 10^{-6}$, and $\varepsilon/D = 0.0018$. Interpolating from Figure 2-21 yields f = 0.03. Now apply the Colebrook equation:

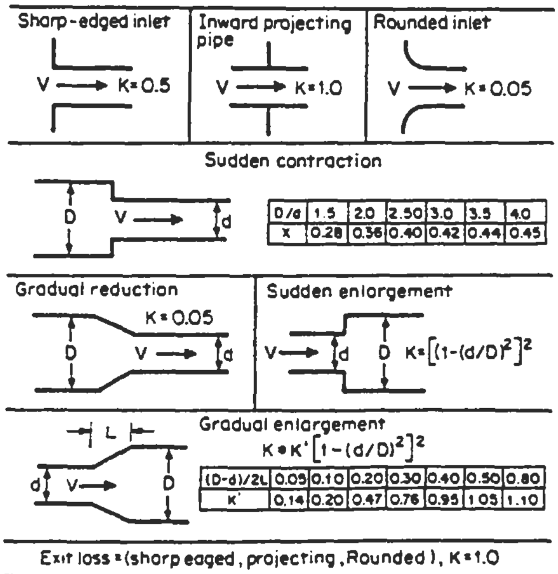
$$\frac{1}{\sqrt{f}} = -2\log_{10}\left[\frac{0.0018}{3.7} + \frac{2.51}{(18{,}150)(0.03)^{0.5}}\right]$$

f = 0.02991

$$P_2 = 2{,}000 + \left(\frac{56.60}{144}\right)\left[(150 - 100) - \frac{(0.02991)(100)(80.21)^2}{2(32.2)(0.0833)}\right] = 610 \text{ psi}$$

In *piping systems* fittings, valves, bends, etc., all cause additional pressure drops. For such components the pressure drop can be estimated by modifying the frictional component of Equation 2-54 with a *resistance coefficient,* K = fL/D, or an *equivalent length,* L/D. Typical resistance coefficients are given in Table 2-12 and typical equivalent lengths are given in Table 2-13. To correctly apply either the resistance coefficient or the equivalent length, the flow must be turbulent.

Table 2-12
Representative Values of Resistance Coefficient K [1]



D/d	1.5	2.0	2.50	3.0	3.5	4.0
K	0.28	0.36	0.40	0.42	0.44	0.45

(D-d)/2L	0.05	0.10	0.20	0.30	0.40	0.50	0.80
K'	0.14	0.20	0.47	0.76	0.95	1.05	1.10

Exit loss = (sharp edged, projecting, Rounded), K = 1.0

Compiled from data given in "Pipe Friction Manual," 3d ed., Hydraulic Institute. 1961.

Table 2-13
Representative Equivalent Length in Pipe Diameters (L/D) of Various Valves and Fittings [1]

Globe valves, fully open	450
Angle valves, fully open	200
Gate valves, fully open	13
$^3/_4$ open	35
$^1/_2$ open	160
$^1/_4$ open	900
Swing check valves, fully open	135
In line, ball check valves, fully open	150
Butterfly valves, 6 in. and larger, fully open	20
90° standard elbow	30
45° standard elbow	16
90° long-radius elbow	20
90° street elbow	50
45° street elbow	26
Standard tee:	
Flow through run	20
Flow through branch	60

Compiled from dta given in "Flow of Fluids," Crane Company Technical Paper 410, ASMF, 1971.

Example 2-14

Water flows from a horizontal 4-in. ID pipe into a horizontal 1-in. ID pipe at a rate of 1,000 gal/min (see Figure 2-22). If the transition is abrupt, what will be the pressure change across the connection?

$$v_1 = \frac{Q}{A_1} = \frac{[(1{,}000 \text{ gal/min})(0.1337 \text{ ft}^3/\text{gal})(60 \text{ s/min})]}{\left[\frac{\pi}{4}\left(\frac{4}{12}\right)^2 \text{ft}^2\right]} = 25.53 \text{ ft/s}$$

$$v_2 = \frac{[(1{,}000)(0.1337)/(60)]}{\left[\frac{\pi}{4}\left(\frac{1}{12}\right)^2\right]} = 408.6 \text{ ft/s}$$

$\gamma_1 = \gamma_2 = 62.4 \text{ lb/ft}^3$

From Table 2-8, $D_{in}/d_{out} = 4$; $k = 0.45$.
From Equation 2-54,

$$P_1 - P_2 = \Delta P = \frac{\gamma}{2g}\left[v_2^2 - v_1^2 + kv_2^2\right] = \frac{\gamma}{2g}\left[v_2^2(1+k) - v_1^2\right]$$

$$= \frac{62.4}{2(32.2)}\left[408.6^2(1+0.45) - 25.53^2\right]\frac{1}{144}$$

$$= 1{,}625 \text{ psi}$$

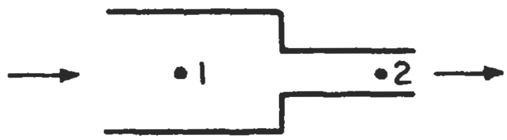


Figure 2-22. Diagram for Example 2-14.

The 1,120 psi of this pressure drop is a dynamic loss due to the change in velocity, and 505 psi is a frictional loss due to the fitting.

Components of a piping system that are connected in *series* produce additive pressure drops, while components that are connected in *parallel* must produce the same pressure drop.

While the modified energy equation provides for calculation of the flowrates and pressure drops in piping systems, the ***impulse-momentum*** equation is required in order to calculate the reaction forces on curved pipe sections. The impulse-momentum equation relates the force acting on the solid boundary to the change in fluid momentum. Because force and momentum are both vector quantities, it is most convenient to write the equations in terms of the scalar components in the three orthogonal directions.

$$\sum F_x = \dot{M}(V_{x_1} - V_{x_2})$$

$$\sum F_y = \dot{M}(V_{y_1} - V_{y_2}) \qquad (2\text{-}62)$$

$$\sum F_z = \dot{M}(V_{z_1} - V_{z_2})$$

where $\dot{M}$ is the fluid mass flow rate, ΣF_x is the sum of forces in the x direction, V_{x_1} is the initial fluid velocity in the x direction, etc.

Example 2-15

Water flows through a 120° reducing bend at a rate of 100 gpm. The inlet diameter of the bend is 2 in. and the outlet diameter is 1 in. (see Figure 2-23). What is the reaction force on the bend?

Assuming that the flow is incompressible,

$$\dot{M} = \frac{(100\ \text{gal/min})(0.1337\ \text{ft}^3/\text{gal})(62.4\ \text{lb/ft}^3)}{\left[(32.2\ \text{ft/s}^2)(60\ \text{s/min})\right]}$$

$$\dot{M} = 0.4318 \frac{\text{lb} - \text{s}^4}{\text{ft}} = 0.4318\ \text{slugs/s}$$

$$A_1 = \frac{\pi}{4}\left(\frac{2}{12}\right)^2 = 0.02182\ \text{ft}^2$$

$$A_2 = \frac{\pi}{4}\left(\frac{1}{12}\right)^2 = 0.005454\ \text{ft}^2$$

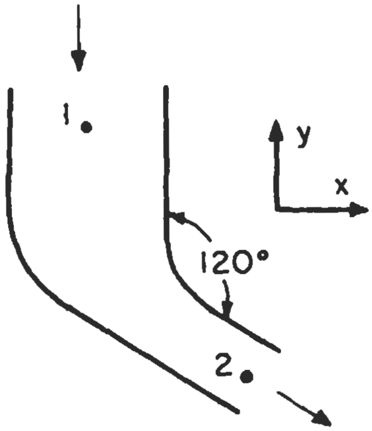


Figure 2-23. Diagram for Example 2-15.

$$v_1 = v_{y_1} = -\frac{Q}{A_1} = \frac{(100)(0.1337)(1/60)}{0.02182} = 10.21 \text{ ft/s}$$

$$v_2 = \frac{Q}{A_2} = \frac{(100)(0.1337)(1/60)}{0.005454} = 40.86 \text{ ft/s}$$

$$v_{y_2} = -v_2 \sin 30° = -20.43 \text{ ft/s}$$

$$v_{x_2} = v_2 \cos 30° = 35.38 \text{ ft/s}$$

$$F_x = \dot{M}(v_{x_1} - v_{x_2}) = 0.4318 \frac{\text{lb} \bullet \text{s}}{\text{ft}}(0 - 35.38 \text{ ft/s}) = -15.28 \text{ lb}$$

$$F_y = \dot{M}(v_{y_1} - v_{y_2}) = 0.4318(-10.21 + 20.43) = 4.413 \text{ lb}$$

Flow through *chokes and nozzles* is a special case of fluid dynamics. For *incompressible fluids* the problem can be handled by mass conservation and Bernoulli's equation. Bernoulli's equation is solved for the pressure drop across the choke, assuming that the velocity of approach and the vertical displacement are negligible. The velocity term is replaced by the volumetric flow rate times the area at the choke throat to yield

$$\Delta P = \frac{Q^2\gamma}{2gC^2A^2} \tag{2-63}$$

C is a constant introduced to account for frictional effects. In general, $0.94 \le C \le 0.98$.

Example 2-16

Assume 100 ft^3/min of water is to be pumped through a nozzle with a throat diameter of 3/4 in. What pressure drop should be expected?

$$Q = \frac{100}{60} = 1.67 \text{ ft}^3/\text{s}$$

$$A = \frac{\pi(0.75^2)}{4(144)} = 3.068 \times 10^{-3} \text{ ft}^2$$

$$\gamma = 62.4 \text{ lb/ft}^3$$

Assume C = 0.95; then

$$\Delta P = \frac{(1.67^2)(62.4)}{2(32.2)(0.95^2)(3.068 \times 10^{-3})^2(144)} = 2{,}210 \text{ psia}$$

To analyze *compressible flow through chokes* it is assumed that the entropy of the fluid remains constant. The equation of *isentropic* flow is

$$P_1V_1^k = P_2V_2^k \qquad (2\text{-}64)$$

where P_1 and V_1 are the pressure and specific volume of the fluid at point 1, immediately upstream of the choke, and P_2 and V_2 are the pressure and specific volume immediately downstream of the choke. Equation 2-64 can be combined with the ideal gas law to provide an estimate for the temperature drop across the choke

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} \qquad (2\text{-}65)$$

where T_2 and T_1 are temperatures in °R. Furthermore, the first law of thermodynamics can also be imposed, yielding the following equation for the volumetric flowrate:

$$Q = 864\ CA\frac{P_1}{(ST_1)^{0.5}}\left\{\frac{k}{k-1}\left[\left(\frac{P_2}{P_1}\right)^{2/k} - \left(\frac{P_2}{P_1}\right)^{(k+1)/k}\right]\right\}^{0.5} \qquad (2\text{-}66)$$

where Q is the volumetric flow rate in scfm, C is a *discharge coefficient* that accounts for friction and velocity of approach (see Figure 2-24). A is the choke area in square inches, P_1 is the inlet pressure in pounds per square inch absolute (psia), P_2 is the outlet pressure in psia, T_1 is the inlet temperature in °R, and S is the specific gravity of the gas.

Equations 2-65 and 2-66 apply only as long as the fluid velocity at the throat of the choke is subsonic. Sonic velocity is the speed of a pressure wave in a fluid. Once *sonic velocity* is achieved, the effects of the downstream pressure can no longer be transmitted to the upstream side of the choke.

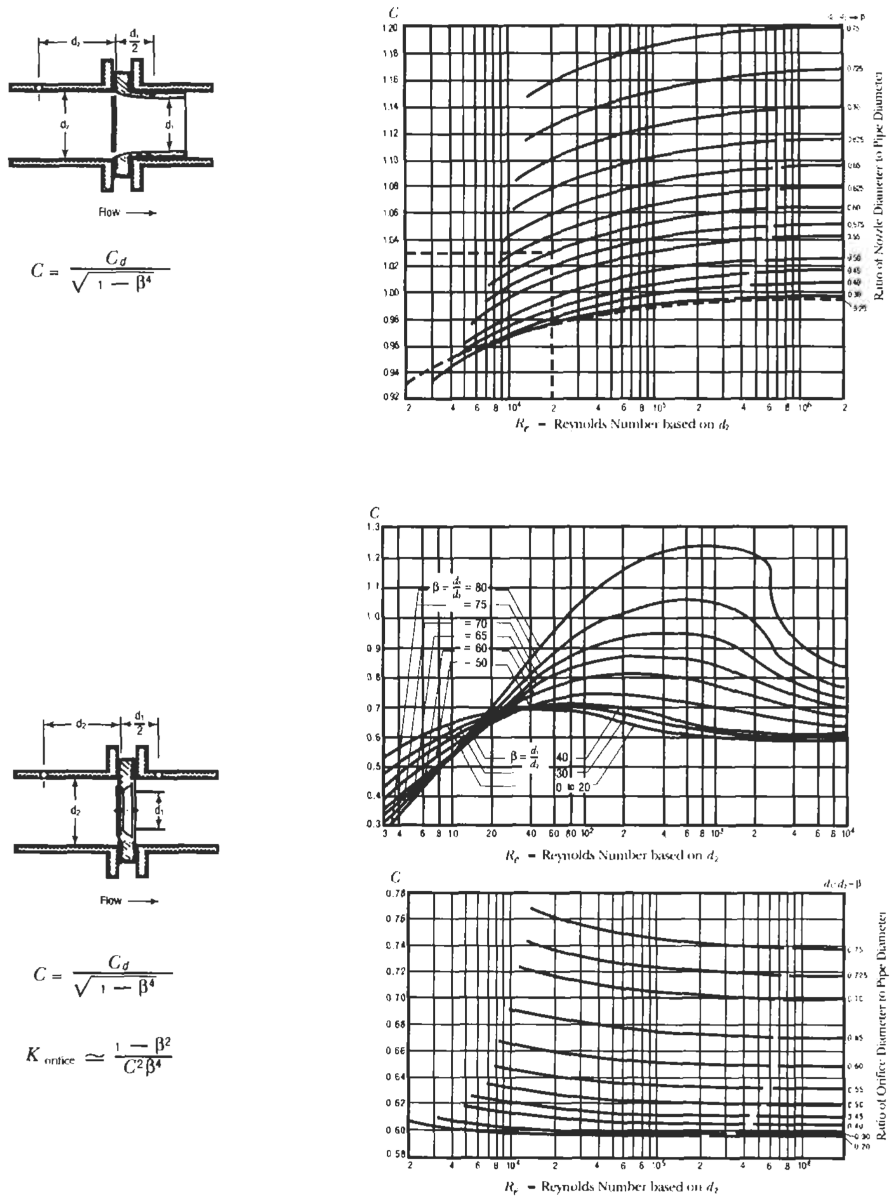


Figure 2-24. Flow coefficient for nozzles and orifices (from Brown, p. 226, Fig. 3-19).

Thus there is a critical pressure ratio beyond which the flow at the throat is always sonic. This is termed *critical flow.*

$$\frac{P_2}{P_1} = \left(\frac{2}{k+1}\right)^{k/(k-1)} \tag{2-67}$$

If the pressure ratio is less than or equal to that specified by Equation 2-67, the flow will be sonic at the choke throat and the temperature at the throat can be found from

$$T_2 = T_1\left(\frac{2}{k+1}\right) \tag{2-68}$$

The flow rate can be found from

$$Q = 610CA\frac{P_1}{(ST_1)^{0.5}}\left[k\left(\frac{2}{k+1}\right)^{(k+1)/(k-1)}\right]^{0.5} \tag{2-69}$$

For critical flow the discharge coefficient is dependent upon the geometry of the choke and its diameter or the ratio β of its diameter to that of the upstream pipe (see Figure 2-24).

Example 2-17

A 0.6 gravity hydrocarbon gas flows from a 2-in. ID pipe through a 1-in. ID orifice plate. The upstream temperature and pressure are 75°F and 800 psia, respectively. The downstream pressure is 200 psia. Does heating need to be applied to assure that frost does not clog the orifice? What will be the flow rate?

Check for Critical Flow. From Table 2-8 it is determined that k = 1.299. Checking the pressure ratio, Equation 2-67, gives

$$\frac{P_2}{P_1} = 0.25 < \left(\frac{2}{k+1}\right)^{k/(k-1)} = 0.546$$

Therefore critical flow conditions exist and the fluid velocity in the choke is sonic.

Temperature in the Choke. From Equation 2-68 the temperature in the throat is calculated as

$$T_2 = T_1\left(\frac{2}{k+1}\right) = 535\left(\frac{2}{1+1.299}\right) = 465.4°\,R = 5.4°\,F$$

Therefore, if the gas contains water, icing or hydrate formation may occur causing the throat to clog. A heating system may be needed.

Flowrate. Assume that the discharge coefficient is C = 1.0. The choke area is A = $\pi(1/2)^2 = 0.785$ in.^{2}. Thus, from Equation 2-69,

$$Q = 610(1.0)(0.785)\frac{800}{[(0.6)(535)]^{0.5}}\left[1.299\left(\frac{2}{1.299+1}\right)^{(1.299+1)/(1.299-1)}\right]^{0.5}$$

$$= 14{,}021 \text{ scfm}$$

Now check on the discharge coefficient.

The Reynolds number for gases can be calculated directly in terms of flowrate and gas gravity as

$$R_e = \frac{28.8Qs}{\mu d} \tag{2-70}$$

where Q is in scfm, s is the specific gravity of the gas, d is the pipe hydraulic diameter in inches, and μ is in centipoise. From Figure 2-20 the viscosity of the gas is

$$\mu = 0.0123 \text{ cp}$$

and

$$R_e = \frac{(28.8)(14021)(0.6)}{(0.0123)(2)} = 9{,}850{,}000$$

From Figure 2-24, using $\beta = 0.5$, the value of the discharge coefficient is read as C = 0.62, and a new estimate of Q is:

$$Q = 0.62(14{,}021) = 8{,}693 \text{ scfm}$$

A further iteration produces no change in the estimated flow rate for this case.

In subcritical flow the discharge coefficient is affected by the velocity of approach as well as the type of choke and the ratio of choke diameter to pipe diameter. Discharge coefficients for subcritical flow are given in Figure 2-24 as a function of the diameter ratio and the upstream Reynolds number. Since the flow rate is not initially known, it is expedient to assume C = 1, calculate Q, use this Q to calculate the Reynold's number, and then use the charts to find a better value of C. This cycle should be repeated until the value of C no longer changes.

Example 2-18

A 0.65 gravity naturally gas (K = 1.25) flows from a two-in. line through a 1.5-in. nozzle. The upstream temperature is 90°F. The upstream pressure is 100 psia while the downstream pressure is 80 psia. Is icing a potential problem? What will be the flowrate?

Check for critical flow using Equation 2-67.

$$\frac{P_2}{P_1} = 0.8 > \left[\frac{2}{k+1}\right]^{k/(k-1)} = 0.549$$

The flow is clearly subcritical.

Check the outlet temperature using Equation 2-65.

$$T_2 = 550(0.8^{0.25/1.25}) = 506.86°R = 66°F$$

There will be no icing.

Calculate the flowrate.

$$A = \pi\left(\frac{1.5}{2}\right)^2 = 1.767 \text{ in.}^2$$

Assuming C = 1 and applying Equation 2-66 gives

$$Q = (864)(1)(1.767)\frac{100}{[(0.65)(530)]^{1/2}}\left\{\frac{1.25}{1.25-1}\left[\left(\frac{80}{100}\right)^{2/1.25} - \left(\frac{80}{100}\right)^{(1.25+1)/1.25}\right]\right\}^{1/2}$$

$$= 3214 \text{ scfm}$$

From Figure 2-20 the viscosity of the gas is

$\mu = 0.0106$ cp

and from Equation 2-70

$$R_e = \frac{(28.8)(3{,}214)(0.65)}{(0.0106)(2)} = 2.84 \times 10^6$$

From Figure 2-24, using $\beta = 0.75$, the value of the discharge coefficient is read as c = 1.2. Now a new estimate of Q can be found as

$$Q = \left(\frac{1.2}{1}\right)3214 = 3{,}857 \text{ scfm}$$

Because further increases in the flowrate (see Figure 2-24) will produce no increase in the discharge coefficient, it is unnecessary to do any further iterations.

For further information on this subject refer to Reference 1 and References 6–9.

STRENGTH OF MATERIALS

The principles of strength of materials are applied to the design of structures to assure that the elements of the structures will operate reliably under a known set of loads. Thus the field encompasses both the calculation of the strength and deformation of members and the measurement of the mechanical properties of engineering materials.

Stress and Strain

Consider a bar of length L and uniform cross-sectional area A to which an axial, uniformly distributed load with a magnitude, P, is applied at each end (Figure 2-25). Then within the bar there is said to be *uniaxial stress* σ, defined as the load, or force per unit area

$$\sigma = \frac{P}{A} \tag{2-71}$$

If the load acts to elongate the bar, the stress is said to be *tensile (+)*, and if the load acts to compress the bar, the stress is said to be *compressive (−)*. For all real materials,

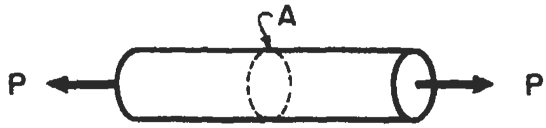


Figure 2-25. Uniaxial loading of a bar.

an externally applied load will produce some *deformation.* The ratio of the deformation to the undeformed length of the body is called the *strain* ε. In the simple case illustrated in Figure 2-25, the strain is

$$\varepsilon = \delta/L \tag{2-72}$$

where δ is the longitudinal deformation. The strain is tensile or compressive depending upon the sign of δ. The relationship between stress and strain in an axially loaded bar can be illustrated in a stress–strain curve (Figure 2-26). Such curves are experimentally generated through *tensile tests.*

In the region where the relationship between stress and strain is linear, the material is said to be *elastic,* and the constant of proportionality is E, *Young's modulus,* or the *elastic modulus.*

$$\sigma = E\varepsilon \tag{2-73}$$

Equation 2-73 is called *Hooke's law.*

In the region where the relationship between stress and strain is nonlinear, the material is said to be *plastic.* Elastic deformation is recoverable upon removal of the load, whereas plastic deformation is permanent. The stress at which the transition occurs, σ_y, is called the *yield strength* or *yield point* of the material, and the maximum

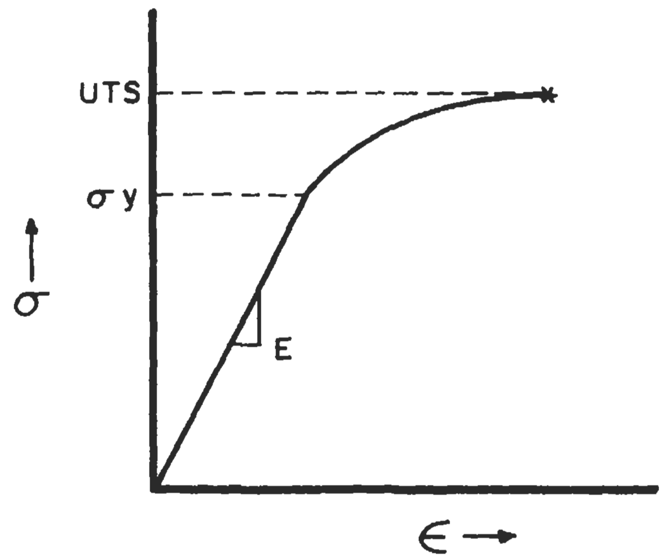


Figure 2-26. Idealized stress–strain curve.

stress is called the *ultimate tensile strength,* UTS, of the material. Standard engineering practice is to define the yield point as 0.2% permanent strain.

When a bar is elongated axially, as in Figure 2-25, it will contract laterally. The negative ratio of the lateral strain to the axial strain is called *Poisson's ratio* ν. For *isotropic* materials, materials that have the same elastic properties in all directions, Poisson's ratio has a value of about 0.3.

Now consider a block to which a uniformly distributed load of magnitude P is applied parallel to opposed faces with area A (Figure 2-27). These loads produce a *shear stress* within the material τ.

$$\tau = P/A \tag{2-74}$$

Note that in order for the block of Figure 2-27 to be in static equilibrium, there must also be a load P applied parallel to each of the faces B. Thus any given shear stress always implies a second shear stress of equal magnitude acting perpendicularly to the first so as to produce a state of static equilibrium. The shear stress will produce a deformation of the block, manifested as a change in the angle between the face perpendicular to the load and the face over which the load is applied. This change in angle is called the *shear strain* γ.

$$\gamma = \Delta\alpha \tag{2-75}$$

For an elastic material the shear stress is related to the shear strain through a constant of proportionality G, called the *shear modulus.* The shear strain is dimensionless, and the shear modulus has units of force per unit area.

$$\tau = G\gamma \tag{2-76}$$

The shear modulus is related to Young's modulus and Poisson's ratio by

$$G = \frac{E}{2(1+\nu)} \tag{2-77}$$

In practice, loads are not necessarily uniformly distributed nor uniaxial, and cross-sectional areas are often variable. Thus it becomes necessary to define the stress at a point as the limiting value of the load per unit area as the area approaches zero. Furthermore, there may be tensile or compressive stresses (σ_x, σ_y, σ_z) in each of three orthogonal directions and as many as six shear stresses (τ_{xy}, τ_{yx}, τ_{xz}, τ_{zx}, τ_{yz}, τ_{zy}). The

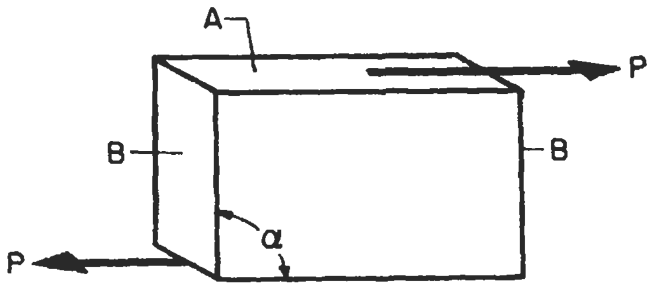


Figure 2-27. Shear loading of a block.

direction of the shear stress is indicated by two subscripts, the first of which indicates the direction normal to the plane in which the load is applied, and the second of which indicates the direction of the load. Note that for static equilibrium to exist, $\tau_{xy} = \tau_{yx}$, $\tau_{xz} = \tau_{zx}$, and $\tau_{yz} = \tau_{zy}$.

If a multidimensional state of stress exists, the Poisson's ratio effect causes the tensile and compressive strains to be dependent upon each of the components of stress.

$$\varepsilon_x = \frac{1}{E}\left[\sigma_x - \nu(\sigma_y + \sigma_z)\right] \tag{2-78}$$

$$\varepsilon_y = \frac{1}{E}\left[\sigma_y - \nu(\sigma_x + \sigma_z)\right] \tag{2-79}$$

$$\varepsilon_z = \frac{1}{E}\left[\sigma_z - \nu(\sigma_y + \sigma_z)\right] \tag{2-80}$$

Likewise the stresses may be written in terms of the components of strain.

$$\sigma_x = \frac{E}{(1+\nu)(1-2\nu)}\left[(1-\nu)\varepsilon_x + \nu(\varepsilon_y + \varepsilon_z)\right] \tag{2-81}$$

$$\sigma_y = \frac{E}{(1+\nu)(1-2\nu)}\left[(1-\nu)\varepsilon_y + \nu(\varepsilon_x + \varepsilon_z)\right] \tag{2-82}$$

$$\sigma_z = \frac{E}{(1+\nu)(1-2\nu)}\left[(1-\nu)\varepsilon_z + \nu(\varepsilon_x + \varepsilon_y)\right] \tag{2-83}$$

The components of the shear stress all obey Equation 2-74.

While the foregoing discussion of stress and strain is based on a Cartesian coordinate system, any orthogonal coordinate system may be used.

Elementary Loading Configurations

Torsion of a Cylinder

Consider a uniform cylindrical bar or tube to which some balanced torque T is applied (Figure 2-28). The bar will be subject to a *torsional stress*, or shear stress $\tau_{z\theta}$, which increases with the radial position within the bar.

$$\tau_{z\theta} = Tr/J \tag{2-84}$$

where r is the radial distance from the z axis, and J is the polar moment of inertia. The polar moment of inertia for a hollow cylinder with an internal radius r_i and an external radius, r_o, is

$$J = \frac{\pi}{2}\left(r_o^4 - r_i^4\right) \tag{2-85}$$

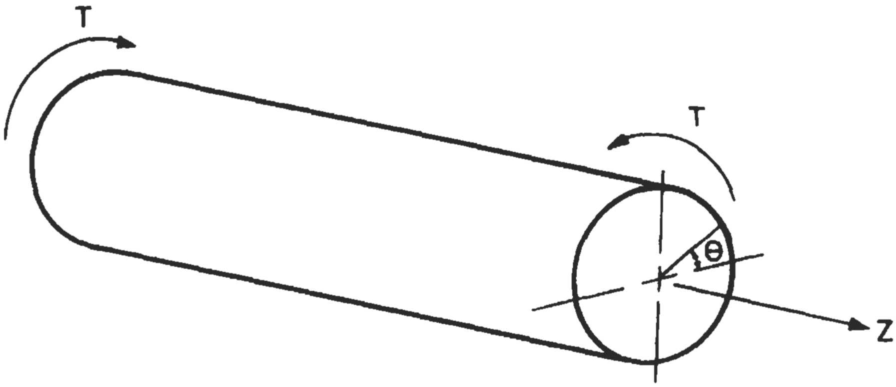


Figure 2-28. Torsional loading of a right circular cylinder.

The strain due to the torque T is given by

$$\gamma_{z\theta} = \frac{2(1+\nu)}{E}\frac{Tr}{J} \tag{2-86}$$

and the total angular deflection between any two surfaces perpendicular to the z axis is

$$\Theta = 2(1+\nu)\frac{TL}{EJ} \tag{2-87}$$

where L is the distance between the two surfaces.

Example 2-19

A uniform steel bar 3 ft long and 2.5 in. in diameter is subjected to a torque of 800 ft-lb. What will be the maximum torsional stress and the angular deflection between the two ends?

Assume $E = 30 \times 10^6$ psi and $\nu = 0.3$.

Then

$$J = \frac{\pi}{2}(1.25^4 - 0^4) = 3.83 \text{ in.}^4$$

$$\tau_{z\theta} = \frac{800 \text{ ft-lb}}{} \Big| \frac{12 \text{ in.}}{\text{ft}} \Big| \frac{1.25 \text{ in.}}{} \Big| \frac{}{3.83 \text{ in.}^4} = 3133 \text{ psi}$$

$$\theta = 2(1+0.3)\frac{800 \text{ ft-lb}}{} \Big| \frac{12 \text{ in.}}{\text{ft}} \Big| \frac{3 \text{ ft}}{} \Big| \frac{12 \text{ in.}}{\text{ft}} \Big| \frac{\text{in.}^2}{30 \times 10^6 \text{ lb}} \Big| \frac{}{3.83 \text{ in.}^4}$$

$$= 7.8 \times 10^{-3} \text{rad} \frac{360°}{2\pi \text{ rad}} = 0.45°$$

Transverse Loading of Beams

A beam subjected to a simple transverse load (Figure 2-29a) will bend. Furthermore, if the beam is cut (Figure 2-29b) and free-body diagrams of the remaining sections are constructed, then a shear force V and a moment M must be applied to the cut ends to maintain static equilibrium.

The magnitude of the shearing force and the moment can be determined from the conditions of static equilibrium of the beam section. Thus, for the cut shown in Figure 2-29b, the shear force and the moment on the left-hand section are

$$V = -R_A = (-b/L)P \tag{2-88}$$

$$M = R_A x = (b/L)Px \tag{2-89}$$

If the cut had been to the right of the point of application of load P, then the shear force and the moment on the left-hand section would be

$$V = -R_A + P = (-b/L)P + P = (a/L)P \tag{2-90}$$

$$M = R_A x - P(x-a) = P\left(\frac{bx}{L} - x + a\right) = \frac{a}{L}P(L-x) \tag{2-91}$$

In the loading configuration of Figure 2-29, the beam will bend in the concave upward direction, thus putting the lowermost fiber in tension and the uppermost fiber in compression. The magnitude of this axial stress is

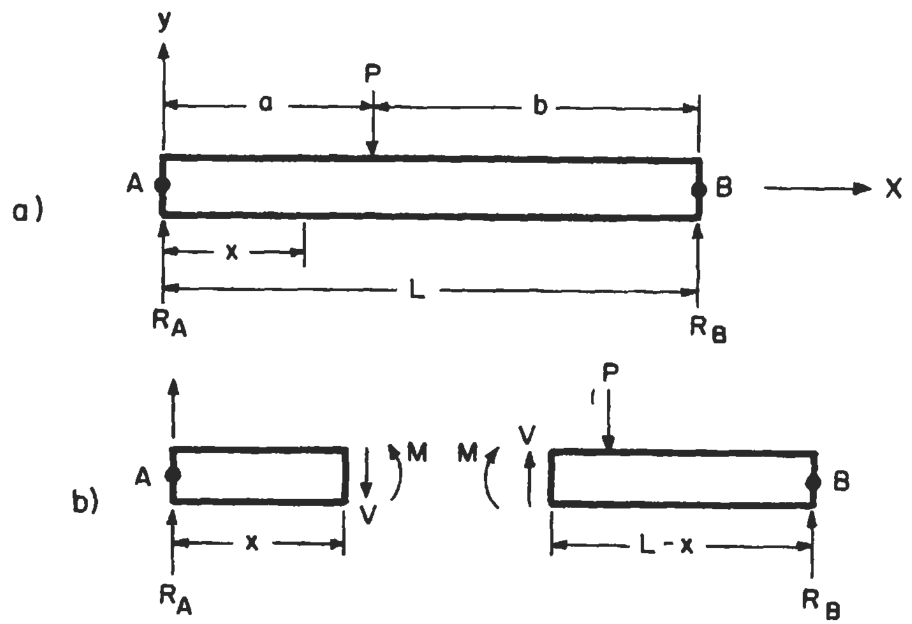


Figure 2-29. Transverse loading of a beam.

$$\sigma_x = -\frac{My}{I_z} \tag{2-92}$$

where y is the distance from the *neutral axis* of the beam (the axis along which $\sigma_x = 0$) and I_z is the *areal moment of inertia* about the z axis. The neutral axis and the *centroidal axis* will coincide. The position of the centroidal axis of common areas is given in Table 2-14. The areal moment of inertia about any axis, w for instance, is defined by

$$I_w = \int_A r^2 \, dA \tag{2-93}$$

Table 2-14
Properties of Cross-Sections

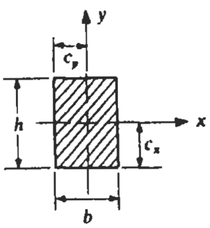
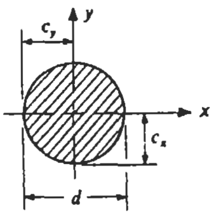
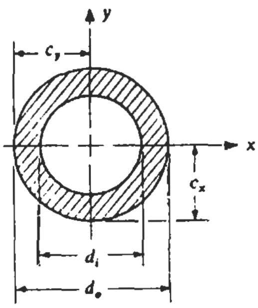
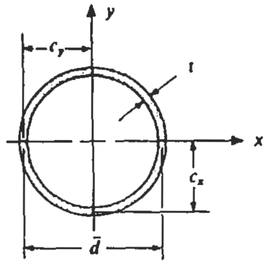
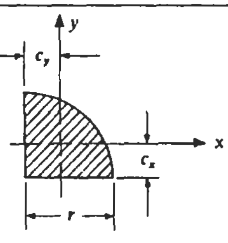
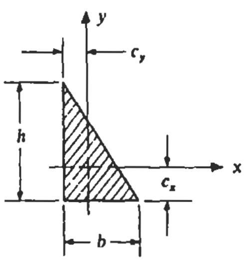
Section	Area	Centroid	Area moments of inertia
 Rectangle	bh	$c_x = \frac{h}{2}$ $c_y = \frac{b}{2}$	$I_x = \frac{bh^3}{12}$ $I_y = \frac{hb^3}{12}$
 Circle	$\frac{\pi d^2}{4}$	$c_x = \frac{d}{2}$ $c_y = \frac{d}{2}$	$I_x = \frac{\pi d^4}{64}$ $I_y = \frac{\pi d^4}{64}$ $I_z = J = \frac{\pi d^4}{32}$
 Thick-walled tube	$\frac{\pi}{4}(d_o^2 - d_i^2)$	$c_x = \frac{d_o}{2}$ $c_y = \frac{d_o}{2}$	$I_x = \frac{\pi}{64}(d_o^4 - d_i^4)$ $I_y = \frac{\pi}{64}(d_o^4 - d_i^4)$ $I_z = J = \frac{\pi}{32}(d_o^4 - d_i^4)$

Table 2-14
(continued)

 Thin-walled tube ($\bar{d} \gg t$)	$\pi \bar{d} t$	$c_x = \frac{\bar{d}}{2}$ $c_y = \frac{\bar{d}}{2}$	$I_x = \frac{\pi}{8} t \bar{d}^3$ $I_y = \frac{\pi}{8} t \bar{d}^3$ $I_z = J = \frac{\pi}{4} t \bar{d}^3$
 Circular quadrant	$\frac{\pi}{4} r^2$	$c_x = \frac{4r}{3\pi}$ $c_y = \frac{4r}{3\pi}$	$I_x = \left(\frac{\pi}{16} - \frac{4}{9\pi}\right) r^4$ $I_y = \left(\frac{\pi}{16} - \frac{4}{9\pi}\right) r^4$ $P_{xy} = \left(\frac{1}{8} - \frac{4}{9\pi}\right) r^4$
 Triangle	$\frac{1}{2} bh$	$c_x = \frac{h}{3}$ $c_y = \frac{b}{3}$	$I_x = \frac{bh^3}{36}$ $I_y = \frac{hb^3}{36}$ $P_{xy} = -\frac{b^2 h^2}{72}$

where A is the cross-sectional area of the body and r is the perpendicular distance from axis w to the differential element of area dA. Values for the areal moments of inertia of common cross-sections are given in Table 2-14.

The beam is also subject to a shear stress that varies over the beam cross-section.

$$\tau_{xy} = \frac{VQ}{I_z b} \tag{2-94}$$

where b is the width of the beam. *The moment area about the z axis* Q is defined as

$$Q = \int_{y_0}^{y_{max}} y \, dA \tag{2-95}$$

where y_0 is the location of the shear stress.

Example 2-20

Assuming that the beam in Figure 2-29 has a rectangular cross-section with a height of 1 ft and a width of 3 in. and given that L = 10 ft, a = 4 ft, and P = 1,000 lb, what are the maximum values of the shear and tensile stresses within the beam?

Tensile Stress. The maximum moment occurs at the point of application of the load P and has the value

$$M = \frac{ba}{L} = \frac{(4)(6)}{10} 1{,}000 = 2{,}400 \text{ ft-lb} = 28{,}800 \text{ in.-lb}$$

The areal moment of inertia of the beam can be found from Table 2-14 as

$$I_z = \frac{1}{12} bh^3 = \frac{1}{12}(3)(12^3) = 432 \text{ in.}^4$$

The maximum tensile stress occurs at the outer fiber of the beam (at $y = -6$ in.) and has the value

$$\sigma_{x_{max}} = -\frac{My}{I_z} = -\frac{(28{,}800)(-6)}{432} = 400 \text{ lb/in.}^2$$

Shear Stress. For a rectangular cross-section, the maximum value of Q occurs at the neutral axis, and, because the width b of the beam is a constant 3 in., the maximum value of the shear stress occurs at the neutral axis.

$$Q = \int_0^6 yb\,dy = \frac{1}{2} by^2\Big|_0^6 = \frac{1}{2}(3)(6)^2 = 54 \text{ in.}^3$$

$$V = -\frac{4}{10} \times 1{,}000 = -400 \text{ lb}$$

$$\tau_{xy} = \frac{(-400)(54)}{(3)(432)} = -16.67 \text{ lb/in.}^2$$

Thin-Walled Pressure Vessels

A thin-walled pressure vessel is one in which the wall thickness t is small when compared to the local radius of curvature r. At a point in the wall of the vessel where the radius of curvature varies with the direction, the wall stresses are

$$\frac{\sigma_\theta}{r_\theta} + \frac{\sigma_\alpha}{r_\alpha} = \frac{p}{t} \tag{2-96}$$

where p is the net internal pressure and θ and α indicate any two orthogonal directions tangent to the vessel surface at the point in question. For a spherical vessel, $r_\theta = r_\alpha = r$, and Equation 2-96 reduces to

$$\sigma_\theta = \sigma_\alpha = \frac{Pr}{2t} \tag{2-97}$$

For a cylindrical vessel, the radius of curvature in the axial direction is infinite, and the stress in the direction of the circumference, called the *hoop stress,* is

$$\sigma_\theta = \frac{Pr}{t} \tag{2-98}$$

The stress in the axial direction in a cylindrical vessel is found by taking a cross-section perpendicular to the longitudinal axis and imposing the conditions of static equilibrium. This yields

$$\sigma_z = \frac{Pr}{2t} \tag{2-99}$$

Prediction of Failure

For most practical purposes, the onset of plastic deformation constitutes failure. In an axially loaded part, the yield point is known from testing (see Tables 2-15 through 2-18), and failure prediction is no problem. However, it is often necessary to use uniaxial tensile data to predict yielding due to a multidimensional state of stress. Many failure theories have been developed for this purpose. For elastoplastic materials (steel, aluminum, brass, etc.), the *maximum distortion energy theory* or *von Mises theory* is in general application. With this theory the components of stress are combined into a single effective stress, denoted as σ_e, which can be compared to known data for uniaxial yielding. The ratio of the measure yield stress to the effective stress is known as the factor of safety.

$$\sigma_e = \left\{\frac{1}{2}\left[(\sigma_x - \sigma_y)^2 + (\sigma_x - \sigma_z)^2 + (\sigma_y - \sigma_z)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{xz}^2)\right]\right\}^{1/2} \tag{2-100}$$

For brittle materials such as glass or cast iron, the maximum shear-stress theory is usually applied.

Example 2-21

A cylindrical steel pressure vessel (AISI SAE 1035, cold rolled) with a wall thickness of 0.1 in. and an inside diameter of 1 ft is subject to an internal pressure of 1,000 psia and a torque of 10,000 ft-lb (see Figure 2-30). What is the effective stress at point A in the wall? What is the factor of safety in this design?

Hoop stress:

$$\sigma_\theta = \frac{(1{,}000\ \text{psi})(6\ \text{in.})}{(0.1\ \text{in.})} = 60{,}000\ \text{psi}$$

Axial stress:

$$\sigma_z = \frac{(1{,}000\ \text{psi})(6\ \text{in.})}{2(0.1\ \text{in.})} = 30{,}000\ \text{psi}$$

(text continues on page 207)

Table 2-15
Mechanical Properties of Metals and Alloys [10]

FERROUS ALLOYS

Ferrous alloys comprise the largest volume of metal alloys used in engineering. The actual range of mechanical properties in any particular grade of alloy steel depends on the particular history and heat treatment. The steels listed in this table are intended to give some idea of the range of properties readily obtainable. Many hundreds of steels are available. Cost is frequently an important criterion in the choice of material; in general the greater the percentage of alloying elements present in the alloy, the greater will be the cost.

				Typical mechanical properties				
No.	Material	Nominal composition	Form and condition	Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
	IRON							
1	Ingot iron (Included for comparison)	Fe 99.9	Hot-rolled Annealed	29 19	45 38	26 45	90 67	
	PLAIN CARBON STEELS							
2	AISI-SAE 1020	C 0.20 Mn 0.45 Si 0.25 Fe bal.	Hot-rolled Hardened (water-quenched, 1000°F-tempered)	30 62	55 90	25 25	111 179	Bolts, crankshafts, gears, connecting rods; easily weldable
3	AISI 1025	C 0.25 Fe bal. Mn 0.45	Bar stock Hot-rolled Cold-drawn	 32 54	 58 64	 25 15	 116 126	
4	AISI-SAE 1035	C 0.35 Mn 0.75	Hot-rolled Cold-rolled	39 67	72 80	18 12	143 163	Medium-strength, engineering steel
5	AISI-SAE 1045	C 0.45 Fe bal. Mn 0.75	Bar stock Annealed Hot-rolled Cold-drawn	 73 45 77	 80 82 91	 12 16 12	 170 163 179	
6	AISI-SAE 1078	C 0.78 Fe bal. Mn 0.45	Bar stock Hot-rolled; spheroidized Annealed	 55 72	 100 94	 12 10	 207 192	
7	AISI-SAE 1095	C 0.95 Fe bal. Mn 0.40						
8	AISI-SAE 1120	C 0.2 Mn 0.8 S 0.1	Cold-drawn	58	69	—	137	Free-cutting, leaded, resulphurized steel; high-speed, automatic machining
	ALLOY STEELS							
9	ASTM A202/56	C 0.17 Mn 1.2 Cr 0.5 Si 0.75	Stress-relieved	45	75	18	—	Low alloy; boilers, pressure vessels

Table 2-15
(continued)

				Typical mechanical properties				
No.	Material	Nominal composition	Form and condition	Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
10	AISI 4140	C 0.40 Si 0.3 Cr 1.0 Mo 0.2 Mn 0.9	Fully-tempered Optimum properties	95 132	108 150	22 18	240 —	High strength; gears, shafts
11	12% Manganese steel	12% Mn C	Tempered 600°F Rolled and heat-treated stock	200 44	220 160	10 40	— 170	Machine tool parts; wear, abrasion-resistant
12	VASCO 300	Ni 18.5 Ti 0.6 Co 9.0 C 0.03 Mo 4.8	Solution treatment 1500°F; aged 900°F	110	150	18	—	Very high strength, maraging, good machining properties in annealed state
13	TI (AISI)	W 18.0 V 1.0 Cr 4.0 C 0.7	Quenched; tempered				R(c)	High speed tool steel, cutting tools, punches, etc.
14	M2 (AISI)	W 6.5 Mo 5.0 Cr 4.0 C 0.85 V 2.0	Quenched; tempered				65–66	M-grade, cheaper, tougher
15	Stainless steel type 304	Ni 9.0 C 0.08 Cr 19.0 max	Annealed; cold-rolled	35 to 160	85 to 185	60 8	160 to 400	General purpose, weldable; *nonmagnetic austenitic* steel
16	Stainless steel type 316	Cr 18.0 C 0.10 Ni 11.0 max Mo 2.5 Fe bal.	Annealed	30 to 120	90 to 150	50 8	165 275	For severe corrosive media, under stress; nonmagnetic austenitic steel
17	Stainless steel type 431	Cr 16.0 Si 1.0 Ni 2.0 C 0.20 Mn 1.0 Fe bal.	Annealed Heat-treated	85 150	120 195	25 20	250 400	Heat-treated stainless steel, with good mechanical strength; magnetic
18	Stainless steel 17 4 PH	Cr 17.0 Co 0.35 Ni 4.0 C 0.07 Cu 4.0 Fe bal.	Annealed	110	150	10	363	Precipitation hardening, heat-resisting type; retains strength up to approx. 600°F

No.	Material	Nominal composition	Form and condition	Typical mechanical properties: Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
	CAST IRONS AND CAST STEELS							
	These alloys are used where large and or intricate-shaped articles are required or where over-all dimensional tolerances are not critical. Thus the article can be produced with the fabrication and machining costs held to a minimum. Except for a few heat-treatable cast steels, this class of alloys does not demonstrate high-strength qualities.							
	CAST IRONS							
19	Cast gray iron ASTM A48 48, Class 25	C 3.4 Si 1.8 Mn 0.5	Cast (as cast)	—	25 min	0.5 max	180	Engine blocks, fly-wheels, gears, machine-tool bases
20	White	C 3.4 Si 0.7 Mn 0.6	Cast	—	25	0	450	
21	Malleable iron ASTM A47	C 2.5 Si 1.0 Mn 0.55	Cast (annealed)	33	52	12	130	Automotives, axle bearings, track wheels, crankshafts
22	Ductile or nodular iron (Mg-containing) ASTM A339 ASTM A395	C 3.4 P 0.1 Mn 0.40 max Ni 1% Mg 0.06 Si 2.5 Fe bal.	Cast Cast (as cast) Cast (quenched, tempered)	53 68 108	70 90 135	18 7 5	170 235 310	Heavy-duty machines, gears, cams, crankshafts
23	Ni-hard type 2	C 2.7 Si 0.6 Mn 0.5 Ni 4.5 Cr 2.0 Fe bal.	Sand-cast Chill-cast (tempered)	— —	55 75	— —	550 625	Strength, with heat- and corrosion-resistance
24	Ni-resist type 2	C 3.0 Si 2.0 Mn 1.0 Ni 20.0 Cr 2.5 Fe bal.	Cast (as cast)	—	27	2	140	
	CAST STEELS							
25	ASTM A27-62 (60-30)	C 0.3 Mn 0.6 Si 0.8 Ni 0.5 Cr 0.4 Mo 0.2		30	60	24	—	Low alloy, medium strength, general application
26	ASTM A148-60 (105-85)			85	105	17	—	High strength; structural application

Table 2-15
(continued)

No.	Material	Nominal composition	Form and condition	Typical mechanical properties: Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
27	Cast 12 Cr alloy (CA-15)	C 0.15 max, Mn 1.00 max Si 1.50 max, Cr 11.5–14 Ni 1.00 max, Fe bal.	Air-cooled from 1800°F; tempered at 600°F Air-cooled from 1800°F; tempered at 1400°F	150 75	200 100	7 30	390 185	Stainless, corrosion-resistant to mildly corrosive alkalis and acids
28	Cast 29 9 alloy (CE-30) ASTM A296 63T	C 0.30 max, Mn 1.50 max Si 2.00 max, Cr 26–30 Ni 8–11, Fe bal.	As cast	60	95	15	170	Greater corrosion resistance, especially for oxidizing condition
29	Cast 28 7 alloy (HD) ASTM A297-63T	C 0.50 max, Mn 1.50 max Si 2.00 max, Cr 26–30 Ni 4–7, Fe bal.	As cast	48	85	16	190	Heat-resistant
	SUPER ALLOYS							
	NICKEL BASE							
30	Hastelloy X	Co 1.5 max, Fe 18.5 Mo 9.0 Cr 22.0, C 0.15 max (wrought) W 0.6 C 0.20 max (cast), Ni bal.	Wrought sheet Mill-annealed As investment cast	52 — 46.5	113.2 67 —	43 17 —	194 172 —	
31	Hastelloy C	Cr 16.0, Fe 6.0 W 4.0, C 0.15 max Mo 17.0, Ni bal.	Sand-cast (annealed) Rolled (annealed) Investment cast	50 71 50	78 130 80	5 45 10	199 204 215	

The advent of engineering applications requiring high temperature and high strength, as in jet engines and rocket motors, has led to the development of a range of alloys collectively called super alloys. These alloys require excellent resistance to oxidation together with strength at high temperatures, typically 1800°F in existing engines. These alloys are continually being modified to develop better specific properties, and therefore entries in this group of alloys should be considered "fluid." Both wrought and casting-type alloys are represented. As the high temperature properties of cast materials improve, these alloys become more attractive, since great dimensional precision is now attainable in investment castings.

				Typical mechanical properties				
No.	Material	Nominal composition	Form and condition	Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
	NICKEL BASE (Cont.)							
32	Inconel 713C	Ni (+Co) bal. Mo 4.5 Al 6.0 Cr 13.0 Cb 2.0 Ti 0.6	Investment cast	102	120	6	—	
33	In 100	C 18.0 Mo 3.0 Al 55.0 V 1.0 Cr 10.0 Ti 4.7 Co 15.0	Cast					
34	Tax 8	C 125.0 Mo 4.0 W 4.0 Ta 8.0 Cr 6.0 Al 6.0 Zr 1.0 V 2.5	Cast					
35	Nimonic 90	Ni (+ Co) 57.00 Mn 0.50 S 0.007 Cu 0.05 Al 1.65 Co 16.90 C 0.05 Fe 0.45 Si 0.20 Cr 20.55 Ti 2.60	Annealed; wrought	90	155	—	260	General elevated temperature applications
36	Inconel X	Ni (+ Co) 72.85 Mn 0.65 S 0.007 Cu 0.05 Al 0.75 Cb (+ Ta) 0.85 C 0.04 Fe 6.80 Si 0.30 Cr 15.0 Ti 2.50	Annealed Annealed; age-hardened	50 115	115 175	50 25	150 300	
37	Waspaloy	C 0.08 Mo 4.3 Co 13.5 Cr 19.5 Ti 3.0	Cold-rolled	270	275	8	Rc 51	
38	Rene 41	C 0.09 Mo 10.0 Al 1.5 Cr 19.0 Ti 3.1 Co 11.0	Wrought	100	145	—	—	

Table 2-15 (continued)

No.	Material	Nominal composition	Form and condition	Typical mechanical properties: Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
39	Udimet 700	C 0.08 Cr 15.0 Mo 5.0 Ti 3.5 Al 4.3 Co 18.5	Cold-rolled	280	285	6	Rc 53	
40	T. D. Nickel	Ni 97.5 ThO_2 2.4	Extended and cold-worked	85	100	13	—	High temperature; jet engine parts
	COBALT BASE							
41	Haynes Stellite alloy 25 (L605)	C 0.15 max Cr 20.0 W 15.0 Ni 10.0 Co bal. Mn 1.5	Wrought sheet; mill annealed	63	140	60	244	Wrought products
42	Haynes Stellite alloy 21 AMS 5385 (cast)	C 0.25 Mo 5.5 Ni 2.5 Co bal. Cr 28.5	As investment cast	82	103	8	313 max	For castings

ALUMINUM ALLOYS

Although the strength of aluminum alloys is in general less than that attainable in ferrous alloys or copper-base alloys, their major advantage lies in their high strength-to-weight ratio due to the low density of aluminum. Aluminum alloys have good corrosion resistance for most applications except in alkaline solutions.

No.	Material	Nominal composition	Form and condition	Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
43	3003 ASTM B221	Cu 0.12 Al bal. Mn 1.2	Annealed-O Cold-rolled-H14 Cold-rolled-H18	6 21 27	16 22 29	40 16 10	28 40 55	Good formability, weldable, medium strength; chemical equipment
44	2017 ASTM B221	Mn 0.5 Mg 0.5 Cu 4.0 Al bal.	Annealed-O Heat-treated-T4	10 40	26 62	22 22	45 105	High strength; structural parts, aircraft, heavy forgings
45	2024 ASTM B211	Cu 4.5 Mg 1.5 Mn 0.6 Al bal.	Heat-treated-T4	47	68	19	120	
46	5052 ASTM B211	Cr 0.25 Al bal. Mg 2.5	Annealed-O Cold-rolled and stabilized-H34	13 31	28 38	30 14	47 68	Medium strength, good fatigue properties; street-light standards
47	ASTM B209		Cold-rolled and stabilized-H38	37	42	8	77	

				Typical mechanical properties				
No.	Material	Nominal composition	Form and condition	Yield strength (0.2% offset), 1000 lb/sq in.	*Tensile* strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
48	7075	Cu 1.6 Mg 2.5	Annealed-O	15	33	17	60	High strength, good
	ASTM B211	Cr 0.3 Al bal.	Heat-treated and					corrosion resistance
		Zn 5.6	artificially aged-T6	73	83	11	150	
49	380	Si 9.0 Al bal.	Die-cast	24	48	3	—	General purpose die-casting
	ASTM SC84B	Cu 3.5						
50	195	Si 0.8 Al bal.	Sand-cast: heat-treated-T4	16	32	8.5	60	Structural elements, aircraft,
	ASTM C4A	Cu 4.5	Sand-cast: heat-treated and					and machines
			artificially aged-T6	24	36	5	75	
51	214	Mg 3.8 Al bal.	Sand-cast-F	12	25	9	50	Chemical equipment, marine
	ASTM G4A							hardware, architectural
52	220	Mg 10.0 Al bal.	Sand-cast: heat-treated-T4	26	48	16	75	Strength with shock
	ASTM G10A							resistance: aircraft

COPPER ALLLOYS

Because of their corrosion resistance and the fact that copper alloys have been used for many thousands of years, the number of copper alloys available is second only to the ferrous alloys. In general copper alloys do not have the high-strength qualities of the ferrous alloys, while their density is comparable. The cost per strength-weight ratio is high; however, they have the advantage of ease of joining by soldering, which is not shared by other metals that have reasonable corrosion resistance.

53	Copper	Cu 99.9	Annealed	10	32	45	42	Bus-bars, switches,
	ASTM B152	plus	Cold-drawn	40	45	15	90	architectural, roofing,
	ASTM B124, R133		Cold-rolled	40	46	5	100	screens
	ASTM B1, B2, B3							
54	Gilding metal	Cu 95.0 Zn 5.0	Cold-rolled	50	56	5	114	Coinage, ammunition
	ASTM B36							
55	Cartridge 70-30 brass	Cu 70.0 Zn 30.0	Cold-rolled	63	76	8	155	Good cold-working
	ASTM B14							properites: radiator covers,
	ASTM B19							hardware, electrical
	ASTM B36							
	ASTM B134							
	ASTM B135							
56	Phosphor bronze 10%	Cu 90.0 Sn 10.0	Spring temper	—	122	1	241	Good spring qualities,
	ASTM B103	P 0.25						high-fatigue strength
	ASTM B139							
	ASTM B159							

Table 2-15 (continued)

No.	Material	Nominal composition	Form and condition	Typical mechanical properties: Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
57	Yellow brass (high brass) ASTM B36 ASTM B134 ASTM B135	Cu 65.0 Zn 35.0	Annealed Cold-drawn Cold-rolled (HT)	18 55 60	48 70 74	60 15 10	55 115 180	Good corrosion resistance: plumbing, architectural
58	Manganese bronze ASTM B138	Cu 58.5 Zn 39.2 Fe 1.0 Sn 1.0 Mn 0.3	Annealed Cold-drawn	30 50	60 80	30 20	95 180	Forgings
59	Naval brass ASTM B21	Cu 60.0 Zn 39.25 Sn 0.75	Annealed Cold-drawn	22 40	56 65	40 35	90 150	Condensor tubing: high resistance to salt-water corrosion
60	Muntz metal ASTM B111	Cu 60.0 Zn 40.0	Annealed	20	54	45	80	Condensor tubes: valve stress
61	Aluminum bronze ASTM B169, alloy A ASTM B124 ASTM B150	Cu 92.0 Al 8.0	Annealed Hard	25 65	70 105	60 7	80 210	
62	Beryllium copper 25 ASTM B194 ASTM B197 ASTM B196	Be 1.9 Cu bal. Co or Ni 0.25	Annealed, solution-treated Cold-rolled Cold-rolled	32 104 70	70 110 190	45 5 3	B60 (Rockwell) B81 C40	Bellows, fuse clips, electrical relay parts, valves, pumps
63	Free-cutting brass	Cu 62.0 Zn 35.5 Pb 2.5	Cold-drawn	44	70	18	B80 (Rockwell)	Screws, nuts, gears, keys
64	Nickel silver 18" Alloy A (wrought) ASTM B122, No. 2	Cu 65.0 Zn 17.0 Ni 18.0	Annealed Cold-rolled Cold-drawn wire	25 70	58 85 105	40 4	70 170	Hardware, optical goods, camera parts
65	Nickel silver 13" (cast) 10A ASTM B149, No. 10A	Ni 12.5 Pb 9.0 Sn 2.0 Cu bal. Zn 20.0	Cast	18	35	15	55	Ornamental castings, plumbing: good machining qualities
66	Cupronickel 10" ASTM B111 ASTM B171	Cu 88.35 Ni 10.0 Fe 1.25 Mn 0.4	Annealed Cold-drawn tube	22 57	44 60	45 15	— —	Condensor, salt-water piping

				Typical mechanical properties				
No.	Material	Nominal composition	Form and condition	Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
67	Cupronickel	Cu 70.0 Ni 30.0	Wrought					Heat-exchanger process equipment, valves
68	Red brass (cast) ASTM B30, No. 4A	Cu 85.0 Zn 5.0 Pb 5.0 Sn 5.0	As-cast	17	35	25	60	
69	Silicon bronze ASTM B30, alloy 12A	Si 4.0 Fe 2.0 Zn 4.0 Al 1.0 Mn 1.0	Castings					Cheaper substitute for tin bronze
70	Tin bronze ASTM B30, alloy 1B	Sn 8" Zn 4.0	Castings					Bearings, high-pressure bushings, pump impellers
71	Navy bronze		Cast					
			TIN AND LEAD-BASE ALLOYS					
	Major uses for these alloys are as "white" metal bearing alloys, extruded cable sheathing, and solders. Tin forms the basis of pewter used for culinary applications.							
72	Lead-base Babbitt ASTM B23, alloy 19	Pb 85.0 Sn 5.0 Sb 10.0 As 0.6 Cu 0.5	Chill cast	—	10	5	19	Bearings, light loads and low speeds
73	Arsenical-lead Babbitt ASTM B23, alloy 15	Pb 83.0 Sn 1.0 Sb 16.0 As 1.1 Cu 0.6	Chill cast	—	10.3	2	20	Bearings, high loads and speeds, diesel engines, steel mills
74	Chemical lead	Pb 99.9 Cu 0.06 Bi 0.005 max	Rolled 95%	1.9	2.5	50	5	
75	Antimonial lead (hard lead)	Pb 94.0 Sb 6.0	Chill cast Rolled 95%	— —	6.8 4.1	22 47	(500 kg) 9	Good corrosion resistance and strength
76	Calcium lead	Pb 99.9 Ca 0.025 Cu 0.10	Extruded and aged	—	4.5	25	—	Cable sheathing, creep-resistant pipe
77	Tin Babbitt alloy ASTM B23-61, grade 1	Sb 4.5 Sn bal. Cu 4.5	Chill cast	—	9.3	2	17	General bearings and die-casting
78	Tin die-casting alloy ASTM B102-52	Sb 13.0 Sn bal. Cu 5.0	Die-cast	—	10	1	29	Die-casting alloy

Table 2-15 (continued)

No.	Material	Nominal composition	Form and condition	Typical mechanical properties: Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
79	Pewter	Sn 91.0 Sb 7.0 Cu 2.0	Rolled sheet, annealed	—	8.6	40	9.5	Ornamental and household items
80	Solder 50 50	Sn 50.0 Pb 50.0	Cast	4.8	6.1	60	14	General-purpose solder
81	Solder	Sn 20.0 Pb 80.0	Cast	3.6	5.8	16	11	Coating and joining, tilling seams on automobile bodies

MAGNESIUM ALLOYS

Because of their low density these alloys are attractive for use where weight is at a premium. The major drawback to the use of these alloys is their ability to ignite in air (this can be a problem in machining); they are also costly. Magnesium alloys are used in both the wrought and die-cast forms, the latter being the most frequently used form.

No.	Material	Nominal composition	Form and condition	Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
82	Magnesium alloy AZ31B	Zn 1.0 Mn 0.20 Al 3.0 min. Mg bal.	Rolled-plate (strain-hardened, then partially annealed)	24	37	18	—	Structural applications of medium strength
			Rolled-sheet (strain-hardened, then partially annealed)	32	42	15	73	
			Annealed	22	37	21	56	
			Extruded	28	38	14	—	
83	Magnesium alloy AZ80A	Zn 0.5 Mn 0.15 Al 8.5 min Mg bal.	Extruded	36	49	11	60	General extruded and forged products
			Extruded (age-hardened)	39	53	6	82	
			Forged (age-hardened)	34	50	6	72	
84	Magnesium alloy AZ92A	Zn 2.0 Mn 0.10 Al 9.0 min Mg bal.	Sand-cast (as cast)	14	24	6	50	
			Sand-cast (solution heat-treated)	14	40	12	55	
			Sand-cast (solution heat-treated and aged)	19	40	5	83	
			Sand-cast (age-hardened)	16	30	18	—	
			Sand-cast and tempered	22	40	3	81	Pressure-tight sand and permanent mold castings: high UTS and good yield strength
85	Magnesium alloy ZK60A	Zn 5.7 Mg bal. Zr 0.55	Extruded	43	52	12	82	

No.	Material	Nominal composition	Form and condition	Typical mechanical properties: Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
86	Magnesium alloy	Zn 0.6 Mn 0.13	Die-cast (as cast)	22	33	3	67	General die-casting
	AZ91A and AZ91B	Al 9.0 min						applications
		Mg bal.						
			BERYLLIUM					
87	Beryllium			27	33	1–3	—	Windows, X-ray tubes
			Hot-pressed	38	51			
			Cross-rolled	40	60	10–40	—	Moderator- and reflector-cladding nuclear reactors; heat-shield and structural-member missiles
				60	90			

NICKEL ALLOYS

Nickel and its alloys are expensive and used mainly either for their high-corrosion resistance *in many environments or for high-temperature* and strength applications. (See Super Alloys, above.)

No.	Material	Nominal composition	Form and condition	Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
88	Nickel (cast)	Ni 95.6 Cu 0.5 Fe 0.5 Mn 0.8 Si 1.5 C 0.8	As cast	25	57	22	110	Good corrosion-resistance applications
89	K Monel	Ni (+Co) 65.25 C 0.15	Annealed	45	100	40	155	High strength and corrosion resistance; aircraft parts, valve stems, pumps
		Mn 0.60 Fe 1.00	Annealed, age-hardened	100	155	25	270	
		S 0.005 Si 0.15	Spring	140	150	5	300	
		Cu 29.60 Al 2.75	Spring, age-hardened	160	185	10	335	
		Ti 0.45 —						
90	A nickel	Ni (+Co) 99.40 C 0.06	Annealed	20	70	40	100	Chemical industry for resistance to strong alkalis, plating nickel
	ASTM B160	Fe 0.15	Hot-rolled	25	75	40	110	
	ASTM B161	Mn 0.25 Si 0.05	Cold-drawn	70	95	25	170	
	ASTM B162	S 0.005	Cold-rolled	95	105	5	210	
		Cu 0.05						
91	Duranickel	Ni (+Co) 93.90 C 0.15	Annealed	45	100	40	160	High strength and corrosion resistance; pump rods, shafts, springs
		Fe 0.15	Annealed, age-hardened	125	170	25	330	
		Mn 0.25 Si 0.55	Spring	—	175	5	320	
		S 0.005 Al 4.50	Spring, age-hardened	—	205	10	370	
		Cu 0.05						
		Ti 0.45						

Table 2-15 (continued)

No.	Material	Nominal composition	Form and condition	Typical mechanical properties: Yield strength (0.2% offset), 1000 lb/sq in.	Tensile strength, 1000 lb/sq in.	Elongation, in 2 in., %	Hardness, Brinell	Comments
92	Cupronickel 55-45 (Constantan)	Cu 55.0 Ni 45.0	Annealed Cold-drawn Cold-rolled	30 50 65	60 65 85	45 30 20	— — —	Electrical-resistance wire; low temperature coefficient, high resistivity
93	Nichrome	Ni 80.0 Cr 20.0						Heating elements for furnaces
94	"S" Monel	Ni 60.0 Cu 29.0 Fe 2.50 max Mn 1.5 max Si 4.0 Al 0.5 max	Sand-casting	80-115	110-145	2	270-350	High-strength casting alloy; good bearing properties for valve seats
	TITANIUM ALLOYS The main application for these alloys is in the aerospace industry. Because of the low density and high strength of titanium alloys, they present excellent strength-to-weight ratios.							
95	Commerical titanium ASTM B265-58T	Ti 99.4	Annealed at 1100 to 1350°F (593 to 732°C)	70	80	20	—	Moderate strength, excellent fabricability; chemical industry pipes
96	Titanium alloy ASTM B265 58T 5 Ti 6 Al 4V		Water-quenched from 1750°F (954°C); aged at 1000 F (538 C) for 2 hr	160	170	13	—	High-temperature strength needed in gas-turbine compressor blades
97	Titanium alloy Ti 4 Al 4Mn		Water-quenched from 1450°F (788°C); aged at 900°F (482°C) for 8 hr	170	185	13	—	Aircraft forgings and compressor parts
98	Ti Mn alloy ASTM B265 58T 7	Fe 0.5 Ti bal. Mn 7.0 8.0	Sheet	140	150	18	—	Good formability, moderate high-temperature strength; aircraft skin
	ZINC ALLOYS A major use for these alloys is for low-cost die-cast products such as household fixtures, automotive parts, and trim.							
99	Zinc ASTM B69	Cd 0.35 Zn bal. Pb 0.08	Hot-rolled	—	19.5	65	38	Battery cans, grommets lithographer's sheet
100	Zilloy-15	Cu 1.00 Zn bal. Mg 0.010	Hot-rolled Cold-rolled	— —	29 36	20 25	61 80	Corrugated roofs, articles with maximum stiffness
101	Zilloy 40	Cu 1.00 Zn bal.	Hot-rolled Cold-rolled	— —	24 31	50 40	52 60	Weatherstrip, spun articles
102	Zamac-5 ASTM 25	Zn (99.99% pure remainder) Al 3.5–4.3 Cu 0.75 1.25 Mg 0.03-0.08	Die-cast	—	47.6	7	91	Die-casting for automobile parts, padlocks; used also for die material
	ZIRCONIUM ALLOYS These alloys have good corrosion resistance but are easily oxidized at elevated temperatures in air. The major application is for use in nuclear reactors.							
103	Zirconium, commercial	O_2 0.07 C 0.15 Hf 1.90 Zr bal.	Annealed	40	65	27	B80 (Rockwell)	
104	Zircaloy-2	Hf 0.02 Ni 0.05 Fe 0.15 Other 0.25 Sn 1.46 Zr bal.	Annealed	50	75	22	B90 (Rockwell)	Nuclear power-reactor cores at elevated temperatures

Table 2-16
Typical Properties of Glass-Fiber-Reinforced Resins [10]

Property	Base resin: Polyester	Phenolic	Epoxy	Melamine	Polyurethane
Molding quality	Excellent	Good	Excellent	Good	Good
Compression molding					
Temperature, °F	170 to 320	280 to 350	300 to 330	280 to 340	300 to 400
Pressure, psi	250 to 2000	2000 to 4000	300 to 5000	2000 to 8000	100 to 5000
Mold shrinkage, in./in.	0.0 to 0.002	0.0001 to 0.001	0.001 to 0.002	0.001 to 0.004	0.009 to 0.03
Specific gravity	1.35 to 2.3	1.75 to 1.95	1.8 to 2.0	1.8 to 2.0	1.11 to 1.25
Tensile strength, 1000 psi	25 to 30	5 to 10	14 to 30	5 to 10	4.5 to 8
Elongation, %	0.5 to 5.0	0.02	4		10 to 650
Modulus of elasticity, 10^{-5} psi	8 to 20	33	30.4	24	
Compression strength, 1000 psi	15 to 30	17 to 26	30 to 38	20 to 35	20
Flexural strength, 1000 psi	10 to 40	10 to 60	20 to 26	15 to 23	7 to 9
Impact, Izod, ft-lb/in. or notch	2 to 10	10 to 50	8 to 15	4 to 6	No break
Hardness, Rockwell	M70 to M120	M95 to M100	M100 to M108		M28 to R60
Thermal expansion, per °C	2 to 5×10^{-5}	1.6×10^{-5}	1.1 to 3.0×10^{-5}	1.5×10^{-5}	10 to 20×10^{-5}
Volume resistivity at 50% RH, 23°C, ohm-cm	1×10^{14}	7×10^{12}	3.8×10^{15}	2×10^{11}	2×10^{11} to 10^{14}
Dielectric strength, ⅛ in. thickness, v/mil	350 to 500	140 to 370	360	170 to 300	330 to 900
Dielectric constant					
At 60 hz	3.8 to 6.0	7.1	5.5	9.7 to 11.1	5.4 to 7.6
At 1 khz	4.0 to 6.0	6.9			5.6 to 7.6
Dissipation factor					
At 60 hz	0.01 to 0.04	0.05	0.087	0.14 to 0.23	0.015 to 0.048
At 1 khz	0.01 to 0.05	0.02			0.043 to 0.060
Water absorption, %	0.01 to 1.0	0.1 to 1.2	0.05 to 0.095	0.09 to 0.21	0.7 to 0.9
Sunlight (change)	Slight	Darkens	Slight	Slight	None to slight
Chemical resistance	Fair**	Fair**	Excellent	Very good†	Fair
Machining qualities	Good		Good	Good	Good

Note: Filament-wound components with high glass content, highly oriented, have higher strengths. The decreasing order of tensile strength is: roving, glass cloth, continuous mat, and chopped-strand mat.

**Attacked by strong acids or alkalies.
†Attacked by strong acids.
*From "Reinforced Thermosets," C.A. Spang and G.J. Davis. *Machine Design*, 40(29): 32. Dec. 12. 1968.

(text continued from page 194)

Torsion:

$$J = \pi/2(6.05^4 - 6.00^4) = 68.71 \text{ in.}^4$$

$$\tau_{r\theta} = \frac{10{,}000 \text{ ft-lb} \mid 12 \text{ in.} \mid 6.05 \text{ in.}}{\mid \text{ft} \mid 68.71 \text{ in.}^4} = 10{,}566 \text{ psi}$$

Effective stress:

$$\sigma_e = \left\{\frac{1}{2}\left[(60{,}000 - 30{,}000)^2 + (60{,}000 - 0)^2 + (30{,}000 - 0)^2 + 6(10{,}566)^2\right]\right\}^{1/2}$$

$$= 55{,}090 \text{ psi}$$

From Table 2-15, the yield strength for AISI SAE 1035, cold rolled is 67,000 psi. Thus the factor of safety is

$$SF = \frac{67{,}000}{55{,}090} = 1.22$$

For further information on this subject refer to Reference 1 and References 10–14.

Table 2-17
Allowable Unit Stresses for Lumber [10]

SPECIES, SIZES, ALLOWABLE STRESSES, AND MODULUS OF ELASTICITY

Normal Loading Conditions: Moisture Content Not Over 19 Percent

Species and grades (visual grading)*	Sizes, nominal	Typical grading agency, 1968*	Allowable unit stresses, psi*: Extreme fiber in bending	Tension parallel to grain	Compression perpendicular	Compression parallel	Modulus of elasticity, psi
Idaho white pine	2 × 4	W	850	500	240	1,050	1,120,000
	2 × 6 and wider		1,200	800	240	1,100	1,400,000
Ponderosa pine	2 × 4	W	850	500	280	1,000	950,000
	2 × 6 and wider		1,150	800	280	1,000	1,190,000
Lodgepole pine	2 × 4	W	1,400	850	250	1,100	1,030,000
	4 × 4		1,350	800	240	1,050	1,000,000
Southern pine	2 × 4	S	1,810	1,190	405	1,190	1,800,000
	4 × 4	S	1,810	1,190	405	1,300	1,700,000
Douglas fir	2 × 4	W	1,700	1,000	385	1,510	1,800,000
	2 × 6 and wider	W	1,900	1,250	385	1,800	1,810,000
Western hemlock	2 × 4	W	1,450	850	245	1,350	1,210,000
	2 × 6 and wider		1,650	1,100	245	1,450	1,520,000
Western spruce	2 × 4	W	1,150	650	220	950	920,000
	2 × 6 and wider		1,050	700	220	1,000	1,150,000
Western cedar	2 × 4	W	850	500	295	1,150	860,000
	2 × 6 and wider		1,200	800	295	1,150	1,070,000
Redwood (unseasoned)	2" and 4" "Construction"	R	1,640	—	305	1,190	1,240,000

Note: Allowable unit stresses in horizontal shear are in the range of 75 to 150 psi.

*There is no single grade designation that applies to all lumber. Values in the table apply approximately to "No. 1," although this designation is often modified by terms such as dense or dry. For grades better than No. 1, such terms as structural, heavy, select, dense, etc. are used. Lower grades are No. 2. No. 3 factory, light industrial, etc., but there are seldom more than four grades of a single size in a given species. The allowable stresses are for "repetitive member" users.

*Most lumber is graded by the following agencies, although there are other grading organizations.
W = Western Wood Products Association
S = Southern Pine Inspection Bureau
R = Redwood Inspection Service

*Load applied to joists or planks. For beam or stringer grades, stresses are for load applied to the narrow face.

*For engineered uses the allowable stresses are slightly lower; for kiln-dried lumber slightly higher. For short-term loads, such as wind, earthquake, or impact, higher unit stresses are allowed

REFERENCES

"Wood Handbook," U.S. Department of Agriculture Handbook No. 72, 1955.
"Timber Construction Manual," American Institute of Timber Construction, John Wiley & Sons, 1966.
"National Design Specification for Stress-Grade Lumber," National Forest Products Association, Washington D.C., 1968.

Table 2-18
Physical, Mechanical, and Thermal Properties of Common Stones [1]

Type of Stone	Density, lb per cu ft	Compressive strength × 10^{-3}, psi	Rupture modulus × 10^{-3}, psi (ASTM C99 52)	Shearing strength × 10^{-3}, psi	Young's modulus × 10^{-6}, psi	Modulus of rigidity × 10^{-6}, psi	Poisson's ratio	Abrasion-hardness index (ASTM C241-51)	Porosity, volume percent	48-hr water absorption (ASTM C97-47)	Thermal conductivity, Btu per ft per hr per deg F	Coefficient of thermal expansion × 10^{-6}, per deg F
Granite	160–190	13–55	1.4–5.5	3.5–6.5	4–16	2–6	0.05–0.2	37–88	0.6–3.8	0.02–0.58	20–35	3.6–4.6
Marble	165–179	8–27	0.6–4.0	1.3–6.5	5–11.5	2–4.5	0.1–0.2	8–42	0.4–2.1	0.02–0.45	8–36	3.0–8.5
Slate	168–180	9–10	6–15	2.0–3.6	6–16	2.5–6	0.1–0.3	6–12	0.1–1.7	0.01–0.6	12–26	3.3–5.6
Sandstone	119–168	5–20	0.7–2.3	0.3–3.0	0.7–10	0.3–4	0.1–0.3	2–26	1.9–27.3	2.0–12.0	4–40	3.9–6.7
Limestone	117–175	2.5–28	0.5–2.0	0.8–3.6	3–9	1–4	0.1–0.3	1–24	1.1–31.0	1.0–10.0	20–32	2.8–4.5

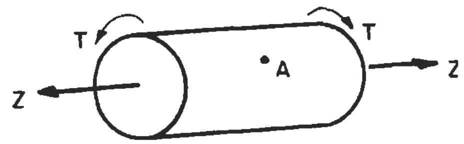


Figure 2-30. Diagram for Example 2-21.

THERMODYNAMICS

Thermodynamics centers around the concept of "energy in transit," but is considerably more encompassing in its applications. The science of thermodynamics deals very broadly with the concepts of how things work, why some things cannot work, and why some things do not work as intended. Three "laws" of thermodynamics have been formulated, which can be summarized as follows:

1. *The First Law of Thermodynamics:* A statement of the principle of conservation of energy.
2. *The Second Law of Thermodynamics:* Deals with the concept of entropy, which serves as a means of determining whether or not a process is possible.
3. *The Third Law of Thermodynamics:* Defines the zero entropy state for any substance in a single, pure quantum state as the absolute zero of temperature.

Units of Energy

Measurements of energy are made in terms of absolute joules, but engineering practice has persistently retained the thermochemical calorie as the unit of energy. The two are related by the definition:

$$1 \text{ calorie} = 4.1840 \text{ absolute joules} \tag{2-101}$$

which is often referred to as the mechanical equivalent of heat.

The following table expresses the relationship between several other useful energy units and the calorie:

1 Btu = 252.16 cal
1 kWhr = 8.6056×10^5 cal
1 hp-hr = 6.4162×10^5 cal
1 ft(lb_f) = 0.324 cal
1 ℓ-atm = 24.218 cal
A = work function (Helmholtz free energy), Btu/lb_m or Btu
C = heat capacity, Btu/lb_m°R
C_p = heat capacity at constant pressure
C_v = heat capacity at constant volume
F = (Gibbs) free energy, Btu/lb_m or Btu
g = acceleration due to gravity = 32.174 ft/s^2
g_c = conversion factor between force and mass = 32.174 (lb_m)(ft/s^2)/lb_f
h, H = enthalpy or heat content, Btu/lb_m or Btu
κ = ratio C_p/C_v
Mw = molecular weight, lb_m/lb_m-mole
m, M = mass of fluid, lb_m
$\dot{m}$, $\dot{M}$ = mass flow rate, lb_m/s

P = absolute pressure, lb_f/ft^2
$\dot{P}_s$ = entropy production rate, Btu/°R•s
Q = heat transferred to system across a system boundary, Btu/lb_m or Btu
$\dot{Q}$ = rate of heat transfer, Btu/s
R = universal gas constant, $lb_f\text{-}ft^3$/mole•°R
s, S = entropy, Btu/lb_m°R or Btu/°R
T = absolute temperature, °R
u, U = internal energy, Btu/lb_m or Btu
V = volume, ft^3/lb_m or ft^3
v = flow velocity, ft/s
W = work done by a system against its surroundings, Btu/lb_m or Btu
Z = height from center of gravity of a fluid mass to a fixed base level, ft

The First Law of Thermodynamics

The differrential form of the first law as applied to a *closed* system, for which there is no exchange of matter between the system and its surroundings, is given by

$$dU = \delta Q - \delta W \tag{2-102}$$

where dU represents an infinitesimal increase in the internal energy of the system, δQ is the heat absorbed by the system from its surroundings, and δW is the work done by the system on its surroundings. The state of a system is defined by its temperature, pressure, specific volume, and chemical composition. The change in internal energy expressed by Equation 2-102 depends only upon the difference between the final and initial states and not upon the process or processes that occurred during the change. The heat and work terms, on the other hand, are dependent upon the process path. For a change from a state A to a state B, the first law becomes

$$\Delta U = U_B - U_A = Q - W \tag{2-103}$$

Work interchange between a system and its surroundings can take on any of a variety of forms including mechanical shaft work, electrical work, magnetic work, surface tension, etc. For many applications, the only work involved is that of compression or expansion against the surroundings, in which case the work term in Equation 2-102 becomes

$$\delta W = PdV$$

or

$$W = \int_{V_A}^{V_B} P\, dV \tag{2-104}$$

where V_B is the final volume and V_A the initial volume of the system, and P is the system pressure. Thus, for a *constant pressure* process:

$$W = P\Delta V = P(V_B - V_A) \quad \text{(constant pressure process)} \tag{2-105}$$

or, combining Equations 2-103 and 2-104:

$$\Delta U = U_B - U_A = Q - PV_B + PV_A \tag{2-106}$$

or

$$Q = (U_B + PV_B) - (U_A + PV_A) \tag{2-107}$$

The combination of properties (U + PV) occurs so frequently in thermodynamics that it is given a special symbol, H, and termed the "enthalpy" or "heat content" of the system. Thus Equation 2-107 can be written as

$$Q = \Delta(U + PV) = H_B - H_A = \Delta H \text{ [constant pressure process]} \tag{2-108}$$

Enthalpy is a *property* of the system independent of the path selected. Processes can be conveniently represented graphically. For example, a P–V diagram can be used to illustrate the work done when a system undergoes a change in state (see Figure 2-31). In each of the cases depicted in Figure 2-31, the work is equal to the shaded area under the P–V curve as shown.

Since the mass is fixed for a closed system, the equations in this discussion will be valid for the entire mass (M) or on a unit mass basis.

The First Law of Thermodynamics Applied to Open Systems

An open system is one which exchanges mass with its surroundings in addition to exchanging energy. For open systems, the first law is formulated from a consideration of the conservation of energy principle which can be stated as follows:

$$\begin{pmatrix}\text{Net increase}\\ \text{of stored energy}\\ \text{of system}\end{pmatrix} = \begin{pmatrix}\text{Stored energy}\\ \text{of mass}\\ \text{entering}\end{pmatrix} - \begin{pmatrix}\text{Stored energy}\\ \text{of mass}\\ \text{leaving}\end{pmatrix} + \begin{pmatrix}\text{Net energy}\\ \text{entering as}\\ \text{heat and all}\\ \text{forms of work}\end{pmatrix}$$

Consider the arbitrary open thermodynamic system illustrated in Figure 2-32. The foregoing statement of the first law for this open system can be written as

$$m_f U_f - m_i U_i + \frac{m_f v_f^2 - m_i v_i^2}{2g_c} + \frac{g}{g_c}(m_f Z_f - m_i Z_i)$$

$$= \int\left(H_1 + \frac{v_1^2}{2g_c} + \frac{g}{g_c}Z_1\right)\delta m_1 - \int\left(H_2 + \frac{v_2^2}{2g_c} + \frac{g}{g_c}Z_2\right)\delta m_2 + Q - W \tag{2-109}$$

where δm refers to a differential mass of fluid, and the subscripts f and i refer to the entire system in its final state and initial state, respectively. Clearly, for a closed system defined as one which exchanges no mass with its surroundings, Equation 2-109 reduces to Equation 2-103.

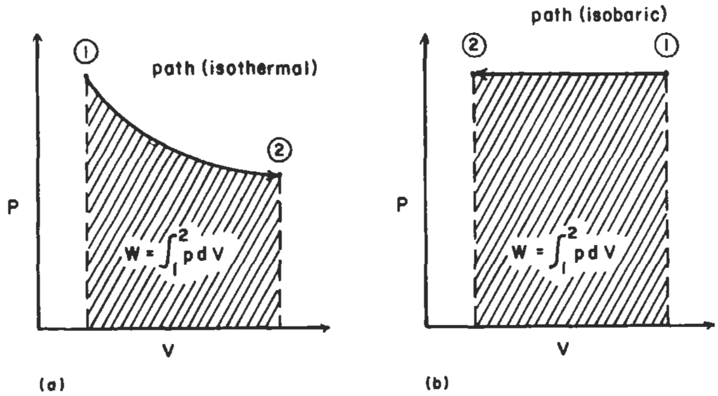


Figure 2-31. P–V process diagrams: (a) isothermal expansion; (b) isobaric compression.

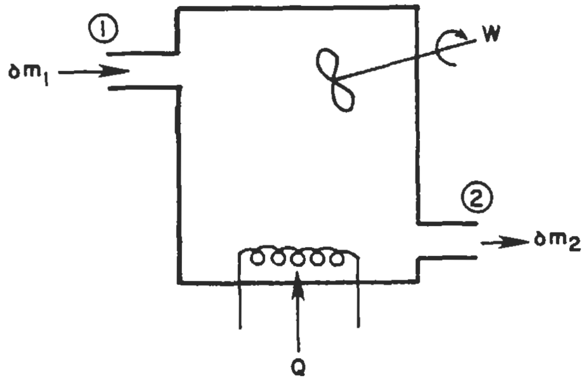


Figure 2-32. An open thermodynamic system.

For an open system at *steady state,* as in the case of turbines, compressors, pumps, etc., Equation 2-109 can be written (for unit mass flow rate) as

$$\underset{\substack{\text{Increase}\\ \text{in enthalpy}}}{\Delta H} + \underset{\substack{\text{Increase}\\ \text{in kinetic}\\ \text{energy}}}{\frac{\Delta v^2}{2g_c}} + \underset{\substack{\text{Increase}\\ \text{in potential}\\ \text{energy}}}{\frac{g}{g_c}\Delta Z} = \underset{\substack{\text{Net heat}\\ \text{exchanged with}\\ \text{surroundings}}}{Q} - \underset{\substack{\text{Net work}\\ \text{exchanged with}\\ \text{surroundings}}}{W_{net}} \tag{2-110}$$

where W_{net} is the net useful work (or shaft work) done by the fluid.

Example 2-22. Isobaric Compression of an Ideal Gas

One pound-mole of an ideal gas is compressed at a constant pressure of 1 atm in a piston-like device from an initial volume of 1.5 ft^3 to a final volume of 0.5 ft^3. The internal energy is known to decrease by 20 Btu. How much heat was transferred to or from the gas?

Solution

This is an isobaric (constant pressure) process in a closed system. Equation 2-103 applies.

$$Q = \Delta U + W = \Delta U + \int_{V_1}^{V_2} P\,dV$$

$$= -20 \text{ Btu} + P(V_2 - V_1)$$

$$= -20 \text{ Btu} + (1 \text{ atm})\left(\frac{14.7 \text{ lb}_f}{\text{in.}^2 \text{ atm}}\right)\left(\frac{144 \text{ in.}^2}{\text{ft}^2}\right)(0.5\text{ft}^3 - 1.5\text{ft}^3)\left(\frac{1 \text{ Btu}}{778 \text{ ft} \bullet \text{lb}_f}\right)$$

$$= -20 \text{ Btu} - 2.72 \text{ Btu}$$

$$= -22.72 \text{ Btu}$$

Because Q is negative, 22.72 Btus of heat were transferred from the gas to its surroundings.

Example 2-23. Hydroelectric Power System

A hydroelectric power plant proposes to use 1,500 ft^3/s of river water to generate electricity. The water enters the system at 1 atm and 50°F and is discharged at 1 atm and 50.4°F after passing through a turbine generator. The discharge point is 600 ft below the inlet. The increase in enthalpy of the water is known to be 0.36 Btu/lb_m. Assuming 70% efficiency for the conversion, what power output can be expected from the power plant?

Solution

The following assumptions pertain to this *open* system:

1. Steady-state flow.
2. No heat transferred between system and surroundings.
3. Change in kinetic energy of the flow streams is negligible.

With these assumptions, we take as a reference point the discharge level of the water, and apply Equation 2-110. Thus $Z_2 = 0$, $Z_1 = 600$ ft and the energy balance becomes

$$\Delta H + \frac{g}{g_c}\Delta Z = -W_{net}$$

$$\text{where } \Delta H = \left(0.36\frac{\text{Btu}}{\text{lb}_m}\right)\left(\frac{1{,}500\ \text{ft}^3}{\text{s}}\right)\left(\frac{62.4\ \text{lb}_m}{\text{ft}^3}\right)\left(\frac{3{,}600\ \text{s}}{\text{hr}}\right)$$

$$= 1.213\times10^8\ \text{Btu/hr}$$

$$= 35{,}553\ \text{kW}$$

$$\text{and } \frac{g}{g_c}\Delta Z = \left(\frac{32.2\frac{\text{ft}}{\text{s}^2}}{32.2\frac{\text{ft}\bullet\text{lb}_m}{\text{lb}_f\bullet\text{s}^2}}\right)(0-600\ \text{ft})\left(\frac{1{,}500\ \text{ft}^3}{\text{s}}\right)\left(\frac{62.4\ \text{lb}_m}{\text{ft}^3}\right)\left(\frac{3{,}600\ \text{s}}{\text{hr}}\right)$$

$$= -2.599\times10^8\ \text{Btu/hr}$$

$$= -76{,}163\ \text{kW}$$

Therefore,

$$W_{net} = -35{,}553\ \text{kW} + 76{,}163\ \text{kW} = 40{,}610\ \text{kW}$$

At 70% efficiency, this would yield

$$W_{actual} = (0.70)(40{,}610) = 28{,}427\ \text{kW} = 28.427\ \text{mega-Watts.}$$

Entropy and the Second Law

The second law of thermodynamics provides a basis for determining whether or not a process is possible. It is concerned with availability of the energy of a given system for doing work. All natural systems proceed towards a state of equilibrium and, during any change process, useful work can be extracted from the system. The property called *entropy*, and given the symbol S or s, serves as a quantitative measure of the extent to which the energy of a system is "degraded" or rendered unavailable for doing useful work.

For any *reversible process*, the sum of the changes in entropy for the system and its surroundings is zero. All natural or *real* processes are *irreversible* and are accompanied by a net *increase* in entropy.

Several useful statements have been formulated concerning the second law that are helpful in analyzing thermodynamic systems, such as:

- No thermodynamic cycle can be more efficient than a reversible cycle operating between the same temperature limits.
- The efficiency of all reversible cycles absorbing heat from a single-constant higher temperature and rejecting heat at a single-constant lower temperature must be the same.
- Every real system tends naturally towards a state of maximum probability.
- For any actual process, it is impossible to devise a means of restoring to its original state every system participating in the process.
- For any reversible process, the increase in entropy of any participating system is equal to the heat absorbed by that system divided by the absolute temperature at which the transfer occurred. That is, for a system, i,

$$dS_i = \frac{\delta Q_i}{T_i} \quad \text{(reversible processes)} \qquad (2\text{-}111)$$

Alternatively, for an ideal reversible process, the sum of all the changes in entropy must be zero or

$$\sum dS_i = \sum \frac{\delta Q_i}{T_i} = 0 \quad \text{(reversible processes)} \qquad (2\text{-}112)$$

Because all *real* processes are *irreversible* as a result of friction, electrical resistance, etc., any processes involving real systems experience an *increase in entropy*. For such systems

$$\sum dS_i > 0 \quad \text{(irreversible processes)} \qquad (2\text{-}113)$$

The entropy change of a system during any process depends only upon its initial and final states and not upon the path of the process by which it proceeds from its initial to its final state. Thus one can devise a reversible idealized process to restore a system to its initial state following a change and thereby determine $\Delta S = S_{final} - S_{initial}$. This is one of the most useful aspects of the concepts of a reversible process.

Entropy Production: Flow Systems

In general, for all real processes, there is a net production of entropy and Equation 2-113 applies. Since many practical engineering processes involve open systems, it is useful to develop a generalized expression of the second law applied to such systems.

For the generalized control volume shown in Figure 2-33, and entropy balance can be stated as follows:

$$\begin{pmatrix}\text{Rate of entropy}\\ \text{in}\end{pmatrix} - \begin{pmatrix}\text{Rate of entropy}\\ \text{out}\end{pmatrix} + \begin{pmatrix}\text{Rate of entropy}\\ \text{production}\end{pmatrix} = \begin{pmatrix}\text{Rate of entropy}\\ \text{accumulation}\end{pmatrix}$$

$$\left[\sum_{in}(\dot{M}S) + \sum_{in}\frac{\delta\dot{Q}_i}{T_i}\right] - \left[\sum_{out}\dot{M}S + \sum_{out}\frac{\delta\dot{Q}_i}{T_i}\right] + \dot{P}_S = \frac{d(SM)}{dt} \qquad (2\text{-}114)$$

In Equation 2-114, $\dot{P}_S$ is the rate of entropy production within the control volume; symbols with dots refer to the time rate of change of the quantity in question. The second law requires that the rate of entropy production be positive.

$$\dot{P}_S \geq 0 \qquad (2\text{-}115)$$

Heat Capacity

The heat capacity of a substance is extremely important in thermodynamic analysis involving both the first and second laws.

Heat capacity per unit mass is defined by the relationship

$$C \equiv \frac{\delta Q}{dT} \qquad (2\text{-}116)$$

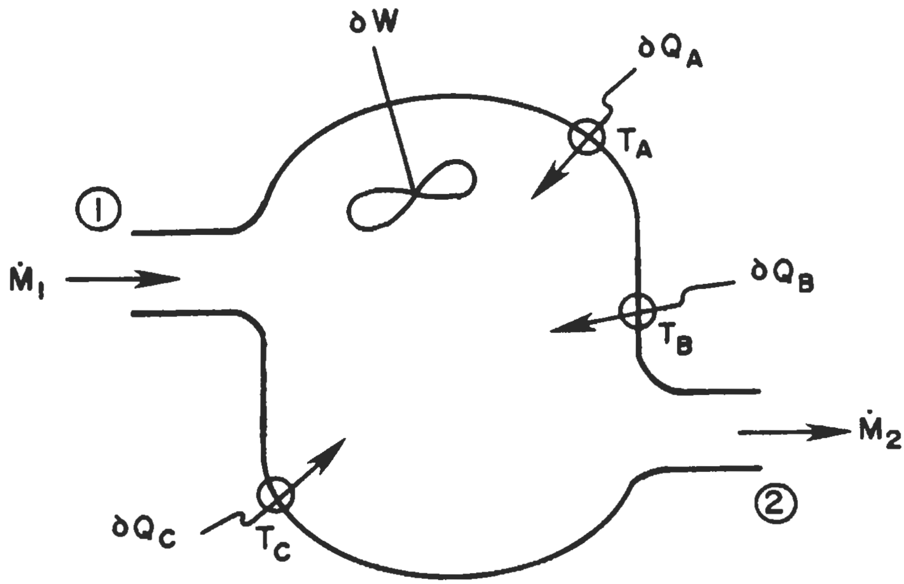


Figure 2-33. Generalized control volume for a flow system.

where δQ is the heat absorbed by unit mass of the system over the infinitesimal temperature change, dT.

For *constant pressure* when only P–V work is involved, the first law yields

$$C_P = \left(\frac{\partial H}{\partial T}\right)_P \tag{2-117}$$

For constant volume

$$C_V = \left(\frac{\partial U}{\partial T}\right)_V \tag{2-118}$$

C_P is related to C_V by the following expression:

$$C_P - C_V = \left[P + \left(\frac{\partial U}{\partial V}\right)_T\right]\left(\frac{\partial V}{\partial T}\right)_P \tag{2-119}$$

For an ideal gas for which $PV = RT/M_w$, with M_w representing the molecular weight, the enthalpy and internal energy are functions only of temperature and Equation 2-119 becomes

$$C_P - C_V = P\left(\frac{\partial V}{\partial T}\right)_P = \frac{R}{M_w} \tag{2-120}$$

The ratio of the heat capacity at constant pressure to that at constant volume is

$$\frac{C_P}{C_V} = \kappa \tag{2-121}$$

Again, for an ideal gas, this ratio becomes

$$\kappa = 1 + \frac{R}{C_V M_w} \tag{2-122}$$

Application of the Second Law

Heat Engines

The purpose of a heat engine is to remove heat Q_1 from a thermal reservoir at a higher absolute temperature T_1; extract useful work W; and reject heat Q_2 to a second thermal reservoir at a lower absolute temperature T_2. The device used to obtain the useful work is the heat engine.

Considering an "ideal" heat engine as the system, the first law as applied to the engine undergoing a series of reversible changes in a cyclical fashion becomes

$$\Sigma \Delta U_i = 0, \qquad \text{or} \qquad W = Q_1 - Q_2 \tag{2-123}$$

The second law yields

$$\sum \Delta S_i = 0, \quad \text{or} \quad \frac{Q_1}{T_1} - \frac{Q_2}{T_2} = 0 \quad \text{or} \quad \frac{Q_1}{Q_2} = \frac{T_1}{T_2} \tag{2-124}$$

To obtain the fraction of the heat input Q_1 that is converted to useful work, Equations 2-123 and 2-124 are combined to give

$$\frac{W}{Q_1} = \frac{T_1 - T_2}{T_1} \tag{2-125}$$

This important result is called the *Carnot engine efficiency* and yields the *maximum thermal efficiency* that can be achieved by *any* heat engine cycle operating between any two given temperature limits. Heat engines have been proposed to operate within the temperature gradients of the ocean as a means of harnessing the vast amounts of renewable energy available from that source.

Heat Pumps

A heat pump, which is the opposite of a heat engine, uses work energy to transfer heat from a cold reservoir to a "hot" reservoir. In households, the cold reservoir is often the surrounding air or the ground while the hot reservoir is the home. For an ideal heat pump system with Q_1 and T_1 referring to the hot reservoir and Q_2 and T_2 referring to the cold reservoir, the work required is, from the first and second laws,

$$\frac{W}{Q_1} = \frac{T_1 - T_2}{T_1} \tag{2-126}$$

Application of this result shows that if 100 units of heat Q_1 are needed to maintain a household at 24°C (297°K) by "pumping" heat from the outside surroundings at 0°C (273°K), it would require a minimum of $(24 \times 100/297) = 8.08$ units of work energy.

Refrigeration Machines

Refrigerating machines absorb heat Q_2 from a cold reservoir at temperature T_2, and discharge heat Q_1, into a "hot" reservoir at T_1. To accomplish this, work energy must also be absorbed. The minimum required work is obtained as shown before, using the first and second laws:

$$\frac{W}{Q_2} = \frac{T_1 - T_2}{T_1} \tag{2-127}$$

Reversible Work of Expansion or Compression

Many systems involve only work of expansion or compression of the system boundaries. For such systems the first law is written for unit mass of fluid as the basis:

$$dU = \delta Q - PdV \tag{2-128}$$

where $\int_{V_1}^{V_2} P\,dV$ represents the reversible work of compression or expansion.

From the second law, for a reversible process,

$$dS = \frac{\delta Q}{T} \quad \text{or} \quad \delta Q = TdS \tag{2-129}$$

Combining Equations 2-128 and 2-129 gives

$$dU = TdS - PdV \quad \text{(reversible, P–V work only)} \tag{2-130}$$

Reversible Isobaric Processes

The second law is written as

$$\delta Q = TdS$$

but, since the heat capacity $C = \delta Q/dT$, the second law becomes

$$CdT = TdS \tag{2-131}$$

Thus, for constant pressure processes, the entropy increase is written as

$$dS = C_P \frac{dT}{T} = C_P d(\ell nT) \quad \text{(reversible isobaric process)} \tag{2-132}$$

Reversible Constant Volume Processes

A constant-volume process is called *isochoric* and, for such processes, the entropy increase is written as

$$dS = C_V \frac{dT}{T} = C_V d(\ell n\ T) \quad \text{(reversible isochoric process)} \tag{2-133}$$

Reversible Isothermal Changes: Maximum Work

The variation of entropy with volume and pressure under conditions of constant temperature is determined by using Equation 2-130:

$$\left(\frac{\partial U}{\partial V}\right)_T = T\left(\frac{\partial S}{\partial V}\right)_T - P \tag{2-134}$$

or rearranging:

$$\left(\frac{\partial S}{\partial V}\right)_T = \frac{1}{T}\left[\left(\frac{\partial U}{\partial V}\right)_T + P\right] \quad \text{(constant T)} \tag{2-135}$$

The variation of entropy with pressure is likewise written as

$$\left(\frac{\partial S}{\partial P}\right)_T = \frac{1}{T}\left[P\left(\frac{\partial V}{\partial P}\right)_T + \left(\frac{\partial U}{\partial P}\right)_T\right] \quad \text{(constant T)} \tag{2-136}$$

or

$$\left(\frac{\partial S}{\partial P}\right)_T = \frac{1}{T}\left[-V + \left(\frac{\partial H}{\partial P}\right)_T\right] \quad \text{(constant T)} \qquad (2\text{-}137)$$

Combining the first law Equation 2-102 with the second law equation 2-111 yields the expression

$$dU = TdS - \delta W = d(TS) - \delta W$$

or

$$\delta W = -d(U - TS)$$

which, upon integration between states 1 and 2, yields

$$W = -\Delta(U - TS) \quad \text{(reversible isothermal processes)} \qquad (2\text{-}138)$$

The combination of properties U – TS occurs so frequently in thermodynamic analysis that it is given a special name and symbol, namely A, the *work function* or *maximum work* (because it represents the maximum work per unit mass, obtainable during any isothermal reversible change in any given system). Therefore, it is seen that

$$W_{max} = -\Delta A \quad \text{(reversible isothermal process)} \qquad (2\text{-}139)$$

Note that the maximum work depends only upon the initial and final states of a system and not upon the path.

Maximum Useful Work: Free Energy

The first and second law expressions can be combined and written for constant temperature, constant pressure processes:

$$dU = TdS - \delta W = TdS - PdV - \delta W' \qquad (2\text{-}140)$$

where $\delta W'$ represents all work energy exchanged with the surroundings except P–V work that is written as PdV. Therefore, solving for $\delta W'$ gives

$$-\delta W' = dU + PdV - TdS \qquad (2\text{-}141)$$

or, because both T and P are constant,

$$-\delta W' = dU + d(PV) - d(TS)$$

or

$$-\delta W' = -d(U + PV - TS) \qquad (2\text{-}142)$$

By integration this becomes

$$W' = -\Delta(U + PV - TS)$$

$$= -\Delta(H - TS) \quad \text{(constant temperature and pressure)} \qquad (2\text{-}143)$$

This expression shows that the maximum possible *useful work* (i.e., reversible work) that can be obtained from any process occurring at constant temperature and pressure is a function of the initial and final states only and is independent of the path. The combination of properties U + PV – TS or H – TS occurs so frequently in thermodynamic analysis that it is given a special name and symbol, F, the *free energy* (sometimes called the Gibbs Free Energy). Using this definition, Equation 2-143 is written

$$W'_{max} = -\Delta F \tag{2-144}$$

Because F is a function of temperature and pressure, its differential can be written as

$$dF = \left(\frac{\partial F}{\partial P}\right)_T dP + \left(\frac{\partial F}{\partial T}\right)_P dT \tag{2-145}$$

Since F = U + PV – TS, we can also write

$$dF = dU + PdV + VdP - TdS - SdT \tag{2-146}$$

Using Equation 2-130, this becomes

$$dF = VdP - SdT \tag{2-147}$$

Comparison with Equation 2-145 shows that

$$\left(\frac{\partial F}{\partial P}\right)_T = V \tag{2-148}$$

$$\left(\frac{\partial F}{\partial T}\right)_P = -S \tag{2-149}$$

Example 2-24

An inventor claims to have devised a CO_2 compressor that requires no shaft work. The device operates at steady state by transferring heat from a feed stream of 2 lb_m/s of CO_2 at 150 psia and 100°F. The CO_2 is compressed to a final pressure of 500 psia and a temperature of 40°F. Kinetic and potential energy effects are negligible. A cold source at –140°F "drives" the device at a heat transfer rate of 60 Btu/sec. Check the validity of the inventor's claim.

Solution

The device will be impossible if it violates either the first or second law of thermodynamics. From Figure 2-34 the inlet and outlet properties are:

State 1	State 2
T_1 = 100°F	T_2 = 40°F
P_1 = 150 psia	P_2 = 500 psia
h_1 = 315 Btu/lb_m	h_2 = 285 Btu/lb_m
S_1 = 1.318 Btu/lb_m°R	S_2 = 1.215 Btu/lb_m°R

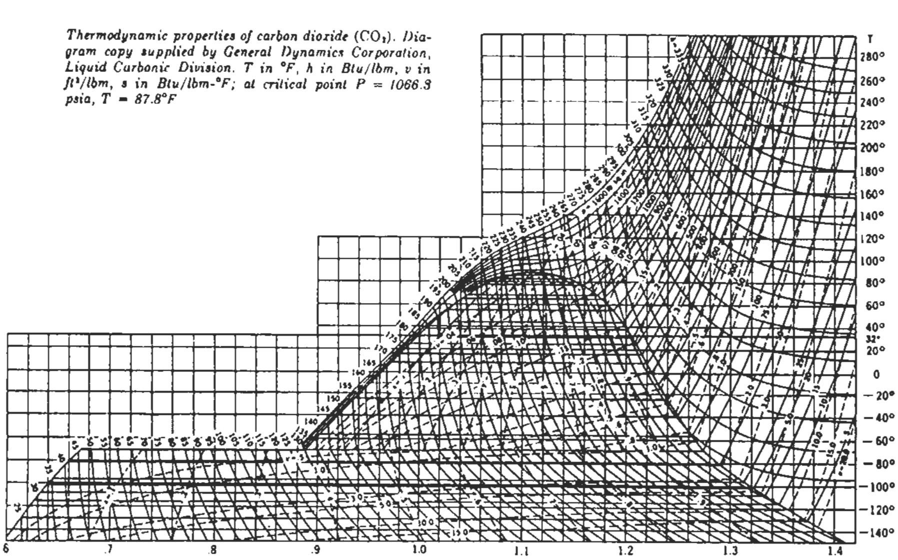


Figure 2-34. Thermodynamic properties of carbon dioxide [21]).

Referring to Figure 2-35 (process diagram), the first law for this steady-state flow system becomes

$$\dot{M}(h_2 - h_1) + \dot{Q} = (2)(285 - 315) + 60 = 0 \text{ Btu/s}$$

Because energy is not created, the device does not violate the first law. Application of the second law (Equation 2-114) yields

$$\dot{P}_S = \dot{M}(S_2 - S_1) + \frac{\dot{Q}_C}{T_C} = (2)(1.215 - 1.318) + \frac{60}{(-140 + 460)}$$

$$= -0.206 + 0.1875 = -0.0185 \text{ Btu/s}\bullet°\text{R}$$

Because the rate of entropy production is negative, the device violates the second law and is therefore impossible. Note that the device would be theoretically possible if the final pressure were specified as 400 psia or less by the inventor. That is, at P_2 = 400 psia, T_2 = 40°F, h_2 = 290 Btu/lb_m, and S_2 = 1.25 Btu/lb_m°R, the entropy production rate would be

$$\dot{P}_S = 2(1.25 - 1.318) + 0.1875 = +0.0515 \text{ Btu/s}\bullet°\text{R}$$

Because entropy is produced in this case, the device is theoretically possible.

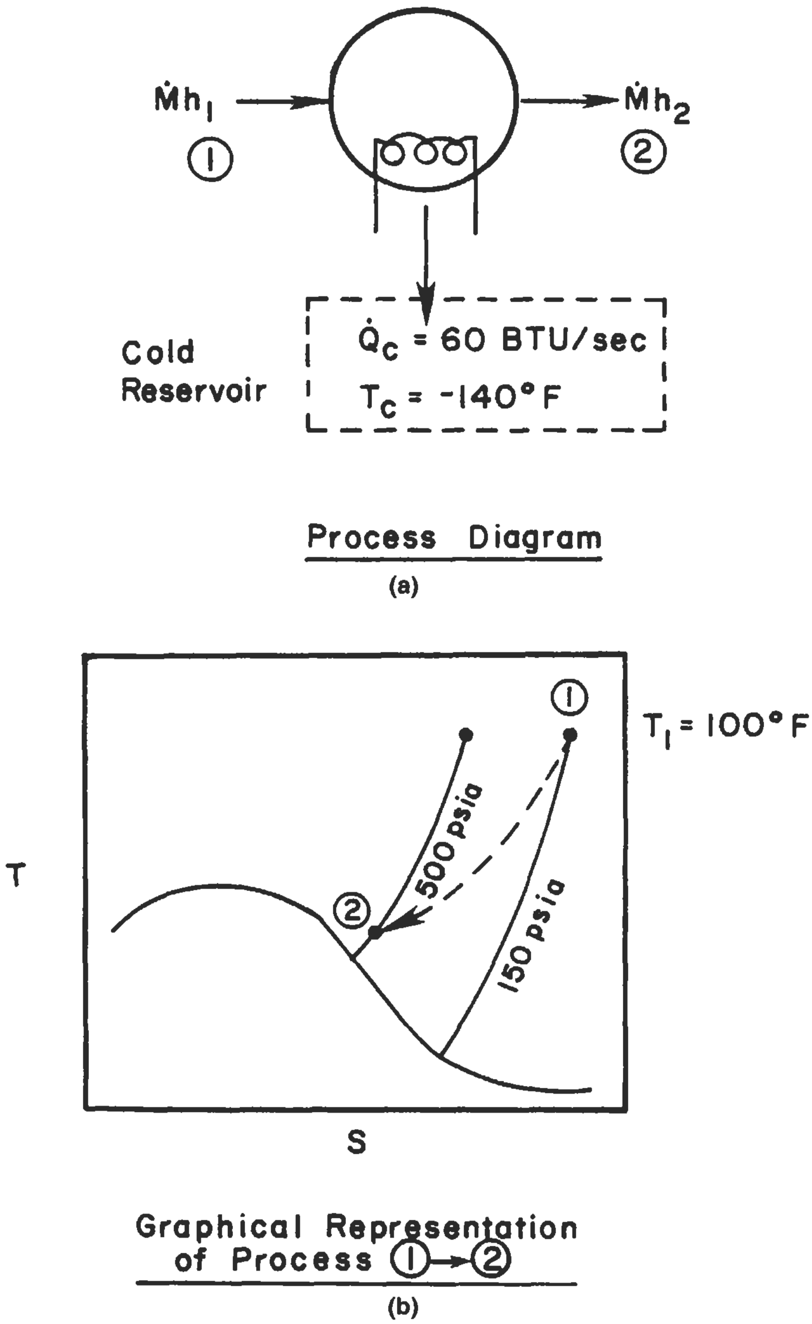


Figure 2-35. (a) Process diagram. (b) Graphical representation of Processes 1 and 2.

Summary of Thermodynamic Equations

The thermodynamic relations formulated earlier for a pure substance are summarized in Table 2-19 with unit mass of fluid as the basis. Several additional important relationships can be derived from them and these are shown in the third column of Table 2-19.

By mathematical manipulation, numerous additional relationships can be derived from those given in Table 2-19. Of particular significance are expressions that relate enthalpy H and internal energy U to the measurable variables, P, V, and T. Thus, choosing the basis as one pound mass,

$$\left(\frac{\partial H}{\partial P}\right)_T = V - T\left(\frac{\partial V}{\partial T}\right)_P \tag{2-150}$$

and

$$\left(\frac{\partial U}{\partial V}\right)_T = -P + T\left(\frac{\partial P}{\partial T}\right)_V \tag{2-151}$$

Equations 2-150 and 2-151 apply to any substance or system and are called *equations of state* because they completely determine the state of a system in terms of its thermodynamic properties.

Thermal Properties for Selected Systems

For practical applications of the numerous thermodynamic relationships, it is necessary to have available the properties of the system. In general, a given property of a pure substance can be expressed in terms of any other two properties to completely define the state of the substance. Thus one can represent an equation of state by the functional relationship:

$$P = f(T, V) \tag{2-152}$$

which indicates that the pressure is a function of the temperature and specific volume.

Plots of the properties of various substances as well as tables and charts are extremely useful in solving engineering thermodynamic problems. Two-dimensional representations of processes on P–V, T–S, or H–S diagrams are especially useful in analyzing cyclical processes. The use of the P–V diagram was illustrated earlier. A typical T–S diagram for a Rankine vapor power cycle is depicted in Figure 2-36.

For the Rankine cycle, the area enclosed by the line segments connecting points 1, 2, 3, 4, 1 on Figure 2-36 represents the net heat transferred into the system per unit mass, because

$$Q_{net} = \int_{S_4}^{S_1} T\,dS - \int_{S_2}^{S_3} T\,dS \tag{2-153}$$

(text continues on page 226)

Table 2-19
Summary of Thermodynamic Relations (Basis: Unit mass of Fluid)

Function or Definition	Common Name or Terminology	Differential Equation	Derived Relationships Among Variables
$C_P = \left(\frac{\delta Q}{\partial T}\right)_P = \left(\frac{\partial H}{\partial T}\right)_P = T\left(\frac{\partial S}{\partial T}\right)_P$	Heat Capacity at Constant Pressure	$\delta Q_P = C_P dT$	$C_P = \left(\frac{\partial U}{\partial T}\right)_P + P\left(\frac{\partial V}{\partial T}\right)_P$
$C_V = \left(\frac{\delta Q}{\partial T}\right)_V = \left(\frac{\partial U}{\partial T}\right)_V = T\left(\frac{\partial S}{\partial T}\right)_V$	Heat Capacity at Constant Volume	$\delta Q_V = C_V dT$	$C_V = \left(\frac{\partial U}{\partial P}\right)_V \left(\frac{\partial P}{\partial T}\right)_V$
$\kappa = \frac{C_P}{C_V}$	Heat Capacity Ratio		$C_P - C_V = T\left(\frac{\partial P}{\partial T}\right)_V \left(\frac{\partial V}{\partial T}\right)_P$
			$C_P - C_V = \frac{R}{M_w}$ |Ideal Gas|
$\Delta U = Q - W$	Internal Energy (First Law of Thermodynamics)	$dU = TdS - PdV$	$\left(\frac{\partial T}{\partial V}\right)_S = -\left(\frac{\partial P}{\partial S}\right)_V$
			$\left(\frac{\partial U}{\partial S}\right)_V = T$
			$\left(\frac{\partial U}{\partial V}\right)_S = -P$

$H = U + PV$	Enthalpy	$dH = TdS + VdP$	$\left(\frac{\partial T}{\partial P}\right)_S = \left(\frac{\partial V}{\partial S}\right)_P$
			$\left(\frac{\partial H}{\partial S}\right)_P = T$
			$\left(\frac{\partial H}{\partial P}\right)_S = V$
$A = U - TS$	Work function or maximum work (Helmholtz free energy)	$dA = -SdT - PdV$	$\left(\frac{\partial S}{\partial V}\right)_T = \left(\frac{\partial P}{\partial T}\right)_V$
			$\left(\frac{\partial A}{\partial T}\right)_V = -S$
			$\left(\frac{\partial A}{\partial V}\right)_T = -P$
$F = H - TS$	Free energy or Gibbs Free Energy	$dF = -SdT + VdP$	$\left(\frac{\partial S}{\partial P}\right)_T = -\left(\frac{\partial V}{\partial T}\right)_P$
			$\left(\frac{\partial F}{\partial T}\right)_P = -S$
			$\left(\frac{\partial F}{\partial P}\right)_T = V$

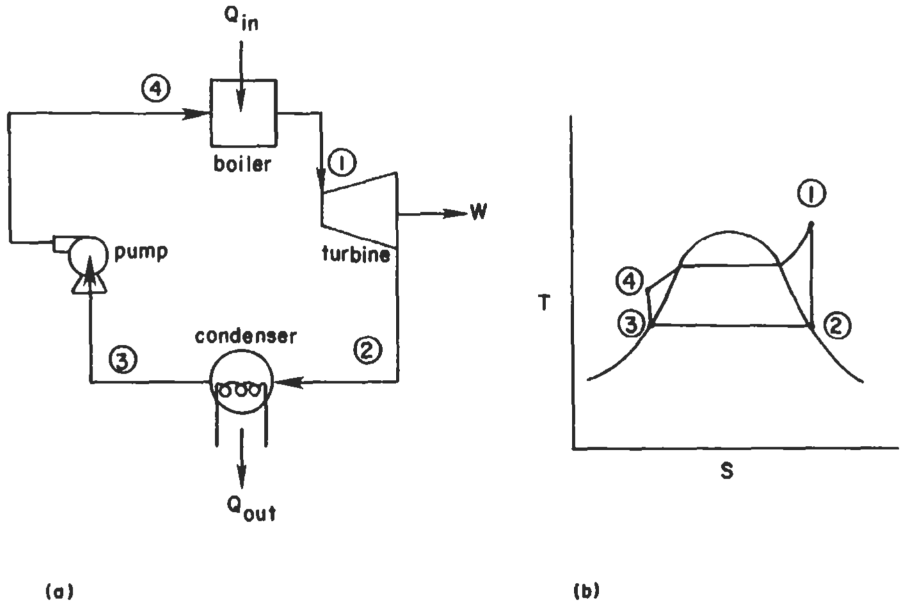


Figure 2-36. (a) Schematic diagram of system. (b) T–S diagram of process.

(text continued from page 223)

The efficiency of power cycles such as the Rankine cycle is given by the ratio of the net work out to the heat added. Thus from Figure 2-36 the efficiency is

$$\eta = \frac{W_{net}}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} \tag{2-154}$$

The H–S plot is called a Mollier diagram and is particularly useful in analyzing throttling devices, steam turbines, and other fluid flow devices. A Mollier diagram for steam is presented in Figure 2-37 (standard engineering units) and in Figure 2-38 in SI units.

Thermodynamic properties may be presented in various ways, including:

- Equations of state (e.g., perfect gas laws, Van der Waals equation, etc.).
- Charts or graphs.
- tables.

Tables 2-20–2-25 present thermodynamic properties for several pure substances commonly encountered in petroleum engineering practice.

For further information on this subject, refer to References 15 through 26.

(text continues on page 240)

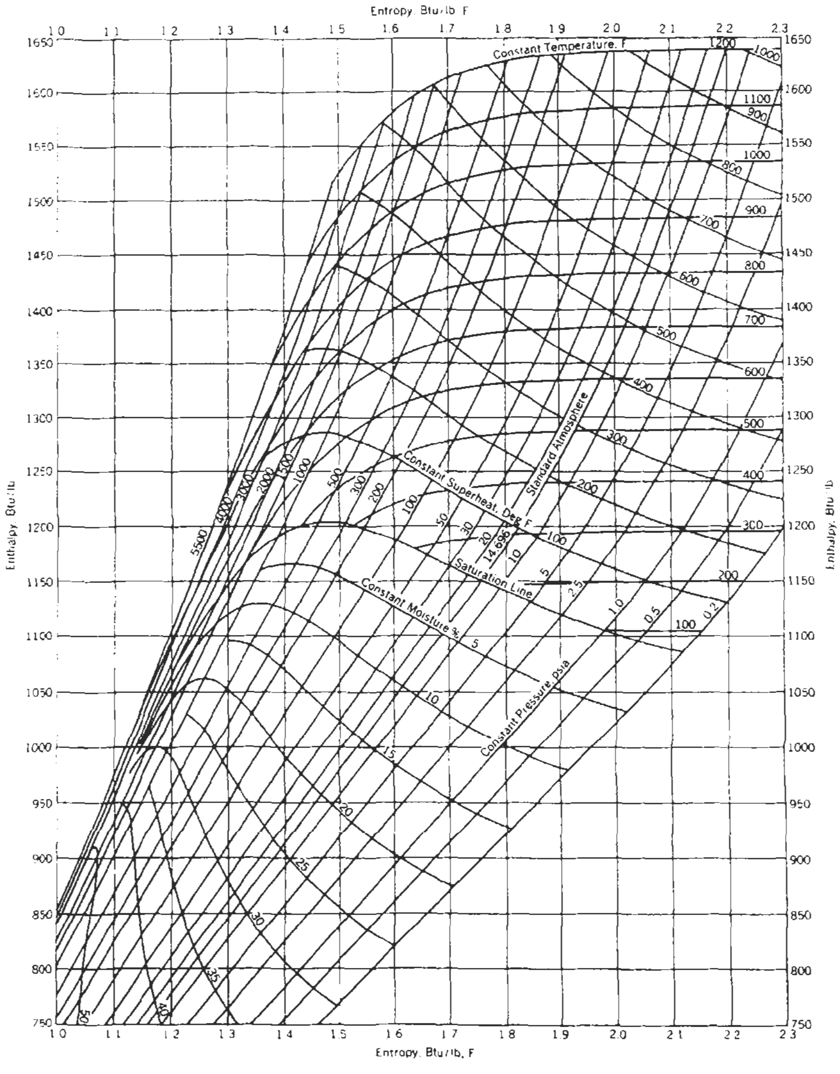


Figure 2-37. Mollier diagram for steam.

SOURCE: *Steam Tables in SI-Units*, Springer-Verlag, 1984.

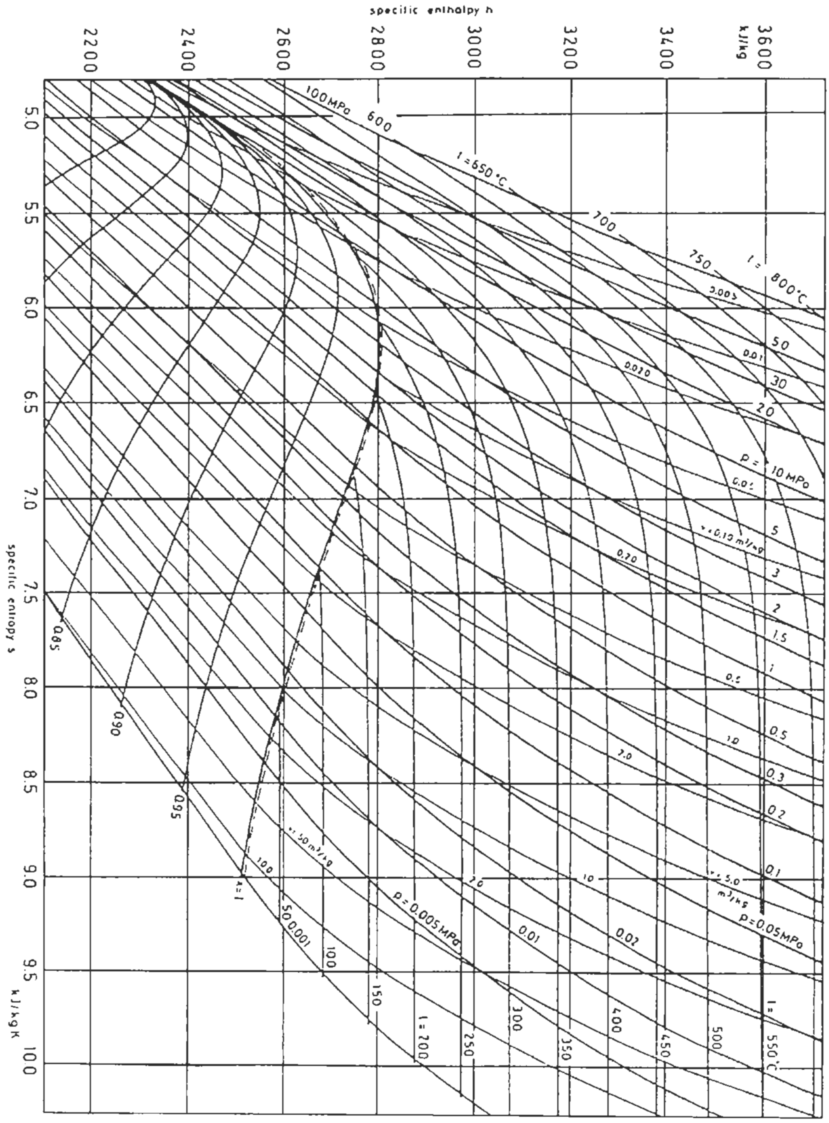


Figure 2-38. Mollier (enthalpy-entropy) diagram for steam.
SOURCE: *Steam, its generation and use,* The Babcock $ Wilcox Co., 1978.

Table 2-20
Properties of Saturated Steam
(h_f and s_f are measured from 32°F)

Abs press., psi	Temp, deg F	Specific volume		Enthalpy			Entropy			Internal energy
		Liquid	Vapor	Liquid	Evap	Vapor	Liquid	Evap	Vapor	Evap
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0	0.1326	1.8456	1.9782	974.6
1.2	107.92	0.01616	280.9	75.87	1032.7	1108.6	0.1435	1.8193	1.9628	970.3
1.1	113.26	0.01618	243.0	81.20	1029.6	1110.8	0.1528	1.7971	1.9498	966.7
1.6	117.99	0.01620	214.3	85.91	1026.9	1112.8	0.1610	1.7776	1.9386	963.5
1.8	122.23	0.01621	191.8	90.14	1024.5	1114.6	0.1683	1.7605	1.9288	960.6
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2	0.1749	1.7451	1.9200	957.9
2.2	129.62	0.01624	158.85	97.52	1020.2	1117.7	0.1809	1.7311	1.9120	955.5
2.4	132.89	0.01626	146.38	100.79	1018.3	1119.1	0.1864	1.7183	1.9047	953.3
2.6	135.94	0.01627	135.78	103.83	1016.5	1120.3	0.1916	1.7065	1.8981	951.2
2.8	138.79	0.01629	126.65	106.68	1014.8	1121.5	0.1963	1.6957	1.8920	949.2
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6	0.2008	1.6855	0.8863	947.3
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3	0.2198	1.6427	1.8625	939.3
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1	0.2347	1.6094	1.8441	933.0
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2	0.2472	1.5820	1.8292	927.5
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9	0.2581	1.5586	1.8167	922.7
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3	0.2674	1.5383	1.8057	918.4
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4	0.2759	1.5203	1.7962	914.6
10	193.21	0.01659	38.42	161.17	982.1	1143.3	0.2835	1.5041	1.7876	911.1
11	197.75	0.01662	35.14	165.73	979.3	1145.0	0.2903	1.4897	1.7800	907.8
12	201.96	0.01665	32.40	169.96	976.6	1146.6	0.2967	1.4763	1.7730	904.8
13	205.88	0.01667	30.06	173.91	974.2	1148.1	0.3027	1.4638	1.7665	901.9
14	209.56	0.01670	28.04	177.61	971.9	1149.5	0.3083	1.4522	1.7605	899.3
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	897.5
15	213.03	0.01672	26.29	181.11	969.7	1150.8	0.3135	1.4415	1.7549	896.7
16	216.32	0.01674	24.75	184.42	967.6	1152.0	0.3184	1.4313	1.7497	894.3
17	219.44	0.01677	23.39	187.56	965.5	1153.1	0.3231	1.4218	1.7449	892.0
18	222.41	0.01679	22.17	190.56	963.6	1154.2	0.3275	1.4128	1.7403	889.9
19	225.24	0.01681	21.08	193.42	961.9	1155.3	0.3317	1.4043	1.7360	887.8
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	885.8
21	230.57	0.01685	19.192	198.79	958.4	1157.2	0.3395	1.3885	1.7280	883.9
22	233.07	0.01687	18.375	201.33	956.8	1158.1	0.3431	1.3811	1.7242	882.0
23	235.49	0.01689	17.627	203.78	955.2	1159.0	0.3466	1.3740	1.7206	880.2
24	237.82	0.01691	16.938	206.14	953.7	1159.8	0.3500	1.3672	1.7172	878.5
25	240.07	0.01692	16.303	208.42	952.1	1160.6	0.3533	1.3606	1.7139	876.8
26	242.25	0.01694	15.715	210.62	950.7	1161.3	0.3564	1.3544	1.7108	875.2
27	244.36	0.01696	15.170	212.75	949.3	1162.0	0.3594	1.3484	1.7078	873.6
28	246.41	0.01698	14.663	214.83	947.9	1162.7	0.3623	1.3425	1.7048	872.1
29	248.40	0.01699	14.189	216.86	946.5	1163.4	0.3652	1.3368	1.7020	870.5
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	869.1
31	252.22	0.01702	13.330	220.73	944.0	1164.7	0.3707	1.3260	1.6967	867.7
32	254.05	0.01704	12.940	222.59	942.8	1165.4	0.3733	1.3209	1.6941	866.3
33	255.84	0.01705	12.572	224.41	941.6	1166.0	0.3758	1.3159	1.6917	864.9
34	257.08	0.01707	12.226	226.18	940.3	1166.5	0.3783	1.3110	1.6893	863.5
35	259.28	0.01708	11.898	227.91	939.2	1167.1	0.3807	1.3063	1.6870	862.3
36	260.95	0.01709	11.588	229.60	938.0	1167.6	0.3831	1.3017	1.6848	861.0
37	262.57	0.01711	11.294	231.26	936.9	1168.2	0.3854	1.2972	1.6826	859.8
38	264.16	0.01712	11.015	232.89	935.8	1168.7	0.3876	1.2929	1.6805	858.5
39	265.72	0.01714	10.750	234.48	934.7	1169.2	0.3898	1.2886	1.6784	857.2

SOURCE: Baumeister T., and Marks, L. S., eds., *Standard Handbook for Mechanical Engineers*, Seventh edition, McGraw-Hill Book Co., New York, 1967.
Note: Specific volume in cu. ft. per lb_m, enthalphy and internal energy in Btu per lb_m, entropy in Btu per lb_m°R

Table 2-20
(continued)

Abs press, psi	Temp, deg F	Specific volume		Enthalpy			Entropy			Internal energy
		Liquid	Vapor	Liquid	Evap	Vapor	Liquid	Evap	Vapor	Evap
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	856.1
41	268.74	0.01716	10.258	237.55	932.6	1170.2	0.3940	1.2803	1.6743	855.0
42	270.21	0.01717	10.029	239.04	931.6	1170.7	0.3960	1.2764	1.6724	853.8
43	271.64	0.01719	9.810	240.51	930.6	1171.1	0.3980	1.2726	1.6706	852.7
44	273.05	0.01720	9.601	241.95	929.6	1171.6	0.4000	1.2687	1.6687	851.6
45	274.44	0.01721	9.401	243.36	928.6	1172.0	0.4019	1.2650	1.6669	850.5
46	275.80	0.01722	9.209	244.75	927.7	1172.4	0.4038	1.2613	1.6652	849.5
47	277.13	0.01723	9.025	246.12	926.7	1172.9	0.4057	1.2577	1.6634	848.4
48	278.45	0.01725	8.848	247.47	925.8	1173.3	0.4075	1.2542	1.6617	847.4
49	279.74	0.01726	8.678	248.79	924.9	1173.7	0.4093	1.2508	1.6601	846.4
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	845.4
51	282.26	0.01728	8.359	251.37	923.0	1174.4	0.4127	1.2432	1.6569	844.3
52	283.49	0.01729	8.208	252.63	922.2	1174.8	0.4144	1.2409	1.6553	843.3
53	284.70	0.01730	8.062	253.87	921.3	1175.2	0.4161	1.2377	1.6538	842.4
54	285.90	0.01731	7.922	255.09	920.5	1175.6	0.4177	1.2346	1.6523	841.5
55	287.07	0.01732	7.787	256.30	919.6	1175.9	0.4193	1.2316	1.6509	840.6
56	288.23	0.01733	7.656	257.50	918.8	1176.3	0.4209	1.2285	1.6494	839.7
57	289.37	0.01734	7.529	258.67	917.9	1176.6	0.4225	1.2255	1.6480	838.7
58	290.50	0.01736	7.407	259.82	917.1	1176.9	0.4240	1.2226	1.6466	837.8
59	291.61	0.01737	7.289	260.96	916.3	1177.3	0.4255	1.2197	1.6452	836.9
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	836.0
61	293.79	0.01739	7.064	263.20	914.7	1177.9	0.4285	1.2140	1.6425	835.2
62	294.85	0.01740	6.957	264.30	913.9	1178.2	0.4300	1.2112	1.6412	834.3
63	295.90	0.01741	6.853	265.38	913.1	1178.5	0.4314	1.2085	1.6399	833.4
64	296.94	0.01742	6.752	266.45	912.3	1178.8	0.4328	1.2059	1.6387	832.6
65	297.97	0.01743	6.655	267.50	911.6	1179.1	0.4342	1.2032	1.6374	831.8
66	298.99	0.01744	6.560	268.55	910.8	1179.4	0.4356	1.2006	1.6362	831.0
67	299.99	0.01745	6.468	269.58	910.1	1179.7	0.4369	1.1981	1.6350	830.2
68	300.98	0.01746	6.378	270.60	909.4	1180.0	0.4383	1.1955	1.6338	829.4
69	301.96	0.01747	6.291	271.61	908.7	1180.3	0.4396	1.1930	1.6326	828.6
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	827.8
71	303.88	0.01749	6.124	273.60	907.2	1180.8	0.4422	1.1881	1.6303	827.0
72	304.83	0.01750	6.044	274.57	906.5	1181.1	0.4435	1.1857	1.6292	826.3
73	305.76	0.01751	5.966	275.54	905.8	1181.3	0.4447	1.1834	1.6281	825.5
74	306.68	0.01752	5.890	276.49	905.1	1181.6	0.4460	1.1810	1.6270	824.7
75	307.60	0.01753	5.816	277.43	904.5	1181.9	0.4472	1.1787	1.6259	824.0
76	308.50	0.01754	5.743	278.37	903.7	1182.1	0.4484	1.1764	1.6248	823.3
77	309.40	0.01754	5.673	279.30	903.1	1182.4	0.4496	1.1742	1.6238	822.5
78	310.29	0.01755	5.604	280.21	902.4	1182.6	0.4508	1.1720	1.6228	821.7
79	311.16	0.01756	5.537	281.12	901.7	1182.8	0.4520	1.1698	1.6217	821.0
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	820.3
81	312.89	0.01758	5.408	282.91	900.4	1183.3	0.4543	1.1654	1.6197	819.6
82	313.74	0.01759	5.346	283.79	899.7	1183.5	0.4554	1.1633	1.6187	818.9
83	314.59	0.01760	5.285	284.66	899.1	1183.8	0.4565	1.1612	1.6177	818.2
84	315.42	0.01761	5.226	285.53	898.5	1184.0	0.4576	1.1592	1.6168	817.5
85	316.25	0.01761	5.168	286.39	897.8	1184.2	0.4587	1.1571	1.6158	816.8
86	317.07	0.01762	5.111	287.24	897.2	1184.4	0.4598	1.1551	1.6149	816.1
87	317.88	0.01763	5.055	288.08	896.5	1184.6	0.4609	1.1530	1.6139	815.4
88	318.68	0.01764	5.001	288.91	895.9	1184.8	0.4620	1.1510	1.6130	814.8
89	319.48	0.01765	4.948	289.74	895.3	1185.1	0.4630	1.1491	1.6121	814.1

Table 2-20
(continued)

Abs press, psi	Temp, deg F	Specific volume		Enthalpy			Entropy			Internal energy
		Liquid	Vapor	Liquid	Evap	Vapor	Liquid	Evap	Vapor	Evap
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	813.4
91	321.06	0.01767	4.845	291.38	894.1	1185.5	0.4651	1.1452	1.6103	812.8
92	321.83	0.01768	4.796	292.18	893.5	1185.7	0.4661	1.1433	1.6094	812.2
93	322.60	0.01768	4.747	292.98	892.9	1185.9	0.4672	1.1413	1.6085	811.5
94	323.36	0.01769	4.699	293.78	892.3	1186.1	0.4682	1.1394	1.6076	810.9
95	324.12	0.01770	4.652	294.56	891.7	1186.2	0.4692	1.1376	1.6068	810.2
96	324.87	0.01771	4.606	295.34	891.1	1186.4	0.4702	1.1358	1.6060	809.6
97	325.61	0.01772	4.561	296.12	890.5	1186.6	0.4711	1.1340	1.6051	808.9
98	326.35	0.01772	4.517	296.89	889.9	1186.8	0.4721	1.1322	1.6043	808.3
99	327.08	0.01773	4.474	297.65	889.4	1187.0	0.4731	1.1304	1.6035	807.7
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	807.1
102	329.25	0.01775	4.350	299.90	887.6	1187.5	0.4759	1.1251	1.6010	805.9
104	330.66	0.01777	4.271	301.37	886.5	1187.9	0.4778	1.1216	1.5994	804.7
106	332.05	0.01778	4.194	302.82	885.4	1188.2	0.4796	1.1182	1.5978	803.5
108	333.42	0.01780	4.120	304.26	884.3	1188.6	0.4814	1.1149	1.5963	802.4
110	334.77	0.01782	4.049	305.66	883.2	1188.9	0.4832	1.1117	1.5948	801.2
112	336.11	0.01783	3.981	307.06	882.1	1189.2	0.4849	1.1085	1.5934	800.0
114	337.42	0.01784	3.914	308.43	881.1	1189.5	0.4866	1.1053	1.5919	798.9
116	338.72	0.01786	3.850	309.79	880.0	1189.8	0.4883	1.1022	1.5905	797.8
118	339.99	0.01787	3.788	311.12	879.0	1190.1	0.4900	1.0992	1.5891	796.7
120	341.25	0.01789	3.728	312.44	877.9	1190.4	0.4916	1.0962	1.5878	795.6
122	342.50	0.01791	3.670	313.75	876.9	1190.7	0.4932	1.0933	1.5865	794.5
124	343.72	0.01792	3.614	315.04	875.9	1190.9	0.4948	1.0903	1.5851	793.4
126	344.94	0.01793	3.560	316.31	874.9	1191.2	0.4964	1.0874	1.5838	792.3
128	346.13	0.01794	3.507	317.57	873.9	1191.5	0.4980	1.0845	1.5825	791.3
130	347.32	0.01796	3.455	318.81	872.9	1191.7	0.4995	1.0817	1.5812	790.2
132	348.48	0.01797	3.405	320.04	872.0	1192.0	0.5010	1.0790	1.5800	789.2
134	349.64	0.01799	3.357	321.25	871.0	1192.2	0.5025	1.0762	1.5787	788.2
136	350.78	0.01800	3.310	322.45	870.1	1192.5	0.5040	1.0735	1.5775	787.2
138	351.91	0.01801	3.264	323.64	869.1	1192.7	0.5054	1.0709	1.5763	786.2
140	353.02	0.01802	3.220	324.82	868.2	1193.0	0.5069	1.0682	1.5751	785.2
142	354.12	0.01804	3.177	325.98	867.2	1193.2	0.5083	1.0657	1.5740	784.3
144	355.21	0.01805	3.134	327.13	866.3	1193.4	0.5097	1.0631	1.5728	783.3
146	356.29	0.01806	3.094	328.27	865.3	1193.6	0.5111	1.0605	1.5716	782.3
148	357.36	0.01808	3.054	329.39	864.5	1193.9	0.5124	1.0580	1.5705	781.4
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694	780.5
152	359.46	0.01810	2.977	331.61	862.7	1194.3	0.5151	1.0532	1.5683	779.5
154	360.49	0.01812	2.940	332.70	861.8	1194.5	0.5165	1.0507	1.5672	778.5
156	361.52	0.01813	2.904	333.79	860.9	1194.7	0.5178	1.0483	1.5661	777.6
158	362.03	0.01814	2.869	334.86	860.0	1194.9	0.5191	1.0459	1.5650	776.8
160	363.53	0.01815	2.834	335.93	859.2	1195.1	0.5204	1.0436	1.5640	775.8
162	364.53	0.01817	2.801	336.98	858.3	1195.3	0.5216	1.0414	1.5630	775.0
164	365.51	0.01818	2.768	338.02	857.5	1195.5	0.5229	1.0391	1.5620	774.1
166	366.48	0.01819	2.736	339.05	856.6	1195.7	0.5241	1.0369	1.5610	773.2
168	367.45	0.01820	2.705	340.07	855.7	1195.8	0.5254	1.0346	1.5600	772.3
170	368.41	0.01822	2.675	341.09	854.9	1196.0	0.5266	1.0324	1.5590	771.4
172	369.35	0.01823	2.645	342.10	854.1	1196.2	0.5278	1.0302	1.5580	770.5
174	370.29	0.01824	2.616	343.10	853.3	1196.4	0.5290	1.0280	1.5570	769.7
176	371.22	0.01825	2.587	344.09	852.4	1196.5	0.5302	1.0259	1.5561	768.8
178	372.14	0.01826	2.559	345.06	851.6	1196.7	0.5313	1.0238	1.5551	767.9

Table 2-20
(continued)

Abs press, psi	Temp, deg F	Specific volume		Enthalpy			Entropy			Internal energy
		Liquid	Vapor	Liquid	Evap	Vapor	Liquid	Evap	Vapor	Evap
180	373.06	0.01827	2.532	346.03	850.8	1196.9	0.5325	1.0217	1.5542	767.1
182	373.96	0.01829	2.505	347.00	850.0	1197.0	0.5336	1.0196	1.5532	766.2
184	374.86	0.01830	2.479	347.96	849.2	1197.2	0.5348	1.0175	1.5523	765.4
186	375.75	0.01831	2.454	348.92	848.4	1197.3	0.5359	1.0155	1.5514	764.6
188	376.64	0.01832	2.429	349.86	847.6	1197.5	0.5370	1.0136	1.5506	763.8
190	377.51	0.01833	2.404	350.79	846.8	1197.6	0.5381	1.0116	1.5497	763.0
192	378.38	0.01834	2.380	351.72	846.1	1197.8	0.5392	1.0096	1.5488	762.1
194	379.24	0.01835	2.356	352.64	845.3	1197.9	0.5403	1.0076	1.5479	761.3
196	380.10	0.01836	2.333	353.55	844.5	1198.1	0.5414	1.0056	1.5470	760.6
198	380.95	0.01838	2.310	354.46	843.7	1198.2	0.5425	1.0037	1.5462	759.8
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	759.0
205	383.86	0.01842	2.234	357.58	841.1	1198.7	0.5461	0.9971	1.5432	757.1
210	385.90	0.01844	2.183	359.77	839.2	1199.0	0.5487	0.9925	1.5412	755.2
215	387.89	0.01847	2.134	361.91	837.4	1199.3	0.5512	0.9880	1.5392	753.2
220	389.86	0.01850	2.087	364.02	835.6	1199.6	0.5537	0.9835	1.5372	751.3
225	391.79	0.01852	2.0422	366.09	833.8	1199.9	0.5561	0.9792	1.5353	749.5
230	393.68	0.01854	1.9992	368.13	832.0	1200.1	0.5585	0.9750	1.5334	747.7
235	395.54	0.01857	1.9579	370.14	830.3	1200.4	0.5608	0.9708	1.5316	745.9
240	397.37	0.01860	1.9183	372.12	828.5	1200.6	0.5631	0.9667	1.5298	744.1
245	399.18	0.01863	1.8803	374.08	826.8	1200.9	0.5653	0.9627	1.5280	742.4
250	400.95	0.01865	1.8438	376.00	825.1	1201.1	0.5675	0.9588	1.5263	740.7
260	404.42	0.01870	1.7748	379.76	821.8	1201.5	0.5719	0.9510	1.5229	737.3
270	407.78	0.01875	1.7107	383.42	818.5	1201.9	0.5760	0.9436	1.5196	733.9
280	411.05	0.01880	1.6511	386.98	815.3	1202.3	0.5801	0.9363	1.5164	730.7
290	414.23	0.01885	1.5954	390.46	812.1	1202.6	0.5841	0.9292	1.5133	727.5
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	724.3
320	423.29	0.01899	1.4485	400.39	803.0	1203.4	0.5952	0.9094	1.5046	718.3
340	428.97	0.01908	1.3645	406.66	797.1	1203.7	0.6022	0.8970	1.4992	712.4
360	434.140	0.01917	1.2895	412.67	791.4	1204.1	0.6090	0.8851	1.4941	706.8
380	439.60	0.01925	1.2222	418.45	785.8	1204.3	0.6153	0.8738	1.4891	701.3
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	695.9
420	449.39	0.0194	1.1061	429.4	775.2	1204.6	0.6272	0.8527	1.4799	690.8
440	454.02	0.0195	1.0556	434.6	770.0	1204.6	0.6329	0.8426	1.4755	685.7
460	458.50	0.0196	1.0094	439.7	764.9	1204.6	0.6383	0.8330	1.4713	680.7
480	462.82	0.0197	0.9670	444.6	759.9	1204.6	0.6436	0.8237	1.4673	675.7

Table 2-20
(continued)

Abs press, psi	Temp, deg F	Specific volume		Enthalpy			Entropy			Internal energy
		Liquid	Vapor	Liquid	Evap	Vapor	Liquid	Evap	Vapor	Evap
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	1118.6
520	471.07	0.0198	0.8915	454.1	750.1	1204.2	0.6536	0.8060	1.4596	1118.4
540	475.01	0.0199	0.8578	458.6	745.4	1204.0	0.6584	0.7976	1.4560	1118.3
560	478.85	0.0200	0.8265	463.0	740.8	1203.8	0.6631	0.7893	1.4524	1118.2
580	482.58	0.0201	0.7973	467.4	736.1	1203.5	0.6676	0.7813	1.4489	1118.0
600	486.21	0.0201	0.7698	471.6	731.6	1203.2	0.6720	0.7734	1.4454	1117.7
620	489.75	0.0202	0.7440	475.7	727.2	1202.9	0.6763	0.7658	1.4421	1117.5
640	493.21	0.0203	0.7198	479.8	722.7	1202.5	0.6805	0.7584	1.4389	1117.3
660	496.58	0.0204	0.6971	483.8	718.3	1202.1	0.6846	0.7512	1.4358	1117.0
680	499.88	0.0204	0.6757	487.7	714.0	1201.7	0.6886	0.7441	1.4327	1116.7
700	503.10	0.0205	0.6554	491.5	709.7	1201.2	0.6925	0.7371	1.4296	1116.3
720	506.25	0.0206	0.6362	495.3	705.4	1200.7	0.6963	0.7303	1.4266	1116.0
740	509.34	0.0207	0.6180	499.0	701.2	1200.2	0.7001	0.7237	1.4237	1115.6
760	512.36	0.0207	0.6007	502.6	697.1	1199.7	0.7037	0.7172	1.4209	1115.2
780	515.33	0.0208	0.5843	506.2	692.9	1199.1	0.7073	0.7108	1.4181	1114.8
800	518.23	0.0209	0.5687	509.7	688.9	1198.6	0.7108	0.7045	1.4153	1114.4
820	521.08	0.0109	0.5538	513.2	684.8	1198.0	0.7143	0.6983	1.4126	1114.0
840	523.88	0.0210	0.5396	516.6	680.8	1197.4	0.7177	0.6922	1.4099	1113.6
860	526.63	0.0211	0.5260	520.0	676.8	1196.8	0.7210	0.6862	1.4072	1113.1
880	529.33	0.0212	0.5130	523.3	672.8	1196.1	0.7243	0.6803	1.4046	1112.6
900	531.98	0.0212	0.5006	526.6	668.8	1195.4	0.7275	0.6744	1.4020	1112.1
920	534.59	0.0213	0.4886	529.8	664.9	1194.7	0.7307	0.6687	1.3995	1111.5
940	537.16	0.0214	0.4772	533.0	661.0	1194.0	0.7339	0.6631	1.3970	1111.0
960	539.68	0.0214	0.4663	536.2	657.1	1193.3	0.7370	0.6576	1.3945	1110.5
980	542.17	0.0215	0.4557	539.3	653.3	1192.6	0.7400	0.6521	1.3921	1110.0
1,000	544.61	0.0216	0.4456	542.4	649.4	1191.8	0.7430	0.6467	1.3897	1109.4
1,050	550.57	0.0218	0.4218	550.0	639.9	1189.9	0.7504	0.6334	1.3838	1108.0
1,100	556.31	0.0220	0.4001	557.4	630.4	1187.8	0.7575	0.6205	1.3780	1106.4
1,150	561.86	0.0221	0.3802	564.6	621.0	1185.6	0.7644	0.6079	1.3723	1104.7
1,200	567.22	0.0223	0.3619	571.7	611.7	1183.4	0.7711	0.5956	1.3667	1103.0
1,250	572.42	0.0225	0.3450	578.6	602.4	1181.0	0.7776	0.5836	1.3612	1101.2
1,300	577.46	0.0227	0.3293	585.4	593.2	1178.6	0.7840	0.5719	1.3559	1099.4
1,350	582.35	0.0229	0.3148	592.1	584.0	1176.1	0.7902	0.5604	1.3506	1097.5
1,400	587.10	0.0231	0.3012	598.7	574.7	1173.4	0.7963	0.5491	1.3454	1095.4
1,450	591.73	0.0233	0.2884	605.2	565.5	1170.7	0.8023	0.5379	1.3402	1093.3
1,500	596.23	0.0235	0.2760	611.6	556.3	1167.9	0.8082	0.5269	1.3351	1091.0
1,600	604.90	0.0239	0.2548	624.1	538.0	1162.1	0.8196	0.5053	1.3249	1086.7
1,700	613.15	0.0243	0.2304	636.3	519.6	1155.9	0.8306	0.4843	1.3149	1081.8
1,800	621.03	0.0247	0.2179	648.3	501.1	1149.4	0.8412	0.4637	1.3049	1076.8
1,900	628.58	0.0252	0.2021	660.1	482.4	1142.4	0.8516	0.4433	1.2949	1071.4
2,000	635.82	0.0257	0.1878	671.7	463.4	1135.1	0.8619	0.4230	1.2849	1065.0
2,200	649.46	0.0268	0.1625	694.8	424.4	1119.2	0.8820	0.3826	1.2646	1053.1
2,400	662.12	0.0280	0.1407	718.4	382.7	1101.1	0.9023	0.3411	1.2434	1038.6
2,600	673.94	0.0295	0.1213	743.0	337.2	1080.2	0.9232	0.2973	1.2205	1021.9
2,800	684.99	0.0310	0.1035	770.1	284.7	1054.8	0.9459	0.2487	1.1946	1001.2
3,000	695.36	0.0346	0.0858	802.5	217.8	1020.3	0.9731	0.1885	1.1615	972.7
3,200	705.11	0.0444	0.0580	872.4	62.0	934.4	1.0320	0.0532	1.0852	898.4
3,206.2	705.40	0.0503	0.0503	902.7	0	902.7	1.0580	0	1.0580	872.9

Table 2-21
Superheated Steam Tables
(v = specific volume, cu ft per lb; h = enthalpy, Btu per lb; s = entropy)

Pressure, psia (Saturation temp. deg F)		Temperature of steam, deg F								
		340	380	420	460	500	550	600	650	700
20	v	23.60	24.82	26.04	27.25	28.46	29.97	31.47	32.97	34.47
(227.96)	h	1210.8	1229.7	1248.7	1267.6	1286.6	1310.5	1334.4	1358.6	1382.9
	s	1.8053	1.8285	1.8505	1.8716	1.8918	1.9160	1.9392	1.9671	1.9829
40	v	11.684	12.315	12.938	13.555	14.168	14.930	15.688	16.444	17.198
(267.25)	h	1207.0	1226.7	1246.2	1265.5	1284.8	1309.0	1333.1	1357.4	1361.9
	s	1.7252	1.7493	1.7719	1.7934	1.8140	1.8385	1.8619	1.8843	1.9058
60	v	7.708	8.143	8.569	8.988	9.403	9.917	10.427	10.935	11.441
(292.71)	h	1203.0	1223.6	1243.6	1263.4	1283.0	1307.4	1331.8	1356.3	1380.9
	s	1.6766	1.7135	1.7250	1.7470	1.7678	1.7927	1.8162	1.8388	1.8605
80	v	5.718	6.055	6.383	6.704	7.020	7.410	7.797	8.180	8.562
(312.03)	h	1198.8	1220.3	1240.9	1261.1	1281.1	1305.8	1330.5	1355.1	1379.9
	s	1.6407	1.6669	1.6909	1.7134	1.7346	1.7598	1.7836	1.8063	1.8281
100	v	4.521	4.801	5.071	5.333	5.589	5.906	6.218	6.527	6.835
(327.81)	h	1194.3	1216.8	1238.1	1258.8	1279.1	1304.2	1329.1	1354.0	1378.9
	s	1.6117	1.6391	1.6639	1.6869	1.7085	1.7340	1.7581	1.7610	1.8029
120	v	......	3.964	4.195	4.418	4.636	4.902	5.165	5.426	5.683
(341.25)	h	......	1213.2	1235.3	1256.5	1277.2	1302.6	1327.7	1352.8	1377.8
	s	......	1.6156	1.6413	1.6649	1.6869	1.7127	1.7370	1.7801	1.7822
140	v	......	3.365	3.569	3.764	3.954	4.186	4.413	4.638	4.861
(353.02)	h	......	1209.4	1232.3	1254.1	1275.2	1300.9	1326.4	1351.6	1376.8
	s	......	1.5950	1.6217	1.6458	1.6683	1.6945	1.7190	1.7423	1.7645
160	v	......	2.914	3.098	3.273	3.443	3.648	3.849	4.048	4.244
(363.53)	h	......	1205.5	1229.3	1251.6	1273.1	1299.3	1325.0	1350.4	1375.7
	s	......	1.5766	1.6042	1.6291	1.6519	1.6785	1.7033	1.7268	1.7491
180	v	......	2.563	2.732	2.891	3.044	3.230	3.411	3.588	3.764
(373.06)	h	......	1201.4	1226.1	1249.1	1271.0	1297.6	1323.5	1349.2	1374.7
	s	......	1.5596	1.5884	1.6139	1.6373	1.6642	1.6894	1.7130	1.7355
200	v	......	......	2.438	2.585	2.726	2.895	3.060	3.221	3.380
(381.79)	h	......	......	1222.9	1246.5	1268.9	1295.8	1322.1	1348.0	1373.6
	s	......	......	1.5738	1.6001	1.6240	1.6513	1.6767	1.7006	1.7232
220	v	......	......	2.198	2.335	2.465	2.621	2.772	2.920	3.066
(389.86)	h	......	......	1219.5	1243.8	1266.7	1294.1	1320.7	1346.8	1372.6
	s	......	......	1.5603	1.5874	1.6117	1.6395	1.6652	1.6892	1.7120
260	v	......	......	1.8257	1.9483	2.063	2.199	2.330	2.457	2.582
(404.42)	h	......	......	1212.4	1238.3	1262.3	1290.5	1317.7	1344.3	1370.4
	s	......	......	1.5354	1.5642	1.5897	1.6184	1.6447	1.6692	1.6922
300	v	......	......	1.5513	1.6638	1.7675	1.8891	2.005	2.118	2.227
(417.33)	h	......	......	1204.8	1232.5	1257.6	1286.8	1314.7	1341.8	1368.3
	s	......	......	1.5126	1.5434	1.5701	1.5998	1.6268	1.6517	1.6751
350	v	......	......	......	1.3984	1.4923	1.6010	1.7036	1.8021	1.8980
(431.72)	h	......	......	......	1224.8	1251.5	1282.1	1310.9	1338.5	1365.5
	s	......	......	......	1.5197	1.5481	1.5792	1.6070	1.6325	1.6563
400	v	......	......	......	1.1978	1.2851	1.3843	1.4770	1.5654	1.6508
(444.59)	h	......	......	......	1216.5	1245.1	1277.2	1306.9	1335.2	1362.7
	s	......	......	......	1.4977	1.5281	1.5607	1.5894	1.6155	1.6398

SOURCE: Baumeister T., and Marks, L. S., eds., *Standard Handbook for Mechanical Engineers*, Seventh edition, McGraw-Hill Book Co., New York, 1967.

Table 2-21
(continued)

Pressure, psia (Saturation temp. deg F)		Temperature of steam, deg F								
		500	550	600	650	700	750	800	900	1,000
450	v	1.1231	1.2154	1.3005	1.3810	1.4584	1.5337	1.6074	1.7516	1.8928
(456.28)	h	1238.4	1272.0	1302.8	1331.9	1359.9	1387.3	1414.3	1467.7	1521.0
	s	1.5095	1.5437	1.5735	1.6003	1.6250	1.6481	1.6699	1.7108	1.7486
500	v	0.9927	1.0798	1.1591	1.2333	1.3044	1.3732	1.4405	1.5715	1.6996
(467.01)	h	1231.3	1266.7	1298.6	1328.4	1357.0	1384.8	1412.1	1466.0	1519.6
	s	1.4919	1.5279	1.5588	1.5663	1.6115	1.6350	1.6571	1.6982	1.7363
550	v	0.8852	0.9686	1.0431	1.1124	1.1783	1.2419	1.3038	1.4241	1.5414
(476.94)	h	1223.7	1261.2	1294.3	1324.9	1354.0	1382.3	1409.9	1464.3	1518.2
	s	1.4751	1.5131	1.5451	1.5734	1.5991	1.6228	1.6452	1.6868	1.7250
600	v	0.7947	0.8753	0.9463	1.0115	1.0732	1.1324	1.1899	1.3013	1.4096
(486.21)	h	1215.7	1255.5	1289.9	1321.3	1351.1	1379.7	1407.7	1462.5	1516.7
	s	1.4586	1.4990	1.5323	1.5613	1.5875	1.6117	1.6343	1.6762	1.7147
700	v	......	0.7277	0.7934	0.8525	0.9077	0.9601	1.0108	1.1082	1.2024
(503.10)	h	......	1243.2	1280.6	1313.9	1345.0	1374.5	1403.2	1459.0	1513.9
	s	......	1.4722	1.5084	1.5391	1.5665	1.5914	1.6147	1.6573	1.6963
800	v	......	0.6154	0.6779	0.7328	0.7833	0.8308	0.8763	0.9633	1.0470
(518.23)	h	......	1229.8	1270.7	1306.2	1338.6	1369.2	1398.6	1455.4	1511.0
	s	......	1.4467	1.4863	1.5190	1.5476	1.5734	1.5972	1.6407	1.6801
900	v	......	0.5264	0.5873	0.6393	0.6863	0.7300	0.7716	0.8506	0.9262
(531.98)	h	......	1215.0	1260.1	1298.0	1332.1	1363.7	1393.9	1451.8	1508.1
	s	......	1.4216	1.4653	1.5002	1.5303	1.5570	1.5814	1.6257	1.6656
1,000	v	......	0.4533	0.5140	0.5640	0.6084	0.6492	0.6878	0.7604	0.8294
(544.61)	h	......	1198.3	1248.8	1289.5	1325.3	1358.1	1389.2	1448.2	1505.1
	s	......	1.3961	1.4450	1.4825	1.5141	1.5418	1.5670	1.6121	1.6525
1,100	v	......	......	0.4632	0.5020	0.5445	0.5830	0.6191	0.6866	0.7503
(556.30)	h	......	......	1236.7	1280.5	1318.3	1352.4	1384.3	1444.5	1502.2
	s	......	......	1.4251	1.4656	1.4989	1.5276	1.5535	1.5995	1.6405
1,200	v	......	......	0.4016	0.4498	0.4909	0.5277	0.5617	0.6250	0.6843
(567.22)	h	......	......	1223.5	1271.0	1311.0	1364.4	1379.3	1440.7	1499.2
	s	......	......	1.4052	1.4491	1.4843	1.5142	1.5409	1.5879	1.6293
1,400	v	......	......	0.3174	0.3668	0.4062	0.4403	0.4714	0.5281	0.5805
(587.10)	h	......	......	1193.0	1250.6	1295.5	1334.0	1369.1	1433.1	1493.2
	s	......	......	1.3639	1.4171	1.4567	1.4893	1.5177	1.5666	1.6093
1,600	v	......	......	......	0.3027	0.3417	0.3743	0.4034	0.4553	0.5027
(604.90)	h	......	......	......	1227.3	1278.7	1320.9	1358.4	1425.3	1487.0
	s	......	......	......	1.3800	1.4303	1.4660	1.4964	1.5476	1.5914
1,800	v	......	......	......	0.2506	0.2907	0.3225	0.3502	0.3986	0.4421
(621.03)	h	......	......	......	1200.3	1260.3	1307.0	1347.2	1417.4	1480.8
	s	......	......	......	1.3515	1.4044	1.4438	1.4765	1.5301	1.5752
2,000	v	......	......	......	0.2058	0.2489	0.2806	0.3074	0.3532	0.5935
(635.82)	h	......	......	......	1167.0	1240.0	1292.0	1335.5	1409.2	1474.5
	s	......	......	......	1.3139	1.3783	1.4223	1.4576	0.5139	1.5380
2,200	v	......	......	......	0.1633	0.2135	0.2457	0.2721	0.3159	0.3538
(649.46)	h	......	......	......	1121.0	1217.4	1276.0	1323.3	1400.8	1468.2
	s	......	......	......	1.2665	1.3515	1.4010	1.4393	1.4986	1.5465

Table 2-22
Steam Table for Use in Condenser Calculations

Temp, deg F, t	Abs pressure: Psi (p)	Abs pressure: In. Hg (p)	Specific volume: Sat vapor, V_g	Enthalpy: Sat liquid, h_f	Enthalpy: Evap, h_{fg}	Enthalpy: Sat vapor, h_g	Entropy: Sat liquid, s_f	Entropy: Sat vapor, s_g
50	0.17811	0.3626	1703.2	18.07	1065.6	1083.7	0.0361	2.1264
52	0.19182	0.3906	1587.6	20.07	1064.4	1084.5	0.0400	2.1199
54	0.20642	0.4203	1481.0	22.07	1063.3	1085.4	0.0439	2.1136
56	0.2220	0.4520	1382.4	24.06	1062.2	1086.3	0.0478	2.1072
58	0.2386	0.4858	1291.1	26.06	1061.0	1087.1	0.0517	2.1010
60	0.2563	0.5218	1206.7	28.06	1059.9	1088.0	0.0555	2.0948
62	0.2751	0.5601	1128.4	30.05	1058.8	1088.9	0.0593	2.0886
64	0.2951	0.6009	1055.7	32.05	1057.6	1089.7	0.0632	2.0826
66	0.3164	0.6442	988.4	34.05	1056.5	1090.6	0.0670	2.0766
68	0.3390	0.6903	925.9	36.04	1055.5	1091.5	0.0708	2.0706
70	0.3631	0.7392	867.9	38.04	1054.3	1092.3	0.0745	2.0647
72	0.3886	0.7912	813.9	40.04	1053.2	1093.2	0.0783	2.0588
74	0.4156	0.8462	763.8	42.03	1052.1	1094.1	0.0820	2.0530
76	0.4443	0.9046	717.1	44.03	1050.9	1094.9	0.0858	2.0473
78	0.4747	0.9666	673.6	46.02	1049.8	1095.8	0.0895	2.0416
80	0.5069	1.0321	633.1	48.02	1048.6	1096.6	0.0932	2.0360
82	0.5410	1.1016	595.3	50.01	1047.5	1097.5	0.0969	2.0304
84	0.5771	1.1750	560.2	52.01	1046.4	1098.4	0.1005	2.0249
86	0.6152	1.2527	527.3	54.00	1045.2	1099.2	0.1042	2.0195
88	0.6556	1.3347	496.7	56.00	1044.1	1100.1	0.1079	2.0141
90	0.6982	1.4215	468.0	57.99	1042.9	1100.9	0.1115	2.0087
92	0.7432	1.5131	441.3	59.99	1041.8	1101.8	0.1151	2.0034
94	0.7906	1.6097	416.2	61.98	1040.7	1102.6	0.1187	1.9981
96	0.8407	1.7117	392.8	63.98	1039.5	1103.5	0.1223	1.9929
98	0.8935	1.8192	370.9	65.97	1038.4	1104.4	0.1259	1.9877
100	0.9492	1.9325	350.4	67.97	1037.2	1105.2	0.1295	1.9826
102	1.0078	2.0519	331.1	69.96	1036.1	1106.1	0.1330	1.9775
104	1.0695	2.1775	313.1	71.96	1034.9	1106.9	0.1366	1.9725
106	1.1345	2.3099	296.2	73.95	1033.8	1107.8	0.1401	1.9675
108	1.2029	2.4491	280.3	75.95	1032.7	1108.6	0.1436	1.9626
110	1.2748	2.5955	265.4	77.94	1031.6	1109.5	0.1471	1.9577
112	1.3504	2.7494	251.4	79.94	1030.4	1110.3	0.1506	1.9529
114	1.4298	2.9111	238.2	81.93	1029.2	1111.1	0.1541	1.9481
116	1.5130	3.0806	225.8	83.93	1028.1	1112.0	0.1576	1.9433
118	1.6006	3.2589	214.2	85.92	1026.9	1112.8	0.1610	1.9386
120	1.6924	3.4458	203.27	87.92	1025.8	1113.7	0.1645	1.9339
122	1.7888	3.6420	192.95	89.92	1024.6	1114.5	0.1679	1.9293
124	1.8897	3.8475	183.25	91.91	1023.4	1115.3	0.1714	1.9247
126	1.9955	4.0629	174.10	93.91	1022.3	1116.2	0.1748	1.9202
128	2.1064	4.2887	165.47	95.91	1021.1	1117.0	0.1782	1.9156
130	2.2225	4.5251	157.34	97.90	1020.0	1117.9	0.1816	1.9112
132	2.3440	4.7725	149.66	99.90	1018.8	1118.7	0.1849	1.9067
134	2.4712	5.0314	142.42	101.90	1017.6	1119.5	0.1883	1.9023
136	2.6042	5.3022	135.58	103.90	1016.4	1120.3	0.1917	1.8980
138	2.7432	5.5852	129.12	105.89	1015.3	1121.2	0.1950	1.8937
140	2.8886	5.8812	123.01	107.89	1014.1	1122.0	0.1984	1.8894
142	3.0440	6.1903	117.23	109.89	1012.9	1122.8	0.2016	1.8851
144	3.1990	6.5132	111.77	111.89	1011.7	1123.6	0.2049	1.8809
146	3.365	6.850	106.60	113.89	1010.6	1124.5	0.2083	1.8768
148	3.537	7.202	101.71	115.89	1009.4	1125.3	0.2116	1.8726
150	3.718	7.569	97.07	117.89	1008.2	1126.1	0.2149	1.8685

SOURCE: Baumeister T., and Marks, L. S., eds., *Standard Handbook for Mechanical Engineers*, Seventh edition, McGraw-Hill Book Co., New York, 1967.

Table 2-23
Properties of Carbon Dioxide
(h_f and s_f are measured from 32°F)

Temp. deg F, t	Pressure, psia, p	Density, lb per cu ft: Sat liquid	Sat vapor	Enthalpy, Btu: Sat liquid, h_f	Vaporization, h_{fg}	Sat vapor, h_g	Entropy: Sat liquid, s_f	Sat vapor, s_g
−40	145.87	69.8	1.64	−38.5	136.5	98.0	−0.0850	0.2400
−35	161.33	69.1	1.83	−35.8	134.3	98.5	−0.0793	0.2367
−30	177.97	68.3	2.02	−33.1	132.1	99.0	−0.0735	0.2336
−25	195.85	67.6	2.23	−30.4	129.8	99.4	−0.0676	0.2306
−20	215.02	66.9	2.44	−27.7	127.5	99.8	−0.0619	0.2277
−15	235.53	66.1	2.66	−24.9	125.0	100.1	−0.0560	0.2250
−10	257.46	65.3	2.91	−22.1	122.4	100.3	−0.0500	0.2220
− 5	280.85	64.5	3.17	−19.4	120.0	100.6	−0.0440	0.2198
0	305.76	63.6	3.46	−16.7	117.5	100.8	−0.0381	0.2173
5	332.2	62.8	3.77	−14.0	115.0	101.0	−0.0322	0.2151
10	360.4	61.9	4.12	−11.2	112.2	101.0	−0.0264	0.2124
15	390.2	61.0	4.49	− 8.4	109.4	101.0	−0.0204	0.2100
20	421.8	60.0	4.89	− 5.5	106.3	100.8	−0.0144	0.2071
25	455.3	59.0	5.33	− 2.5	103.1	100.6	−0.0083	0.2043
30	490.6	58.0	5.81	+ 0.4	99.7	100.1	−0.0021	0.2012
35	528.0	57.0	6.35	3.5	95.8	99.3	+0.0039	0.1975
40	567.3	55.9	6.91	6.6	91.8	98.4	0.0099	0.1934
45	608.9	54.7	7.60	9.8	87.5	97.3	0.0160	0.1892
50	652.7	53.4	8.37	12.9	83.2	96.1	0.0220	0.1852
55	698.8	52.1	9.27	16.1	78.7	94.8	0.0282	0.1809
60	747.4	50.7	10.2	19.4	74.0	93.4	0.0345	0.1767
65	798.6	49.1	11.3	22.9	68.9	91.8	0.0412	0.1724
70	852.4	47.3	12.6	26.6	62.7	89.3	0.0482	0.1665
75	909.3	45.1	14.2	30.9	54.8	85.7	0.0562	0.1587
80	969.3	42.4	16.2	35.6	44.0	79.6	0.0649	0.1464
85	1032.7	38.2	19.1	41.7	27.5	69.2	0.0761	0.1265
88	1072.1	32.9	25.4	Critical point at 88.43°F				

SOURCE: Baumeister T., and Marks, L. S., eds., *Standard Handbook for Mechanical Engineers,* Seventh edition, McGraw-Hill Book Co., New York, 1967.

Table 2-24
Properties of Propane and Butane

	Propane (C_3H_4) (Heat measurement are from 0°F)						Butane (C_4H_{10}) (Heat measurements are from 0°F)					
			Enthalpy, Btu per lb		Entropy				Enthalpy, Btu per lb		Entropy	
Temp, deg F	Pressure, psia	Specific volume of vapor, cu ft per lb	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g	Pressure, psia	Specific volume of vapor, cu ft per lb	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−70	7.37	12.9	−37.0	152.5	−0.086	0.400						
−60	9.72	9.93	−32.0	155.0	−0.074	0.393						
−50	12.6	7.74	−26.5	158.0	−0.061	0.389						
−40	16.2	6.13	−21.5	160.0	−0.049	0.384						
−30	20.3	4.93	−16.0	163.0	−0.036	0.380						
−20	25.4	4.00	−11.0	165.0	−0.024	0.377						
−10	31.4	3.26	−5.5	168.0	−0.012	0.374						
0	38.2	2.71	0	170.5	0.000	0.371	7.3	11.10	0	170.5	0.000	0.371
+10	46.0	2.27	5.5	173.5	0.012	0.370	9.2	8.95	5.5	174.0	0.011	0.370
20	55.5	1.90	11.0	176.0	0.024	0.368	11.6	7.23	10.5	177.5	0.022	0.370
30	66.3	1.60	17.0	179.0	0.035	0.366	14.4	5.90	16.0	181.5	0.033	0.371
40	78.0	1.37	23.0	182.0	0.047	0.366	17.7	4.88	21.5	185.0	0.044	0.371
50	91.8	1.18	29.0	185.0	0.059	0.365	21.6	4.07	27.0	188.5	0.056	0.373
60	107.1	1.01	35.0	188.0	0.070	0.364	26.3	3.40	33.0	192.5	0.067	0.374
70	124.0	0.883	41.0	190.5	0.082	0.364	31.6	2.88	38.5	196.0	0.078	0.375
80	142.8	0.770	47.5	193.5	0.093	0.364	37.6	2.46	44.5	199.5	0.089	0.376
90	164.0	0.673	54.0	196.5	0.105	0.364	44.5	2.10	51.0	203.0	0.100	0.377
100	187.0	0.591	60.5	199.0	0.116	0.363	52.2	1.81	57.0	206.5	0.111	0.378
110	212.0	0.521	67.0	201.0	0.128	0.363	60.8	1.58	63.5	210.5	0.122	0.380
120	240.0	0.459	73.5	202.5	0.140	0.363	70.8	1.38	70.0	213.5	0.134	0.382
130	......	......	......	.....	......	.....	81.4	1.21	76.5	217.0	0.145	0.384
140	......	......	......	.....	......	.....	92.6	1.07	83.5	221.0	0.157	0.386

SOURCE: Baumeister T., and Marks, L. S., eds., *Standard Handbook for Mechanical Engineers*, Seventh edition, McGraw-Hill Book Co., New York, 1967.

Table 2-25
Properties of Freon 11 and Freon 12

	Freon 11 (CChF) (Heat measurements are from –40°F)						Freon 12 ($CChF_2$) (Heat measurements are from –40°F)					
			Enthalpy, Btu per lb		Entropy				Enthalpy, Btu per lb		Entropy	
Temp, deg F	Pressure, psia	Specific volume of vapor, cu ft per lb	Liquid h_f	Vapor h_g	Liquid s_f	Vapor S_g	Pressure, psia	Specific volume of vapor, cu ft per lb	Liquid h_f	Vapor h_g	Liquid S_f	Vapor S_g
–40	0.739	44.2	0.00	87.48	0.0000	0.2085	9.3	3.91	0.00	73.50	0.0000	0.1752
–30	1.03	32.3	1.97	88.67	0.0046	0.2064	12.0	3.09	2.03	74.70	0.00471	0.1739
–20	1.42	24.1	3.94	89.87	0.0091	0.2046	15.3	2.47	4.07	75.87	0.00940	0.1727
–10	1.92	18.2	5.91	91.07	0.0136	0.2030	19.2	2.00	6.14	77.05	0.01403	0.1717
0	2.55	13.9	7.89	92.27	0.0179	0.2015	23.9	1.64	8.25	78.21	0.01869	0.1709
10	3.35	10.8	9.88	93.48	0.0222	0.2003	29.3	1.35	10.39	79.36	0.02328	0.1701
15	3.82	9.59	10.88	94.09	0.0244	0.1997	32.4	1.23	11.48	79.94	0.02556	0.1698
20	4.34	8.52	11.87	94.69	0.0264	0.1991	35.7	1.12	12.55	80.49	0.02783	0.1695
25	4.92	7.58	12.88	95.30	0.0285	0.1986	39.3	1.02	13.66	81.06	0.03008	0.1692
30	5.56	6.75	13.88	95.91	0.0306	0.1981	43.2	0.939	14.76	81.61	0.03233	0.1689
35	6.26	6.07	14.88	96.51	0.0326	0.1976	47.3	0.862	15.87	82.16	0.03458	0.1686
40	7.03	5.45	15.89	97.11	0.0346	0.1972	51.7	0.792	17.00	82.71	0.03680	0.1683
45	7.88	4.90	16.91	97.72	0.0366	0.1968	56.4	0.730	18.14	83.26	0.03903	0.1681
50	8.80	4.42	17.92	98.32	0.0386	0.1964	61.4	0.673	19.27	83.78	0.04126	0.1678
55	9.81	4.00	18.95	98.93	0.0406	0.1960	66.7	0.622	20.41	84.31	0.04348	0.1676
60	10.9	3.63	19.96	99.53	0.0426	0.1958	72.4	0.575	21.57	84.82	0.04568	0.1674
70	13.4	2.99	22.02	100.73	0.0465	0.1951	84.8	0.493	23.90	85.82	0.05009	0.1670
80	16.3	2.49	24.09	101.93	0.0504	0.1947	98.8	0.425	26.28	86.80	0.05446	0.1666
90	19.7	2.09	26.18	103.12	0.0542	0.1942	114.3	0.368	28.70	87.74	0.05882	0.1662
100	23.6	1.76	28.27	104.30	0.0580	0.1938	131.6	0.319	31.16	88.62	0.06316	0.1658
110	28.1	1.50	30.40	105.47	0.0617	0.1935	150.7	0.277	33.65	89.43	0.06749	0.1654
120	33.2	1.28	32.53	106.63	0.0654	0.1933	171.8	0.240	36.16	90.15	0.07180	0.1649
130	39.0	1.10	34.67	107.78	0.0691	0.1931	194.9	0.208	38.69	90.76	0.07607	0.1644

SOURCE: Baumeister T., and Marks, L. S., eds., *Standard Handbook for Mechanical Engineers*, Seventh edition, McGraw-Hill Book Co., New York, 1967.

(text continued from page 226)

GEOLOGICAL ENGINEERING

Geology is the study of the earth, its internal and surface composition, structure, and the earth processes that cause changes in composition and structure. The earth is constantly changing. The processes within the earth and the history of these processes are important factors in determining how minerals deposits were formed, where they accumulated, and how they have been preserved. The geology of the present composition and structure of the earth and, secondarily, the history of the processes that resulted in the present geology have become very important in the prediction of where accumulations of economically valuable hydrocarbons (oil and gas) may be found. Studies of surface geological fractures and the past processes coupled with such surface geophysical investigation techniques as seismic, gravity, magnetic, radioactive, electrical, and geochemical are used to locate probable subsurface target regions that might contain economically valuable accumulations of hydrocarbons. However, only by drilling a borehole from the surface to these subsurface regions is it possible to definitely assess whether hydrocarbons exist there. The borehole provides a direct fluid communication from these subsurface regions to the surface, where the fluid (if present) can be assessed for its economic value.

Not only is geology important in exploring for hydrocarbons, but also engineers must study the present composition and structure of the earth to successfully drill the borehole itself. Further, once hydrocarbons have been found and have proven to be economically recoverable, studies of the physical and chemical aspects of earth in such regions are important to the follow-on production and reservoir engineering. These studies help ensure that the accumulated hydrocarbons are recovered in an economic manner [24].

General Rock Types

The earth is composed of three general rock types: igneous, sedimentary, and metamorphic [24].

- *Igneous* rocks are the original rocks of the earth and were solidified from the molten mixture of materials that made up the earth prior to its cooling. Igneous rocks are very complex assemblages of minerals. Usually such rocks are very dense and have very few pores (or voids) which can accumulate or pass any type of fluid.
- *Sedimentary* rocks are aggregates of particles broken away from other rock masses which are exposed at or near the earth's surface to weathering processes. These particles are then transported by water, wind, or ice (glaciers) motion to new locations where they assemble eventually into a new rock mass. The origin of the rock mass prior to action by the weathering process can be igneous rock, another sedimentary rock, or metamorphic rock (see next rock type). During the process of weathering and transportation (particularly by water) compounds are precipitated chemically from the original rock mass or from materials within the water itself. In lake or seawater environments organisms provide such compounds. As the particles are deposited in slower moving waters, the chemical compounds provide a type of cement that ultimately binds the particles into a sedimentary rock mass. Sedimentary rock masses that are formed from water erosion and deposition by water in lake or ocean environments were originally laid down (i.e., deposited) in horizontal or near horizontal (deltas) layers.
- *Metamorphic* rocks are formed from either igneous, sedimentary, or possibly other metamorphic rock masses. These original rock masses are subjected to heat,

pressure, or chemically active gases or liquids which significantly alter the original rock to a new crystalline form.

In general, it is found from surface observations that about 75% of the land surface area is composed of sedimentary rocks. Most of the remaining land area is composed of igneous rock, with metamorphic rocks being a rather small percentage of the earth's surface-exposed rock. If the outer 10 mi of thickness of the earth is considered, it is estimated that 95% of this volume of rocks (including those exposed at the surface) are igneous rocks. Igneous rocks are the original rocks of the earth and are, therefore, the ancestors of all other rock types [24,25].

Historical Geology

It is estimated that the earth's age is in the neighborhood of 4 to 7 billion years. These estimates are basically derived from carbon-14, potassium-40, uranium-235, and uranium-238 dating of earth rocks and meteorites. The meteorites give important data as to the age of our solar system. Geologic time is felt to be represented by the presence of rock intervals in the geologic column (layers of rock formations in vertical depth) or by the absence of equivalent rocks in correlative columns in adjacent locations [25,26]. The two basic factors that are used to determine geologic time are:

1. The principle of uniformity, which states that internal and external processes affecting the earth today have been operating unchanged and at the same rates throughout the developmental history of earth. This means that a historical geologic event preserved in the rock record can be identified, in terms of its elapsed time of development, with similar events occurring at the present time. Therefore, rates of deposition, erosion, igneous emplacement, and structural development will be preserved in the geologic column and can be compared with current similar processes [25,26].
2. Relative time, which is based upon the occurrence of geologic events relative to each other. Dating in this manner requires the development of a sequence of events that could be established on the basis of obvious consecutive criteria. The identification of a geologic continuum such that the events within the sequence are sufficiently identifiable (e.g., uplifting, erosion, deposition) and widespread to have practical significance. Such dating in relative time allows events to be identified throughout the world [25,26]:
   - Superposition is fundamental to the study of layered rocks. This means that in a normal layered (sedimentary) rock sequence, the oldest rocks were deposited first and are at the bottom of the sequence. The younger rocks were deposited last and are at the top of the sequence.
   - Succession of flora and fauna refers to the deposition of sedimentary material, which will include the remains of plant and animal life that existed at the time of the deposition of these rock particles. The fossils of these plants and animals will be found in the rock formations that result from the deposition. The presence, absence, or change of the plant and animal life within a sequence of the geologic column provide important information that allows for the correlation of rock formations (and, thereby, relative time) from location to location. Also, the fossil records within sequences give important information regarding the evolution of life through geologic time.
   - Inclusion of one rock type (usually igneous) into surrounding rocks is invariably younger than the rocks it intrudes. Inclusions are useful in determining relative geologic ages.

- The term *cross-cutting relations* refers to pre-existing rocks that can be affected by later faulting or folding of the rock mass. Layered rock sequences must be in existence before the cross-cutting events occur. Cross-cutting is useful in determining relative geologic ages.
- Physiographic development of the surface of the earth refers to the landforms and shapes of the landscape. These surface features are subject to continuous change from constructive (e.g., uplift, volcanic activity, and deposition of sediments) and destructive (e.g., erosion) processes. Landform modifications are continuous and sequential. These modifications establish a predictable continuity that can be helpful in determining certain aspects of relative geologic ages.

The relationship between time units, time-rock units, and rock units is as follows:

Time Units	Time-Rock Units	Rock Units
Eon		
Era	Erathem	Group
Period	System	Formation
Epoch	Series	Member
Age	Stage	

This system for keeping track of these important units is used as the basis for the standard geologic time and the evolution of the animal life on earth. (See also Tables 2-26 and 2-27.) Table 2-28 gives the relationship between geologic time and important physical and evolutionary events that are used to aid in the identification of rock units in relative geologic time [26].

Petroleum Geology

Basically, circumstantial evidence has lead geologists and other scientists to believe that nearly all hydrocarbon (liquid and gas) deposits within the earth are strongly associated with sedimentary rocks. This evidence suggests that hydrocarbons in the earth are altered organic material derived from microscopic plant and animal life. This microscopic plant and animal life that has thrived on land and in lake and ocean regions of the earth through geologic time has been deposited with finely divided clastic sediments. In general it is felt that most of the important deposition (relative to hydrocarbon accumulations) has occurred in the marine environment in ocean margins. The organic material is buried and protected by the clay and silt sediments that accompany this material during deposition. The amount of organic material available during deposition dictates the amount of hydrocarbons that might be produced. The amount of sediments available for burial of the organic material depends upon the amount of erosion that occurs on the landmasses. The more rapid the burial of the organic material by the sediments, the more efficient the alterations through geologic time of the organic material to basic hydrocarbons.

Conversion of the organic material appears to be assisted by the pressure of burial depth, the temperature resulting from burial depth, and the type of bacterial action in a closed nonoxidizing chemical system. If burial action is not sufficient to eliminate oxygen, aerobic bacteria will act upon the organic material and destroy it. If burial is sufficient to nearly eliminate oxygen from reaching the organic material, anaerobic bacterial action involving oxygen from dissolved sulfates begins and a reducing

Table 2-26
The Standard Geologic Column [24]

Relative Geologic Time				
Era	*Period*		*Epoch*	*Atomic Time**
Cenozoic	Quaternary		Holocene	
			Pleistocene	2–3
	Tertiary		Pliocene	12
			Miocene	26
			Oligocene	37–38
			Eocene	53–54
			Paleocene	65
Mesozoic	Cretaceous		Late Early	136
	Jurassic		Late Middle Early	190–195
	Triassic		Late Middle Early	225
Paleozoic	Permian		Late Early	280
	Carboniferous Systems	Pennsylvanian	Late Middle Early	
		Mississippian	Late Early	345
	Devonian		Late Middle Early	395
	Silurian		Late Middle Early	430–440
	Ordovician		Late Middle Early	500
	Cambrian		Late Middle Early	600
Precambrian				3600

*Estimated ages of time boundaries (millions of years)

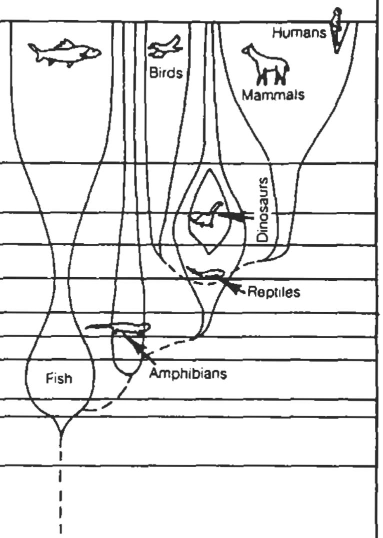
environment develops. It is felt that this reducing environment is necessary for altering the buried organic material and for beginning the process that ultimately leads to the accumulation of the hydrocarbon deposits found in the great sedimentary basins of the world [26–29].

Source Rocks

Abundant deposits of fine silt, clay, and organic material in ocean margins are the most likely sedimentary deposits that will have the required reducing environment for the alteration of the organic material. Such deposits eventually result in shale rock formations after the cementing of the particles takes place over geologic time. It is felt, therefore, that shale rock is the most likely source rock for hydrocarbons. The best source rocks are considered to be black shales originally deposited in a nonoxidizing, quiet marine environment. The deposition of organic material in nearly pure carbonate deposits may represent a possible second type of sedimentary deposit.

Table 2-27
Geologic Time and Evolution of Ancient Life [26]

Era	Approx. Age in Millions of Years (Radioactivity)	Period or System Period refers to a time measure; system refers to the rocks deposited during a period.		
Cenozoic		Recent (Holocene)		Neogene
		Pleistocene		
	7	Tertiary	Pliocene	
	26		Miocene	
	37-38		Oligocene	Paleogene
	53-54		Eocene	
			Paleocene	
	65			
Mesozoic		Cretaceous		
	136	Jurassic		
	190-195	Triassic		
	225			
Paleozoic		Permian		
	280	Pennsylvanian	Carboniferous	
	310	Mississippian		
	345	Devonian		
	395	Silurian		
	430-440	Ordovician		
	500	Cambrian		
	570			
Precambrian	700	First multi-celled organisms		
	3,400	First one-celled organisms		
	4,000	Approximate age of oldest rocks discovered		
	4,500	Approximate age of meteorites		



These deposits would result in carbonate rock (e.g., limestone). A third source rock possibility would be evaporite rocks (e.g., salt, gypsum, anhydrite), which often contain large organic concentrations when originally deposited [26–29].

Migration

The hydrocarbons in some altered form migrate from the source beds through other more porous and permeable beds to eventually accumulate in a rock called the *reservoir rock*. The initially altered (i.e., within the source beds) organic material may continue to alter as the material migrates. The hydrocarbon movement is probably the result of hydrodynamic pressure and gravity forces. As the source beds are compacted by increased burial pressures, the water and altered organic material are expelled. Water movement carries the hydrocarbons from the source beds into the reservoir, where the hydrocarbon establishes a position of equilibrium for the hydrodynamic and structural conditions [26–29].

Table 2-28
Geologic Time and Important Events [26]

Uniform Time Scale		Subdivisions Based on Strata/Time			Radiometric Dates (millions of years ago)	Outstanding Events	
			Systems/Periods	Series/Epochs		In Physical History	In Evolution of Living Things
0 Phanerozoic	Phanerozoic	Cenozoic	Quaternary	Recent or Holocene Pleistocene	0	Several glacial ages	Homo sapiens
			Tertiary	Pliocene	2?		Later hominids
				Miocene	6	Colorado River begins	Primitive hominids Grasses; grazing mammals
580				Oligocene	22	Mountains and basins in Nevada	
				Eocene	36	Yellowstone Park volcanism	Primitive horses
				Paleocene	58		
		Mesozoic	Cretaceous	(Many)	63	Rocky Mountains begin Lower Mississippi River begins	Spreading of mammals Dinosaurs extinct Flowering plants Climax of dinosaurs
			Jurassic		145		Birds
			Triassic		210	Atlantic Ocean begins	Conifers, cycads, primitive mammals Dinosaurs
Precambrian		Paleozoic	Permian		255	Appalachian Mountains climax	Mammal-like reptiles
			Pennsylvanian (Upper Carboniferous)		280		Coal forests, insects, amphibians, reptiles
			Mississippian (Lower Carboniferous)		320		
			Devonian		360		Amphibians
			Silurian		415		Land plants and land animals
			Ordovician		465	Appalachian Mountains begin	Primitive fishes
			Cambrian		520		Marine animals abundant
	Precambrian (Mainly igneous and metamorphic rocks; no worldwide subdivisions.)				580 1,000 2,000 3,000	Oldest dated rocks	Primitive marine animals Green algae Bacteria, blue green algae
~4,650	Birth of Planet Earth				4,650		

Accumulation

The accumulation of hydrocarbons occurs when, during migration through a porous and permeable rock, the migration process is impeded by the existence of an impermeable rock formation on the boundary (in the path of the migrating fluid) of the rock through which the migrating fluid is attempting to transverse. Once the hydrocarbons are trapped, the rock in which they have been trapped becomes the reservoir rock. Eventually the hydrocarbons accumulate in the reservoir rock and become stratified in accordance to their fluid phases and the amounts of formation water within the reservoir. The quantity of gas dissolved in the oil depends on the pressure and temperature in the reservoir. The hydrocarbons will accumulate in the highest portion of the reservoirs with the gaseous portion at the top, oil next, and water at the bottom (see Figure 2-39) [26–29].

Structural Geology

All sediments are deposited in basically the horizontal plane. Therefore, source beds and the beds that will eventually become reservoir rocks are originally nearly horizontal. Eventually, after these deposition beds became cemented into rock formations, tectonic events within the earth applied forces to the bedded rock formations within the sedimentary basins of the world. The causes of these tectonic events through geologic time in the historical past of the earth are not entirely understood. It is speculated that as the earth cooled, the outer surface of the sphere cooled from a liquid to a solid. The cooling process itself released gases which in turn released condensed liquids (water). These gases and liquids ultimately resulted in our atmosphere and abundant water at the earth's surface. The cooling solid crust of the earth likely cooled differentially, i.e., cooled more rapidly on the outer layers than on the inner layers. Most surface rock materials will shrink as they cool from the molten state. Therefore, the outer layers of the earth shrank at a greater rate than the inner layers, causing large cracks to form in the surface outer layers. The formation of the cracks and the differential cooling of the layers of the earth would

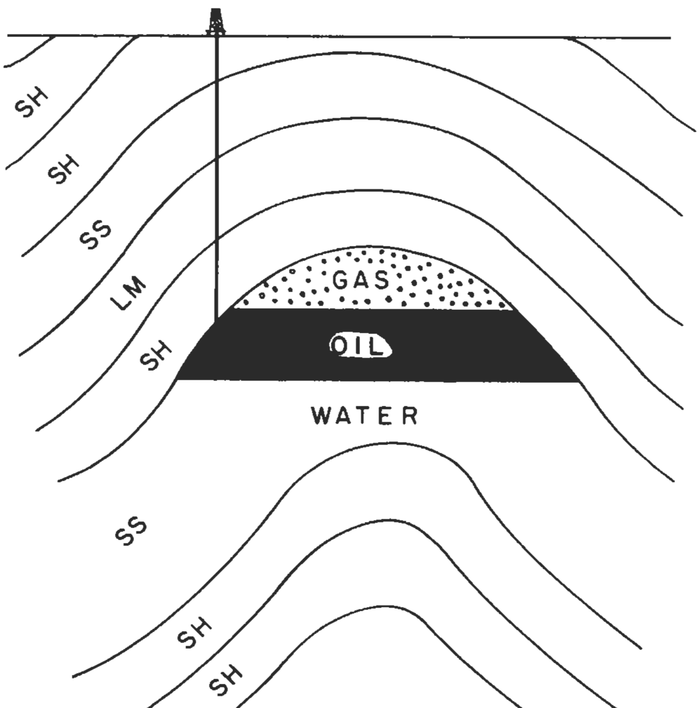


Figure 2-39. Anticlinal hydrocarbon accumulation.

likely cause a curling effect on the edges of large surfaces (plates) that had cracked. This curling effect at the edges of the plates (or at intermediate positions) could be partially responsible for the mountain building processes that have occurred throughout geology time.

The solid plates that resulted from the cooling process at the surface of the earth were able to "float" on the remaining molten inner portion of the earth. Because of the rotational motion of the earth about its own axis and the earth's motion in the solar system, inertial and gravitational forces have produced great interactive forces between the plates. It is speculated that these interactive forces have led to plate contact and situations where one plate has slid over another. The great forces created by plate tectonics are likely responsible for the forces that have resulted in the folding and faulting of the earth's crust [30].

Of interest is the detailed structural configurations that can form in sedimentary rock formations, because hydrocarbons are nearly always found in sedimentary rocks. The structural features of interest are faults and folds [26].

1. Faults are breaks in the earth's crust along which there has been measurable movement of one side of the break relative to the other. Fault terminology is as follows (see Figure 2-40):
   - *Dip*—the angle the fault plane makes with the horizontal, measured from the horizontal to the fault plane.
   - *Strike*—a line on the horizontal surface represented by the intersection of the fault plane and the horizontal surface. The strike line is always horizontal, and since it has direction, it is measured either by azimuth or bearing. Strike is always perpendicular to the dip.
   - *Heave*—the horizontal component of movement of the fault.
   - *Throw*—the vertical component of movement of the fault.
   - *Slip*—the actual linear movement along the fault plane.
   - *Hanging wall*—the block located above and bearing down on the fault surface.
   - *Footwall*—the block that occupies the position beneath the fault, regardless of whether the hanging wall has moved up or down.

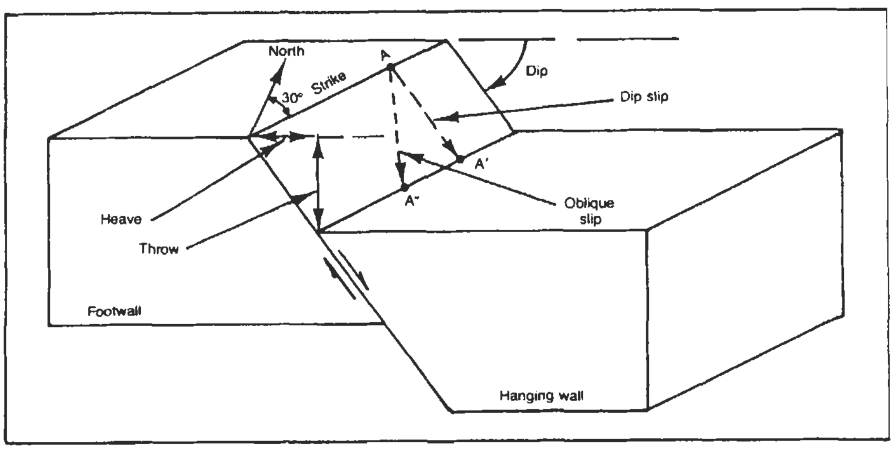


Figure 2-40. Fault terminology [26].

- *Normal faulting*—basically dominated by tension and gravity forces results in the hanging wall being displaced downward relative to the footwall.
- *Reverse faulting*—basically dominated by compression forces and, therefore, the hanging wall is moved up relative to the footwall. The reverse fault that dips at 30° or less becomes a thrust fault.
- *Strike-slip faulting*—occurs when the blocks slide laterally relative to each other.

2. Folds in layered rock formations consist of the deformation of layers without faulting. Folds are formed by compressional forces within the crust. Often faulting accompanies extensive folding. Fold terminology is as follows (see Figure 2-41).
   - *Anticline*—a fold with upward convexity.
   - *Syncline*—a fold that is concave upward.

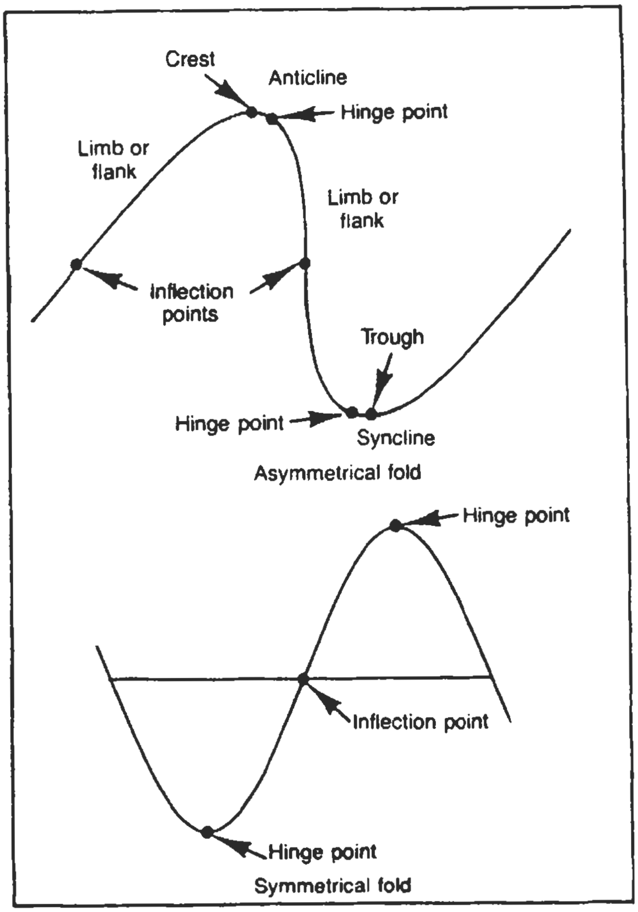


Figure 2-41. Folding terminology [26].

- *Hinge point*—the point of maximum curvature of a fold. The *hinge surface* is the locus of hinge lines within the fold.
- *Inflection point*—occurs when bed curvature in one direction changes to bed curvature in the opposite direction.
- *Limbs (or flanks) of a fold*—those portions adjacent to the inflection points of the fold.
- *Symmetrical fold*—a fold whose shape is a mirror image across the hinge point.
- *Asymmetrical fold*—a fold whose shape is not a mirror image across the huge point.
- *Recumbent fold*—characterized by a horizontal or nearly horizontal hinge surface (see Figure 2-42).
- *Overturned fold*—when the hinge surface is depressed below the horizontal (see Figure 2-42).
- *Concentric (parallel) folds*—rock formations parallel to each other such that their respective thicknesses remain constant (see Figure 2-43).

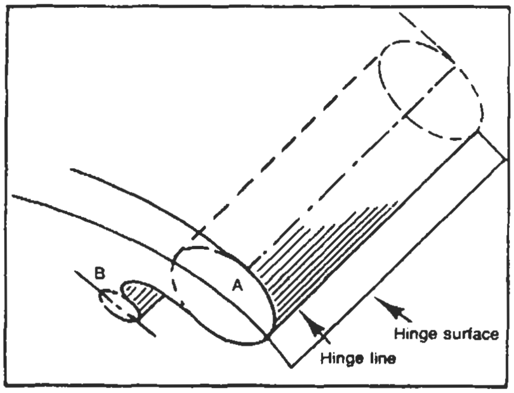


Figure 2-42. Recumbent (A) and overturned (B) folds [26].

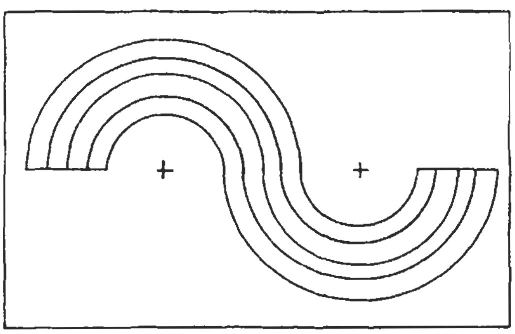


Figure 2-43. Concentric folds [26].

- *Nonparallel folds*—rock formations that do not have constant thickness along the fold (see Figure 2-44).
- *Similar folds*—folds that have the same geometric form, but where shear flow in the plastic beds has occurred (see Figure 2-45).

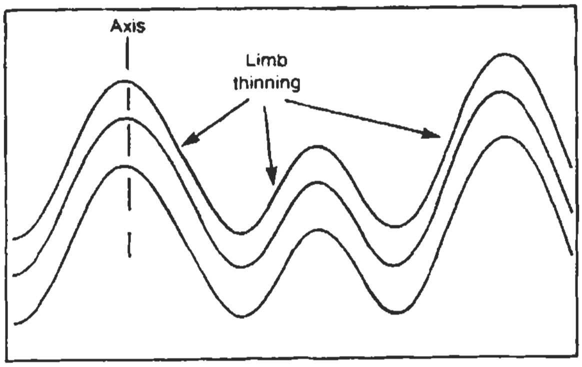


Figure 2-44. Nonparallel folds [26].

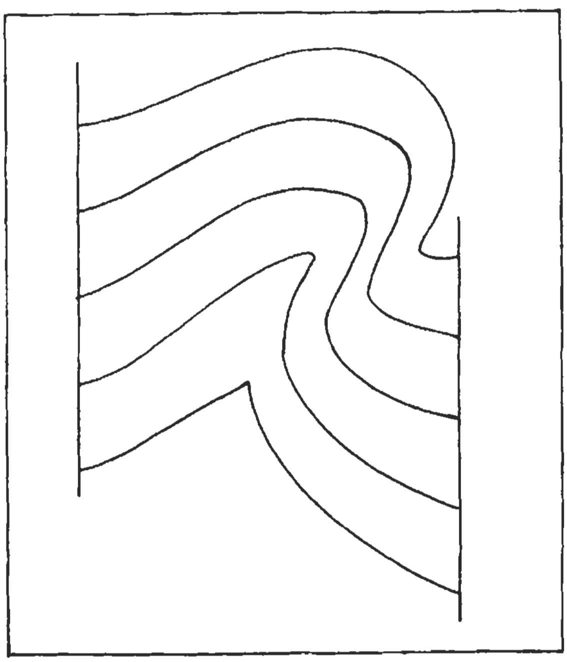


Figure 2-45. Similar folds [26].

- *Disharmonic folds*—folds in layered rock that have variable thickness and competence and, thereby, fold in accordance to their ability (see Figure 2-46).

Traps

A trap provides an impermeable barrier to the migration of hydrocarbons moving through a porous and permeable sedimentary bed. The trap allows the hydrocarbons to accumulate within the trap. The trap is a prerequisite for the formation of an accumulation of hydrocarbons and, therefore, for a reservoir [26].

1. Structural traps result from deformation of rock formations. Such deformations are the result of folding or faulting.
   - *Anticline trap*—a simple fold that traps hydrocarbons in its crest (see Figure 2-39). Anticlines are some of the most common productive structures. Overthrust anticlines are the most complex of this type of trap (see Figure 2-47). Such structural traps are associated with complex faulting and are prevalent in the overthrust belt provinces in the United States and Canada.
   - *Fault traps*—involve the movement of the reservoir rock formation to a position where the formation across the fault plane provides a seal preventing further migration of hydrocarbons (see Figure 2-48).
   - *Salt-related traps*—formed when the plastic salt formations deform into domelike structures under the overburden forces of the beds above the salt beds. Such plastic flowing (and bulging) of the salt beds deforms the rock formations above producing anticline structures and faults in the rock formation astride the domelike structures (see Figures 2-49 and 2-50).
2. Stratigraphic traps are formed by depositional and sedimentary factors. In such traps the depositional process and the follow-on cementing process, which changes the sediment bed into a rock, create porosity and permeability alterations in geometric forms that provide traps.

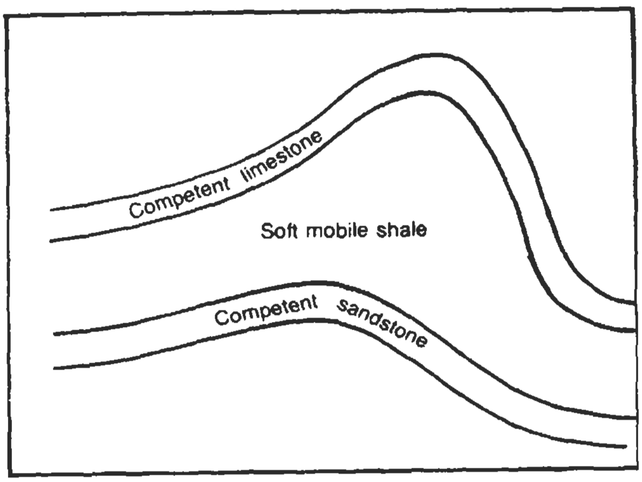


Figure 2-46. Disharmonic folds [26].

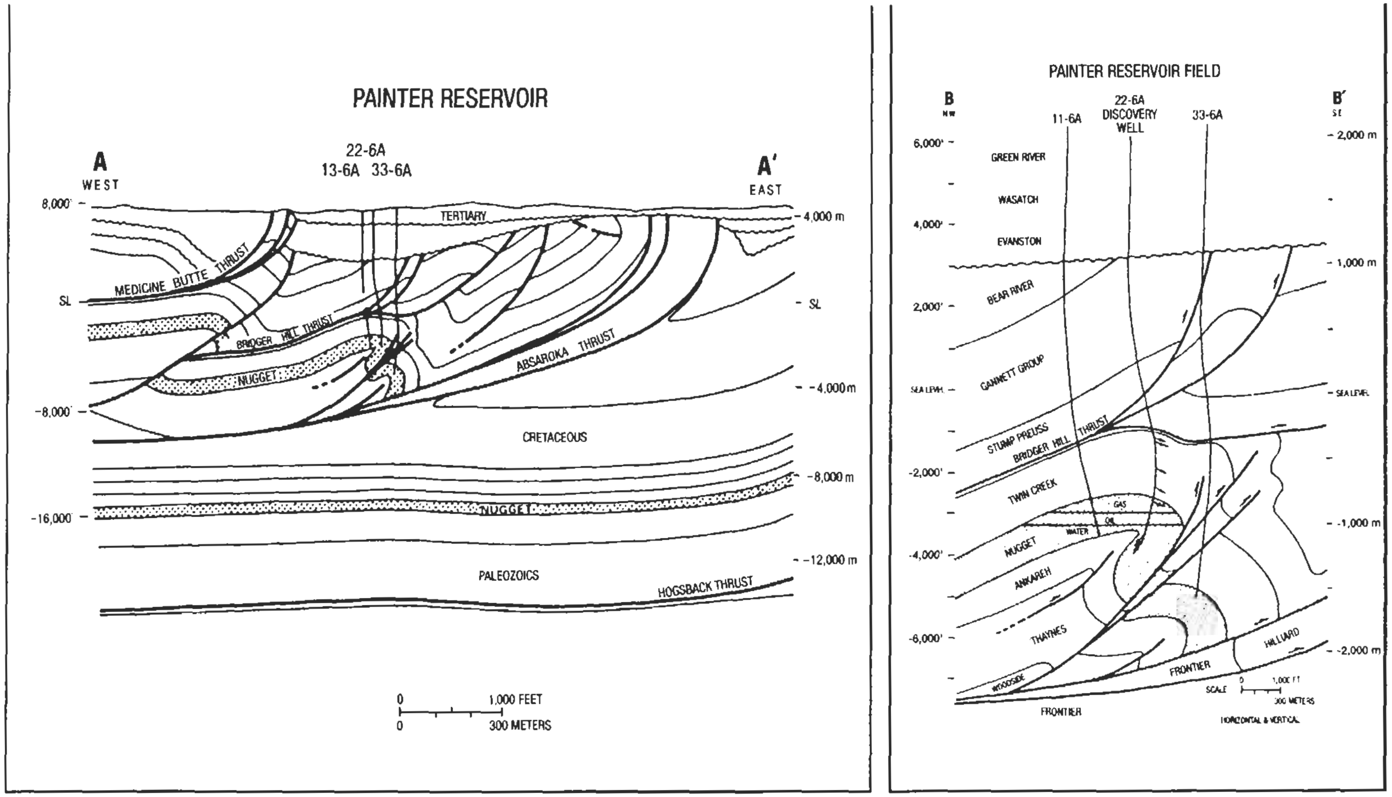


Figure 2-47. Overthrust structural trap (Painter Reservoir, Wyoming) [26].

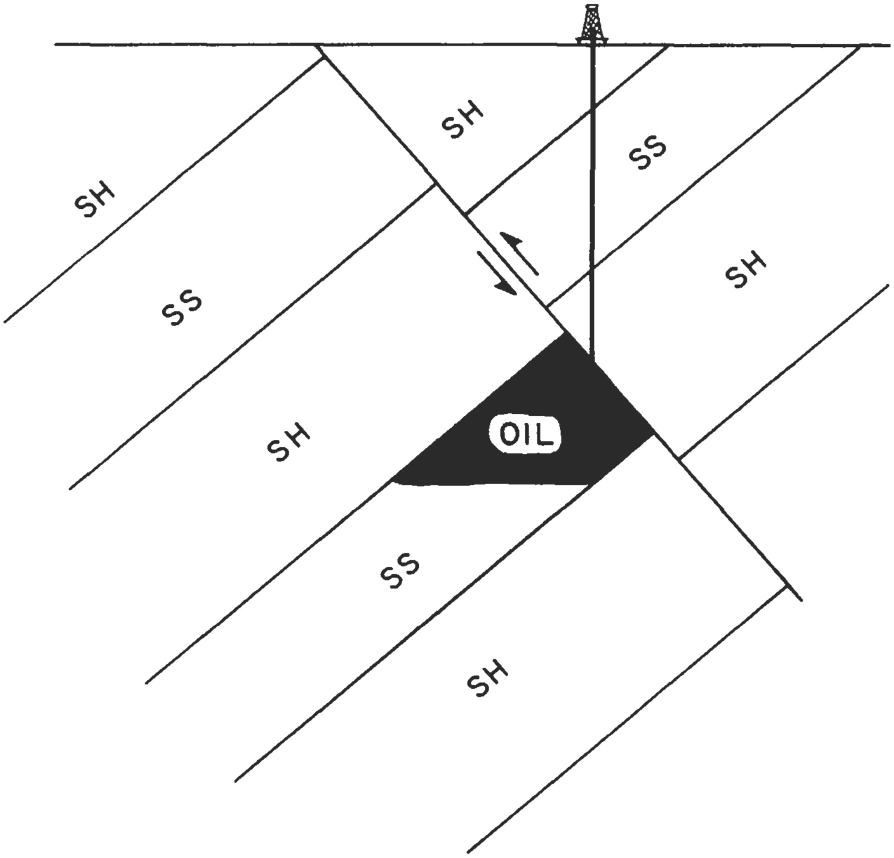


Figure 2-48. Fault structural traps.

- *Sand body traps*—finite sand bodies such as channel sands, river delta sands, and sea or ocean beach or barrier bar sands. These sand body traps are deposited over well-defined regions. As deposition continues on a wider regional basis of shale-forming deposits, the sand body becomes enclosed by shale and thus becomes a trap for fluids, particularly hydrocarbons (see Figure 2-51).
- *Reef traps*—important hydrocarbon-producing geological features. The porosity and permeability in reefs can be excellent. Again, as in sand body traps, reef traps are finite bodies that are deposited over well-defined regions. Continued deposition of silt and clay materials will eventually enclose such features in shale, allowing them to trap fluids, particularly hydrocarbons (see Figure 2-52).
- *Unconformity traps*—the sedimentary rock result of a period of deposition. A typical example would be when a sequence of horizontal beds is tilted, eroded,

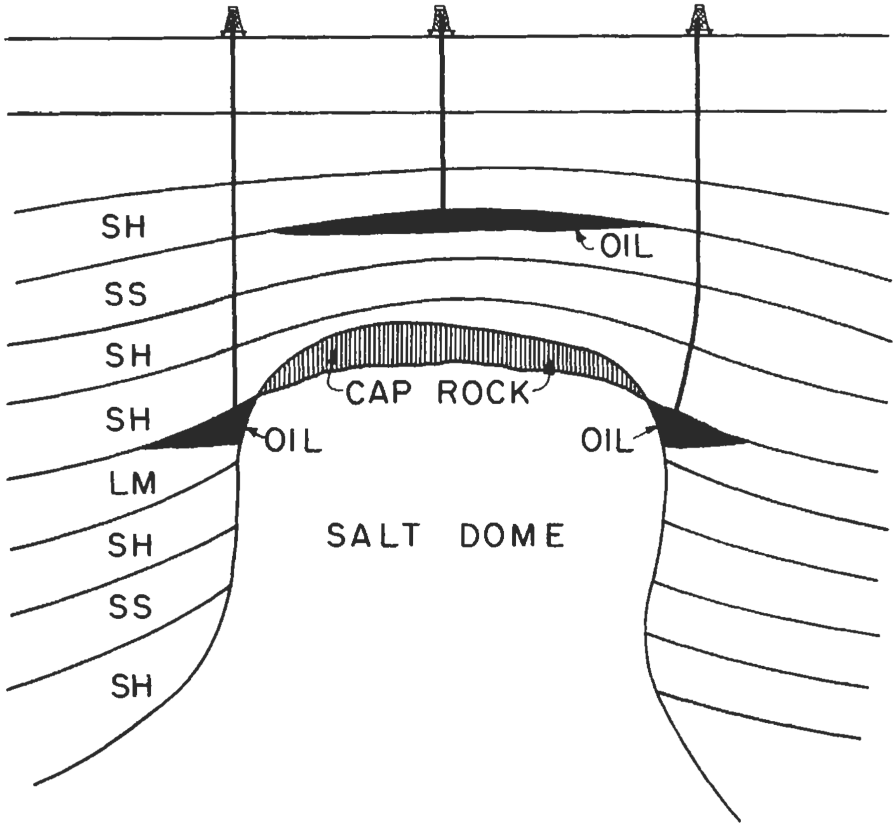


Figure 2-49. Salt dome structural traps.

and subsequently, overlaid by younger flat-lying beds. If flat-lying beds are impermeable, the permeable tilted beds will be a potential reservoir (see Figure 2-53).

- *Combination traps*—sedimentary trap features that result from both stratigraphic and structural mechanisms. There can be many combinations for stratigraphic and structural traps. An example of such a trap would be a reef feature overlaying a porous and permeable sandstone, but in which the sequence has been faulted (see Figure 2-54). Without the fault, which has provided an impregnable barrier, the hydrocarbons would have migrated further up dip within the sandstone.

Basic Engineering Rock Properties

The petroleum industry is concerned with basic properties of sedimentary rocks. Although the following discussions can be applied to other rocks, the emphasis will be on sedimentary rocks.

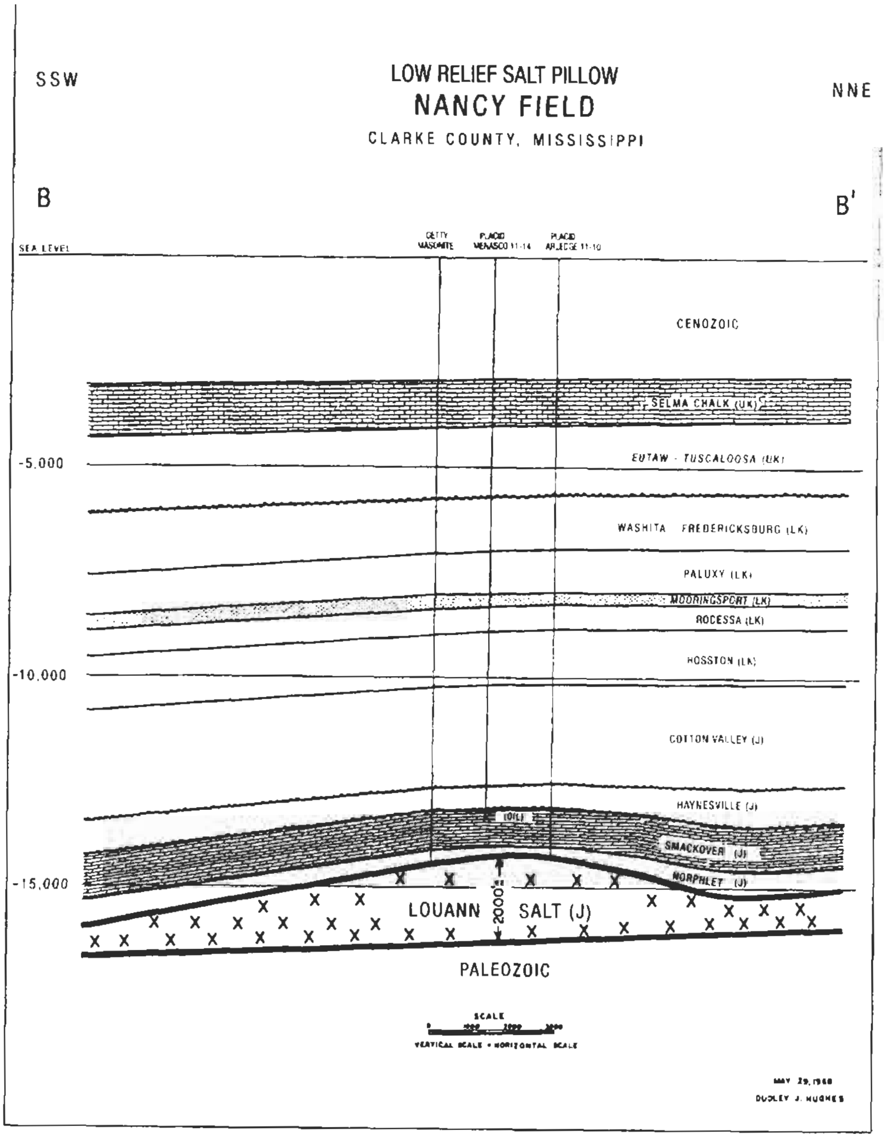


Figure 2-50. Low relief salt pillow trap (Nancy Field, Mississippi). (from Gulf Coast Association of Geological Societies).

Sedimentary deposits are usually carried to the region of deposition by water and are deposited in water. (In some cases deposits are carried by wind or ice.) It is within these water-deposited sediments that hydrocarbons are likely generated from the plant and animal life that exists in these environments. Two principal properties of the sedimentary rocks that form from such deposits are porosity and permeability.

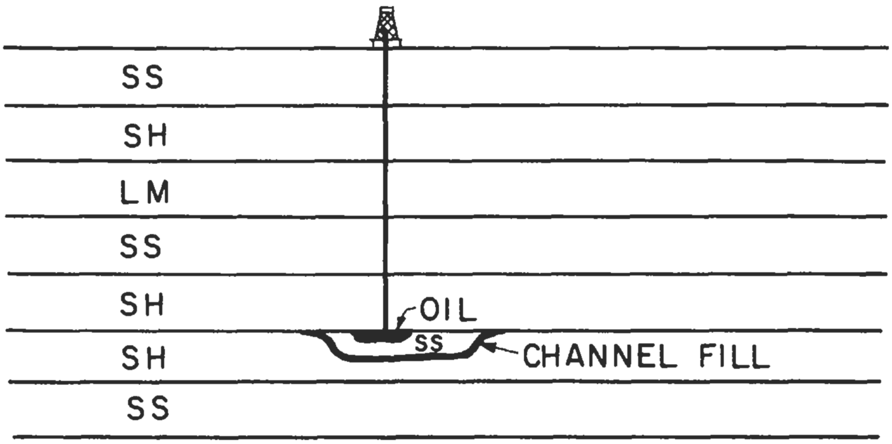


Figure 2-51. River channel sand trap.

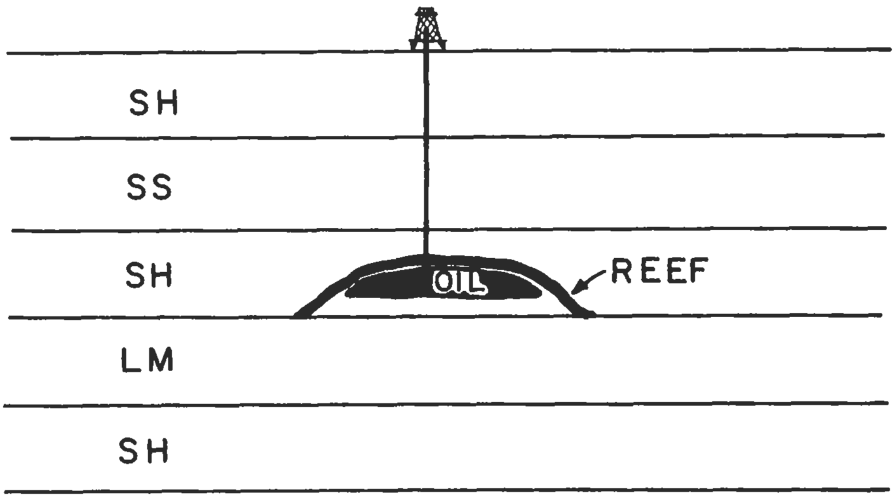


Figure 2-52. Reef trap.

Porosity

Porosity is a measure of the void space within a rock, which is expressed as a fraction (or percentage) of the bulk volume of that rock [31].

The general expression for porosity ϕ is

$$\phi = \frac{V_b - V_s}{V_b} = \frac{V_p}{V_b} \tag{2-155}$$

where V_b is the bulk volume of the rock, V_s is the volume occupied by solids (also called grain volume), and V_p is the pore volume.

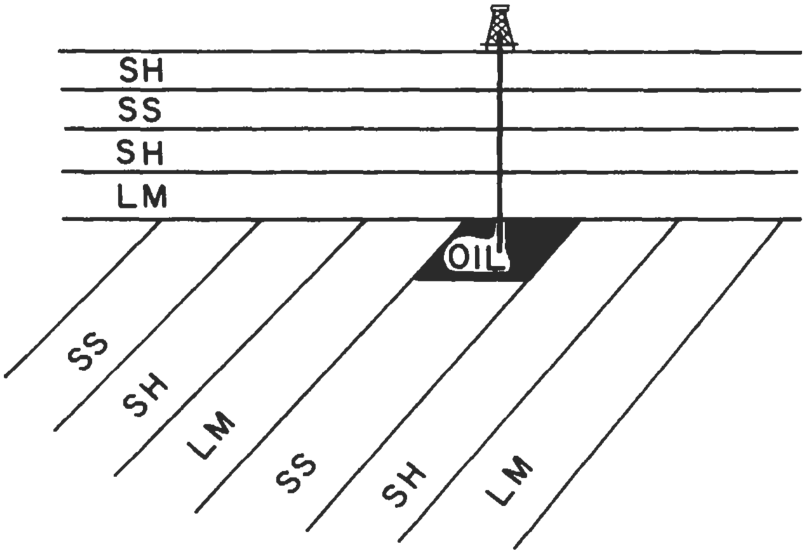


Figure 2-53. Unconformity trap.

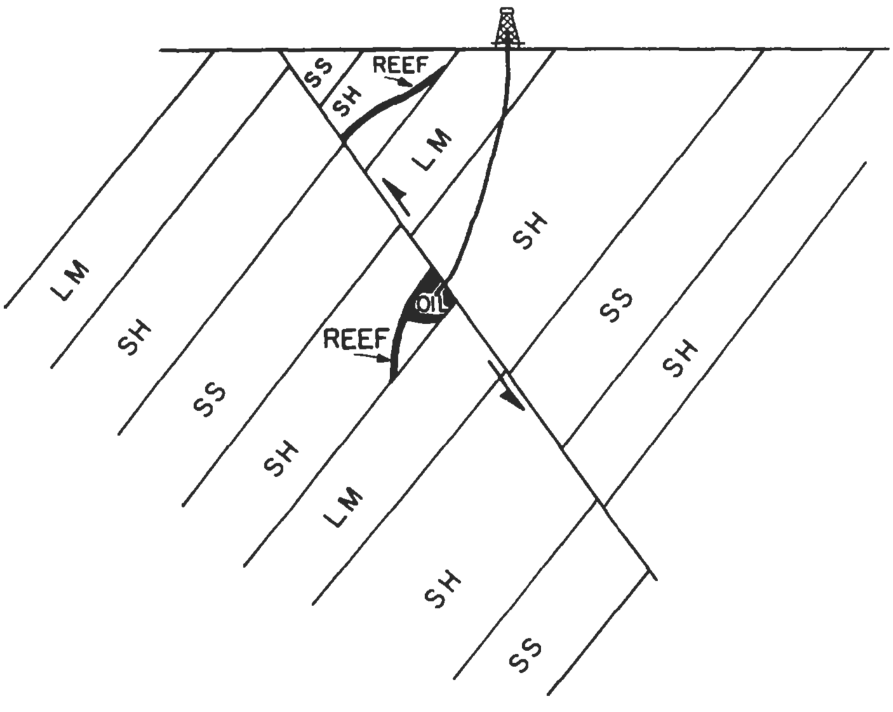


Figure 2-54. Combination trap.

From an engineering point of view, porosity is classified as:

- *Absolute porosity*—total porosity of a rock, regardless of whether or not the individual voids are connected.
- *Effective porosity*—only that porosity due to voids that are interconnected.

It is the effective porosity that is of interest. All further discussion of porosity will pertain to effective porosity.

From a geologic point of view, porosity is classified as:

1. *Primary porosity*—porosity formed at the time the sediment was deposited. Sedimentary rocks that typically exhibit primary porosity are the *clastic* (also called *fragmental* or *detrital*) rocks, which are composed of erosional fragments from older beds. These particles are classified by grain size.
2. *Secondary porosity*—voids formed after the sediment was deposited. The magnitude, shape, size, and interconnection of the voids bears little or no relation to the form of the original sedimentary particles. Secondary porosity is subdivided into three classes.
   - *Solution porosity* refers to voids formed by the solution of the more soluble portions of the rock in the presence of subsurface migrating (or surface percolating) waters containing carbonic and other organic acids. Solution porosity is also called *vugular porosity* where individual holes are called vugs.
   - Fractures, fissures, and joints are openings in sedimentary rocks formed by the structural (mechanical) failure of the rock under loads caused by earth crust tectonics. This form of porosity is extremely hard to evaluate quantitatively due to its irregularity.
   - Dolomitization is the process by which limestone ($CaCO_3$) is transformed into dolomite $CaMg(CO_3)_2$. During the transformation (which occurs under pressure), crystal reorientation occurs, which results in porosity in dolomite.

The typical value of porosity for a clean, consolidated, and reasonably uniform sand is 20%. The carbonate rocks (limestone and dolomite) normally exhibit lower values, e.g., 6–8%. These are approximate values and do not fit all situations. The principal factors that complicate intergranular porosity magnitudes are uniformity of grain size, degree of cementation, packing of the grains, and particle shape.

Permeability

Permeability is defined as a measure of a rock's ability to transmit fluids. In addition to a rock's being porous, sedimentary rock can also be permeable. Permeability refers to the property of a rock that allows fluids to flow through its pore network at practical rates under reasonable pressure differentials. The quantitative definition of permeability was first given in an empirical relationship developed by the French hydrologist Henry D'Arcy who studied the flow of water through unconsolidated sands [31].

This law in differential form is

$$v = -\frac{k}{\mu}\frac{dp}{d\ell} \tag{2-156}$$

where v is the apparent flow velocity (cm/sec), μ is the viscosity of the flowing fluid (centipoise), p is pressure (atmospheres), ℓ is the length (cm), k is permeability of the porous media (darcies).

Consider the linear flow system of Figure 2-55. The following assumptions are necessary to establish the basic flow equations:

- Steady-state flow conditions.
- Pore volume is 100% filled with flowing fluid; therefore, k is the absolute permeability.
- Viscosity of the flowing fluid is constant. In general, this is not true for most real fluids. However, the effect is negligible if μ at the average pressure is used.
- Isothermal conditions prevail.
- Flow is horizontal and linear.
- Flow is laminar.

Using the foregoing restrictions:

$$v = \frac{q}{A} \tag{2-157}$$

where q is the volumetric rate of fluid flow (cm^3/s); A is the total cross-sectional area perpendicular to flow direction (cm^2).

This further assumption concerning velocity and the volumetric rate of flow restricts flow to the pores and not the full area. Therefore, v is an apparent velocity. The actual velocity, assuming a uniform medium, is

$$V_{actual} = \frac{V_{apparent}}{\phi} \tag{2-158}$$

where ϕ is porosity defined in Equation 2-155.

Substituting Equation 2-157 into Equation 2-156 yields

$$\frac{q}{A} = -\frac{k}{\mu}\frac{dp}{d\ell} \tag{2-159}$$

Separation of variables and using limits from Figure 2-55 gives

$$\frac{q}{A}\int_0^{\ell} d\ell = -\frac{k}{\mu}\int_{p_1}^{p_2} dp \tag{2-160}$$

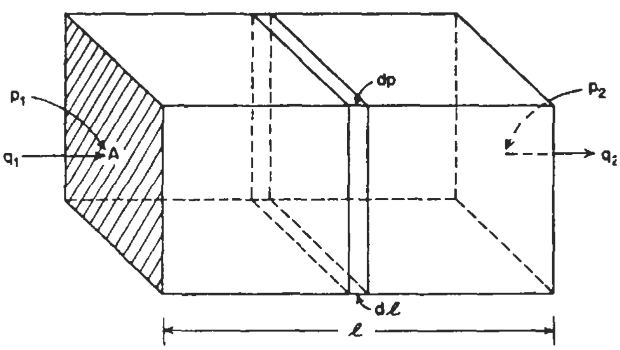


Figure 2-55. Linear flow system.

Integrating the preceding yields

$$q = \frac{kA(p_1 - p_2)}{\mu \ell} \tag{2-161}$$

or

$$k = \frac{q\mu\ell}{A\Delta p} \tag{2-162}$$

Equations 2-161 and 2-162 are the basic forms of the permeability relationship, and the following example serves to define the darcy unit:

If $q = 1\ cm^3/s$

$A = 1\ cm^2$

$\mu = 1\ cp$

$\Delta p/\ell = 1\ atm/cm$

then from Equation 2-162

$k = 1$ darcy

A permeability of 1 darcy is much higher than that commonly found in sedimentary rock, particularly reservoir rocks. Consequently, a more common unit is the millidarcy, where

1 darcy = 1,000 millidarcies

Typical values for sedimentary rock permeability for the flow of hydrocarbons and other fluids are 100 millidarcies (md) or greater. Rocks exhibiting permeabilities of 50 md or less are considered tight, relative to the flow of most fluids.

Subsurface Temperature

The earth is assumed to contain a molten core; therefore, it is logical to assume that the temperature should increase with depth below the surface. This temperature-depth relationship is commonly assumed to be a linear function.

$$t_d = t_s + \beta d \tag{2-163}$$

where t_d is the temperature of the rock at depth, d(°F), t_s is the average surface temperature (°F), β is the temperature gradient (°F/ft), and d is the depth (ft).

In general there is considerable variation in the geothermal gradient throughout the United States and the world. Also, in many regions of the world where there is evidence of rather thin crust, the relationship between temperature at depth and depth may not be approximated by the linear function given in Equation 2-163. The increase in temperature with depth has important consequences for drilling and production equipment that is used in the petroleum industry. The viscosity of drilling and production fluids will, in general decrease with high temperatures. The sitting time for well cement will generally decrease with increased temperature, as will the strength and fracture toughness of steels used in drilling, well completion, and production.

Temperature logs can be taken as wells are drilled and the temperature gradient determined for the particular region. These temperature logs taken at depth are used to determine the types of drilling fluids used as drilling progresses. Also, the temperatures at depth will determine the cements used in well completion operations.

The average geothermal gradient used in most areas of the United States for initial predictions of subsurface temperatures is a value of 0.016°F/ft [32].

Example 2-25

Determine the temperature at 20,000 ft using the approximate geothermal gradient $\beta \approx 0.016$ °F/ft.

Assume $t_s \approx 60$°F.

$$t_d = t_s + 0.016d$$

$$t_d = 60 + 0.016(20{,}000)$$

$$t_d = 380°F$$

Subsurface Fluid Pressure (Pore Pressure Gradient). The total overburden pressure is derived from the weight of the materials and fluids that lie above any particular depth level in the earth. Of interest to the petroleum industry are the sedimentary rocks derived from deposits in water, particularly, in seawater. Such sedimentary rocks contain rock particle grains and saline water within the pore spaces.

Total theoretical maximum overburden pressure, P (lb/ft^2), is

$$P_{ob} = \frac{W_r + W_{sw}}{A} \tag{2-164}$$

where W_r is weight of rock particle grains (lb), W_{sw} is weight of water (lb), and A is area (ft^2).

The term W_r can be approximated by

$$W = (1 - \phi)\, Ad\gamma_m \tag{2-165}$$

where ϕ is the fractional porosity, d is depth (ft), and γ_m is average mineral specific weight (lb/ft^3).

The term W can be approximated by

$$W_{sw} = \phi\, Ad\gamma_{sw} \tag{2-166}$$

where γ_{sw} is the average saltwater specific weight (lb/ft^3).

Substitution of Equations 2-165 and 2-166 into Equation 2-164 yields

$$P_{ob} = (1 - \phi)d\gamma_m + \phi d\gamma_{sw} \tag{2-167}$$

Equation 2-167 can be rewritten in terms of the specific gravities of average minerals S_m and salt water S_{sw}:

$$P = (1 - \phi)dS_m\gamma_w + \phi dS_{sw}\gamma_w \tag{2-168}$$

where γ_w is the specific weight of fresh water (i.e., 62.4 lb/ft^3).

The average specific gravity of minerals in the earth's crust is taken to be 2.7. The average specific gravity of saltwater is taken to be 1.07. If the average sedimentary rock porosity is assumed to be 10%, then the total theoretical maximum overburden pressure gradient $(lb/ft^2)/ft$ becomes

$$\frac{P_{ob}}{d} = (1 - 0.10)(2.7)(62.4) + 0.10(1.07)(61.4)$$

$$\frac{P_{ob}}{d} = 158.3 \tag{2-169}$$

Equation 2-169 can be expressed in normal gradient terms of psi/ft. Equation 2-169, which is the theoretical maximum overburden pressure gradient, becomes

$$\frac{p_{ob}}{d} = 1.10 \tag{2-170}$$

where p_{ob} is pressure in (psi).

The foregoing theoretical overburden pressure gradient assumes that the sedimentary deposits together with the saline water are a mixture of materials and fluid. Such a mixture could be considered as a fluid with a new specific weight of

$$\gamma_f = 158.3 \text{ lb/ft}$$

which in terms of drilling mud units would be

$$\bar{\gamma}_f = \frac{158.3}{7.48} = 21.1 \text{ ppg}$$

where 1 ft^3 = 7.48 gal.

After sedimentary deposits are laid down in saline water, cementing processes commence. These cementing processes take place through geologic time. In general the cementing process results in sedimentary rock that has structural competency. The cementing of the particles results in a structure that contains the saline in the rock pores. Therefore, if the sedimentary column of rock below the surface were a mature rock column in which cementing is complete throughout the entire column, the theoretical minimum pressure gradient basically would be the saltwater gradient $(lb/ft^2)/ft$.

$$\frac{P}{d} = 1.07(62.4)$$

or

$$\frac{P}{d} = 66.8 \tag{2-171}$$

Equation 2-171 can be expressed in normal gradient terms of psi/ft. Equation 2-171, which is the minimum pressure gradient, becomes

$$\frac{p}{d} = 0.464 \qquad (2\text{-}172)$$

The foregoing minimum pressure gradient assumes also that the sedimentary column pores are completely filled with saline water and that there is communication from pore to pore within the rock column from surface to depth.

Figure 2-56 shows a plot of the theoretical maximum overburden pressure and the theoretical minimum pressure as a function of depth. Also plotted are various bottomhole fluid pressures from actual wells drilled in the Gulf Coast region [33]. These experimentally obtained pressures are the measurements of the pressures in the fluids that result from a combination of rock overburden and the fluid hydraulic column to the surface. These data show the bottomhole fluid pressure extremes. The abnormally high pressures can be explained by the fact that the sedimentary basins in the Gulf Coast region are immature basins and are

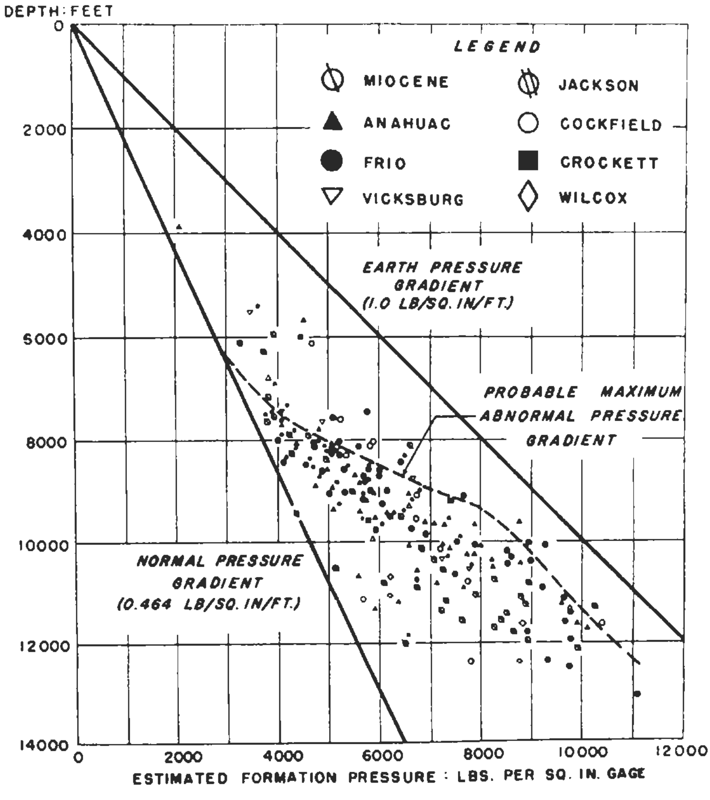


Figure 2-56. Magnitude of abnormal pressure encountered in Gulf Coast region (from Cannon and Sullins, "Problems Encountered in Drilling Abnormal Pressure Formations," in *API Drilling and Production Practices,* 1946).

therefore unconsolidated relative to older basins. In such basins the cementing process is by no means complete, which results in pressures at depths approaching the maximum theoretical overburden pressures.

The fluid pressure in the rock at the bottom of a well is commonly defined as pore pressure (also called formation pressure, or reservoir pressure). Depending on the maturity of the sedimentary basin, the pore pressure will reflect geologic column overburden that may include a portion of the rock particle weight (i.e., immature basins), or a simple hydrostatic column of fluid (i.e., mature basins). The pore pressure and therefore its gradient can be obtained from well log data as wells are drilled. These pore pressure data are fundamental for the solution of engineering problems in drilling, well completions, production, and reservoir engineering.

Because the geologic column of sedimentary rock is usually filled with saline water, the pore pressure and pore pressure gradient can be obtained for nearly the entire column. Figure 2-57 shows a typical pore pressure gradient versus depth plot for a Gulf Coast region well.

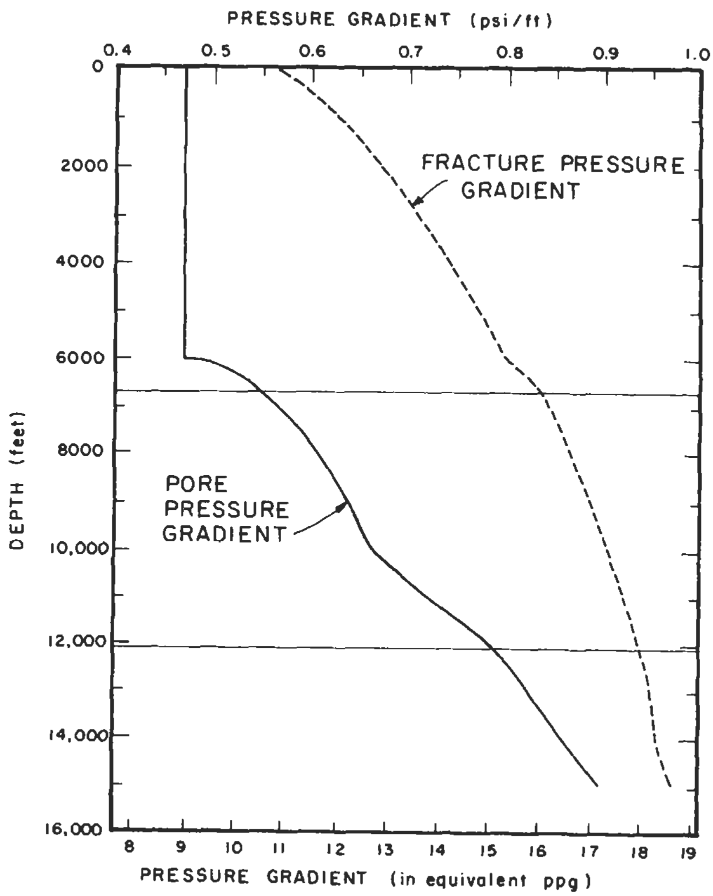


Figure 2-57. Pressure gradients vs. depth (Gulf Coast example).

Subsurface Rock Fracture Pressure (Fracture Pressure Gradient). The subsurface rock fracture pressure can be approximated by utilizing the known pore pressure at the same depth. The relationship between rock fracture pressure p_f (psi) and pore pressure p_p (psi) is [34]

$$p_f = (\sigma_{ob} - p_p)\left(\frac{v}{1-v}\right) + p_p \tag{2-173}$$

where σ_{ob} is overburden stress (psi) and v is Poisson's ratio.

The subsurface rock fracture pressure gradient is

$$\frac{p_f}{d} = \left(\frac{\sigma_{ob}}{d} - \frac{p_p}{d}\right)\left(\frac{v}{1-v}\right) + \frac{p_p}{d} \tag{2-174}$$

where d is the depth to the subsurface zone (ft).

Figure 2-58 shows the variation of Poisson's ratio versus depth for two general locations, the West Texas region and the Gulf Coast region.

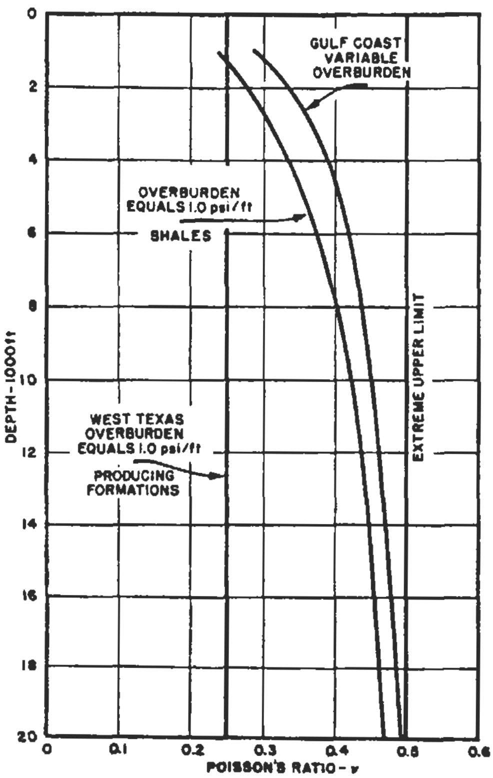


Figure 2-58. Poisson's ratio vs. depth (from Eaton, "Fracture Gradient Prediction and Its Application in Oilfield Operations," Journal of Petroleum Technology, October 1969).

The constant value of 0.25 for Poisson's ratio versus depth reflects the geology and the rock mechanics of the mature sedimentary basin in the West Texas region. Since mature basins are well cemented, the rock columns of West Texas will act as compressible, brittle, elastic materials.

The Cenozoic portions of the Gulf Coast sedimentary basins are immature; therefore, little cementing of the sediments has taken place. Poisson's ratio varies with depth for such sedimentary columns, reflecting the variation of properties through the column. At great depth (i.e., approaching 20,000 ft), Poisson's ratio approaches that of incompressible, plastic materials (i.e., 0.5) [35].

Figure 2-59 gives typical total overburden stress gradients versus depths for several regions in North America [36].

The rock fracture pressure gradient at depth can be approximated by using Equation 2-174 and the variable Poisson's ratios versus depth data (Figure 2-58) and the variable total overburden stress gradients versus depth data (Figure 2-59).

Example 2-26

In Figure 2-57 the pore pressure gradient has been given as a function of depth for a typical Gulf Coast well. Determine the approximate fracture pressure gradient for a depth of 10,000 ft. From Figure 2-57, the pore pressure gradient at 10,000 ft is

$$\frac{p_p}{d} = 0.66 \text{ psi/ft}$$

From Figure 2-58, Poisson's ratio at 10,000 ft is (i.e., Gulf Coast curve)

$$v = 0.45$$

From Figure 2-59, the total overburden stress gradient is (i.e., Gulf Coast curve)

$$\frac{\sigma_{ob}}{d} = 0.95 \text{ psi/ft}$$

Substituting the foregoing values into Equation 2-174 yields

$$\frac{p_f}{d} = (0.95 - 0.66)\left(\frac{0.45}{1 - 0.45}\right) + 0.66 = 0.90 \text{ psi/ft}$$

This value of 0.90 psi/ft falls on the dashed line of Figure 2-57. The entire dashed line (fracture pressure gradient) in Figure 2-57 has been determined by using Equation 2-174.

In general, Equation 2-174 can be used to approximate fracture pressure gradients. To obtain an adequate approximation for fracture pressure gradients, the pore pressure gradient must be determined from well log data. Also, the overburden stress gradient and Poisson's ratio versus depth must be known for the region.

There is a field operation method by which the fracture pressure gradient can be experimentally verified. Such tests are known as leak-off tests. The leak-off test will be discussed in Chapter 4.

Basic Engineering Soil Properties

The surface rock formations of the earth are continually exposed to the weathering process of the atmosphere that surrounds the earth. The weathering process through

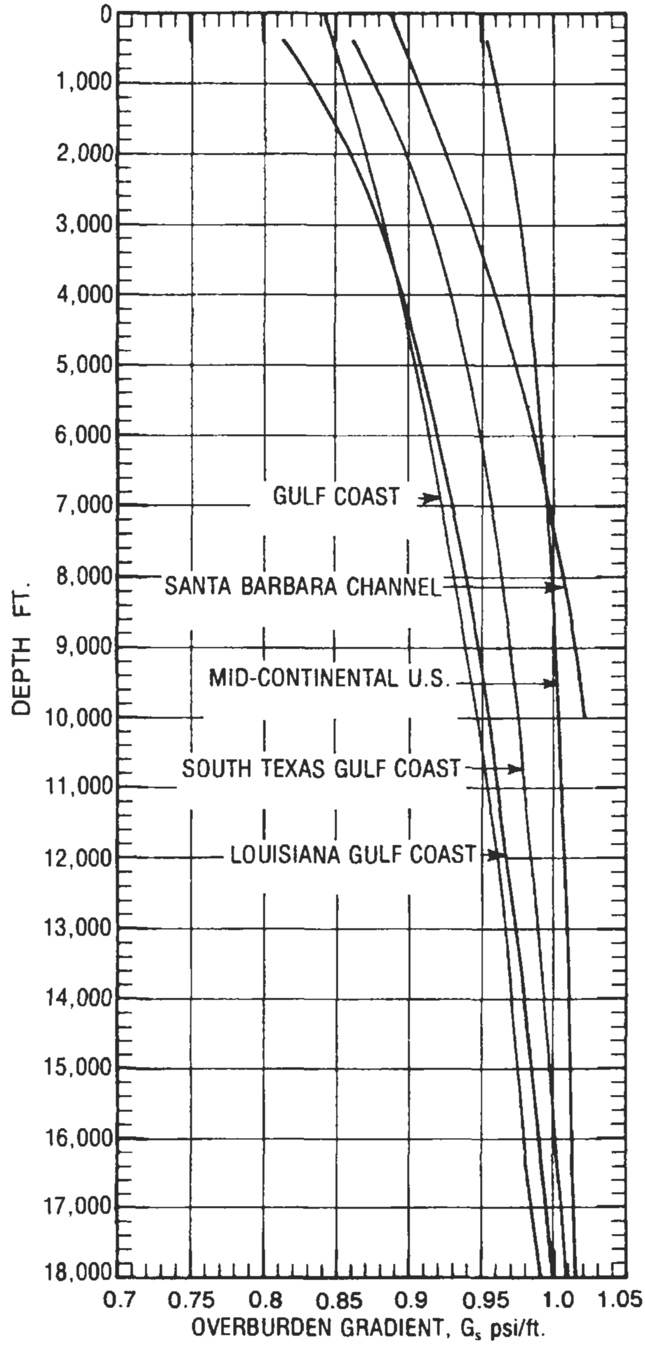


Figure 2-59. Total overburden stress gradient vs. depth (from *Engineering of Modern Drilling,* Energy Publication Division of Harcourt Brace Jovanovich, New York, 1982, p. 82).

time can change the rock exposed at the surface. Such changes are both mechanical and chemical. The altered surface rock is often used by lifeforms, particularly by plant life that directly draws nourishment from minerals at the surface. As the weathering process proceeds at the surface of the earth, rock at the surface disintegrates into small separate particles and is carried off and deposited at various locations around the original rock. Some particles are carried great distances to the sea to become source sediments for new rock formations. Some particles move only a few feet or a mile or two to be deposited with fragments from other nearby rock locations. These weathered particles, once deposited on the surface of the earth in land locations, are often referred to as soil [37].

To a farmer, soil is the substance that supports plant life. To a geologist soil is an ambiguous term that refers to the material that supports life plus the loose rock from which it was derived. To the engineer, soil has a broader meaning.

Soil, from the engineering point of view, is defined as any unconsolidated material composed of discrete solid particles that has either liquids or gases in the voids between the individual particles.

In general, soil overlays rock formations, and the soil is related to the rock since the rock was its source. Where the soil ends (in depth) and rock begins is not a well-defined interface. Basically, the depth to which soil is found is that depth where excavation by land methods can be employed. The area where the removal of material requires drilling, wedging, and blasting is believed to be the beginning of rock (in the engineering sense). The engineering properties of soil are of importance to petroleum engineering because it is soil that the drilling engineer first encounters as drilling is initiated. But, more important, it is soil that must support the loads of the drilling rig through an appropriately designed foundation. Further, the production engineer must support the well head surface equipment on soil through an appropriately designed foundation.

Soil Characteristics and Classification

The engineer visualizes a soil mass as an ideal, real, physical body incapable of resisting tensile stresses.

The ideal soil is defined as a loose, granular medium that is devoid of cohesion but possesses internal friction. In contrast, an ideal cohesive medium is one that is devoid of internal friction. Real soils generally fall between the foregoing two limiting definitions.

Soils can consist of rock, rock particles, mineral materials derived from rock formations, and/or organic matter.

- Bedrock is composed of competent, hard, rock formations that underlie soils. Bedrock is the foundation engineer's description of transition from soils to rock at depth. Such rock can be igneous, sedimentary, or metamorphic. Bedrock is very desirable for foundation placement.
- Weathered rock is bedrock that is deteriorating due to the weathering process. Usually this is confined to the upper layers of the bedrock.
- Boulders are rock fragments over 10 in. in diameter found in soils.
- Cobbles are rock fragments from 2–4 in. in diameter found in soils.
- Pebbles are rock fragments from about 4 mm to 2 in. in diameter found in soils.
- Gravel denotes unconsolidated rock fragments from about 2 mm to 6 in. in size.
- Sand consists of rock particles from 0.05–2 mm in size.
- Silt and clay are fine-grained soils in which individual particle size cannot be readily distinguished with the unaided eye. Some classification systems distinguish these particles by size, other systems use plasticity to classify these particles.

Plasticity is defined as the ability of such particle groups to deform rapidly without cracking or crumbling. It also refers to the ability of such groups to change volume with relatively small rebound when the deforming force is removed.

- Silt, in one particle classification system, consists of rock particles from 0.005 to 0.05 mm in size.
- Clay, in one particle classification system, consists of inorganic particles less than 0.005 mm in size. In another system, clay is a fine-grained inorganic soil that can be made plastic by adjusting the water content. When dried, clay exhibits considerable strength (i.e., clay loses its plasticity when dried and its strength when wetted). Also, it will shrink when dried and expand when moisture is added.

Figure 2-60 shows a classification system developed by the Lower Mississippi Valley Division, U.S. Corps of Engineers. Percentages are based on dry weight. A mixture with 50% or more clay is classified as clay; with 80% or more silt, as silt; and with 80% or more sand, as sand. A mixture with 40% clay and 40% sand is a sandy clay. A mixture with 25% clay and 65% silt is a clay-silt (see intersection of dashed lines in Figure 2-60).

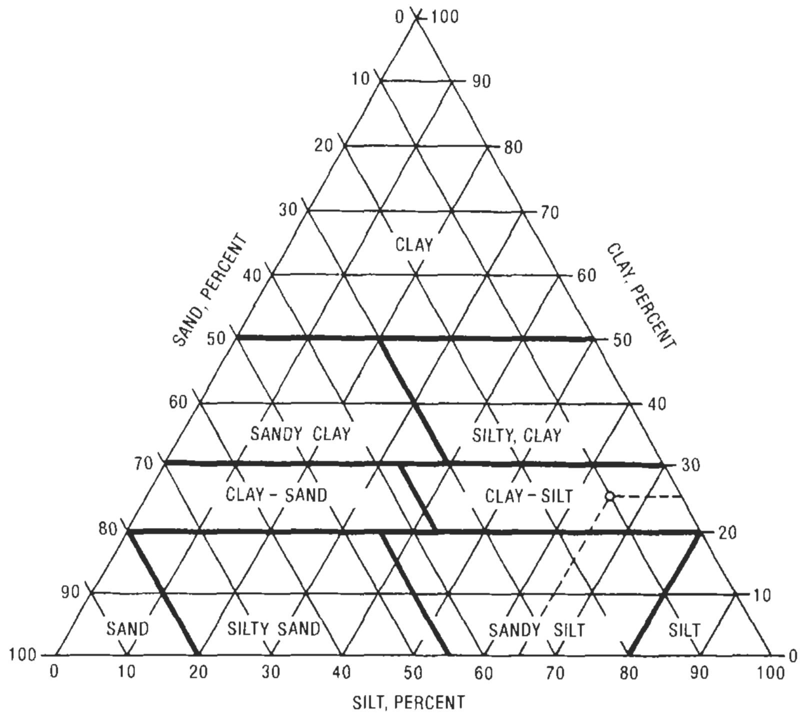


Figure 2-60. Classification chart for mixed soils (from Lower Mississippi Valley Division, U.S. Corps of Engineers).

Index Properties of Soils

Easily observed physical properties of soils often are useful indexes of behavior. These index properties include texture and appearance, specific weight, moisture content, consistency, permeability, compressibility, and shearing strength [37,38].

- Soil texture, or appearance, depends on particle size, shape, and gradation. Therefore, using the classification in Figure 2-60 the soil texture can be specified as sandy clay or clay-sand.
- Soil specific weight is the measure of the concentration of packing of particles in a soil mass. It is also an index of compressibility. Less dense, or loosely packed, soils are much more compressible under loads. Soil specific weight may be expressed numerically as soil ratio and porosity (porosity for soils being basically the same definition as that for rocks discussed earlier in this section). Soil porosity e is

$$e = \frac{V_b - V_s}{V_b} \tag{2-175}$$

where V_b is bulk volume of undisturbed soil (ft^3), and V_s is volume of solids in soil (ft).

The specific weight (unit weight) (lb/ft^3) of undisturbed soil is

$$\gamma = \frac{W_s}{W_b} \tag{2-176}$$

where W_s is the weight of the soil solids relative to the undistributed soil bulk volume (lb).

Relative density D_d (a percent) is a measure of the compactness of a soil with void ratio e when the maximum void ratio is e_{max} and the minimum e_{min}. Relative density is

$$D_d = \frac{e_{max} - e}{e_{max} - e_{min}}(100) \tag{2-177}$$

Percentage compaction usually is used to measure soil density in a fill situation. Generally, the maximum Proctor specific weight (dry, lb/ft^3), determined by a standard laboratory test, is set up as the standard for the soil. Normally, 90–100% compaction is specified.

- Moisture content, or water content, is an important influence on soil behavior. Water content w, dry-weight basis percent, is

$$W = \frac{W_w}{W_s} \tag{2-178}$$

where W_w is weight of water in soil (lb) and W_s is weight of solids in soil (lb).

Total net weight of soils, W (lb), is

$$W_e = W_w + W_s \tag{2-179}$$

Degree of saturation, S (percent) is

$$S = \frac{V_w}{V_b - V_s}(100) \tag{2-180}$$

where V is volume of water in soil (ft^3).

Saturation, porosity, and moisture content are related by

$$S_e = WG \tag{2-181}$$

where G is the specific gravity of solids in the soil mass.

- Consistency describes the condition of fine-grained soils: soft, firm, or hard. Shearing strength and bearing capacity vary significantly with consistency. In consistency there are four states: liquid, plastic, semisolid, and solid.
- Permeability is the ability of a soil to conduct or discharge water under a pressure, or hydraulic gradient (permeability for soils being basically the same definition as that for rocks discussed earlier in this section). For soils the definition of coefficient of permeability is slightly different than that discussed earlier for petroleum reservoir rocks. Since civil engineers and hydrologists are always dealing with water, the coefficient of permeability, or more precisely, the hydraulic conductivity, k′ (cm/s)* is

$$k' = \frac{\gamma_w}{\mu_w} k \tag{2-182}$$

where k is permeability as defined in Equation 2-162, γ_w is the specific weight of water, and μ_w is the viscosity of water.

Therefore, substituting Equation 2-162 into Equation 2-182 yields

$$k' = \frac{q}{iA} \tag{2-183}$$

where i (cm/cm) is the hydraulic gradient and is expressed by

$$i = \frac{\Delta p}{\ell \gamma_w} \tag{2-184}$$

and A (cm^2) is the total cross-sectional area of soil through which flow occurs.

Table 2-29 gives some typical values for hydraulic conductivity and drainage characteristics for various soil types.

- Soil compressibility is important for foundation engineering because it indicates settlement. Settlement or deformation of the soil under the foundation occurs because of change of position of particles in a soil mass.
- Shearing strength is the shear stress in a soil mass at failure of the soil mass (usually cracking), or when continuous displacement can occur at constant stress.

*Note: k′, hydraulic conductivity, is used for water flow in rocks as well as in soils.

Table 2-29
Hydraulic Conductivity and Drainage Characteristics of Soils [37]

Soil Type	Approximate Coefficient of Permeability *k*, cm per sec	Drainage Characteristic
Clean gravel	5–10	Good
Clean coarse sand	0.4–3	Good
Clean medium sand	0.05–0.15	Good
Clean fine sand	0.004–0.02	Good
Silty sand and gravel	10^{-5}–0.01	Poor to good
Silty sand	10^{-5}–10^{-4}	Poor
Sandy clay	10^{-6}–10^{-5}	Poor
Silty clay	10^{-6}	Poor
Clay	10^{-7}	Poor
Colloidal Clay	10^{-9}	Poor

Shearing strength of a soil is usually important in determining bearing capacity of the soil. Shearing stress, τ (lb/in.^{2}), of a soil is expressed as

$$\tau = c + \sigma_n \tan \theta \tag{2-185}$$

where C is cohesion of the soil (lb/in.^{2}), σ_n is the normal, effective stress perpendicular to shear surface (lb/in.^{2}), and θ is the angle of internal friction of soil.

The shear strength of coarse particle soils like gravel and sand depend on the interlocking of their particles and, thereby, on intergranular friction. The shear strength of pure clay soils depends basically on cohesion. Figure 2-61 gives a graphical representation of the two limits of soil types, i.e., coarse particle soils are denoted as θ soils, pure clay soils are denoted as c soils. Most real soils are a combination, i.e., (θ-c) soils. Some approximate friction angles for typical cohesionless soil types are as follows [37]:

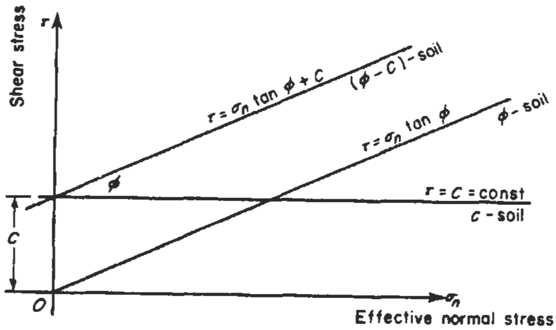


Figure 2-61. Graphic shear strength of soils.

Soil Type	ϕ, deg	tan ϕ
Silt, or uniform fine to medium sand	26–30	0.5–0.6
Well-graded sand	30–34	0.6–0.7
Sand and gravel	32–36	0.6–0.7

Site Investigations and Laboratory Tests

Prior to the construction of a foundation, field investigation should be carried out to determine surface and subsurface conditions at the site.

Site Investigations. Numerous techniques are used for site investigations. The techniques vary in cost from relatively low-cost visual investigations to costly subsurface explorations and laboratory tests [39,40].

- Visual inspection is an essential primary step. Such inspections should provide data on surface soils, surface waters, and slopes.
- Probing, driving a rod or pipe into the soil and measuring the penetration resistance, obtains initial subsurface information. This is a low-cost method, but in general it is likely to supply inadequate information about subsurface conditions, especially on the depth and nature of bedrock.
- Augers provide subsurface data by bringing up material for detailed examination. Augers disturb the soil; therefore, little or no information can be obtained on the character of the soil in its natural undisturbed state.
- Test pits permit visual examination of the soil in place. Such pits also allow manual sampling of "undisturbed" soil samples. These samples can be taken from the side walls of the pit.
- "Dry" spoon sampling is a technique that is often used in conjunction with auger drilling. At certain depths in the augered borehole a spoon is driven into the undisturbed bottom of the borehole. The spoon sampler is a specified size (usually a 2-in. OD). The number of blows per foot to the spoon samples frequently are recorded, indicating the resistance of the soil. The spoon sampler is driven into the bottom of the hole with a free-falling weight. A 140-lb weight falling 30 in. onto the 2 in. O.D. spoon sampler is the standard method of driving the sampler into the borehole bottom. Table 2-30 shows a system of correlation of this technique of sampling. Figure 2-62 shows a typical subsurface soils log that describes the soils encountered at depth and number of blows per 6 in. (instead of per foot) on the spoon sampler.

Laboratory Tests. In addition to the site investigations just described and the on-site tests which can be carried out through test pits and augering, often specific laboratory tests are required to identify soils and determine their properties. Such laboratory tests are conducted on the soil samples that are recovered from various subsurface depths. These laboratory tests are used when there are questions as to the structural supporting capabilities of the soils at a particular site. Numerous types of laboratory tests are available that can aid the engineer in designing adequate foundation support for heavy or dynamic loads. In general, however, few tests are necessary for most foundation designs [37–40].

- Mechanical analyses determine the particle-size distribution in a soil sample. The distribution of coarse particles is determined by sieving, and particles finer than a 200 or 270-mesh sieve and found by sedimentation.

Table 2-30
Correlation of Standard-boring-spoon Penetration with Soil Consistency and Strength (2 in OD Spoon, 140 lb)

Soil Consistency	Number of Blows per Ft on Spoon	Unconfined Compressive Strength, Tons/ft^2
Sand:		
Loose	15	
Medium compact	16–30	
Compact	30–50	
Very compact	Over 50	
Clay:		
Very soft	3 or less	0.3 or less
Soft	4–12	0.3–1.0
Stiff	12–35	1.0–4
Hard	Over 35	4 or more

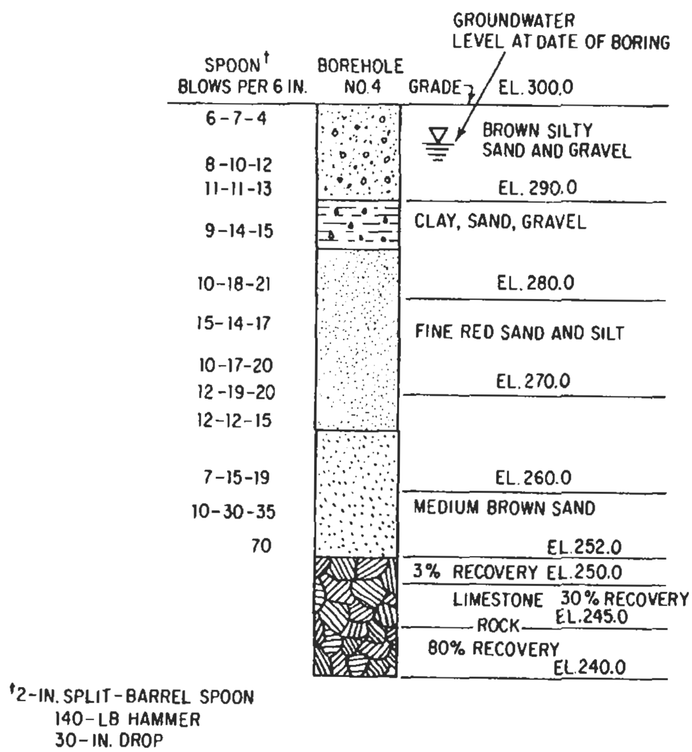


Figure 2-62. Typical soils boring log.

- Specific weight determinations measure the relative volumes of voids and solids in a soil.
- Compaction tests, such as the standard Proctor, determine the maximum specific weight or minimum void ratio that can be obtained for a soil, particularly a soil which is to be used for a fill. Specific weights of at least 95% of maximum are usually specified for compacted fills.
- In-place specific weight tests are used to correlate field compaction results with specified engineering requirements for specific weight.
- Moisture-content determinations provide data for estimating soil compaction and compressibility. If a soil is saturated, no volume change can occur without intake or discharge of water.
- Atterberg-limit tests determine the water content influence in defining liquid, plastic, semisolid and solid states of fine-grained soils. Permeability tests may be carried out in the laboratory or in the field. Such tests are used to determine the hydraulic conductivity coefficient k'.
- Confined compression tests are used to determine information pertaining to the behavior of foundations where large volume changes of soil can occur under compression but in the vertical dimension only.
- Unconfined compression tests are used to estimate the shearing strength of cohesive soils.
- Consolidation tests are made on saturated silts and clays to determine the rate of volume change under constant load.
- Direct shear tests are made in the laboratory to obtain data for determining the bearing capacity of soils and the stability of embankments.
- Triaxial compression tests are another means of determining shearing strength of a soil. A complex device is used to apply pressure along the sides of a cylindrical specimen and axially down the axis of the cylindrical specimen. In general, triaxial tests are superior to direct shear tests since there is better control over intake and discharge of water from the specimen.
- California bearing ratio tests are used to evaluate subgrades for pavements. These tests may be carried out in the field or in the laboratory. Such tests determine the resistance to penetration of a subgrade soil relative to that of a standard crushed-rock base.
- Plate bearing tests are field tests that are also used to evaluate subgrades for pavements.

Foundation Loads and Pressures. Foundations should be designed to support the weight of the structure, the live load, and the load effect on the structure and its foundation due to such other loads as wind. In general, for foundation designs, a safety factor of 3 is used for dead loads or live loads independently. A safety factor of 2 is used for combination loads including transient loads [38,40].

In general, a foundation is designed for settlement and for pressure distribution. In designing for settlement the usual practice is to ignore transient loads. To keep differential settlements small, foundations are designed to apportion the pressure (between the foundation and the soil) equally over the soil. The assumption is that equal intensities of pressure will produce equal settlement. The accuracy of this assumption will vary with the soil uniformity beneath the foundation, the shape of the foundation, and the distribution of the load on the foundation. Pressures used in calculating bearing capacity or settlement are those in excess of the pressure due to the weight of the soil above. Thus, one should consider pressures composed by the adjacent foundation on the region below the foundation under design. Usual practice in foundation design is to assume that bearing pressure at the bottom of a foundation or on a parallel plane below the foundation is constant for concentrically loaded foundations. This assumption may not be entirely accurate, but more accurate and more complicated theories usually are not justified because of the lack of future knowledge of loading conditions or by the present knowledge of soil conditions. The assumption of constant pressure distribution has been the basis of design of many foundations that have performed satisfactorily for decades. This is especially true for rigid foundations on soils with allowable bearing pressures of 6,000 lb/ft^2 or more. Another commonly used assumption in foundation design is that the pressure spreads out with depth from the bottom of the foundation at an angle of 30° from the vertical or at a slope of 1 to 2 (see Figure 2-63). Therefore, for a total load on a foundation of P, the pressure at the base of the footing would be assumed to be $p = P/A$, where A is the area of the foundation itself. As shown in Figure 2-63b, the pressure at a depth h would be taken to be P/A or, for a square foundation, as $P/(b + h)^2$. When a weak layer of soil underlies a stronger layer on which the foundation is founded, this method may be used to estimate the pressure on the lower layer. This method can be adopted to determine the total pressure resulting from overlapping pressure distribution from adjacent foundations.

Approximate allowable bearing pressures on sedimentary rock and soils may be taken from Table 2-31 [1 and 37]. Where questionable surface and subsurface soil conditions exist allowable bearing pressures can be determined with the aid of field sampling, field tests (both surface and subsurface through borings), and laboratory tests.

Spread Foundations. The purpose of the spread foundation is to distribute loads over a large enough area so that soil can support the loads safely and without excessive settlement. Such foundations are made of steel-reinforced concrete. When concrete is used for a foundation, it should be placed on undisturbed soil. All vegetation should be removed from the surface; therefore, the upper few inches of soil should be removed before concrete is laid down. The area of the foundation must be large enough to ensure that the bearing capacity of the soil is not exceeded or that maximum settlement is within acceptable limits. If details are known of the subsurface soil conditions, the foundation must be sized so that differential settlement will not be excessive. For

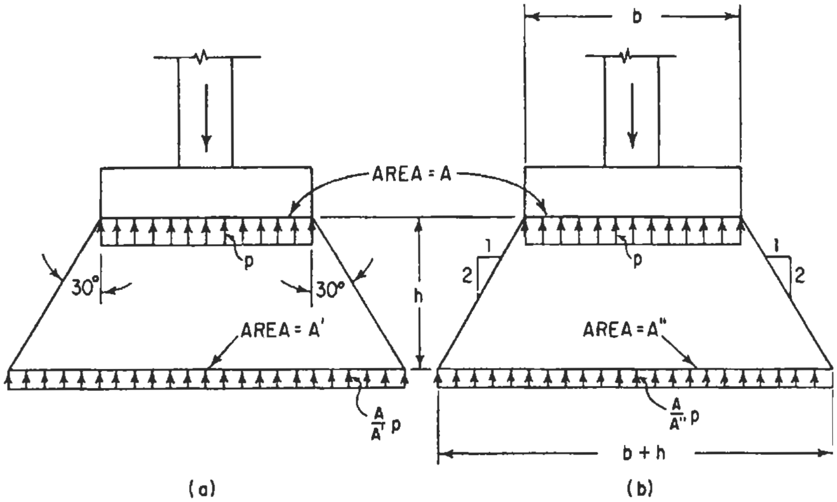


Figure 2-63. Assumed pressure distribution under a foundation: (a) 30° spread; (b) 1 × 2 spread [37].

Table 2-31
Allowable Bearing Capacities of Sedimentary Rock and Soils [37]

Rock or Soil	Allowable Bearing Capacity lbs/ft²
Sedimentary rock: hard shales, siltstones, sandstones, requiring blasting to remove	20,000 to 30,000
Hardpan, cemented sand and gravel, difficult to remove by picking	16,000 to 20,000
Soft rock, disintegrated ledge; in natural ledge, difficult to remove by picking	10,000 to 20,000
Compact sand and gravel, requiring picking to remove	8,000 to 12,000
Hard clay, requiring picking for removal	8,000 to 10,000
Gravel, coarse sand, in natural thick beds	8,000 to 10,000
Loose, medium and coarse sand; fine compact sand	3,000 to 8,000
Medium clay, stiff but capable of being spaded	4,000 to 8,000
Fine loose sand	2,000 to 4,000
Soft clay	2,000

uniform soil conditions (both horizontal and vertical), this is accomplished by designing the foundation such that the unit pressure under the foundation is uniform for the working loads (usually dead load plus normal live loads) [38,40].

Example 2-27

It is intended that a spread foundation be designed for a concentric load of 300,000 lb (dead load plus live load). This foundation is to be placed on the surface (brown silty sand and gravel) of the soil and bedrock column shown in Figure 2-61. If a square foundation can be made to support the 300,000-lb load, what should be the dimensions of this foundation?

- The surface layer of soil (i.e., brown silty sand and gravel) has a minimum number of blows per foot on the 2-in. split-barrel spoon of 8 (see Figure 2-62). According to Table 2-30, this soil would have the classification of a loose silty sand and gravel layer. Table 2-31, indicates the allowable bearing capacity could be as low as 3,000 lb/ft². To improve the soil conditions where the foundation is to be laid, the initial few feet of soil are to be removed to expose the subsurface layer which has a minimum number of blows per foot on the 2-in. split-barrel spoon of 16. This would require a removal of approximately 4 to 5 ft of soil. The brown silty sand and gravel in the layer at about 5 ft of depth can be classified as medium compact. Again referring to Table 2-31 this layer of soil should have an allowable bearing capacity above 3,000 lb/ft². The average between 3,000 and 8,000 lb/ft² is assumed, i.e., 5,500 lb/ft².
- The initial square foundation dimension that would be needed to meet the allowable bearing capacity of 5,500 lb/ft² is

$$A = \frac{300{,}000}{5{,}500} = 54.6 \text{ ft}^2$$

where A is the surface area of the foundation at the point where it contacts the soil. The dimensions of the square foundation are

$b = (54.6)^{1/2} = 7.4$ ft

- There are two potentially weak subsurface soil layers in the boring log in Figure 2-62 that should be checked before the initial foundation design above is accepted.

1. At the depth of approximately 10 ft below the surface, or 5 ft below the intended foundation–soil interface, there is a layer of clay, sand, and gravel that has a minimum number of blows per foot on the 2-in. split-barrel spoon of 18. Referring to Table 2-30, the compressive strength of this layer could be as low as 2,000 lb/ft^2. Assuming a one-by-two spread of the subsurface pressure under the foundation, the pressure at that layer would be

$$P_1 = \frac{300{,}000}{(7.4 + 5)^2} = 1{,}948 \text{ lb/ft}^2$$

For further information on this subject, refer to References 38 through 40.

The subsurface pressure calculated for this layer is below the assumed compressive strength of 2,000 lb/ft^2; therefore, the initial foundation divisions are acceptable relative to the strength of this subsurface layer.

2. At the depth of approximately 40 ft below the surface or 35 ft below the intended foundation-soil interface, there is a layer of medium brown sand that has a minimum number of blows per foot in the 2-in. split-barrel spoon of 14. Referring to Tables 2-30 and 2-31, this soil could be classified as a loose sand having a compressive strength as low as 3,000 lb/ft^2. Assuming a one-by-two spread of the subsurface pressure under the foundation, the pressure at that layer would be

$$P_2 = \frac{300{,}000}{(7.4 + 35)^2} = 167 \text{ lb/ft}^2$$

The subsurface pressure calculated for this layer is well below the assumed compressive strength of 3,000 lb/ft^2; therefore, the initial foundation dimensions are acceptable relative to this subsurface layer also.

- Because the initial foundation dimensions result in acceptable foundation–soil interface bearing pressures and acceptable subsurface pressures on weaker underlying layers of soil, the square foundation of 7.4 ft is the final foundation design.

ELECTRICITY

Electrical Units

There are two principle unit systems used in electrical calculations, the centimeter-gram-second (cgs) and the meter-kilogram-second (mks) or International System (SI). Table 2-32 is a summary of common electrical quantities and their units. The magnitude of the units in Table 2-32 is often not conveniently sized for taking measurements or for expressing values, and Table 2-33 presents the prefixes that modify SI units to make them more convenient.

Energy or *work* W is defined by

$$W = \int_{t_1}^{t_2} p\, dt \qquad (2\text{-}186)$$

Table 2-32
Electrical Units

Quantity	Symbol	MKS and SI Units
Current	I,i	Ampere
Charge (Quantity)	Q,q	Coulomb
Potential	V,v	Volt (V)
Electromotive Force	E,e	Volt (V)
Resistance	R,r	Ohm (Ω)
Resistivity	ℓ	Ohm-cm
Conductance	G,g	Mho, Siemens (Ω^{-1})
Conductivity	γ	Mho/cm
Capacitance	C	Farad (f)
Inductance	L	Henry (h)
Energy (Work)	W	Joule (J), Watthour (Wh) Kilowatthour (KWh)
Power	P,p	Watt
Reactance, Inductive	X_L	Ohm
Reactance, Capacitive	X_C	Ohm
Impedance	Z	Ohm

Table 2-33
SI Unit Dimensional Prefixes

Symbol	Prefix	Multiple
T	tera	10^{12}
G	giga	10^{9}
M	mega	10^{6}
k*	kilo	10^{3}
h*	hecto	10^{2}
da*	deca	10^{1}
d	deci	10^{-1}
c	centi	10^{-2}
m	milli	10^{-3}
	micro	10^{-6}
n	nano	10^{-9}
p	pico	10^{-12}

*May also use capital letter for the symbol

where W is work in joules, p is power in watts, and t is time in seconds.
Relations between common units of energy are as follows:

watt-second = 1 joule
watt-second = 0.239 calorie
watt-second = 0.738 foot-pound

kilowatt-hour = 3413 Btu
kilowatt-hour = 1.34 horsepower-hours
kilowatt-hour = 3.6×10^6 joules
electron-volt = 1.6×10^{-9} joule
erg = 10^{-7} joule

Power (P,p) is the time rate of doing work. For constant current I through an electrical load having a potential drop V, the power is given by

$$P = IV \tag{2-187}$$

where P is power in watts, V is potential drop in volts, and I is current in amperes.

For a time-varying current i and potential drop v, the *average power* P_{av} is given by

$$P_{av} = \frac{1}{t}\int_0^t iv\,dt \tag{2-188}$$

where P_{av} is average power in watts; i is current in amperes, v is potential drop in volts, and t is time in seconds.

Relations between common units of power are as follows:

1 w = 1 j/s
0.239 c/sec
9.48×10^{-4} Btu/s
1/745 hp
0.7375 ft-lb/s

Electric *charge* or *quantity* Q, expressed in units of *coulombs*, is the amount of electricity that passes any section of an electric circuit in one s by a current of one *ampere*. A coulomb is the charge of 6.24×10^{18} electrons.

Current (I,i) is the flow of electrons through a conductor. Two principal classes of current are:

- *Direct* (dc)—the current always flows in the same direction.
- *Alternating* (ac)—the current changes direction periodically.

The unit of current is the *ampere* which is defined as one coulomb per second.

Electric *potential* (V,v), *potential difference*, or *electromotive force* (emf, E, e) have units of *volts* and refer to the energy change when a charge is moved from one point to another in an electric field.

Resistance (R,r) is an element of an electric circuit that reacts to impede the flow of current. The basic unit of resistance is the *ohm* (Ω), which is defined in terms of *Ohm's law* as the ratio of potential difference to current, i.e.,

$$R = \frac{V}{I} \tag{2-189}$$

The resistance of a length of conductor of uniform cross-section is given by

$$R = \frac{\rho \ell}{A} \tag{2-190}$$

where A is cross-section area of the conductor in square meters, ℓ is length of the conductor in meters, and ρ is resistivity of the material in ohm-meters; R is resistance in ohms.

Conductance (G,g) is the reciprocal of resistance and has units of reciprocal ohms or *mhos* (Ω^{-1}) or more properly in SI units, *seimens.*

Conductivity (γ) is the reciprocal of resistivity.

Capacitance (C) is the property that describes the quantity of electricity that can be stored when two conductors are separated by a dielectric material. The unit of capacitance is the *farad.* The capacitance of two equal-area, conducting parallel plates (see Figure 2-64) separated by a dielectric is given by

$$C = \frac{\varepsilon_r \varepsilon_v A}{d} \tag{2-191}$$

where ε_r is dielectric constant of the material between the plates, ε_v is dielectric constant of free space or of a vacuum, A is the area of a plate in square meters, d is distance between the plates in meters, and C is capacitance in farads.

Inductance (L) is the property of an electric circuit that produces an emf in the circuit in response to a change in the rate of current, i.e.,

$$e = -L\frac{di}{dt} \tag{2-192a}$$

where e is emf induced in the circuit in volts, i is current in amperes, t is time in seconds, and L is coefficient of (self) inductance in henries.

For a coil (see Figure 2-65), the inductance is given by

$$L = KN^2 \tag{2-192b}$$

where N is number of turns, K is a constant that depends on the geometry and the materials of construction, and L is coefficient of inductance in henries.

Coils are described later in the section titled "Magnetic Circuits."

Electrical Circuit Elements

Electrical phenomena may generally be classified as *static* or *dynamic.* In *static* phenomena, current or charges do not flow or the flow is only momentary. Most

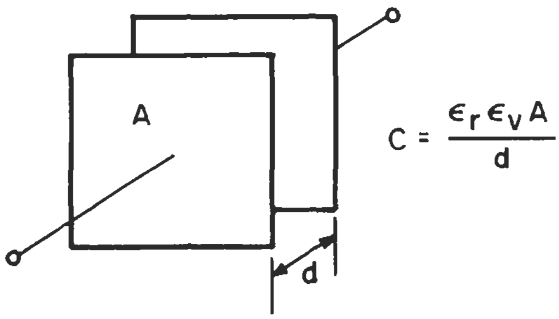


Figure 2-64. Schematic of a parallel plate capacitor.

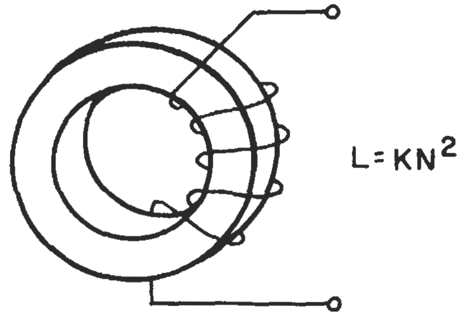


Figure 2-65. Schematic of an induction coil.

practical uses of electricity (at least in the conventional engineering sense) involve *dynamic* systems where current flows for useful periods of time. Systems that allow current to flow for extended periods of time are termed *circuits.* As the name implies, such systems are connected in a loop so that any beginning point in the circuit has electrical continuity with the ending point through *circuit elements.*

The simplest circuit element is the *short circuit.* Figure 2-66a illustrates the concept of a short circuit. A source of emf (labeled v_s) produces a current that flows relatively unimpeded through the conductor resulting in a nearly zero potential drop and an infinite current.

Figure 2-66b illustrates an *open circuit.* The conductor of the short circuit is interrupted and current cannot flow even though the emf produces a finite potential.

Passive Circuit Elements

Circuit elements may be classified as *passive* or *active. Passive elements* may store or transform electrical energy. *Active elements* may transform other forms of energy (including electrical energy from another system) into an increase in the electrical energy of the circuit or they may serve to dissipate circuit energy. In the latter case, the rate of increase or dissipation is controlled by conditions outside of the circuit. A simple example of an active element might be a generator that supplies power (emf) to the circuit. The rate of power generation is dependent upon the speed at which the generator turns, not on conditions in the circuit.

There are three, linear, *passive circuit elements:* resistors, capacitors, and inductors. *Resistors* dissipate energy in the circuit, i.e., electrical energy is transformed to heat energy and is lost from the circuit. *Capacitors* store electrical energy as charges on conductors separated by a dielectric material. *Inductors* store the electrical energy of current as magnetic potential in a manner analogous to the kinetic energy stored in a mass in motion. Table 2-34 summarizes the circuit element characteristics.

Series and Parallel Connection of Circuit Elements

Circuit elements may be connected in either a *series* or *parallel* configuration. In the *series* configuration, the same current flows through each and every element, and the circuit potential drop (or emf that is developed by the voltage source) is the algebraic sum of the potential drops of each individual element. For sources in series, the total emf developed is the algebraic sum of the emfs developed by each individual source.

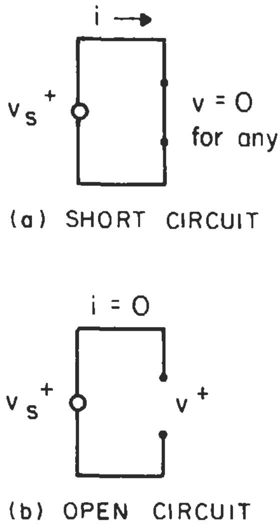


Figure 2-66. Schematic of the simplest circuit elements.

Table 2-34
Circuit Element Characteristics

Element	Unit	Symbol	Characteristic
Resistance (Conductance)	ohm (mho)	i, R, $+v$, (G)	$v = Ri$ $(i = Gv)$
Inductance	henry	i, L, $+v$	$v = L\frac{di}{dt}$ $i = \frac{1}{L}\int_0^t v\,dt + I_0$
Capacitance	farad	i, C, $+v$	$i = C\frac{dv}{dt}$ $v = \frac{1}{C}\int_0^t i\,dt + V_0$
Short circuit		i	$v = 0$ for any i
Open circuit		$+v$	$i = 0$ for any v
Voltage source	volt	$+v_s$	$v = v_s$ for any i
Current source	ampere	i_s	$i = i_s$ for any v

In the *parallel* configuration, the same potential difference occurs across each and every element with the total current being the algebraic sum of the current flowing through each individual circuit element. Table 2-35 summarizes the equivalent resistance, conductance, capacitance, and inductance of series-parallel configurations of resistors, capacitors, and inductors.

Circuit Analysis

There are two fundamental laws used in circuit analysis, called *Kirchoff's laws:*

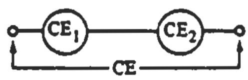
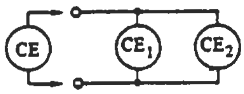
1. At a branch point in an electric circuit, the sum of the currents flowing to the point equals the sum of the currents flowing from the point.
2. The electric potential measured between two points in an electric circuit is the same regardless of the path along which it is measured.

These laws apply to both DC and AC currents.

Transient and AC Circuits

In a transient or an AC circuit we term the sum of resistance, inductance, and capacitance as *impedance.* Using complex notation, the energy storage properties of inductance and capacitance are represented as purely imaginary quantities, while the resistance is represented as a (+) real quantity. Capacitance is represented as the negative imaginary axis, and current through a pure capacitance is said to *lead*

Table 2-35
Series-Parallel Combinations

Circuit Element	Series	Parallel
		
Resistor R_1, R_2	$R = R_1 + R_2$	$R = \dfrac{R_1 R_2}{R_1 + R_2}$ or $R = \left(\dfrac{1}{R_1} + \dfrac{1}{R_2}\right)^{-1}$
Inductor L_1, L_2	$L = L_1 + L_2$	$L = \dfrac{L_1 L_2}{L_1 + L_2}$ or $L = \left(\dfrac{1}{L_1} + \dfrac{1}{L_2}\right)^{-1}$
Capacitor C_1, C_2	$C = \dfrac{C_1 C_2}{C_1 + C_2}$ or $C = \left(\dfrac{1}{C_1} + \dfrac{1}{C_2}\right)^{-1}$	$C = C_1 + C_2$

the potential by 90°. Inductance is represented as the positive imaginary axis, and the current through a pure inductance is said to *lag* the potential by 90°. These relationships may be expressed mathematically for an inductance and resistance in series as

$$Z = R + jX_L \tag{2-193}$$

and for a capacitance and resistance in series as

$$Z = R - jX_C \tag{2-194}$$

where Z is imedance in ohms, j is imaginary unit, X_L is impedance due to inductance or reluctance in *ohms* = $2\pi fL = \omega L$, X_c is impedance due to capacitance or *reactance* in ohms = $1\backslash 2\pi fC = 1/\omega C$, R is resistance in *ohms,* L is coefficient of inductance in henries, C is capacitance in farads, f is frequency of the current (or voltage) in hertz (cycles per second), and ω is angular velocity in radians per second.

For the impedance of the resistance and inductance in series, the current will lag the potential phase angle by

$$\theta = \arctan\left(\frac{X_L}{R}\right) \tag{2-195a}$$

For the impedance of the resistance and capacitance in series, the current will lead the potential phase angle by

$$\theta = \arctan\left(\frac{-X_C}{R}\right) \tag{2-195b}$$

When multiple circuit elements are involved, the resultant phase angle difference between the current and the potential will result from the contribution of each element.

AC Power

The power dissipated in an AC circuit with current of maximum amplitude I_m flowing through a resistance is less than the power produced by a constant DC current of magnitude I_m flowing through the same resistance. For a sinusoidal AC current, the *root mean square* (rms) value of current I is the magnitude of the DC current producing the same power as the AC current with maximum amplitude I_m. The rms value I is given by

$$I = \frac{1}{\sqrt{2}} I_m = 0.707 I_m \tag{2-196}$$

The power dissipated in an AC circuit with only resistive elements is

$$P = I^2R = \frac{V^2}{R} \tag{2-197}$$

where V is the rms value of the potential drop. This resistive power is termed *active power* with units of *watts.*

Reactive circuit elements (e.g., capacitors and inductors) store, not dissipate, energy. While the energy stored is periodically returned to the rest of the circuit, reactive elements do require increased potential or current to flow in the circuit. The power that must be supplied for the reactive elements is termed *reactive* power, and it is calculated as

$$P_x = I^2X = \frac{V^2}{X} \tag{2-198}$$

where P_x is reactive power in volt-amperes reactive or *VAR*, X is reactance in ohms, V is rms potential in volts, and I is current in amperes.

The reactive component of impedance is expressed as

$$X = Z \sin \theta \tag{2-199}$$

where Z is impedance in *ohms*, and θ is leading or lagging phase difference between current and potential.

Note that sin θ may be either positive or negative and lies between 0 and 1 for $|\theta| \leq 90°$.

The *apparent power* is the complex sum of the *active power* and the *reactive power*. By noting that

$$R = Z \cos \theta \tag{2-200}$$

we may calculate

$$P_A = VI \cos \theta + jVI \sin \theta \tag{2-201}$$

where P_A is apparent power in volt-amperes (or *VA*), V is rms potential, I is rms current, and θ is leading or lagging difference in phase angle between current and potential.

The *power factor* cos θ is always a positive fraction between 0 and 1 (as long as $|\theta| \leq 90°$). The smaller the power factor, the greater the current that must be supplied to the circuit for a given active (useful) power output requirement. The increase in current associated with low power factors causes greater line losses or requires an increase in the capacity of the transmission equipment (wire size, transformers, etc.). As a result, for industrial applications there is often a power factor charge in the rate structure for supplying electricity. The usual situation is for loads to be inductive, and the industrial consumer may add capacitance to their circuits to correct the lagging power factor.

Magnetism

Magnetic fields are created by the motion of electric charges. The charge motion may be a current in a conductor or, at the atomic level, the movements of orbital electrons. For certain materials, called *ferromagnetic* materials, the neighboring atoms align themselves so that the magnetic effects of their orbital electrons are additive. When the atoms of a piece of such a ferromagnetic material are aligned, the piece is called a *magnet*. Magnetic fields have north (N) and south (S) poles. When two magnets are brought together, like poles repel and unlike poles attract each other. In other (nonmagnetic) materials, the atoms are aligned randomly and the magnetic effects cancel.

Analogies exist between electric and magnetic fields. The *magnetic flux* (ϕ) is analogous to electric current and has SI units of webers (see Table 2-36). The *magnetic*

Table 2-36
Magnetic Units

Quantity	Symbol	Cgs Units	SI Units
Magnetomotive Force	$\mathfrak{F}$	Gilberts	Amp-Turns (NI)
Magnetic Field Intensity	H	Oersted (Oe)	Amp-Turns/ Meter
Magnetic Flux	ϕ	Maxwell or line	Webers (Wb)
Magnetic Flux Density	B	Gauss	Teslas (Wb/m^2)

flux density (B) is analogous to current density and has units of teslas. One tesla exists when the charge of one coulomb moving normal to the magnetic field with a velocity of one meter per second experiences a force of one newton. In vector notation this is expressed as

$$\bar{f} = q(\bar{u} \times \bar{B}) \qquad (2\text{-}202)$$

where $\bar{f}$ is the force vector in newtons, q is the charge in coulombs, $\bar{u}$ is velocity vector in meters per second, and $\bar{B}$ is magnetic flux density vector in teslas.

If a current flows through a conductor, the magnetic flux is oriented in a direction tangent to a circle whose plane is perpendicular to the conductor. For current flowing in an infinitely long, straight conductor, the magnetic flux density at a point in space outside the conductor is given as

$$B = \frac{\mu i}{2\pi D} \qquad (2\text{-}203)$$

where B is magnetic flux density in teslas, i is current in amperes, D is distance from conductor to the point in space in meters, and μ is permeability in webers per amp-meter or henries per meter.

The *permeability* (μ) is a property of the material surrounding the conductor. The *permeability of free space* (μ_0) is

$$\mu_0 = 4 \times 10^{-7} \text{ henries per meter} \qquad (2\text{-}204)$$

The ratio of the permeability of any material to the permeability of free space is termed the *relative permeability* (μ_r).

The magnetic field intensity (H) is given as

$$H = \frac{B}{\mu} \qquad (2\text{-}205)$$

or for the magnetic field induced by current in an infinite-length straight conductor,

$$H = \frac{i}{2\pi D} \qquad (2\text{-}206)$$

Magnetic Circuits

Previously the analogy between electric fields and magnetic fields was introduced. Likewise, there are analogies between magnetic circuits and electric circuits. Figure 2-67 illustrates these analogies and allows us to define additional terms. In the electric circuit of Figure 2-67a Ohm's law applies, i.e.,

$$\frac{V}{I} = R = \frac{p\ell}{A} = \frac{\ell}{\gamma A} \tag{2-207}$$

In Figure 2-67b, there is an analogous relationship, i.e.,

$$\mathcal{R} = \frac{\mathfrak{F}}{\phi} = \frac{\ell}{\mu A} \tag{2-208}$$

where $\mathfrak{F}$ is magnetic magnetomotive force in ampere-turns, ϕ is magnetic flux in webers, $\mathcal{R}$ is reluctance—the resistance to magentic flux, and μ is magnetic permeability.
The magnetic magnetomotive force ($\mathfrak{F}$) in Figure 2-67 is given by

$$\mathfrak{F} = Ni \tag{2-209}$$

where $\mathfrak{F}$ is magnetomotive force in ampere-turns, N is number of turns, and i is current in amperes.

Transformers

Transformers are electromagnetic devices that allow electrical power supplied at one potential to be transformed into electrical power at another potential. The potential or voltage may be stepped up (increased) or stepped down (decreased). For instance, in the usual transmission of domestic power, the potential in the transmission lines is greater than the load requirements and a step down transformer is used to reduce the potential at the end use point.

Figure 2-68 illustrates the basics of a transformer. First there is a *core,* usually constructed of a material of high magnetic permeability to achieve a high magnetic flux density. The core has two *windings* of conductors, a *primary coil* (designated as

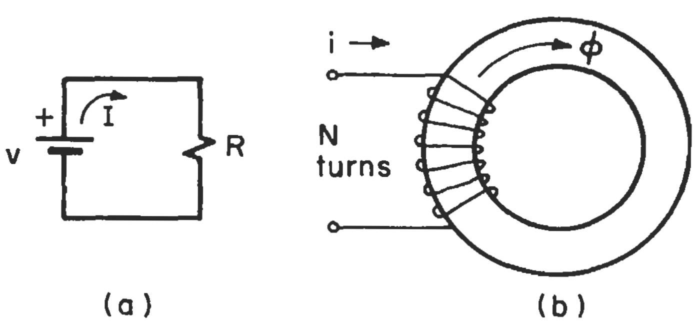


Figure 2-67. Electric and magnetic circuits.

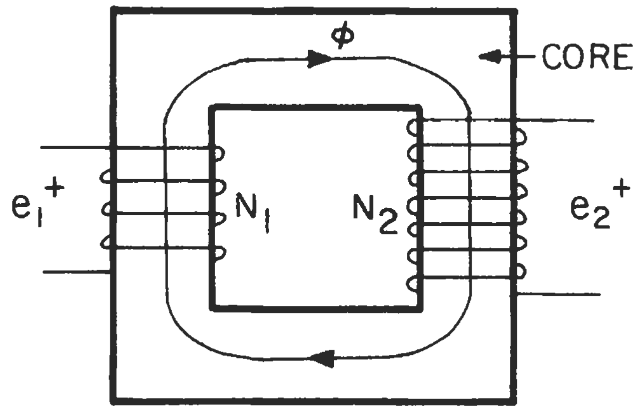


Figure 2-68. Basic transformer operation.

N_1 in the figure) and a *secondary coil* (designated as N_2). Electric current through the primary coil causes a magnetic flux in the core and at the same time an impedance to the current and therefore an induced emf across the primary. The magnetic flux in the core in turn induces an emf across the secondary coil, causing a current to flow. The relation between the emf induced in the primary coil (note that this is not the source emf) and the emf induced in the secondary coil is given by

$$\frac{e_1}{e_2} = \frac{E_1}{E_2} = \frac{N_1}{N_2} \tag{2-210}$$

where e_1, e_2 is AC-induced emfs in volts, E_1, E_2 is rms values of e_1 and e_2, and N_1, N_2 is number of turns on the primary and secondary coils, respectively.

There will be inefficiencies within the transformer, and the voltages at the transformer terminals will vary a little from the previous relationships.

Likewise, the approximate relationship between the primary and secondary currents will be

$$\frac{I_1}{I_2} = \frac{N_2}{N_1} \tag{2-211}$$

Rotating Machines

There is a class of electromechanical equipment in which mechanical energy is converted into electrical energy (or vice versa), all of which use either the response of conductors rotating through a magnetic field or a magnetic field rotating in the presence of stationary conductors. Because the machines rotate, power is transformed in a constant mode rather than the pulsating mode as occurs in similar translational devices.

Figure 2-69 is a schematic of perhaps the simplest rotating machine, the elementary *dynamo*. The elementary dynamo consists of a rectangular-shaped coil, which is free to rotate about an axis. In a practical device, the coil is physically attached to a shaft

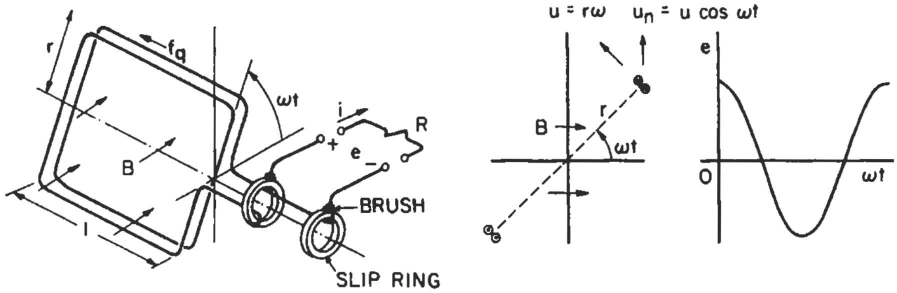


Figure 2-69. Elementary dynamo construction and operation.

at the axis of rotation but is electrically insultated from this shaft. *Slip rings* connected to the coil are also attached but insulated from the shaft. The slip rings and brushes allow electrical contact with an external circuit while the shaft turns.

Consider the effect when the coil turns in the presence of an external magnetic field. An emf will be generated in the coil given by

$$e = NBA\omega \cos wt \tag{2-212}$$

where e is generated emf in volts, N is number of turns of wire in the coil, B is magnetic flux density in teslas, A is area surrounded by the coil = $2r\ell$, r is radius of rotation of the coil, ℓ is length of coil along the axis of rotation, ω is angular velocity in radians per second, and t = time in seconds.

A torque will have to be overcome to maintain the rotation of the coil. This torque is given as

$$\tau_d = NBAi \cos \omega t \tag{2-213}$$

where N, B, A, ω, t are as previously defined, τ_d is developed torque in the elementary dynamo, and i is current.

Note that the direction of the applied torque will dictate the direction of the induced current in the elementary dynamo.

The "external" magnetic field could be due to a magnet or it could be due to the magnetomotive force induced by a current in a conductor (or another stationary coil). The relationship for torque developed when the fields of a stationary (*stator*) coil and a rotating (*rotor*) coil interact is given by

$$\tau_d = k i_s i_r \sin \delta \tag{2-214}$$

where τ_d is torque developed in the dynamo, k is constant term, which includes the number of coil windings, dimensions of the dynamo, velocity of rotation, etc., i_s is current in the stator coil in amperes, i_r is current in the rotor coil in amperes, and δ is angle between the fields, called the *torque* or the *power angle.*

Because the current in each coil induces a magnetic field, the torque relationship may also be given as

$$\tau_d = k' B_s B_r \sin \delta \tag{2-215}$$

where k′ is constant, B_s is magnetic flux density associated with the stator in teslas, and B_r is magnetic flux density associated with the rotor in teslas.

Thus from Equation 2-215 we see that for a given dynamo geometry, the developed torque only depends on the interaction between two magnetic fields and their orientation with respect to each other. One or both of the magnetic fields may be induced by a current. If one of the fields is the field of a magnet, then it may be either in the rotor or the stator. If the rotation results from the imposition of mechanical power on the rotor, the device is called a *generator.* If the rotation is caused by the flow of current, the device is called a *motor,* i.e., converts electric power to mechanical power.

If the rotor of the elementary dynamo is turned in a uniform magnetic field, an AC emf and current are produced. If the speed of rotation is constant, the emf and current are sinusoidal as shown in Figure 2-69.

Figure 2-70 is a schematic of another AC generator called an *alternator.* A DC current is supplied to field windings on the rotor, which are then rotated inside the stator windings, producing the AC emf.

The device of Figure 2-70 can also operate as a motor if a DC current is applied to the rotor windings as in the alternator and an AC current is imposed on the stator windings. As the current to the stator flows in one direction, the torque developed on the rotor causes it to turn until the rotor and stator fields are aligned ($\delta = 0°$). If, at that instant, the stator current switches direction, then mechanical momentum will carry the rotor past the point of field alignment, and the opposite direction of the stator field will cause a torque in the same direction and continue the rotation.

If the slip rings of the elementary dynamo are replaced by a *split-ring commutator,* then a DC emf and current will be generated as shown in Figure 2-71. If the single

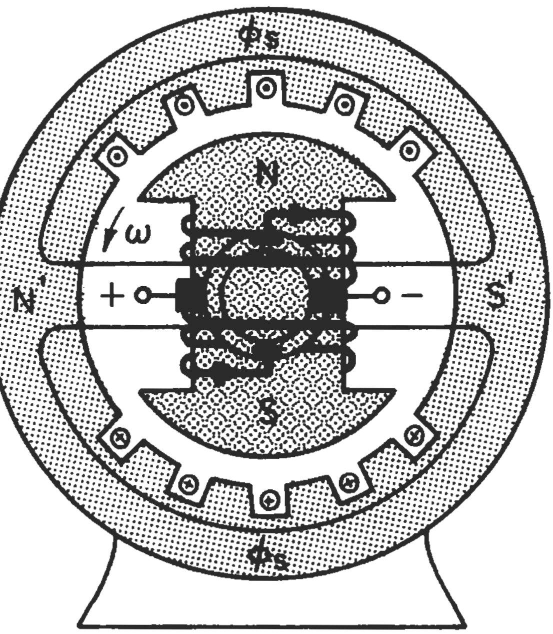


Figure 2-70. An alternator.

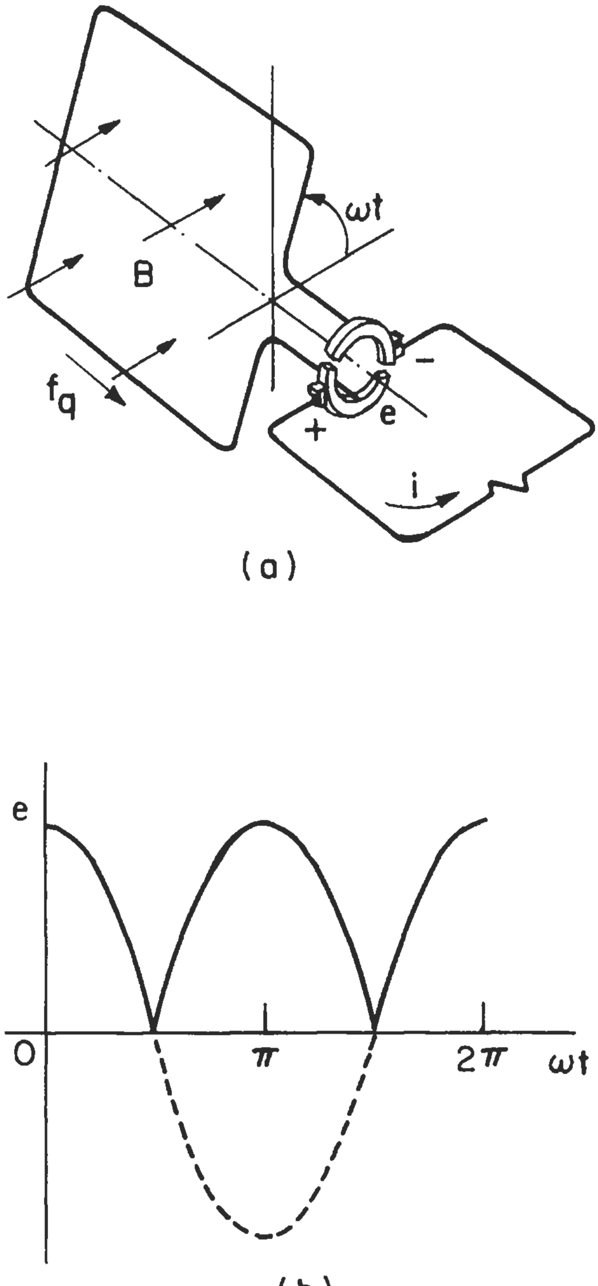


Figure 2-71. Commutator operation and generated voltage waveform.

coil of the elementary dynamo is replaced with multiple coils attached to opposing segments of a multisegment commutator, then the emf generated will be more nearly constant. There will, however, always be a momentary reduction in the emf at the times in the cycle when the spaces in the commutator pass the brushes. By the proper orientation of the rotor (called the *armature*) windings in relation to the commutator segment in contact with the brushes, the torque angle ($\delta = 90°$) can be made to produce maximum torque.

The DC generator can be operated as a motor by imposing a dc current on the armature.

Polyphase Circuits

Circuits that carry AC current employing two, three, or more sinusoidal potentials are called *polyphase* circuits. Polyphase circuits provide for more efficient generation and transmission of power than single-phase circuits. Power in a three- (or more) phase circuit is constant rather than pulsating like the single-phase circuit. As a result, three-phase motors operate more efficiently than single-phase motors.

The usual situation is to have three phases, each generated by the same generator but with a difference of 120° between each phase. Each phase of the generator could be operated independently of the other phases to supply single-phase loads, but to save wiring costs the phases are often run together. The three-phase generator may be connected in either *delta* (Δ) or Y configuration. Figure 2-72 illustrates the two types of generator connections. The coils in the figure are armature windings (see the section on "Rotating Machines"). For the *Y connection* with balanced (equal impedance and impedance phase angle on each phase) load, the *line and coil currents* are *equal*, but the *line emfs* are *three times the coil emfs*. In the *Δ configuration* with balanced loads, the *line current is three times the coil current* while the *coil and line emfs are the same*.

The common connection of all three armature windings in the Y connection allows a fourth, or *neutral*, conductor to be used. This neutral point is often grounded in transmission and distribution circuits. Such a circuit is termed a *three-phase, four-wire* circuit.

In a balanced three-phase circuit, the total power is three times the power in each phase, or

$$P_{total} = 3P_p = 3V_pI_p \cos \theta \tag{2-216}$$

where P_{total} is total circuit power, P_p is phase power, V_p is rms phase potential, I_p is rms phase current, and θ is phase angle difference between phase potential and phase current.

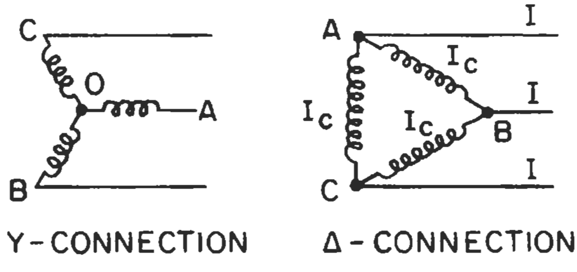


Figure 2-72. Three-phase connections.

Regardless of whether the circuit is connected in Δ or Y, the total power is also

$$P_{total} = \sqrt{3}V_l I_l \cos\theta \tag{2-217}$$

where P_{total} and θ are previously defined, V_l is rms line potential, and I_l is rms line current. Note that θ is not the phase angle difference between line potential and line current.

Just as the armature coils of a three-phase generator may be connected in a Δ or Y configuration, the circuit loads may be connected in a Δ or Y configuration. The Δ-load configuration may be supplied from a source that is connected in either Δ or Y. The Y connection may include a neutral (fourth) wire, connected at the common connection of the circuit.

Power Transmission and Distribution Systems

Electric power is almost always transmitted as three-phase AC current. In domestic use, current is often distributed from a substation at 13,200, 6,600, or 2,300 V, which is stepped down by a transformer close to the point of use to 600, 480, and 240 V for three-phase current for commercial power and 240 and 120 V for single-phase, three-wire current for household power and lights. If DC current is required, synchronous converters or rectifiers are used to convert the AC supply to DC.

Most devices are designed for a constant potential and, as a result, power is usually distributed to loads at constant potential. Two possible configurations for delivering a constant potential to multiple loads are illustrated in Figure 2-73. In the *parallel circuit*, the potential across the load decreases as the distance from the source increases. For the *loop circuit*, potentials are more nearly equal along the length of the circuit.

Transmission lines usually consist of two or more conductors separated by some form of insulation. Such a configuration exhibits a significant resistance, inductance, and capacitance. Figure 2-74 illustrates these effects. Solving the circuit of Figure 2-74 is

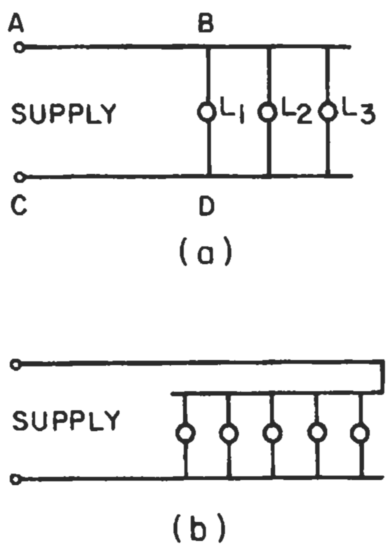


Figure 2-73. Circuits: (a) parallel; (b) loop.

quite involved, and transmission line electrical characteristics are often represented more simply as a lumped parameter model. Figure 2-75 depicts two common lumped models used as equivalent circuits to calculate line losses, changes in phase angle, etc.

The materials for transmission and distribution conductors are usually copper and aluminum. Copper is inexpensive, has a high conductivity, and has sufficient mechanical strength for many uses. Aluminum has the advantage that for a given weight of conductor, it has twice the conductance of copper. A disadvantage of aluminum is that its melting point is lower than copper, while its thermal expansion is greater and stability problems are sometimes encountered. Copper-covered steel is sometimes used in high-voltage transmission for its strength.

Conductor or wire sizes are expressed in terms of the *American Wire Gage* (AWG) system. In this system, the ratio of any wire diameter to the next smaller gage or diameter is 1.123. The AWG sizes range from 40 to 0000. Table 2-37 lists the AWG number, wire dimension, and resistance for solid copper wire. Wires larger than 0000 (as well as smaller wires) are stranded to maintain flexibility.

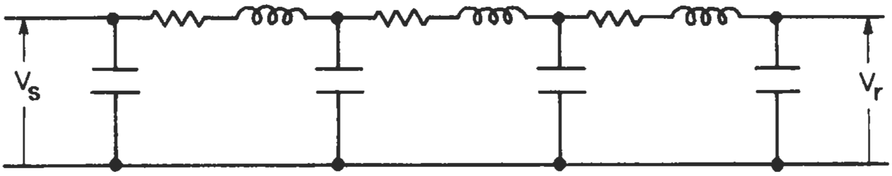


Figure 2-74. Resistance, inductance, and capacitance of transmission lines.

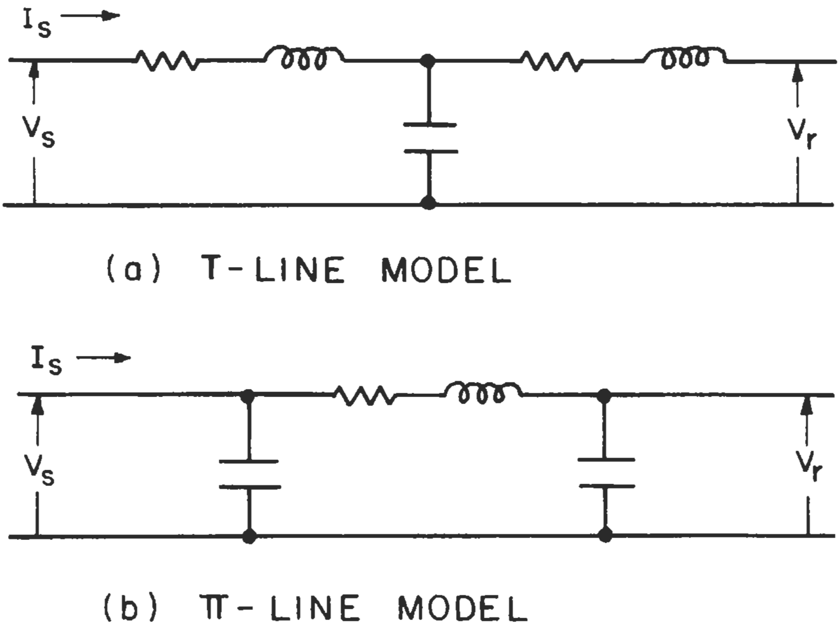


Figure 2-75. Lumped element models of transmission line electrical characteristics.

Table 2-37
Wire Table for Copper

AWG	Diameter, inches	Area, circular mils	Resistance at 20°C (68° F), ohms per 1000 feet of length
0000	0.4600	211,600.	0.04901
000	.4096	167,800.	.06180
00	.3648	133,100.	.07793
0	.3249	105,500.	.09827
1	.2893	83,690.	.1239
2	.2576	66,370.	.1563
3	.2294	52,640.	.1970
4	.2043	41,740.	.2485
5	.1819	33,100.	.3133
6	.1620	26,250.	.3951
7	.1443	20,820.	.4982
8	.1285	16,510.	.6282
9	.1144	13,090.	.7921
10	.1019	10,380.	.9989
11	.09074	8,234.	1.260
12	.08081	6,530.	1.588
13	.07196	5,178.	2.003
14	.06408	4,107.	2.525
15	.05707	3,257.	3.184
16	.05082	2,583.	4.016
17	.04526	2,048.	5.064
18	.04030	1,624.	6.385
19	.03589	1,288.	8.051
20	.03196	1,022.	10.15
21	.02846	810.1	12.80
22	.02535	642.4	16.14
23	.02257	509.5	20.36
24	.02010	404.0	25.67
25	.01790	320.4	32.37
26	.01594	254.1	40.81
27	.01420	201.5	51.47
28	.01264	159.8	64.90
29	.01126	126.7	81.83
30	.01003	100.5	103.2
31	.008928	79.70	130.1
32	.007950	63.21	164.1
33	.007080	50.13	206.9
34	.006305	39.75	260.9
35	.005615	31.32	329.0
36	.005000	25.00	414.8
37	.004453	19.83	523.1
38	.003965	15.72	659.6
39	.003531	12.47	831.8
40	.003145	9.888	1,049.
41	.002800	7.840	1,323.
42	.002494	6.200	1,673.
43	.002221	4.928	2,104.
44	.001978	3.881	2,672.
45	.001760	3.098	3,348.

Interior wiring design and installation for most commercial and industrial uses should follow the *National Electrical Code* (NEC) which has been a national standard since 1970 with the passage of the Occupational Safety and Health Act (OSHA). Some localities, however, may not accept the NEC and require that their own (more stringent) standards be followed.

For further information on this subject, refer to References 1 and 41 through 45.

CHEMISTRY

Introduction

The material in this section is divided into three parts. The first subsection deals with the general characteristics of chemical substances. The second subsection is concerned with the chemistry of petroleum; it contains a brief review of the nature, composition, and chemical constituents of crude oil and natural gases. The final subsection touches upon selected topics in physical chemistry, including ideal gas behavior, the phase rule and its applications, physical properties of pure substances, ideal solution behavior in binary and multicomponent systems, standard heats of reaction, and combustion of fuels. Examples are provided to illustrate fundamental ideas and principles. Nevertheless, the reader is urged to refer to the recommended bibliography [47–52] or other standard textbooks to obtain a clearer understanding of the subject material. Topics not covered here owing to limitations of space may be readily found in appropriate technical literature.

Characteristics of Chemical Compounds

The general nature of chemical compounds and their physical behavior are briefly reviewed in the following subsection.

Chemical Bonds

Two types of chemical bonds, ionic and covalent, are found in chemical compounds. An ionic bond results from the transfer of valence electrons from the atom of an electropositive element (M) to the atom(s) of an electronegative element (X). It is due to coulombic (electrostatic) attraction between the oppositely charged ions, M^+(cation) and X^-(anion). Such ionic bonds are typical of the stable salts formed by combination of the metallic elements (Na, K, Li, Mg, etc.) with the nonmetallic elements (F, Cl, Br, etc.). As an example, the formation of the magnesium chloride molecule from its elemental atoms is shown by the following sequence:

$$Mg \rightarrow Mg^{2+} + 2e$$

$$2Cl + 2e \rightarrow 2Cl^-$$

$$Mg^{2+} + 2Cl^- \leftrightarrow Cl^-Mg^{2+}Cl^- \quad \text{or} \quad MgCl_2$$

where e represents an electron. The ionic charges resulting from the gain or loss of electrons are called electrovalences. Thus, the electrovalence of magnesium is +2 and of chlorine −1. Ionic compounds are mostly inorganic substances.

A nonionic compound is made up of covalent bonds only. A covalent bond results from shared electron pairs between two atoms. It consists of electrostatic attraction between each electron and both nuclei, e.g.,

$$H^{\times} + \cdot H \rightarrow H\overset{\times}{\cdot}H \qquad \text{(hydrogen, } H_2\text{)}$$

$$2H^{\times} + \cdot\overset{\cdot}{\underset{\cdot\cdot}{O}}: \rightarrow H\overset{\times}{\cdot}\overset{\overset{H}{\times\cdot}}{\underset{\cdot\cdot}{O}}: \qquad \text{(water, } H_2O\text{)}$$

where the symbols × and • represent electrons in the outermost shells of the atoms. Covalent compounds are mostly organic. In compounds containing both ionic and covalent bonds (e.g., KNO_3, $MgSO_4$, etc.), the physical properties are primarily determined by the ionic character.

Polarity in Covalent Compounds

A covalent bond will exhibit polarity when it is formed from atoms that differ in electronegativity, i.e., the ability to attract electrons. The order of electronegativity of some elements [50, p. 16] is

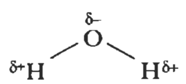
$$F > O > N > Cl > Br > C > S > I > H > B > Si, Zn > Mg > Li, Na > K > Sn$$

As an illustration, in the iodine monochloride (I — Cl) molecule, the electron pair being shared remains closer to the more electronegative Cl atom. This creates fractional negative and positive charges, referred to as formal charges, on the Cl and I atoms, respectively, as illustrated below:

$${}^{\delta+}I \qquad Cl^{\delta-}$$

The symbols δ+ and δ– indicate polarity of the two ends or poles of the electrically neutral molecule. Such a polar molecule constitutes a permanent dipole, i.e., two equal and opposite charges (e) separated by a distance (d) in space. A quantitative measure of the polarity of a molecule is the dipole moment (μ in Debye units), which is defined as the product of the charge (e in electrostatic units) and the distance (d in cm).

Polarity of bonds can lead to polarity of molecules, as shown in the case of the water molecule:



(bond angle = 105°)

Nonpolar molecules such as H_2, N_2, O_2, I_2, and Cl_2 have zero dipole moments, because e = 0. On the other hand, hydrogen fluoride, HF, has a large dipole moment of 1.75 Debye and so is strongly polar. Simple carbon compounds with symmetric arrangement of like atoms (e.g., methane, CH_4, and carbon tetrachloride, CCl_4) have zero dipole moments and so are nonpolar.

In polar compounds, the operative intermolecular forces are dipole–dipole interactions, which refer to the attraction between the positive pole of one molecule and the negative pole of another. For this reason, polar compounds are relatively more stable than nonpolar substances. A particularly strong kind of dipole–dipole attraction is hydrogen bonding, in which a hydrogen atom acts as a bridge between two electronegative atoms, holding one atom by a covalent linkage and the other by purely electrostatic forces, e.g., Hydrogen fluoride $\mathrm{H - F \cdots F - H}$

Water
$$\begin{array}{ccccc} \mathrm{H} & - \mathrm{O} & \cdots & \mathrm{O} - & \mathrm{H} \\ & | & & | & \\ & \mathrm{H} & & \mathrm{H} & \end{array}$$

where the sequence of dots indicates the hydrogen bridge. This bond has a strength of 5 kcal/g-mole versus 50–100 kcal/g-mole for the covalent bond, but it is much stronger than other dipole–dipole interaction [49, p. 27]. For hydrogen bonding to be important, both electronegative atoms must belong to the group: F, O, N.

The intermolecular forces operative in nonpolar compounds are also electrostatic in nature. These weak van der Waals forces involve attraction between nonbonded atoms and are effective over short ranges only.

Physical Properties

The degree of polarity has considerable influence on the physical properties of covalent compounds and it can also affect chemical reactivity. The melting point (mp) and boiling point (bp) are higher in ionic substances due to the strong nature of the interionic forces, whereas the covalent compounds have lower values due to the weak nature of intermolecular forces.

The mp and bp increase in the order of nonpolar to polar to ionic compounds. Associated liquids, in which the molecules are held together by hydrogen bonds, show higher bp than nonassociated polar compounds of similar molecular weights.

Ionic compounds in solution and in molten state are good conductors of electricity. Melts and solutions of covalent substances are nonconducting. Inorganic substances rarely undergo combustion, whereas (organic) covalent compounds do so readily.

Solubility

Ionic compounds can dissolve appreciably in highly polar solvents (e.g., water, liquid ammonia, sulfuric acid). In solution, each ion is surrounded by a cluster of solvent molecules and is said to be solvated; if the solvent is water, the ion is said to be hydrated. To reduce the attraction between solvated ions of opposite charges, the solvent must also have a high dielectric constant. Besides high polarizability and dielectric constant, water can also form hydrogen bonds and, hence, is an excellent solvent. Water can solvate cations at its negative pole and anions via hydrogen bonding.

The solubility of nonionic compounds is largely dictated by their polarity, in accordance with the axiom, "like dissolves like." That is, nonpolar compounds dissolve in nonpolar solvents and polar substances dissolve in polar solvents.

Petroleum Chemistry

Petroleum chemistry is concerned with the origin, composition, and properties of naturally occurring petroleum deposits, whether in liquid (crude oil or petroleum), gaseous (natural gas), or solid (tars and asphalts) form. All of them are essentially mixtures of hydrocarbons. Whereas natural gas contains a few lighter hydrocarbons, both crude oil and tar deposits may consist of a large number of different hydrocarbons that cannot be easily identified for molecular structure or analyzed for composition.

For general literature on the subject of petroleum chemistry consult References 54 through 58.

Crude Oils

Elemental compositional analysis indicates that the ratio of hydrogen to carbon atoms is approximately 1.85:1 in typical crude oils [51, p. 28]; the other elements, chiefly sulfur, nitrogen, and oxygen account for less than 3% by weight in most light crudes. Table 2-38 illustrates the elemental compositions of typical crude oils and asphalts. Sour crudes contain significantly larger amounts of sulfur-containing compounds. Traces of phosphorous and heavy metals such as vanadium, nickel, and iron are also present.

In refining operations, crude oils are subjected to fractional distillation by which they are separated into different fractions according to the boiling point range of the compounds and their end use or application (see Table 2-39).

A "base" designation is given in the refining industry to classify crude oils: (i) A *paraffin-base* crude contains predominantly paraffins and small amounts of naphthenes or asphalt. Upon distillation, it yields fine lubricating oils from the gas-oil fraction and paraffin wax from the solid residue. (ii) An *asphalt-base* crude contains mostly cyclic compounds (primarily naphthenes), which, upon distillation, produce high yields of black, pitchlike, solid residue, asphalt, and heavy fuel oil. (iii) A *mixed-base* crude has characteristics intermediate between the above two categories. (iv) An *aromatic-base* crude contains large amounts of low-molecular weight aromatics together with naphthenes, and small amounts of asphalt and paraffins.

Tars and Asphalts

These are semisolid or solid substances formed in nature from crude oils after the volatile components have evaporated and the remainder has undergone oxidation and polymerization. They are also referred to as bitumens, waxes, and pitch. These materials are believed to consist of mixtures of complex organic molecules of high molecular weight. As with crude oils, which contain thousands of different chemical compounds, an exact chemical analysis for identification and composition is impractical to perform on the solid deposits of petroleum.

Table 2-38
Elemental Composition of Natural Petroleum (Percentage by Weight) [55,56,57]

Element	Most Crude Oils	Typical[c] Crude	Typical[c] Asphalt
Carbon	83–87[a]	84.5	84
Hydrogen	11–15[a]	13.0	10
Sulfur	trace - 8[b]	1.5	3
Nitrogen	trace - 1.6[b]	0.5	1
Oxygen	trace - 1.8[b]	0.5	2
Metals (Ni, V, etc.)	trace - 1,000 ppm[b]	trace - 1,000 ppm	

Source a: Reference 56; Source b: Reference 55; Source c: Reference 57

Table 2-39
Typical Crude Oil Fractions [58]

Crude fraction	Boiling point, °F (melting point)	Approximate chemical composition	Uses
Hydrocarbon gas		C_1–C_2	Fuel gas
	to 100	C_3–C_6	Bottled fuel gas, solvent
Gasoline	100–350	C_5–C_{10}	Motor fuel, solvent
Kerosene	350–450	C_{11}–C_{12}	Jet fuel, cracking stock
Light gas oil	450–580	C_{13}–C_{17}	Diesel fuel, furnace fuel
Heavy gas oil	580–750	C_{18}–C_{25}	Lubricating oil, bunker fuel
Lubricants and waxes	750–950 (100)	C_{26}–C_{38}	Lubricating oil, paraffin wax, petroleum jelly
Residuum	950+ (200+)	C_{38} +	Tars, roofing compounds, paving asphalts, coke, wood preservatives

Natural Gases

Gas collected at the wellhead is mostly methane with decreasing amounts of heavier hydrocarbons. Typical compositions are given in Table 2-40. Natural gases are primarily mixtures of normal alkanes in the C_1 to C_4 range although other paraffins and heavier hydrocarbons may also be present. The nonhydrocarbon content usually includes water vapor, carbon dioxide (CO_2), hydrogen sulfide (H_2S), nitrogen and helium. At conditions of high pressure and low temperature, solid hydrates may form as a result of the chemical combination of H_2O with the hydrocarbons.

Natural gases are classified as sweet or sour (similar to crude oils), depending upon the absence or presence of significant amounts of hydrogen sulfide. A designation of wet gas implies that the natural gas is capable of producing liquid hydrocarbons upon suitable treatment; a dry gas does not have such ability. Processing of natural gas results in a variety of products including pure methane, liquefied petroleum gas (or LPG, which is mostly propane and some n-butane), and gasoline.

Chemistry of Petroleum

Since hydrocarbons form a majority of the types of compounds found in natural petroleum, it is necessary to review briefly the fundamental characteristics of the much larger class of organic compounds. Organic chemistry deals with the chemistry of the compounds of carbon. Covalent bonds, formed by the sharing of one or more electrons between two atoms, are found in organic molecules. Because carbon is a tetravalent atom, single, double, or triple covalent bonds are possible between two carbon atoms. Organic chemistry is based on the structural theory according to which compounds can be grouped into homologous series in which molecules with similar structure exhibit similar physical and chemical properties.

Table 2-40
Composition of Typical Petroleum Gases [58]

Natural Gas	
Hydrocarbon	
Methane	70–98%
Ethane	1–10%
Propane	trace–5%
Butanes	trace–2%
Pentanes	trace–1%
Hexanes	trace–1/2%
Heptanes +	trace–1/2%
Nonhydrocarbon	
Nitrogen	trace–15%
Carbon dioxide*	trace–5%
Hydrogen sulfide*	trace–3%
Helium	up to 5%, usually trace or none

*Occasionally natural gases are found which are predominately carbon dioxide or hydrogen sulfide.

Gas from a well which also is producing petroleum liquid	
Hydrocarbon	
Methane	45–92%
Ethane	4–21%
Propane	1–15%
Butanes	½–7%
Pentanes	trace–3%
Hexanes	trace–2%
Heptanes +	none–1½%
Nonhydrocarbon	
Nitrogen	trace–up to 10%
Carbon dioxide	trace–4%
Hydrogen sulfide	none-trace–6%
Helium	none

Classification of Hydrocarbons

Although organic compounds may be formed from atoms of carbon and other elements, a more restricted class of compounds known as hydrocarbons is made up of carbon and hydrogen atoms only. A broad classification based on structural considerations divides all hydrocarbons into acyclic (open-chain structure) and cyclic (ring structure) compounds which are further divided into certain homologous groups (see Figure 2-76). A simple definition for a homologous series is a family of compounds in which any two successive members differ by a structural unit that is common for the series. The members of such a family show gradual changes in physical properties (e.g., boiling point, melting point, specific gravity, etc.) from one member to the next according to the number of C atoms in the backbone chain.

The nomenclature of organic compounds is based on the convention adopted by the International Union of Pure and Applied Chemistry (IUPAC). Some of these

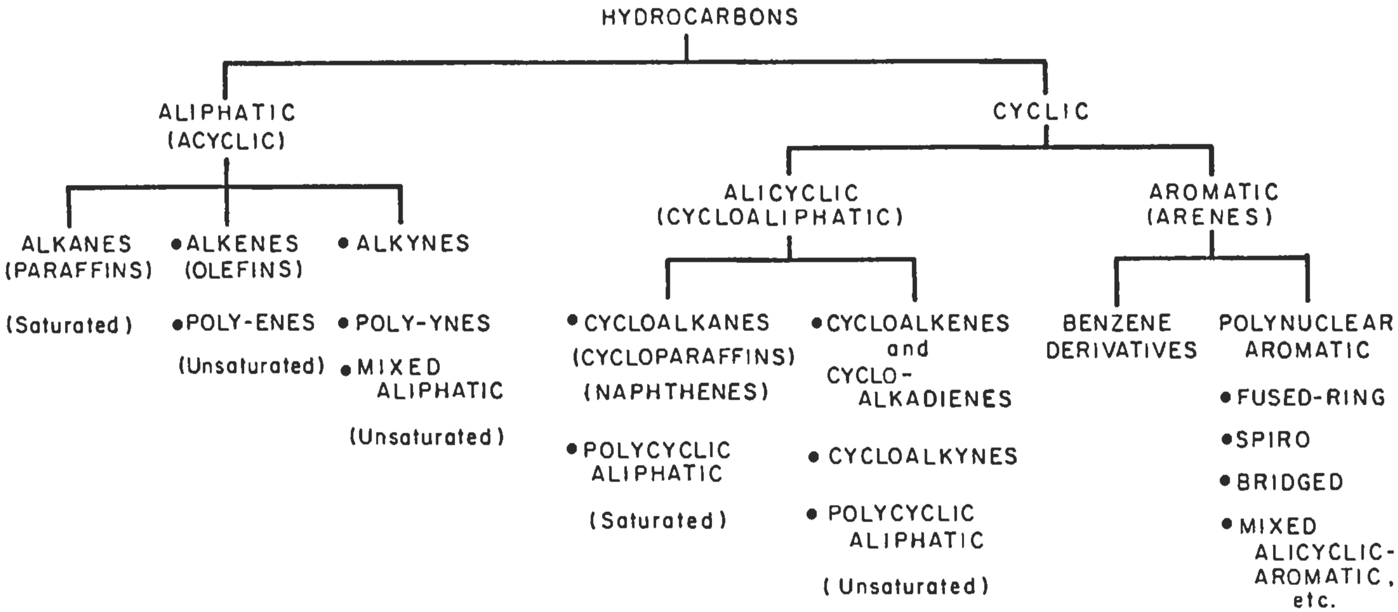


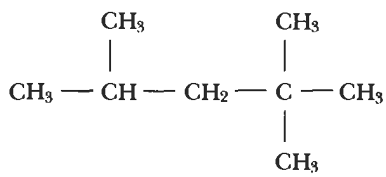
Figure 2-76. Classification of hydrocarbons based on structure.

guidelines are presented in Section C of the *CRC Handbook of Chemistry and Physics* [62] and should be referred to for a better understanding of the rules. A brief discussion of the various classes of hydrocarbons follows.

Aliphatic Hydrocarbons. These are "acyclic" hydrocarbons with an open-chain structure, which can be either straight (i.e., linear) or branched. The former type are called normal (or n-) aliphatic compounds. Unsaturation is manifested in the form of double or triple bonds.

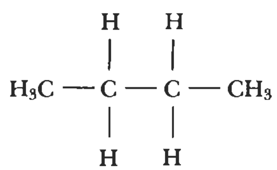
Alkanes. These have the general formula C_nH_{2n+2}, where n is the number of carbon atoms in the alkane molecule, and $n \geq 1$. These are also known as "paraffins" or "saturated aliphatic hydrocarbons," since all of the carbon atoms in the chain are connected by single covalent bonds. Continuous or straight-chain alkanes are called normal paraffins or n-alkanes (e.g., methane, CH_4; ethane, C_2H_6; propane, C_3H_8; n-butane, n-C_4H_{10}; etc.). The corresponding *alkyl* groups, methyl, $-CH_3$; ethyl, $-C_2H_5$; n-propyl, $-C_3H_7$; n-butyl, $-C_4H_9$; etc., are generally represented by the symbol R–.

Branched-chain alkanes, also known as "isoparaffins" or "isoalkanes," are possible when $n \geq 4$. The prefix "iso" is used when two methyl groups are attached to a terminal carbon atom of an otherwise straight chain and the prefix "neo" when three methyl groups are attached in that manner. Branched-chain alkanes are sometimes regarded as normal alkanes with attached substituent alkyl groups. An example is

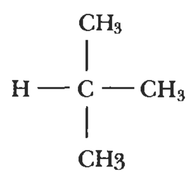


2,2,4-trimethlypentane (isooctane)

Isomers are substances having the same molecular formula and molecular weight, but differing in physical and chemical properties. Since branched and straight-chain alkanes with the same molecular formula can exist as distinct structures having different geometrical arrangement of the atoms, they are termed *structural isomers.* One example is C_4H_{10} (butane) which has two isomers:

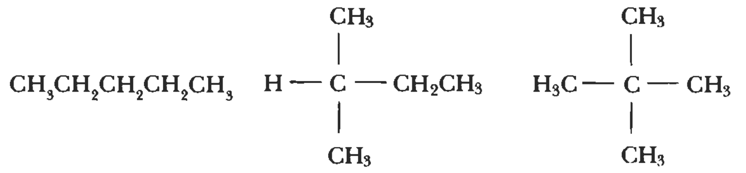


and



[$CH_3CH_2CH_2CH_3$ (n-butane)] [$(CH_3)_3CH$ (isobutane)]

As the carbon number increases in the chain, the number of possible structural isomers grows very rapidly as a result of increased branching possibilities. For example, C_5H_{12} has only three isomers,



[n-C_5H_{12} (n-pentane)] [$(CH_3)_2CHC_2H_5$ (isopentane)] [$(CH_3)_4C$ (neopentane)]

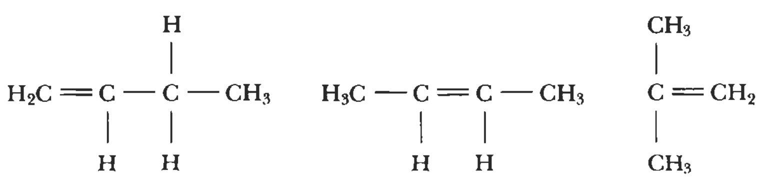
while $C_{10}H_{22}$ (decane) has 75, and $C_{20}H_{42}$ (eicosane) has 366,319 possible isomers.

Whereas n-alkanes exhibit smooth and graded variations in physical properties (see Table 2-41), the branched members do not [49, p. 86]. The structural isomers of any alkane generally show dissimilar physical and chemical characteristics. A branched-chain isomer has a lower boiling point than a straight-chain isomer, and the more numerous the branches, the lower its boiling point. Alkanes are either nonpolar or weakly polar. They are soluble in nonpolar or weakly polar solvents (e.g., benzene, chloroform, ether), and are insoluble in water and other highly polar solvents. Alkanes can dissolve compounds of low polarity. Chemically, paraffins are stable and quite unreactive at ordinary conditions. At high temperatures, they can burn completely in the presence of excess oxygen or air to yield CO_2 and H_2O as products. The combustion reaction is exothermic.

At ambient temperature and pressure, the first four members of the n-alkane series (methane to n-butane) are gases, the next thirteen (n-pentane through n-heptadecane) are liquids, and the higher members from n = 18 on are solids.

Alkenes. These are also called "olefins" and have the general formula C_nH_{2n}, with $n \geq 2$. They contain a single C=C double bond and are named in accordance with the IUPAC convention by specifying the location of the double bond from the terminal carbon atom nearest to it.

The first three members of the olefin series are ethylene, propylene, and butylene (or butene). Structural isomers exist when $n \geq 4$, as a consequence of the positioning of the double bond in normal alkenes as a result of branching in branched alkenes. In addition, *geometric isomers* may be possible owing to restricted rotation of atoms about the C=C bond. For instance, C_4H_8 (butene) has four possible isomers instead of the expected three:

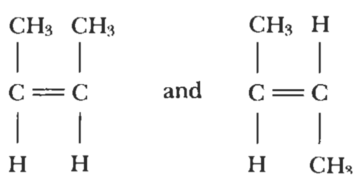


[$CH_2CHCH_2CH_3$ (1-butene)] [$CH_3CHCHCH_3$ (2-butene)] [$(CH_3)_2CCH_2$ (isobutene)]

This occurs because 2-butene, itself, can exist in two different structures, the cis- or the trans- configurations, depending on whether the methyl groups are situated on the same side or on opposite sides of the main chain.

Table 2-41
Physical Properties of Normal Alkanes [47]

Name	Molecular Formula	Melting Point (°C)	Boiling Point (°C)	Sp Gr (liquid)	State Under Atmospheric Conditions
Methane	CH_4	−183	−162	0.4240	Gas
Ethane	C_2H_6	−183	−89	0.5462	Gas
Propane	C_3H_8	−187	−42	0.5824	Gas
n-Butane	C_4H_{10}	−138	0	0.5788	Gas
n-Pentane	C_5H_{12}	−130	36	0.6264	Liquid
n-Hexane	C_6H_{14}	−95	69	0.6594	Liquid
n-Heptane	C_7H_{16}	−91	98	0.6837	Liquid
n-Octane	C_8H_{18}	−57	126	0.702	Liquid
n-Nonane	C_9H_{20}	−54	151	0.7179	Liquid
n-Decane	$C_{10}H_{22}$	−30	174	0.7298	Liquid
n-Undecane	$C_{11}H_{24}$	−26	196	0.7404	Liquid
n-Dodecane	$C_{12}H_{28}$	−10	216	0.7493	Liquid
n-Tridecane	$C_{13}H_{28}$	−6	235	0.7568	Liquid
n-Tetradecane	$C_{14}H_{30}$	6	254	0.7636	Liquid
n-Pentadecane	$C_{15}H_{32}$	10	271	0.7688	Liquid
n-Hexadecane	$C_{16}H_{34}$	18	287	0.7749	Liquid
n-Heptadecane	$C_{17}H_{36}$	22	302	0.7767	Liquid
n-Octadecane	$C_{18}H_{38}$	28	316	0.7767	Solid
n-Nonadecane	$C_{19}H_{40}$	32	330	0.7776	Solid
n-Eicosane	$C_{20}H_{42}$	36	343	0.7777	Solid
n-Heneicosane	$C_{21}H_{44}$	40	356	0.7782	Solid
n-Docosane	$C_{22}H_{46}$	44	369	0.7778	Solid
n-Tricosane	$C_{23}H_{48}$	48	380	0.7797	Solid
n-Tetracosane	$C_{24}H_{50}$	51	391	0.7786	Solid
n-Pentacosane	$C_{25}H_{52}$	53	402	0.7979	Solid
n-Triacontane	$C_{30}H_{62}$	66	450	0.8064	Solid
n-Pentatriacontane	$C_{35}H_{72}$	75	490	0.8126	Solid
n-Tetracontane	$C_{40}H_{82}$	81	525	0.8172	Solid
n-Pentacontane	$C_{50}H_{102}$	92		0.7940	Solid
n-Hexacontane	$C_{60}H_{122}$	104			Solid
n-Dohexacontane	$C_{62}H_{126}$	101			Solid
n-Tetrahexacontane	$C_{64}H_{130}$	102			Solid
n-Heptacontane	$C_{70}H_{142}$	105			Solid



(cis-2-butene) (trans-2-butene)

With the inclusion of these two geometric isomers, butene has a total of four isomers.

Alkenes also form a homologous series; as the carbon number increases, the number of possible isomeric structures for each member increases more rapidly than in the case of the alkane series.

The physical properties of alkenes [49, p. 152] are not very different from the corresponding members of the alkane family. Alkenes are nonpolar or at most weakly polar. They are insoluble in water but are soluble in concentrated H_2SO_4 and liquid HF. Analogous to n-alkanes, normal alkenes dissolve in nonpolar or weakly polar organic liquids such as ethers, CCl_4, and hydrocarbons. In general, the cis- isomer has a slightly higher polarity, a higher boiling point, and a lower melting point than the trans- isomer, but there are exceptions. Chemically, olefins are more reactive than paraffins due to the presence of the double bond, which can be split into two single (stable) bonds.

Alkadienes, alkatrienes, and alkatetraenes (poly-enes). These are unsaturated aliphatic hydrocarbons containing two, three, or four C=C double bonds, respectively. Alkadienes are also called "diolefins" or "dienes," and alkatrienes are also known as "triolefins" or "trienes." Alkenes containing multiple double bonds fall under the general class of "poly-enes." Double bonds that alternate with single bonds in a straight chain are said to be *conjugated.* Examples are

$CH_2 = CH - CH = CH_2$ (1,3-butadiene)
$CH_2 = CH - CH = CH - CH = CH_2$ (1,3,5-hexatriene)

Double bonds that occur together in a straight chain are said to be *cumulated.* Examples are

$CH_2 = C = CH_2$ [allene (propadiene)]
$CH_3 - CH = C = CH - CH_3$ (2,3-pentadiene)

If the double bonds are separated by more than one single bond, they are said to be *isolated* or *unconjugated.* An example is

$CH_2 = CH - CH_2 - CH = CH_2$ (1,4-pentadiene)

Although alkadienes have a higher degree of unsaturation than alkenes, their chemical behavior is similar to alkenes, and their physical properties are similar to alkanes containing the same number of carbon atoms. Common *alkenyl* groups include

$CH_2 = CH -$	$CH_2 = CHCH_2 -$	$CH_3CH = CH -$	$CH_3CH = CHCH_2 -$
(vinyl)	(allyl)	(propenyl)	(crotyl)

Alkynes. These contain a single triple bond and have the general formula C_nH_{2n-2}, with $n \geq 2$. Alkynes are also referred to as "acetylinic compounds." The simple alkynes are alternatively named as derivatives of acetylene, e.g.,

$HC \equiv CH$ [acetylene (or ethyne)]
$CH_3C \equiv CH$ (methylacetylene or propyne)

Butyne, C_4H_6, exists as two isomers:

$CH_3CH_2C \equiv CH$ (1-butyne or ethylacetylene)
$CH_3C \equiv C - CH_3$ (2-butyne or dimethylacetylene)

Physical properties of alkynes [49, p. 251] are essentially similar to those of alkanes and alkenes. These compounds are weakly polar and are insoluble in water, but they are quite soluble in organic solvents of low polarity (e.g., ether, benzene, CCl_4). Chemically, alkynes are more reactive than alkanes but behave like alkenes. The triple bond appears to be less reactive than the double bond in some reagents while more reactive in others. In a chemical reaction, the triple bond is usually broken into a double bond, which may eventually split into single bonds.

Diynes and triynes refer to alkynes containing two or three triple bonds; *poly-ynes* contain multiple triple bonds. A conjugated triyne is a straight-chain hydrocarbon with triple bonds alternating with single bonds. An examples is

$CH_3 - C \equiv C - C \equiv C - C \equiv C - CH_3$ (2,4,6-octatriyne)

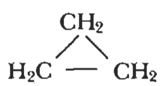
When both double and triple bonds occur in the same molecule, the IUPAC system recommends the use of both endings -ene, and -yne, with the former always preceding the latter in the name. Common *alkynyl* groups are

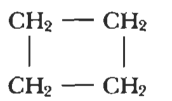
$HC \equiv C -$ and $HC \equiv C - CH_2 -$
(ethynyl) (propargyl)

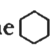
Cyclic Hydrocarbons. These are structures in which the carbon atoms form a ring instead of an open chain. They are also called "carbocyclic" or "homocyclic" compounds. They are divided into two classes: alicyclic (or cycloaliphatic) and aromatic compounds.

Alicyclic Hydrocarbons. These refer to cyclic analogues of aliphatic hydrocarbons and are named accordingly, using the prefix "cyclo-." Their properties are similar to their open-chain aliphatic counterparts. Alicyclic hydrocarbons are subdivided into monocyclic (cycloalkanes, cycloalkenes, cycloalkynes, cycloalkadienes, etc.) and polycyclic aliphatic compounds. Monocyclic aliphatic structures having more than 30 carbon atoms in the ring are known, but those containing 5 or 6 carbon atoms are more commonly found in nature [47, p. 28].

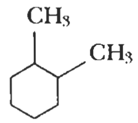
Cycloalkanes (or naphthenes). These are also known as "cycloparaffins or "saturated alicyclic hydrocarbons." They are quite stable compounds with the general formula C_nH_{2n}, with $n \geq 3$ for rings without substituent groups. The first two members are

(cyclopropane)  or 

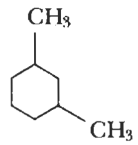
(cyclobutane)  or 

the next two being cyclopentane  and cyclohexane . For convenience, aliphatic rings are represented by polygons.

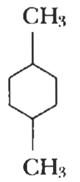
When substituent groups are present, they are identified and their positions indicated by numbers in naming the compound. As an example, dimethylcyclohexane has three structures:



(1,2-dimethylcyclohexane)



(1,3-dimethylcyclohexane)



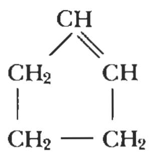
(1,4-dimethylcyclohexane)

Cis-trans isomerism occurs in each of the above disubstituted cycloalkanes.

Physical properties of cycloalkanes [49, p. 284; 50, p. 31] show reasonably gradual changes, but unlike most homologous series, different members exhibit different degrees of chemical reactivity. For example, cyclohexane is the least reactive member in this family, whereas both cyclopropane and cyclobutane are more reactive than cyclopentane. Thus, hydrocarbons containing cyclopentane and cyclohexane rings are quite abundant in nature.

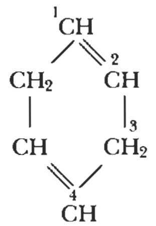
Cycloalkenes and cyclohexadienes. These unsaturated cyclic aliphatic compounds [49, p. 284] have one and two double bonds, respectively, in the ring. Examples are

(1-cyclopentene)

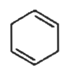


or

(1,4-cyclohexadiene)



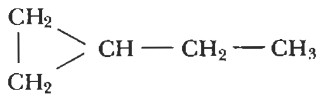
or



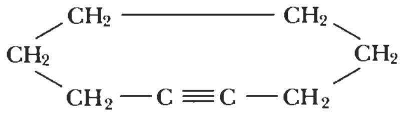
They are chemically as reactive as their straight-chain counterparts. Cycloalkenes can lose their double bond in addition reactions. In scission or cleavage reactions, the ring structure opens up into a straight chain.

Hydrocarbons containing both aliphatic and alicyclic parts may be named by considering either part as the parent structure and the other part as a substituent, e.g.,

(ethylcyclopropane)

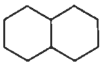


Cycloalkynes. These structures have one triple bond in the carbon ring, as shown in the following example:

(cyclooctyne) 

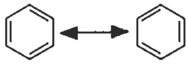
Although it is possible to conceive of alicyclic hydrocarbons containing more than a triple bond or two double bonds in the carbocyclic ring, such ring structures are usually either unstable or have transient existence.

Polycyclic aliphatic hydrocarbons. These may contain two or more rings that share two or more carbon atoms. An example of a fused-ring system is

 [decahydronaphthalene (or decalin)]

The aliphatic rings may be saturated or partially unsaturated. Spiro hydrocarbons also belong to the polycyclic group. More examples may be found in the *CRC Handbook* [63].

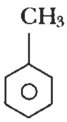
Aromatic Hydrocarbons (or Arenes). These are unsaturated cyclic compounds, usually with benzene or its derivatives as the common building block. Their chemical reactivity is similar to benzene. Benzene, the simplest aromatic hydrocarbon, has the molecular formula C_6H_6. It is a flat and symmetrical molecule with six carbon atoms arranged in a hexagonal ring (bond angle = 120°) with six attached hydrogen atoms. The resonance theory has been proposed to explain the high thermochemical stability of benzene. According to this theory, the benzene ring has six identical hybrid bonds between the carbon atoms, each one intermediate between a single and a double bond. It postulates that the molecule has a hybrid structure between the following two resonance configurations:

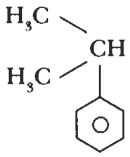
 and is represented by 

It is important to note that benzene does not behave like a typical cyclic olefin in that the benzene ring undergoes ionic substitution rather than addition reactions; the ring also resists hydrogenation and is chemically more inert. Despite this, it is still a common practice to represent benzene with three double bonds as if it were 2,4,6-cyclohexatriene.

It is convenient to divide aromatic hydrocarbons into two groups: (1) benzene derivatives, and (2) polynuclear aromatics containing multiring structures.

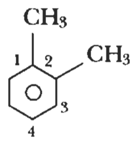
Benzene derivatives. The nomenclature is a combination of the IUPAC system and traditional names. Many of the derivatives are named by the substituent group appearing as the prefix. These may be considered a subclass of the aliphatic-aromatic hydrocarbon family, which contains both aliphatic and aromatic units in its structures. Thus, alkylbenzenes are made up of a benzene ring and alkane units; alkenylbenzenes are composed of a benzene ring and alkene units; and alkynylbenzenes comprise a benzene ring and alkyne units. Examples of alkylbenzenes include

 or $C_6H_5CH_3$ [toluene (or methylbenzene)]

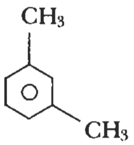


or $C_6H_5CH(CH_3)_2$ [cumene (or isopropylbenzene)]

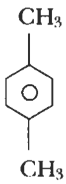
If several groups are attached to the benzene ring, their names as well as their relative positions should be indicated. For example, dimethylbenzene or xylene, $C_6H_4(CH_3)_2$, has three geometric isomers, with prefixes ortho-, meta-, and para-, indicating the relative positions of the two methyl groups.



[ortho-xylene (o-xylene or 1,2-dimethylbenzene)]

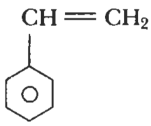


[meta-xylene (m-xylene or 1,3-dimethylbenzene)]

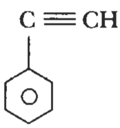


[para-xylene (p-xylene or 1,4-dimethylbenzene)]

Examples of an alkenylbenzene and an alkynylbenzene are given below:

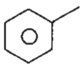


and



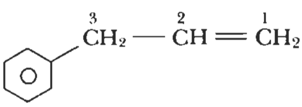
[styrene (vinylbenzene)] (phenylacetylene)

When the benzene ring is regarded as the substituent group, it is called phenyl group and is represented by



or C_6H_5

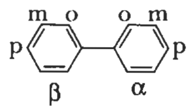
An example is

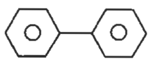


or $CH_2CHCH_2(C_6H_5)$

(3-phenylpropene)

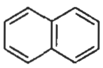
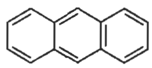
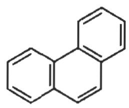
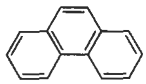
Polynuclear aromatic hydrocarbons. These consist of a variety of complex structures made up of aromatic rings alone, or combinations of aliphatic rings, aromatic rings, and aliphatic chains, etc. One such class of compounds is biphenyl and its derivatives, in which two benzene rings are connected by a single C – C linkage. The structural formula of biphenyl (or phenylbenzene) is



or

or $C_6H_5C_6H_5$

in which the ortho-, meta-, and para- positions of the carbon atoms in the α and β rings are as marked.

Another class of compounds is called condensed-ring or fused-ring systems. These structures contain two or more aromatic rings that share a pair of carbon atoms. Examples include naphthalene, anthracene, and phenanthrene, the latter two being isomeric structures.

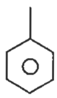



or


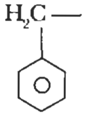
(naphthalene) (anthracene) (phenanthrene)

Other polynuclear hydrocarbons may include bridged hydrocarbons, spiro hydrocarbons, mixed systems containing alicyclic and aromatic rings, and aliphatic chains, etc. Examples may be found in the *CRC Handbook* [63, Section C]. Physical properties of selected polynuclear aromatic compounds are given in [49, p. 967].

Although many of the aromatic compounds based on benzene have pleasant odors, they are usually toxic, and some are carcinogenic. Volatile aromatic hydrocarbons are highly flammable and burn with a luminous, sooty flame. The effects of molecular size (in simple arenes as well as in substituted aromatics) and of molecular symmetry (e.g., xylene isomers) are noticeable in physical properties [48, p. 212; 49, p. 375; 50, p. 41]. Since the hybrid bonds of benzene rings are as stable as the single bonds in alkanes, aromatic compounds can participate in chemical reactions without disrupting the ring structure.

Aromatic groups are called *aryl* groups if they are attached directly to a parent structure. Common aryl groups are


[phenyl or C_6H_5—]



[benzyl or $C_6H_5CH_2$—]

Organic Compounds of Nonhydrocarbon Type

There are numerous families of organic compounds, with structures analogous to hydrocarbons, that contain other atoms (e.g., O, N, S, Cl) besides C and H. Classification is done in accordance with the structural theory on the basis of functional groups present. The atom or atomic grouping that characterizes a particular family and also determines the properties of its members is called a *functional group*. Table 2-42 contains a selected list of common functional groups and examples of

Table 2-42
Selected Functional Groups and Representative Organic Compounds [58]

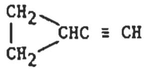
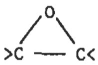
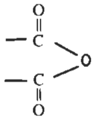
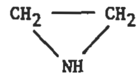
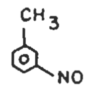
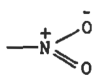
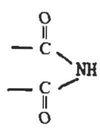
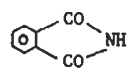
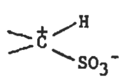
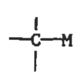
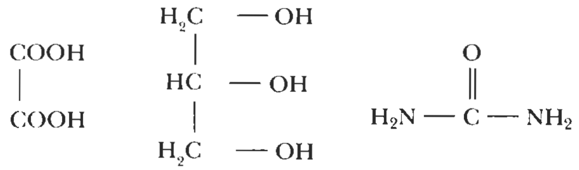
Class	Functional Group	Example: Molecular Formula	Example: Compound Name
Alkene	>C=C<	$CH_3CH = CH_2$	Propylene
Alkyne	-C≡C-		Cyclopropylacetylene
Alcohol	-OH	$CH_3CH_2\overset{\text{OH}}{\overset{\vert}{\text{C}}}HCH_3$	sec-Butyl alcohol
Ether	-O-	$CH_3OC_6H_5$	Methylphenyl ether
Epoxide (Oxirane)			Ethylene oxide
Peroxide	-O-O-	$(CH_3)_3COOC(CH_3)_3$	Di-tert-butyl peroxide
Aldehyde	$-\overset{\text{O}}{\overset{\Vert}{\text{C}}}-\text{H}$	C_6H_5CHO	Benzaldehyde
Ketone	$-\overset{\text{O}}{\overset{\Vert}{\text{C}}}-$	CH_3COCH_3	Acetone (Methyl ketone)
Carboxylic acid	$-\overset{\text{O}}{\overset{\Vert}{\text{C}}}-\text{OH}$	C_6H_5COOH	Benzoic acid
Ester	$-\overset{\text{O}}{\overset{\Vert}{\text{C}}}-\text{O}-$	$CH_3COOCH_2CH_3$	Ethyl acetate
Acid anhydride		$(CH_3CO)_2O$	Acetic anhydride
Halide	-X	$CHCl_3$	Trichloromethane (chloroform)
Acid halide	$-\overset{\text{O}}{\overset{\Vert}{\text{C}}}-\text{X}$	C_6H_5COCl	Benzoyl chloride
Amine	$-NH_2$	$(CH_3)_3N$	Trimethylamine

Table 2-42
(continued)

Class	Functional Group	Example: Molecular Formula	Example: Compound Name
Imine	>C=NH		Ethylene imine
Nitrile	-C≡N	C_6H_5CN	Benzonitrile
Nitroso	-N=O		meta-Nitrosotoluene
Nitro		CH_2ONO_2 / $CHONO_2$ / CH_2ONO_2	Nitroglycerine
Amide	$-\overset{O}{\overset{\Vert}{C}}-NH_2$	$CONH_2$ / $CONH_2$	Oxamide
Imide			Phthalimide
Isocyanate	-N=C=O	$CH_3(CH_2)_4NCO$	n-Amyl isocyanate
Oxime	>C=N-OH	C_6H_5CHNOH	Benzaldoxime
Mercaptan (Thiol)	-SH	C_6H_5SH	Phenyl mercaptan (Thiophenol)
Sulfide (Thioether)	-S-	$CH_3SCH_2CH_3$	Methylethyl sulfide
Disulfide	-S-S-	$CH_3CH_2SSCH_2CH_3$	Ethyldisulfide
Sulfonate (Salt of sulfonic acid)		$C_{12}H_{25}SO_3Na$	Sodium dodecyl sulfonate (Sodium salt of dodecylsulfonic acid)
Organo-metallic		$(CH_3)_4Sn$	Tetramethyltin
	R^-M^+	$\overset{-}{R}\ \overset{++}{Mg}\ \overset{-}{Br}$	Alkylmagnesium bromide

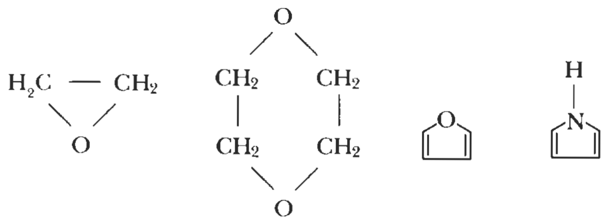
organic compounds containing them. Compounds containing the same functional group form a homologous series and show gradual variations in physical properties as the molecular size increases. An aliphatic group (—R, e.g., alkyl representing methyl, ethyl, isopropyl, etc.), an alicyclic group (e.g., cyclopropyl, cyclobutyl, etc.), or an aromatic group (-ϕ, e.g., aryl representing phenyl, benzyl, etc.), or their combinations may be attached to these functional groups, depending upon the valences to be satisfied.

When compounds contain more than one functional group in their structures, they are referred to as *polyfunctional* compounds. Examples include



(oxalic acid) (glycerol) (urea)

Unlike carbocyclic (or homocyclic) hydrocarbons which have been discussed previously, *heterocyclic* compounds contain in their rings other atoms, such as nitrogen, oxygen, or sulfur, in addition to carbon. Some examples are shown below:



(ethylene oxide) (1,4-dioxane) (furan) (pyrrole)

(thiophene) (pyridine)

Other structures may be found in Reference 63. Physical properties of a limited number of heterocyclic compounds are given in Reference 49, p. 1003.

Isomers and Isomerism

Isomerism is commonly encountered in covalent compounds but is rare among ionic compounds. Isomers can be grouped under two major categories, namely structural isomers and stereoisomers [48, p. 45].

Structural Isomers. These contain different combinations of bonded pairs of atoms. They may be divided into three types: chain, position, and functional isomers, depending on the specific cause of isomerism. *Chain* isomers occur in linear and branched-chain compounds, e.g.,

C_5H_{12} exists as n-pentane, isopentane, and neopentane.

Position isomers differ in the positions of some reference atom, group or bond, e.g., $CH_3CH_2CH_2OH$ (n-propanol) and $CH_3CHOHCH_3$ (isopropanol) are isomers that differ in the location of the -OH group.

$CH_2ClCH_2CH_2CH_2CH_3$	$CH_3CHClCH_2CH_2CH_3$	$CH_3CH_2CHClCH_2CH_3$
(1-chloropentane)	(2-chloropentane)	(3-chloropentane)

are isomers that differ in the position of the attached chlorine atom.

Functional isomers differ in the functional groups present, e.g., C_2H_6O can exist as an alcohol, CH_3CH_2-OH (ethanol) or as an ether, CH_3-O-CH_3 (methylether or dimethylether).

C_3H_6O can exist as an aldehyde, CH_3CH_2-CHO (propionaldehyde) and as a ketone, CH_3-CO-CH_3 (acetone).

Stereoisomers. These contain the same combinations of bonded pairs of atoms, but their orientations in space will be different. There are two types of stereoisomers, geometric and optical. *Geometric* isomers differ in their spatial structures about some reference plane through the molecule. The cis- and trans- forms of 2-butene are examples of geometric isomers, which arise as a consequence of hindered rotation about the double bond. *Optical* isomers are mirror images that occur in asymmetric molecules in the absence of a plane of symmetry. An asymmetric carbon atom is one to which four different atoms or groups are attached and therefore has no point or plane of symmetry. A pair of optical isomers exist for a molecule with an asymmetric center and are referred to as dextro-rotatory (or D-) and laevo-rotatory (or L-) isomers. They have identical physical and chemical properties except for the sign of their optical activity to polarized light. With the observer facing the emerging beam, a dextro-isomer rotates the plane of polarization to the right or clockwise (taken as positive), while a laevo-isomer rotates the plane counterclockwise (negative), e.g.,

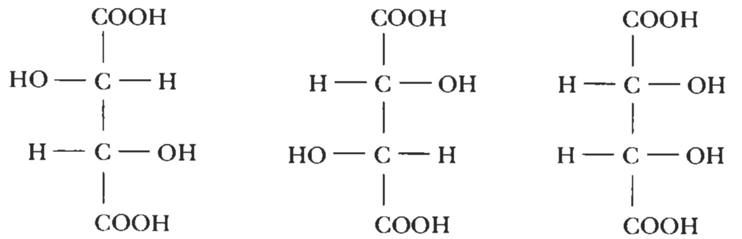
Glyceraldehyde has two optically active isomers, namely,

$$\begin{array}{c} CHO \\ | \\ H - C - OH \\ | \\ CH_2OH \end{array} \quad \text{and} \quad \begin{array}{c} CHO \\ | \\ HO - C - H \\ | \\ CH_2OH \end{array}$$

[(+) D-glyceraldehyde] [(−) L-glyceraldehyde]

The central C atom has two different substituent groups, —H and —OH.

Molecules that contain two or more asymmetric centers exist in more than two stereoisomeric forms. Some are pairs of optically active isomers; others may be symmetric and therefore optically inactive. An example is tartaric acid with two asymmetric central carbon atoms. It has three isomers, two of which are optically active and one inactive,



[(−) tartaric acid] [(+) tartaric acid] [meso-tartaric acid (optically inactive)]

Optical isomers are frequently found in sugars, amino acids, and other biologically important molecules.

Chemical Composition of Crude Oils

Methods of characterizing a crude oil include, among other techniques, the ultimate analysis for elemental composition (see Table 2-38) and the classification, based on a standard distillation procedure, into various boiling fractions and residuum (see Table 2-39). Although the exact chemical constitution of natural petroleum will never be known, pioneering research efforts begun by F. D. Rossini in 1925 under the sponsorship of the American Petroleum Institute (API) and continued by others have yielded certain important results [55,57]. Some of the key findings of these studies on selected crude oils or oil fractions are summarized below. As a supplement to the ensuing discussion, selected examples of the types of compounds present in petroleum are illustrated in Tables 2-43 and 2-44.

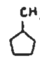
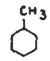
The principal classes of hydrocarbons found in crude oils are paraffins, cycloparaffins (naphthenes), and aromatics. The nonhydrocarbon compounds include those containing N, S, and/or O atoms, and metallic compounds. The molecular weight distribution of a crude oil ranges from 16 for methane to several thousands for asphaltenes. For a given crude oil, the elements are not uniformly distributed over the entire boiling range. The lower boiling point fractions are dominated by saturated hydrocarbons (namely, paraffins and napthenes), whereas the higher boiling fractions and residuum contain large proportions of aromatics and N/O compounds. The sulfur content tends to increase with rising boiling point.

Paraffins (Alkanes). These saturated aliphatic hydrocarbons include normal alkanes as well as branched alkanes (isoalkanes), represented by the formula C_nH_{2n+2}. The paraffin content can vary widely from one crude oil to another. n-alkanes have been found throughout the boiling range of most crude oils, from n = 1 to 78. Some waxy crudes are known to contain higher alkanes, even up to n = 200. The normal paraffins are inert to strong acids, bases and oxidizing agents. The pour point of a crude oil is strongly influenced by the amount and carbon-number distribution of n-alkanes present. For crude oils, it can range from −70 to 110°F (−57 to 43°C). Whereas n-alkanes tend to raise the pour point, other hydrocarbon types lower it.

Identification becomes more difficult in the case of isoparaffins since many structural isomers are possible with increasing carbon number. Nevertheless, all possible isoalkanes from C_4 through C_8 have been found in crude oils, along with several isomers of C_9 and some of C_{10}. Many isoprenoids (e.g., pristane and phytane) which serve as biomarkers to the genesis of petroleum have been detected in significant

Table 2-43
Selected Examples of Cyclic Hydrocarbons Found in Crude Oils (MW = Molecular Weight, NBP = Boiling Point at 760 mm Hg, MP = Melting Point) [Source of Data: Reference 63]

Naphthenes

			
	Methylcyclopentane (C_6H_{12})	Methylcyclohexane (C_7H_{14})	Cycloheptane (C_7H_{14})
MW =	84.2	98.2	98.2
NBP(°C) =	71.8	100.9	118.5
MP(°C) =	−142.4	−126.6	−12

Aromatics

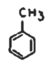
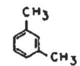
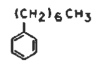
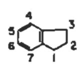
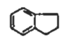
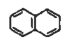
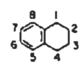
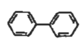
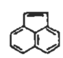
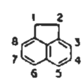
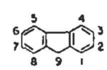
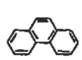
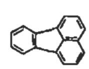
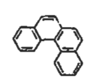
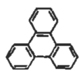
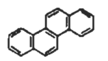
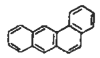
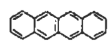
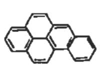
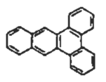
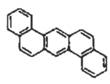
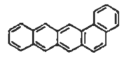
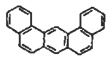
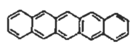
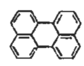
				
	Benzene (C_6H_6)	Toluene (Methylbenzene) (C_7H_8)	m-xylene (1,3-dimethylbenzene) (C_8H_{10})	n-heptylbenzene ($C_{13}H_{20}$)
MW =	78.1	92.1	106.2	176.3
NBP(°C) =	80.1	110.6	139.1	265
MP(°C) =	5.5	−95	−47.9	−37
				
	Indene (Indanaphthene) (C_9H_8)	Indan (2,3-dihydroindene) (C_9H_{10})	Naphthalene ($C_{10}H_8$)	Tetralin (1,2,3,4-Tetrahydronaphthalene) ($C_{10}H_{12}$)
MW =	116.2	118.2	128.2	132.2
NBP(°C) =	182.6	178	218	207.6
MP(°C) =	−1.7	−51.4	80.6	−35.8
				
	Biphenyl (Diphenyl) (Phenylbenzene) ($C_{12}H_{10}$)	Acenaphthylene ($C_{12}H_8$)	Acenaphthene ($C_{12}H_{10}$)	Fluorene ($C_{13}H_{10}$)
MW =	154.2	152.2	154.2	166.2
NBP(°C) =	256	265 - 275	279	294
MP(°C)	71	92.5	96.2	116.5
				
	Phenanthrene ($C_{14}H_{10}$)	Fluoranthene (1,2-Benzacenaphthene) ($C_{16}H_{10}$)	Pyrene (Benzo [d,e,f] phenanthrene) ($C_{16}H_{10}$)	3,4-Benzophenanthrene (Benzo [c] phenanthrene) ($C_{18}H_{12}$)
MW =	178.2	202.3	202.3	228.3
NBP(°C) =	340	375	393	*
MP(°C) =	101	111	156	68

**Table 2-43
(continued)**

				
	Triphenylene (9,10-Benzophenanthrene)	Chrysene (1,2-Benzophenanthrene) (Benzo [a] phenanthrene)	1,2-Benzanthracene (Benz [a] anthracene) (2,3-Benzophenanthrene) (Tetraphene)	Naphthacene (2,3-Benzanthracene) (Tetracene)
	($C_{18}H_{12}$)	($C_{18}H_{12}$)	($C_{18}H_{12}$)	($C_{18}H_{12}$)
MW =	228.3	228.3	228.3	228.3
NBP(°C) =	425	448	435	* (Sublimes)
MP(°C) =	199	255–256	165	357

				
	3,4-Benzopyrene	1,2:3,4-Dibenzanthracene (Dibenz [a,c] anthracene)	1,2:5,6-Dibenzanthracene (Dibenz [a,h] anthracene)	1,2:6,7-Dibenzanthracene (1,2-Benzonaphthacene)
	($C_{20}H_{12}$)	($C_{22}H_{14}$)	($C_{22}H_{14}$)	($C_{22}H_{14}$)
MW =	252.3	278.4	278.4	278.4
NBP(°C) =	*	*	*	*
MP(°C) =	176.5–177.5	205	269–270	263–264

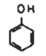
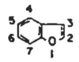
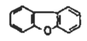
			
	1,2:7,8-Dibenzanthracene (Dibenz [a,j] anthracene)	Pentacene (2,3:6,7-Dibenzanthracene) (Benzo [b] naphthacene)	Perylene
	($C_{22}H_{14}$)	($C_{22}H_{14}$)	($C_{20}H_{12}$)
MW =	278.4	278.4	252.3
NBP (°C) =	*	290-300 (Sublimes)	350-400 (Sublimes)
MP (°C) =	197–198	270–271	277–279

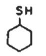
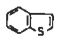
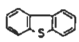
concentrations. 2- and 3-methyl alkanes and pristane appear to be the dominant isoparaffins in crude oils.

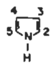
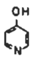
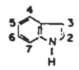
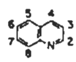
Naphthenes (Cycloparaffins). Cycloalkanes occur in varying amounts in crude oils. The rings are made up of five, six, or seven carbon atoms. Alkyl derivatives of cyclopentane and cyclohexane have been found, but not of cycloheptane. The most common naphthenes are methyl-, and dimethyl-substituted cyclopentane and cyclohexane. The amount of naphthenes can be in excess of 50 wt% of a crude oil, with the lighter boiling fractions containing less and the heavier fractions containing more. Fused polycyclic aliphatic structures such as decalin ($C_{10}H_{18}$) become prevalent in the heavier fractions. Some of these multiring assemblies may contain up to seven rings in a molecule.

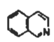
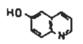
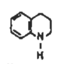
Alkenes (Olefins) and Cycloalkenes. Olefins are unsaturated and highly reactive. Because of their propensity to be reduced to paraffins with hydrogen or to merceptans with hydrogen sulfide, alkenes are found in petroleum only in very low concentrations. Single-ring cycloalkenes have the same formula, C_nH_{2n}, as olefins and so are difficult to identify. Trace concentrations of multiring cycloalkenes such as hopenes and sterenes have been reported to be present in crude oils. Alkynes and cycloalkynes are not commonly found in natural petroleum.

Table 2-44
Selected Examples of NSO Compound Types Found in Crude Oils (MW = Molecular Weight, NBP = Boiling Point at 760 mm Hg, MP = Melting Point) [Source of Data: Reference 63]

		$CH_3(CH_2)_{16}COOH$			
	Phenol (C_6H_6O)	Stearic acid ($C_{18}H_{36}O_2$)	Furan (C_4H_4O)	Benzofuran (C_8H_6O)	Dibenzofuran ($C_{12}H_8O$)
MW =	94.1	284.5	68.1	118.1	168.2
NBP(°C) =	181.8	360 (decomposes)	31.4	174	287
MP(°C) =	43	71.5 – 72.0	-85.7	<-18	86–87

					
	Cyclopentanethiol ($C_5H_{10}S$)	Cyclohexanethiol ($C_6H_{12}S$)	Thiophene (Thiofuran) (C_4H_4S)	Benzothiophene (Thionophthene) (C_8H_6S)	Dibenzothiophene (Diphenylene sulfide) ($C_{12}H_8S$)
MW =	102.2	116.2	84.1	134.2	184.3
NBP(°C) =	131–132	158–160	84.2	221	332–333
MP(°C) =	*	*	-38.3	32	99–100

					
	Pyrrole (Azole) (C_4H_5N)	Pyridine (Azine) (C_5H_5N)	4-Hydroxypyridine (4-Pyridol) (C_5H_5NO)	Indole (1-Benzo [b] pyrrole) (C_8H_7N)	Quinoline (Benzo [b] pyridine) (C_9H_7N)
MW =	67.1	79.1	95.1	117.2	129.2
NBP(°C) =	130–131	115.5	>350	254	238.1
MP(°C) =	*	-42	148.5	52.5	-15.6

				
	Isoquinoline (Benzo [c] pyridine) (C_9H_7N)	6-Hydroxyquinoline (C_9H_7NO)	1,2,3,4-Tetrahydroquinoline ($C_9H_{11}N$)	5,6,7,8-Tetrahydroquinoline ($C_9H_{11}N$)
MW =	129.2	145.2	133.2	133.2
NBP(°C) =	243.2 at 743 mm Hg	>360	251	222
MP(°C) =	26.5	193	20	*

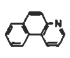
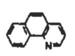
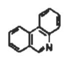
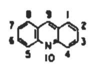
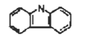
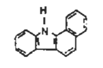
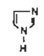
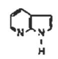
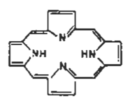
				
	5,6-Benzoquinoline (Benzo [f] quinoline) ($C_{13}H_9N$)	7,8-Benzoquinoline (Benzo [h] quinoline) ($C_{13}H_9N$)	Phenanthridine (3,4-Benzoquinoline) ($C_{13}H_9N$)	Acridine (2,3:5,6-Dibenzopyridine) ($C_{13}H_9N$)
MW =	179.2	179.2	179.2	179.2
NBP(°C) =	350 at 721 mm Hg	338 at 719 mm Hg	349 at 769 mm Hg	345–346
MP(°C) =	94	52	106–107	111

Table 2-44
(continued)

				
	Carbazole (Dibenzopyrrole)	1,2-Benzocarbazole	Imidazole	Pyridazine
	($C_{12}H_9N$)	($C_{16}H_{11}N$)	($C_3H_4N_2$)	($C_4H_4N_2$)
MW =	167.2	217.3	68.1	80.1
NBP (°C) =	355	*	256	207–208
MP (°C) =	247–248	*	89–91	*

		
	7-Azaindole	Porphin (Tetramethenetetrapyrrole)
	($C_7H_6N_2$)	($C_{20}H_{14}N_4$)
MW =	118.1	310.4
NBP (°C) =	*	300 at 12 mm Hg (sublimes)
MP (°C) =	105–107	darkens at 360

Aromatic Hydrocarbons. Nearly all known types of aromatic hydrocarbons have been found in crude oils. The strong odor of crude oil is not imparted by arenes but is due to nonhydrocarbon (NSO) compounds. Benzene and alkylbenzenes, ranging from methyl through decyl groups, have been found in liquid petroleum, together with several C_{11}-alkyl isomers. Indan and tetrahydronaphthalenes as well as some of their methyl isomers have been identified. Biphenyl and its derivatives occur in lower concentrations than naphthalene and its derivatives. Several polynuclear aromatics including phenanthrene, fluoranthene, pyrene, benz[a]anthracene, chrysene, triphenylene, benzopyrenes, perylene, etc., and some of their alkyl derivatives have been detected, but not anthracene. Crude oils were also found to contain aromatic compounds such as acenaphthene, acenaphthylene, flourene, dibenzanthracenes, etc. Among the simpler aromatic molecules, toluene and meta-xylene are the most common; benzene, ethylbenzene, and other alkylbenzenes also occur in significant concentrations in the distillates.

In describing high molecular weight compounds, it is a common practice in petroleum chemistry to regard any compound as aromatic if it contains an aromatic ring. Thus, the composition of the monoaromatic (benzoid) concentrate from crude oils is primarily alkylcycloalkylbenzenes, having one to nine cycloparaffin rings, together with small amounts of alkylbenzenes. The diaromatic (naphthenoid) fraction is largely composed of alkylcycloalkylnaphthalenes, containing one to eight naphthenic rings, and alkylacenaphthylenes, and a minor proportion of alkylnaphthalenes.

Aromatic content can vary considerably between crudes but rarely exceeds 15% of the total crude weight. The aromatic hydrocarbons appear throughout the boiling range but tend to be concentrated in the heavy fractions of petroleum, including the residuum. As a class, aromatics (e.g., toluene and xylenes) have the highest octane ratings among hydrocarbons and, hence, are used as additives to gasoline and other fuel oils. They show the largest viscosity changes with temperature and are, therefore, undesirable in the lubricating oil range. With rising boiling point, the heavy fractions contain increasing amounts of complex polycyclic aromatic compounds that are difficult to characterize. Some polynuclear aromatic molecules such as 3,4-benzopyrene and benz[a]anthracene are regarded as procarcinogens.

Nitrogen, Sulfur and Oxygen Compounds. These are usually abbreviated as *NSO compounds* and sometimes referred to as *asphaltics.* Although present in small amounts, the N, S, and O atoms contribute greatly to the nonhydrocarbon fraction of a crude oil by their incorporation into hydrocarbon molecules. The residuum contains a high percentage of NSO compounds.

Sulfur Compounds. All crude oils contain sulfur in one of several forms including elemental sulfur, hydrogen sulfide, carbonyl sulfide (COS), and in aliphatic and aromatic compounds. The amount of sulfur-containing compounds increases progressively with an increase in the boiling point of the fraction. A majority of these compounds have one sulfur atom per molecule, but certain aromatic and polynuclear aromatic molecules found in low concentrations in crude oil contain two and even three sulfur atoms. Identification of the individual sulfur compounds in the heavy fractions poses a considerable challenge to the analytical chemist.

Alkyl thiols (mercaptans) with normal or branched alkyl groups and with the thiol group in a primary, secondary, or tertiary location have been found in petroleum, together with cycloalkyl thiols, having rings of five or six carbon atoms. Continuous chain or branched alkyl sulfides and cyclic sulfides with four or five carbon atoms in their rings have been detected. Mixed alkyl cycloalkyl sulfides have also been found. Alkylpolycyclic sulfides containing one to eight cycloparaffin rings were identified in certain crudes. Aromatic compounds of sulfur include thiophenes, their benzo- and dibenzo- derivatives, and benzonaphthothiophenes. Thioindans and alkylaryl sulfides are also present.

In general, mercaptans are more malodorous than sulfides and hydrogen sulfide. The presence of significant amounts of sulfur can induce catalyst poisoning during the refining of crude oil.

Nitrogen Compounds. Most crude oils contain nitrogen; a large proportion of it occurs in the high boiling fractions and in the residuum. Examples of the nitrogen compounds present in petroleum include mono-, di-, and tri-alkylpyridines, quinoline and alkyl substituted quinolines, tetrahydroquinolines and dialkylbenz[h]quinolines. Carbazole and methyl- through decyl- substituted carbazoles have also been identified. The high boiling fractions from one crude oil contained a variety of nitrogen compound types (in excess of 0.1 wt% concentration) that included indoles, carbazoles, benzcarbazoles, pyridines, quinolines, and phenanthridines. N/O compounds such as amides, hydroxypyridines and hydroxyquinolines, as well as compounds containing two nitrogens such as azaindoles and azacarbazoles, were also found. Other molecular types including pyrroles, isoquinolines, benzoquinolines, and benzologues of acridine may be present in crude oil. Porphyrins are observed in the residuum, usually in association with metals.

Certain aromatic, nitrogen compounds (e.g., pyridines and quinolines) are basic and can cause coking on acid catalysts during petroleum processing.

Oxygen Compounds. Most crude oils contain only small amounts of oxygen. Oxygen compounds are mainly carboxylic acids, including straight-chain fatty acids, branched-chain acids, naphthenic acids, and dicarboxylic acids. Other molecular types observed in the higher boiling fractions include furans and their benzo-, dibenzo-, and benzonaphtho- derivatives. Oxygen may also be present in the form of phenols, alcohols, esters, and ketones and in combination with nitrogen.

Residuum. This is the undistilled fraction remaining at the end of distillation, which corresponds to an upper limit of ~565°C (~1050°F) at atmospheric pressure, or up to

~675°C (~1250°F) under vacuum. The residuum amounts to a small percentage of a very light crude oil and up to 30–40 wt% of a heavy crude. Its major constituents are resins, asphaltenes, and some high molecular weight oils and waxes. The residuum accounts for most of the total NSO content and the heavy metals. The resins and asphaltenes precipitate out when the residuum (or crude oil) is treated with liquid propane below 70°F. Additional treatment of this precipitate with n-pentane separates the soluble resins from the insoluble asphaltenes. The amount of resins always exceeds the asphaltene content of a crude oil. Resins are light to dark colored and range from thick viscous materials to amorphous solids. Asphaltenes appear as dark brown to black, amorphous solids. Together, they may possess nearly 50% of the total nitrogen and sulfur in the crude oil, predominantly in the form of heterocyclic condensed ring structures containing aromatic and cycloparaffinic rings. Asphaltenes may account for as much as 25 wt% of the residuum (up to 12% of the crude oil). Colorless oils are the most paraffinic, while asphaltenes are the most aromatic. Dark oils and resins show similar degrees of paraffinicity and aromaticity. Up to 40 wt% of saturated hydrocarbons may be present in the residuum; however, this comprises only 1–3 wt% of the total crude. The rest are aromatic and N/O-containing compounds. In the nonasphaltene fraction of the residuum, the typical aromatic structure is a highly substituted, condensed polynuclear aromatic molecule, with an average formula, $C_{100}H_{160}S$. The substituents are fused naphthenic rings, which in turn are substituted with long ($C_{15}-C_{20}$) alkyl side chains, having intermittent methyl branches. The average structure for a N/O compound is similar in features excepting for slightly higher aromaticity and shorter ($C_{10}-C_{15}$) alkyl side chains. Other types of NSO compounds described previously may also be present.

In the asphaltene fraction, pure hydrocarbons become rare at molecular weights above 800, and polar functional groups become very common. Asphaltenes exist as a dispersion of colloidal particles in an oily medium. Their molecular weight distribution can extend up to 200,000 or more. A typical asphaltene molecule has 10–20 condensed aromatic and naphthenic rings, with both alkyl and cycloalkyl substituted side chains. Some of these structures contain free-radical sites that allow complex formation with vanadium, nickel, etc. The observed polar functional groups in the high molecular weight compounds include carboxylic acids, amides, phenols, carbazoles and pyridine benzologs.

Metals. The metals present in crude oils usually exist as complexes of cyclic organic molecules called porphyrins. The parent structure of the porphyrins is tetramethenetetrapyrrole ($C_{20}H_4N_4$), also known as porphin. In a metalloporphyrin, the transition metal atom is held at the center of the porphyrin ring by coordination with the four pyrrole N atoms. Inorganic compounds of metals are probably not related to the genesis of petroleum. Nickel and vanadium are present in petroleum in concentrations of less than 1 ppb to ~1000 ppm, in combined form with porphyrins. Some of the lighter metalloporphyrins are volatile, while the high molecular weight porphyrins appear in the nonvolatile residuum. When oil deposits occur together with saline water, the produced oil-water emulsions contain soluble salts of sodium, calcium, and magnesium. Other metals may be present in the inorganic state as suspended solids in the produced fluids along with any clays or mineral matter derived from the rock matrix and piping.

Figure 2-77 shows how the weight distributions of the different molecular types vary during the fractional distillation of a naphthenic crude oil. Saturated aliphatic hydrocarbons (i.e., paraffins and naphthenes) are the predominant constituents in the light gasoline fraction. As the boiling point is raised, the paraffin content decreases, and the NSO content increases continuously. About 75 wt% of the residuum is composed of aromatics and NSO compounds.

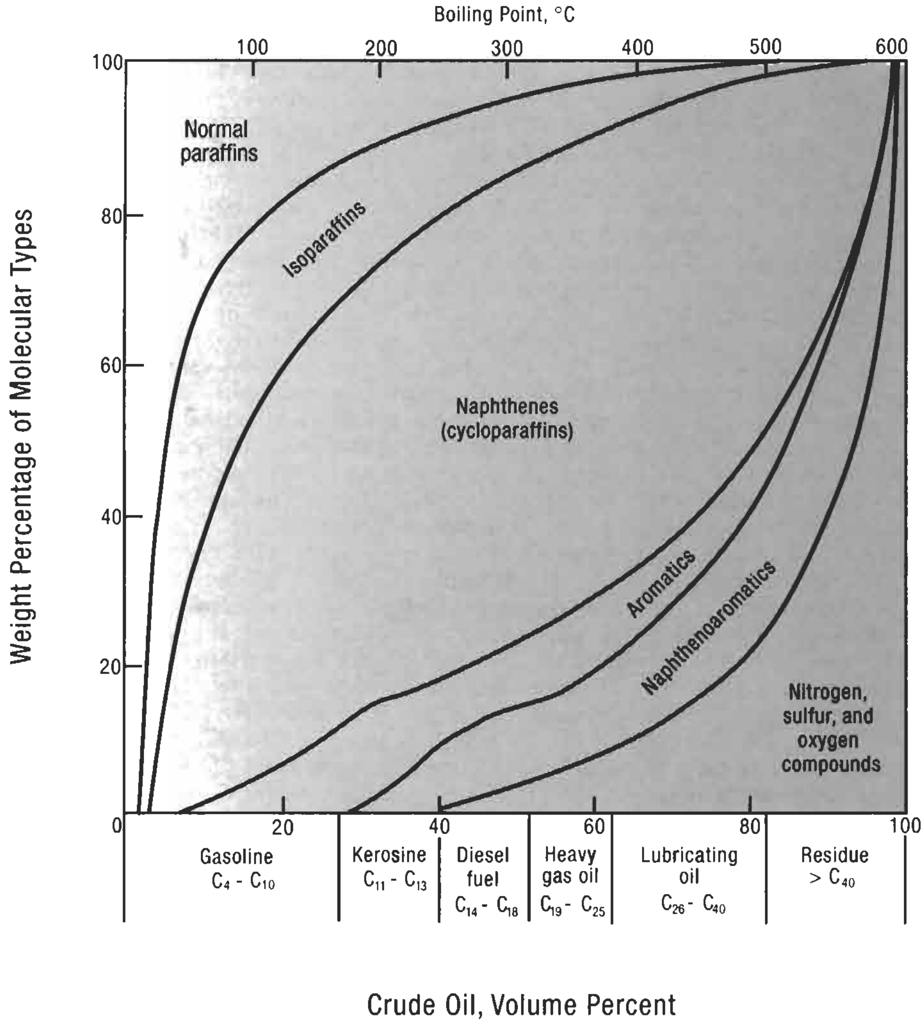


Figure 2-77. Chemical composition of a naphthenic crude oil [57].

Properties of Liquid Hydrocarbon Fuels

Certain properties of a liquid fuel are measured routinely in a laboratory for characterization purposes. Besides density and viscosity, these properties include the pour point, the cloud point, and the flash point. Standard ASTM (American Society for Testing Materials) procedures are available for their determination.

The *pour point* represents the lowest temperature at which the liquid fuel will pour. This is a useful consideration in the transport of fuels through pipelines. To determine the pour point, an oil sample contained in a test tube is heated up to 115°F (46°C) until the paraffin waxes have melted. The tube is then cooled in a bath kept at about 20°F (11°C) below the estimated pour point. The temperature at which the oil does not flow when the tube is horizontally positioned is termed the pour point.

The aniline *cloud point* is a measure of the paraffinicity of a fuel oil. A high value denotes a highly paraffinic oil while a low value indicates an aromatic, a naphthenic, or a highly cracked oil. The *flash point* represents the temperature to which a liquid fuel can be heated before a flash appears on its surface upon exposure to a test flame under specified conditions. A knowledge of the flash point is needed to ensure safe handling and storage without fire hazards.

API gravity. The specific gravity of petroleum or petroleum products is often expressed in terms of "degrees API" on a scale defined by

$$^{\circ}\text{API} = \frac{141.5}{G} - 131.5$$

where G is the specific gravity of the liquid at 60°F, with reference to water at 60°F. Thus, an API gravity of 10° corresponds to G = 1.

U.O.P. Characterization Factor (K). This is used as a qualitative index of the paraffinicity of an oil stock. By definition, $K = (T_B)^{1/3}/G$, where T_B is the average boiling point (°R) at one standard atmosphere, and G is the specific gravity at 60°F. The U.O.P. characterization factor has been found to vary within a homologous series; it is therefore not a true measure of the hydrocarbon type. Nevertheless, graphical correlations have been developed between K, API gravity, molecular weights, T_B, and critical temperatures for a number of petroleum fractions [Refer to K. M. Watson and E. F. Nelson, Ind. Eng. Chem., *25*, p. 880 (1933)]. Characterization factors have also been correlated with kinematic viscosities, API gravity, and T_B.

Physical Chemistry

Basic Definitions

The *formula weight* of an element (or a compound or a species) is obtained as the sum of the weight contributions from the constituent atoms making up its chemical formula. The formula weight of an element is its atomic weight and that of a compound is its molecular weight.

The *equivalent weight* of an ion (or an element) is the ratio of its formula weight to its valence. According to an alternative definition that is also suitable for compounds, an equivalent weight represents the amount of a substance which will react with one atomic weight of hydrogen or its chemical equivalent.

One *gram-atom* (or pound-atom) is the mass in grams (or pounds) of a given element that is numerically equal to its atomic weight. Thus, the number of gram-atoms of an elementary substance is m/A, where m is the mass (in grams) and A, its atomic weight.

One *gram-mole* (or pound-mole) is the mass in grams (or pounds) of a given compound that is numerically equal to its molecular weight (M). Thus, the number of pound-moles of a compound contained in a mass of m pounds is m/M.

One *gram-equivalent* (or pound-equivalent) represents the mass in grams (or pounds) of a material that is numerically equal to its equivalent weight.

Avogadro's number represents the number of atoms (or molecules) present in one gram-atom (or gram-mole) of any elementary substance (or compound). It has a value of 6.02205×10^{23} [63].

The *density* (ρ) of a substance is defined as its mass per unit volume, expressed as g/cm^3, lb/ft^3, etc. The *specific volume* ($\hat{V}$) of a substance is the reciprocal of its density and is expressed as cm^3/g, ft^3/lb, etc.

The *specific gravity* (G) of a substance is the ratio of the density (ρ) of the substance to the density (ρ_{ref}) of a reference substance at specified conditions. That is,

$$G(T, P/T_{ref}, P_{ref}) = \frac{\rho @ T, P}{\rho_{ref} @ T_{ref}, P_{ref}}$$

For solids and liquids, the density is a weak function of pressure and, therefore, the temperatures T and T_{ref} are usually stated. Also, the reference substance is commonly taken as water at 4°C at which $\rho_{ref} = 1.000$ g/cm^3 = 62.43 lb/ft^3. If a single temperature is stated, it implies that both densities have been measured at that temperature.

In the case of gases and vapors, it is imperative that the temperature as well as pressures be clearly indicated in referring to specific gravity. Air is normally chosen as the reference fluid for gaseous substances.

Compositions of Multicomponent Systems

The relative amounts of the individual components (or species) making up a mixture or solution can be expressed in a variety of ways, depending upon the system at hand. A volumetric, mass, or molar basis may be employed to represent the compositions of multicomponent systems.

Volumetric Basis. The volume fraction (v_i) of component i in a mixture is the fraction of the total volume (V) of the mixture that is attributable to that component, at the stated temperature and pressure. Thus, if V_i denotes the actual volume of component i present in a total volume V of the mixture, then

$$v_i = \frac{V_i}{V} \qquad \sum v_i = 1$$

and the volume percent of i = $100v_i$. Instead, if V represents the volume of one mole of the mixture, then V_i must be interpreted as the "partial molar volume of component i" in the mixture.

The above method is commonly used for gases and infrequently for liquid mixtures. At atmospheric conditions when ideal gas behavior is realized, the total volume of the mixture equals the sum of the pure-component volumes (V_i^0). That is, $V = \sum V_i^0$ and $V_i = V_i^0$. In such a case, V_i^0 can be obtained directly from the ideal gas law, without recourse to measurement, and hence, the volumetric composition can be readily computed. On the other hand, in non-ideal (i.e., real) mixtures and solutions, $V \neq \sum V_i^0$ and the measurement of the actual component volumes V_i becomes a difficult undertaking. In these systems, a volume change—either shrinkage or expansion—is experienced upon mixing of the components. In addition, thermal effects may accompany the formation of the mixture or solution. The volumetric composition of liquid mixtures is expected to vary with temperature owing to the density dependence on temperature.

When a gas mixture contains water vapor, the volumetric analysis is stated either on a *wet* basis that includes the water vapor or on a *dry* basis (moisture-free basis) that excludes the water vapor.

Weight (or Mass) Basis. The terms "weight fraction" and "weight percent" are often used as synonymous for "mass fraction" and "mass percent," respectively. If the total mass (m) of a mixture includes an amount m_i of component i, then the mass (or weight) fraction of that component is given by

$$\omega_i = \frac{m_i}{m}$$

and the mass (or weight) percent of i = $100\omega_i$. Note that $\sum m_i = m$ and $\sum \omega_i = 1$.

This representation is ordinarily used for solid and for liquid systems and rarely for gases. In the absence of chemical reactions the mass composition of an isolated system remains unchanged. Any consistent set of units for mass (or volume) may be selected in interpreting the compositions expressed on a mass (or volumetric) basis.

Molar Basis. The mole fraction (x_i) of a component i in a multicomponent mixture is defined as

$$x_i = \frac{\text{Number of moles of i}}{\text{Number of moles of mixture}} = \frac{n_i}{n} = \frac{n_i}{\sum n_i}$$

The mole percent of i = $100x_i$ and $\sum x_i = 1$.

The volume fractions and mole fractions become identical in ideal gas mixtures at fixed conditions of pressure and temperature. In an isolated, nonreactive system, the molar composition does not vary with temperature.

Mixture Properties

The *average molecular weight* (M) of a homogeneous mixture is its total mass (m) divided by the total number of moles (n) of its components. If x_i represents the mole fraction of the i^{th} component whose molecular weight is M_i, then

$$M = \frac{m}{n} = \sum x_i M_i$$

In terms of mass fractions,

$$\frac{1}{M} = \sum\left(\frac{\omega_i}{M_i}\right)$$

The *mass density* (ρ) of the mixture is the ratio of its total mass (m) to its total volume (V), whereas the *molar density* (c) of the mixture is defined as the ratio of the total number of moles (n) to the total volume of the mixture. Thus,

$$\rho = \frac{m}{V} \qquad \text{and} \qquad c = \frac{n}{V}$$

and both are related by $\rho/c = m/n = M$.

Mass and Molar Concentrations of Components

The *mass concentration* (ρ_i) of the i^{th} component (or species) in a homogeneous mixture is defined as the mass of that component present per unit volume of the mixture. The *molar concentration* (c_i) of component i is defined similarly but on a molar basis. Therefore,

$$\rho_i = \frac{m_i}{V} \qquad \text{and} \qquad c_i = \frac{n_i}{V}$$

and both are related by

$$\frac{\rho_i}{c_i} = \frac{m_i}{n_i} = M_i$$

Also,

$$\rho = \sum \rho_i \qquad \text{and} \qquad c = \sum c_i$$

The following is a summary of definitions for the mass and molar compositions of a multicomponent system and the interrelationships between the various quantities.

Notation

m_i =	mass of species i in mixture = $n_i M_i$	(gram of i)
m =	total mass of mixture = $\sum m_i$	(gram of mixture)
n_i =	moles of species i in mixture = m_i/M_i	(g-moles of i)
n =	total moles of mixture = $\sum n_i$	(g-moles of mixture)
M_i =	molecular weight of species i = m_i/n_i	(gram of i/g-mole of i)
V =	total volume of mixture	(cm^3 of mixture)

Mass Compositions

$$\left.\begin{array}{l}\text{Mass fraction}\\ \text{of species i}\end{array}\right\} = \omega_i = \frac{m_i}{\sum m_i} = \frac{m_i}{m} \qquad \left(\sum \omega_i = 1\right)$$

$$\left.\begin{array}{l}\text{Mass concentration}\\ \text{of species i}\end{array}\right\} = \rho_i = \frac{m_i}{V} \qquad (\text{g of i/cm}^3)$$

$$\left.\begin{array}{l}\text{Mass density}\\ \text{of mixture}\end{array}\right\} = \rho = \frac{m}{V} = \sum \rho_i \qquad (\text{g/cm}^3)$$

Molar Compositions

$$\left.\begin{array}{l}\text{Mole fraction}\\ \text{of species i}\end{array}\right\} = x_i = \frac{n_i}{\sum n_i} = \frac{n_i}{n} \qquad \left(\sum x_i = 1\right)$$

$$\left.\begin{array}{l}\text{Molar concentration}\\ \text{of species i}\end{array}\right\} = c_i = \frac{n_i}{V} \qquad (\text{g-moles of i/cm}^3)$$

$$\left.\begin{array}{l}\text{Molar density}\\ \text{of mixture}\end{array}\right\} = c = \frac{n}{V} = \sum c_i \qquad (\text{g-moles/cm}^3)$$

Mixture

$$\left.\begin{array}{l}\text{Molecular weight}\\ \text{of mixture}\end{array}\right\} = M = \frac{M}{n} = \sum x_i M_i = \frac{\rho}{c} \quad \text{(g/g-mole)}$$

Relations Between Various Quantities

$$\omega_i = \frac{m_i}{m} = \frac{\rho_i}{\rho} = \frac{n_i M_i}{\sum n_i M_i} = \frac{x_i M_i}{\sum x_i M_i}$$

$$\rho_i = \frac{m_i}{V} = \rho\omega_i = c_i M_i$$

$$x_i = \frac{n_i}{n} = \frac{c_i}{c} = \frac{m_i/M_i}{\sum (m_i/M_i)} = \frac{\omega_i/M_i}{\sum (\omega_i/M_i)}$$

$$c_i = \frac{n_i}{V} = cx_i = \frac{\rho_i}{M_i}$$

$$M = \frac{\rho}{c} = \frac{m}{n} = \sum x_i M_i = \frac{1}{\sum (\omega_i/M_i)} = \frac{x_i M_i}{\omega_i}$$

$$\sum \omega_i = 1 \qquad \sum x_i = 1$$

$$\rho = \sum \rho_i \qquad c = \sum c_i$$

Compositions of Liquid Systems

Many other types of representation are followed in chemistry in dealing with the concentrations of species in liquid solutions and in mixtures. One component may be arbitrarily selected as a reference substance and the composition of the system expressed in terms of the mass (or volume or moles) of each component present per unit mass (or volume or moles) of the reference substance. For example, in a binary system comprising a solute and a solvent, the composition may be stated as the mass of solute per fixed mass of solvent. These ideas are equally valid when extended to a more general system containing a number of dissolved components. Among the common units used are: (i) grams of solute per gram of solvent; (ii) gram-moles of solute per gram-mole of solvent; (iii) gram-moles of solute per 1000 g of solvent, termed *molality* of the solution; and (iv) parts per million, i.e., parts by weight of a trace component per one million parts by weight of the mixture, abbreviated as ppm.

If a fixed volume of the solution is chosen as the basis, the concentration of a component (or dissolved species) can be expressed in one of the following ways:

(i) grams of solute per unit volume of solution, which is just the mass concentration (e.g., grams of i/cm^3 of solution)
(ii) gram-moles of solute per unit volume of solution, which is equivalent to the molar concentration (e.g., g-moles of i/cm^3 of solution)
(iii) gram-equivalents of solute per one liter of solution, termed *normality*.

The last definition has widespread use in the volumetric analysis of solutions. If a fixed amount of reagent is present in a solution, it can be diluted to any desired normality by application of the general dilution formula $V_1N_1 = V_2N_2$. Here, subscripts 1 and 2 refer to the initial solution and the final (diluted) solution, respectively; V denotes the solution volume (in milliliters) and N the solution normality. The product V_1N_1 expresses the amount of the reagent in gram-milliequivalents present in a volume V_1 ml of a solution of normality N_1. Numerically, it represents the volume of a one normal (1N) solution chemically equivalent to the original solution of volume V_1 and of normality N_1. The same equation $V_1N_1 = V_2N_2$ is also applicable in a different context, in problems involving acid-base neutralization, oxidation-reduction, precipitation, or other types of titration reactions. The justification for this formula relies on the fact that substances always react in titrations, in chemically equivalent amounts.

Example 2-34

The composition of an aqueous solution of H_2SO_4 is given as 40 mole % and its mass density as 1.681 g/cm³ at 20°C. Find the mole density of the solution and also express its composition in the following ways: (i) weight percent, (ii) lb of solute/lb of solvent, (iii) lb-moles of solute/lb of solvent, (iv) g of solute/100 ml of solution, (v) molarity, (vi) normality, and (vii) molality.

Solution

Choose a *basis* of 100 g-moles of solution.

MW of H_2SO_4 (solute) = 98.07; MW of H_2O (solvent) = 18.02

Amount of solute = 40 g-moles = (40)(98.07) = 3922.8 g
Amount of solvent = 60 g-moles = (60)(18.02) = 1081.2 g
Total mass of solution = 5004.0 g

$$\text{Total volume of solution} = \frac{5004 \text{ g}}{1.681 \text{ g/cm}^3} = 2976.8 \text{ cm}^3$$

Molar density $c = n/V = 100/2976.8 = 3.36 \times 10^{-2}$ g-moles/cm³

(i) weight percent of solute = (3922.8)(100)/(5004) = 78.39%
weight percent of solvent = 100 − 78.39 = 21.61%

(ii) $$\frac{\text{lb of solute}}{\text{lb of solvent}} = \frac{3922.8}{1081.2} = 3.628$$

(iii) $$\frac{\text{lb-moles of solute}}{\text{lb of solvent}} = \frac{40}{1081.2} = 3.70 \times 10^{-2}$$

(iv) $$\frac{\text{g of solute}}{\text{100 ml of solution}} = \frac{(3922.8)}{(2976.8)}(100) = 131.78$$

(v) Molarity $= \dfrac{\text{g-moles of solute}}{1000 \text{ ml of solution}} = \dfrac{(40)}{(2976.8)}(1000) = 13.44 \text{ M}$

(vi) Equivalent weight of solute $= \dfrac{\text{MW}}{\text{valence}} = \dfrac{98.07}{2} = 49.03 \text{ g/g-eq.}$

$$\text{Normality} = \frac{\text{g-eq. of solute}}{1000 \text{ ml of solution}}$$

$$= 2(\text{Molarity}) = 26.87\text{N}$$

(vii) Molality $= \dfrac{\text{g-moles of solute}}{1000 \text{ g of solvent}} = \dfrac{(40)(1000)}{(1081.2)} = 37.00 \text{ molal}$

pH Scale and Buffer Solutions. The *pH of a solution* is defined as the logarithm of the reciprocal of the hydrogen ion concentration (or activity) $[H^+]$ of the solution, i.e.,

$$\text{pH} = -\log [H^+]$$

A pOH scale may be defined analogously for the hydroxyl ion concentration by

$$\text{pOH} = -\log [OH^-]$$

Because the ionic product of water $K_w = [H^+][OH^-] = 1.04 \times 10^{-14}$ at 25°C, it follows that pH = 14 – pOH. Thus, a neutral solution (e.g., pure water at 25°C) in which $[H^+] = [OH^-]$ has a pH = pOH = 7. Acids show a lower pH and bases a higher pH than this neutral value of 7. The hydrogen ion concentrations can cover a wide range, from ~1 g-ion/liter or more in acidic solutions to $\sim 10^{-14}$ g-ion/liter or less in alkaline solutions [53, p. 545]. *Buffer* action refers to the property of a solution in resisting change of pH upon addition of an acid or a base. Buffer solutions usually consist of a mixture of a weak acid and its salt (conjugate base) or of a weak base and its salt (conjugate acid).

The ionic strength (μ) of a solution is a measure of the intensity of the electrical field due to the ions present in the solution. It is defined by

$$\mu = \left(\frac{1}{2}\right)\sum c_i z_i^2$$

where c_i is the molal concentration of the ionic species i (g-ions i/1000 g solvent), and z_i is the valence or charge on the ion i.

Example 2-35

Find the ionic strength of (i) 0.05 molal sodium sulfate (Na_2SO_4) solution, and (ii) 0.25 molal nitric acid (HNO_3) and 0.4 molal barium nitrate ($Ba(NO_3)_2$) together in one solution.

Solution

(i) Ions Na^+ and SO_4^{2-} are present in concentrations of 0.10 and 0.05 molal, respectively.

$$\mu = \left(\frac{1}{2}\right)\left[0.10(1)^2 + (0.05)(2)^2\right] = 0.15$$

(ii) Ions H^+ and NO_3^- are present in HNO_3 in concentrations of 0.25 molal each, and ions Ba^{2+} and NO_3^- are present in $Ba(NO_3)_2$ in concentrations of 0.4 and 0.8 molal, respectively.

$$\mu = \left(\frac{1}{2}\right)\left[0.25(1)^2 + (0.25 + 0.80)(1)^2 + (0.4)(2)^2\right] = 1.45$$

Material Balances

Types of Chemical Processes. Chemical processes usually involve the transfer of matter and energy while transforming raw materials into useful products. Process calculations consist of essentially three steps: first, a system is identified; second, a basis of calculations is chosen; and third, appropriate material and energy balances are independently performed on the system, after considering the occurrence of any chemical reactions. In general, a *system* refers to a substance, or a group of substances contained in a volume with defined boundaries, and a *process* refers to changes, whether physical or chemical, occurring in that system. Material and energy balances, as applied to a system, are based upon the principles of conservation of mass and of energy, respectively. In this subsection we shall examine the applications of material balance only; examples of energy balances will be considered later in the subsection on thermochemistry.

All processes may be classified as batch, continuous, or semibatch depending on how materials are transferred into and out of the system. Also, the process operation may be characterized as unsteady state (i.e., transient) or steady state, depending on whether the process variables (e.g., pressure, temperature, compositions, flowrate, etc.) are changing with time or not, respectively. In a *batch* process, the entire feed material (i.e., charge) is added instantaneously to the system marking the beginning of the process, and all the contents of the system including the products are removed at a later time, at the end of the process. In a *continuous* process, the materials enter and leave the system as continuous streams, but not necessarily at the same rate. In a *semibatch* process, the feed may be added at once but the products removed continuously, or vice versa. It is evident that batch and semibatch processes are inherently unsteady state, whereas continuous processes may be operated in a steady or unsteady-state mode. Start-up and shut-down procedures of a steady continuous production process are examples of transient operation.

General Material Balances. According to the law of conservation of mass, the total mass of an isolated system is invariant, even in the presence of chemical reactions. Thus, an *overall* material balance refers to a mass balance performed on the entire material (or contents) of the system. Instead, if a mass balance is made on any component (chemical compound or atomic species) involved in the process, it is termed a *component* (or *species*) material balance. The general mass balance equation has the following form, and it can be applied on any material in any process.

$$\begin{pmatrix}\text{Mass}\\\text{inflow}\\\text{entering}\\\text{the system}\end{pmatrix} - \begin{pmatrix}\text{Mass}\\\text{outflow}\\\text{leaving}\\\text{the system}\end{pmatrix} + \begin{pmatrix}\text{Net internal}\\\text{mass generation}\\\text{due to chemical}\\\text{reactions within}\\\text{the system}\end{pmatrix} = \begin{pmatrix}\text{Net mass}\\\text{accumulation}\\\text{within the}\\\text{system}\end{pmatrix}$$

or symbolically,

$$m_{in} - m_{out} + \mathcal{R} = \Delta m$$

The third term on the left side of the equation has significance in reactive systems only. It is used with a positive sign when material is produced as a net result of all chemical reactions; a negative sign must precede this term if material is consumed by chemical reactions. The former situation corresponds to a source and the latter to a sink for the material under consideration. Since the total mass of reactants always equals the total mass of products in a chemical reaction, it is clear that the reaction (source/sink) term $\mathcal{R}$ should appear explicitly in the equation for component material balances only. The overall material balance, which is equivalent to the algebraic sum of all of the component balance equations, will not contain any $\mathcal{R}$ term.

Differential and Integral Balances. Two types of material balances, differential and integral, are applied in analyzing chemical processes. The differential mass balance is valid at any instant in time, with each term representing a rate (i.e., mass per unit time). A *general differential material balance* may be written on any material involved in any *transient* process, including semibatch and unsteady-state continuous flow processes:

$$\begin{pmatrix}\text{Net rate of}\\\text{mass accumulation}\end{pmatrix} = \begin{pmatrix}\text{Rate of}\\\text{mass inflow}\end{pmatrix} - \begin{pmatrix}\text{Rate of}\\\text{mass outflow}\end{pmatrix} + \begin{pmatrix}\text{Net rate of internal}\\\text{mass generation by}\\\text{chemical reactions}\end{pmatrix}$$

or symbolically

$$\frac{dm}{dt} = \dot{m}_{in}(t) - \dot{m}_{out}(t) + \dot{\mathcal{R}}(t)$$

where each of the rate terms on the right side of the equations can, in general, be functions of time. The actual solution is obtained upon integration over time between the initial and final states of the transient process.

A special case of the above equation applies to a *continuous steady-state flow process* when all of the rate terms are independent of time and the accumulation term is zero. Thus, the differential material balance for any component i in such a process is given by

$$\begin{pmatrix}\text{Rate of mass}\\\text{outflow, } \dot{m}_{out}\end{pmatrix}_i - \begin{pmatrix}\text{Rate of mass}\\\text{inflow, } \dot{m}_{in}\end{pmatrix}_i = \begin{pmatrix}\text{Net rate of internal}\\\text{mass generation, } \dot{\mathcal{R}}\end{pmatrix}_i$$

and the overall material balance is $\dot{m}_{in} = \dot{m}_{out}$. When chemical reactions are absent, we have

$$(\dot{m}_{in})_i = (\dot{m}_{out})_i$$

for a continuous flow process at steady-state.

For any transient process that begins at time t_0 and is terminated at a later time t_f, the *general integral material balance* equation has the form

$$m(t_f) - m(t_0) = \int_{t_0}^{t_f} \dot{m}_{in} dt - \int_{t_0}^{t_f} \dot{m}_{out} dt + \int_{t_0}^{t_f} \dot{\mathcal{R}} dt \qquad (2\text{-}218)$$

Here, $m(t_f)$ is the mass of the system contents at final time t_f and $m(t_0)$ is the mass at initial time t_0. As before, both component and overall mass balances may be written in integral form.

A special case of Equation 2-218 is directly applicable to *batch* processes for which the mass flowrate terms are zero. The integral material balance for any component i in such a process is

$$\begin{pmatrix} \text{Net mass} \\ \text{accumulation of i} \end{pmatrix} = \Delta m_i = m_i(t_f) - m_i(t_0) = \mathcal{R}_i$$

and the overall mass balance is

$$\Delta m = m(t_f) - m(t_0) = 0$$

It must be kept in mind that the reaction term will not occur in the overall mass balance equations of reactive systems because $\Sigma\mathcal{R}_i = 0$, i.e., there is no net mass gain or loss as a result of chemical reactions.

Stoichiometry in Reactive Systems. The use of molar units is preferred in chemical process calculations since the stoichiometry of a chemical reaction is always interpreted in terms of the number of molecules or number of moles. A stoichiometric equation is a balanced representation that indicates the relative proportions in which the reactants and products partake in a given reaction. For example, the following stoichiometric equation represents the combustion of propane in oxygen:

$$\begin{array}{llllll} & C_3H_8(g) + & 5O_2(g) \rightarrow & 3CO_2(g) + & 4H_2(\ell) \\ \upsilon_i = & 1 & 5 & 3 & 4 \end{array}$$

One molecule (or mole) of propane reacts with five molecules (or moles) of oxygen to produce three molecules (or moles) or carbon dioxide and four molecules (or moles) of water. These numbers are called *stoichiometric coefficients* (v_i) of the reaction and are shown below each reactant and product in the equation. In a stoichiometrically balanced equation, the total number of atoms of each constituent element in the reactants must be the same as that in the products. Thus, there are three atoms of C, eight atoms of H, and ten atoms of O on either side of the equation. This indicates that the compositions expressed in gram-atoms of elements remain unaltered during a chemical reaction. This is a consequence of the principle of conservation of mass applied to an isolated reactive system. It is also true that the combined mass of reactants is always equal to the combined mass of products in a chemical reaction, but the same is not generally valid for the total number of moles. To achieve equality on a molar basis, the sum of the stoichiometric coefficients for the reactants must equal the sum of v_i for the products. Definitions of certain terms bearing relevance to reactive systems will follow next.

A *limiting reactant* is that reactant which is present in the smallest stoichiometric amount. In industrial reactions, the reactants are not necessarily supplied in the exact proportions demanded by the stoichiometry of the equation. Under these circumstances, the reaction mixture at the conclusion of the process will include the products together with some of the unreacted reactants present in excess of their stoichiometric requirements. By identifying a limiting reactant, the *percent excess* is calculated for any *excess reactant* as

$$\% \text{ excess} = \frac{(n - n^*)}{n^*} 100$$

where n is number of available moles of excess reactant, and n* is theoretical (i.e., stoichiometric) moles required to react with the limiting reactant.

Conversion (or degree of conversion) refers to the fraction of the feed or fraction of some reactant in the feed which has been converted into products.

The *degree of completion* of a reaction refers to the fraction of the limiting reactant that has been converted into products.

When a single reactant is converted into a single product, the *yield* for the reaction is expressed as the ratio of the moles (or mass) of the product formed to the moles (or mass) of the initial reactant.

In processes involving the combustion of fuels, either pure oxygen or air is supplied in amounts greater than the stoichiometric requirements for complete combustion. The terms "theoretical air" or "theoretical oxygen" are thus frequently encountered in combustion problems. The molar composition of dry air at atmospheric conditions [from *International Critical Tables,* Volume 1, p. 393 (1926)]:

	Mole%
Nitrogen	78.03
Oxygen	20.99
Argon	0.94
Carbon dioxide	0.03
H_2, He, Ne, Kr, Xe	0.01
	100.00
Average molecular weight = 28.97	

For *combustion calculations,* it is acceptable to take the average molecular weight of dry air to be 28.97, and to *assume* a simplified composition of 79.0 mole% nitrogen and 21.0 mole% oxygen (or equivalently, 76.8 wt% nitrogen and 23.2 wt% oxygen).

In summary, the procedure to be adopted in material balance calculations involving reactive systems is as follows:

1. Identify the system and the process.
2. Examine the stoichiometry of the chemical reaction, and identify the limiting reactant and excess reactants.
3. Choose a basis of calculations.
4. Perform an overall material balance and the necessary component material balances so as to provide the maximum number of independent equations. In the event the balance is written in differential form, appropriate integration must be carried out over time, and the set of equations solved for the unknowns.

For a nonreactive system, the material balance may be done either on a mass or on a molar basis.

Example 2-36

Consider the combustion of ethane (C_2H_6) in pure oxygen. If 100 lb of ethane are available and 10% excess oxygen is supplied to ensure complete combustion, calculate (1) the amount of oxygen supplied, and (2) compositions of the reactants and products on mass and molal bases.

Solution

Choose a *basis* of 100 lb of ethane (limiting reactant). The stoichiometric equation is

$$\begin{array}{lcccccc} & C_2H_6 & + & \left(\frac{7}{2}\right) O_2 & \longrightarrow & 2CO_2 & + & 3H_2O & \\ \upsilon_i = & 1 & & \frac{7}{2} & & 2 & & 3 & \text{lb-moles} \\ M_i = & 30.07 & & 32.00 & & 44.01 & & 18.01 & \text{lb/lb-moles} \end{array}$$

Thus, 1 mole of ethane reacts with $\frac{7}{2}$ moles of O_2 to produce 2 moles of CO_2 and 3 moles of H_2O vapor.

lb-moles of C_2H_6 available $= 100/30.07 = 3.326$

Stoichiometric lb-moles of O_2 required $= (\frac{7}{2})(3.326) = 11.640$

With 10% excess, lb-moles of O_2 supplied $= 1.1\ (11.64) = 12.804$

1. Amount of O_2 supplied = 12.804(32.00) = 409.71 lb
2. Calculation of the composition of the feed (reactant) mixture is shown below.

Reactant i	Mass m_i (lb)	Weight %	n_i (lb-moles)	Mole %
Ethane	100.00	24.41	3.326	20.49
Oxygen	409.71	75.59	12.804	79.51
Total	509.71	100.00	16.230	100.00

In the product mixture, amount of excess (unreacted) $O_2 = 0.1(11.64) = 1.164$ lb-moles = 37.25 lb.

Amount of CO_2 formed = (2)(100)/30.07 = 6.651 lb-moles = 292.72 lb
Amount of H_2O formed = (3)(100)/30.07 = 9.977 lb-moles = 179.73 lb

Hence, the composition of the product gas stream can be calculated as follows:

Product j	Mass m_j (lb)	Weight %	n_j (lb-moles)	Mole %
CO_2	292.72	57.43 (88.71)	6.651	37.38 (85.11)
H_2O	179.73	35.26 (—)	9.977	56.08 (—)
O_2	37.25	7.31 (11.29)	1.164	6.54 (14.89)
Total	509.70	100.00	17.792	100.00

The values in parentheses show the product composition on a dry basis, excluding H_2O vapor.

Ideal Gas Behavior

Ideal (or perfect) gas behavior is approached by most vapors and gases in the limit of low pressures and elevated temperatures. Two special forms of restricted utility known as the Boyle's law and the Charles' law preceded the development of the perfect gas law.

Boyle's Law. At constant temperature (T), the volume (V) of a fixed mass of an ideal gas is inversely proportional to the absolute pressure (P). That is,

$$V \propto \frac{1}{P} \qquad \text{or} \qquad PV = \text{constant} \qquad (T = \text{constant})$$

Charles' Law. At constant pressure (P), the volume (V) of a given mass of an ideal gas is directly proportional to the absolute temperature (T):

$$V \propto T \qquad \text{or} \qquad \frac{V}{T} = \text{constant} \qquad (P = \text{constant})$$

A combination of the above two statements results in the equation of state termed the *ideal* (or *perfect*) *gas law:*

$$PV = nRT \qquad \text{or} \qquad P\hat{V} = \frac{RT}{M} \qquad \text{or} \qquad \rho = \frac{PM}{RT}$$

where n is m/M = number of moles of ideal gas, m is mass of gas present, M is molecular weight of gas in mass per mole, $\hat{V}$ is $V/M = 1/\rho$ = specific volume of ideal gas in volume per mass, and R is universal gas constant in energy per mole per absolute degree.

The values of R in different sets of units are given below:

8.314	Joules/g-mole °K
8.314	m^3 Pa/g-mole °K
82.057	cm^3 atm/g-mole °K
83.14	cm^3 bar/g-mole °K
62.36	liter mm Hg/g-mole °K
21.85	$(ft)^3$(in.Hg)/lb-mole °R
0.7302	ft^3 atm/lb-mole °R
10.73	ft^3 psi/lb-mole °R
1545	$(lb_f/ft^2)(ft^3)$/lb-mole °R
1.987	cal/g-mole °K
1.987	Btu/lb-mole °R

At atmospheric conditions, most ordinary gases such as air, nitrogen, etc., exhibit P–V–T behavior that is well represented by the above equation. When n moles of an ideal gas undergo a change of state, then

$$P_1V_1 = nRT_1 \qquad \text{and} \qquad P_2V_2 = nRT_2$$

or

$$\frac{P_1V_1}{P_2V_2} = \frac{T_1}{T_2}$$

where the subscripts 1 and 2 refer to the initial state and final state, respectively.

It is a common practice to evaluate the molal volume ($\tilde{V}$) of an ideal gas at a set of reference conditions known as the *standard state*. If the standard state is chosen to be

$$P = 1 \text{ atm} = 760 \text{ mmHg} = 14.696 \text{ psia}$$

and

$$T = 0°C = 273.16°K = 32°F = 491.69°R$$

Then

$$\tilde{V} = \frac{V}{n} = \frac{RT}{P}$$

represents the volume occupied by 1 mole of any perfect gas, and has the following values at the *s*tandard conditions of *t*emperature and *p*ressure (abbreviated as STP):

$$\tilde{V} = 22.414 \ \ell/\text{g-mole}$$

$$= 22{,}414.6 \text{ cm}^3/\text{g-mole}$$

$$= 359.046 \text{ ft}^3/\text{lb-mole}$$

The *specific gravity* (G) of an ideal gas relative to a reference gas (also assumed ideal) is given by

$$G = \frac{\rho}{\rho_{ref}} = \frac{M}{M_{ref}} \frac{P}{P_{ref}} \frac{T_{ref}}{T}$$

It simplifies to $G = M/M_{ref}$ when $T = T_{ref}$ and $P = P_{ref}$.

Although real gases deviate from ideal gas behavior and therefore require different equations of state, the deviations are relatively small under certain conditions. An error of 1% or less should result if the ideal gas law were used for diatomic gases when $\tilde{V} \geq 5 \ \ell$/gm-mole (80 ft^3/lb-mole) and for other gases and light hydrocarbon vapors when $\tilde{V} \geq 20 \ \ell$/gm-mole (320 ft^3/lb-mole) [61, p. 67].

Example 2-37

A volatile compound of chlorine has been analyzed to contain 61.23% of oxygen (O_2) and 38.77% of chlorine (Cl_2) by weight. At 1 atm and 27°C, 1000 cm^3 of its vapor weighs 7.44 g. Assuming ideal gas behavior for the vapor, estimate its molecular weight and deduce its molecular formula.

Solution

At T = 27°C = 300.2°K and P = 1 atm,

$$\text{Mass density of the vapor} = \rho = \frac{7.44 \text{ g}}{1000 \text{ cm}^3} = \frac{PM}{RT}$$

With R = 82.057 cm^3-atm/g-mole °K, the molecular weight is calculated as

$$M = \frac{\rho RT}{P} = \frac{(7.44)(82.057)(300.2)}{(1000)} = 183.3 \text{ g/g-mole}$$

If the unknown compound is made up of x atoms of chlorine and y atoms of oxygen, its molecular formula will be Cl_xO_y. Because the molecular weights of Cl_2 and O_2 are 70.91 and 32.00 g/g-mole, respectively, we can express the weight composition of the compound as

$$\frac{\text{weight of oxygen}}{\text{weight of chlorine}} = \frac{(y)(32.00/2)}{(x)(70.91/2)} = \frac{0.6123}{0.3877} \quad \text{(given)}$$

or y = 3.5x. Making this substitution in the formula for the molecular weight of the substance, we have

$$183.3 = M = (x)\frac{70.91}{2} + (y)\frac{32.00}{2}$$

or

$$2(183.3) = 70.91x + 32(3.5)x = 182.9x$$

$$\therefore x = 2(183.3)/(182.9) \doteqdot 2$$

With x = 2 and y = 7, the compound has molecular formula Cl_2O_7 with M = 182.9.

Mixtures of Ideal Gases

Two relations are postulated to describe the P-V-T behavior of ideal gas mixtures:

Dalton's Law of Partial Pressures. The total pressure (P) of a gaseous mixture equals the sum of the partial pressures of its components. By definition, the *partial pressure* of any component gas is the hypothetical pressure it would exert by occupying the entire volume (V) of the mixture at the same temperature (T). That is,

$$P = P_1 + P_2 \cdots + P_N = \sum P_i$$

where the partial pressure of component gas i in a N-component gas mixture is

$$P_i = \frac{n_i RT}{V} \qquad i = 1, 2, \cdots, N$$

by the ideal gas law. If the mixture also behaves ideally, then

$$\frac{PV}{RT} = n = \sum n_i = \left(\sum P_i\right)\frac{V}{RT}$$

Thus, the mole fraction of component gas i is

$$y_i = \frac{n_i}{n} = \frac{P_i}{P} = \frac{\text{Partial pressure of i}}{\text{Total pressure of mixture}}$$

Amagat's Law. The total volume of a gaseous mixture equals the sum of the pure-component volumes. By definition, the *pure-component volume* of a component gas in a mixture is the hypothetical volume that the component would occupy at the same temperature and total pressure of the mixture. By Amagat's law,

$$V = \sum V_i = V_1 + V_2 + \ldots$$

If each component gas as well as the mixture obeys the ideal gas law, it follows that the pure-component volume of component i is

$$V_i = \frac{n_i RT}{P} = y_i \frac{nRT}{P} = y_i V$$

or

$$y_i = \frac{n_i}{n} = \frac{V_i}{V}$$

Thus, in an ideal gas mixture, the mole fraction of each component is identical with its volume fraction (by Amagat's law) or the ratio of its partial pressure to the total pressure (by Dalton's law). For both laws to be applicable simultaneously, the mixture and its components must behave ideally.

The behavior of real gases is discussed in the previous section on "Thermodynamics and Heat Transfer."

Example 2-38

A natural gas has the following composition by volume at a temperature of 80°F and a gauge pressure of 40 psig–87.2% of methane, 4.5% of ethane, 3.6% of propane, 1.8 % of n-butane, 1.0% of isobutane, and 1.9% of nitrogen. Assuming the ideal-gas law is applicable, calculate (i) the average molecular weight of the mixture, (ii) density of the natural gas, (iii) specific gravity of the gas, (iv) volume occupied by 100 lb of gas at 1 atm and 60°F, (v) partial pressure of nitrogen, and (vi) pure-component volume of nitrogen per 1,000 ft^3 of gas.

Solution

Start with a *basis* of 1 lb-mole of the natural gas at T = 80°F = 540°R and P = 40 psig = 54.7 psia. The volume percent and mole percent compositions are identical for a perfect-gas mixture.

(i) The molecular weight of the gas mixture is $M = \Sigma y_i M_i$ and its calculation is shown below in tabular form.

Component i	M_i (lb/lb-mole)	y_i	$m_i = y_i M_i$ (lb)
CH_4	16.04	0.872	13.987
C_2H_6	30.07	0.045	1.353
C_3H_8	44.10	0.036	1.588
n-C_4H_{10}	58.12	0.018	1.046
iso-C_4H_{10}	58.12	0.010	0.581
N_2	28.01	0.019	0.532
		1.000	19.087

$\therefore M = \sum m_i = 19.09$ lb/lb-mole

(ii) With R = 10.73 ft³-psia/lb-mole °R, mass density of the mixture

$$\rho = \frac{PM}{RT} = \frac{(54.7)(19.09)}{(10.73)(540)} = 0.180 \text{ lb/ft}^3$$

Molar density $c = \rho/M = 0.180/19.09 = 9.44 \times 10^{-3}$ lb-moles/ft³

(iii) Assuming that air (reference substance) also obeys the ideal-gas law, the specific gravity of the mixture is given by

$$G(80° \text{ F}, 54.7 \text{ psia}) = \frac{\rho}{\rho_{air}} = \frac{M}{M_{air}} = \frac{19.09}{28.97} = 0.659$$

(iv) At T_1 = 60°F = 520°R and P_1 = 1 atm = 14.696 psia, the volume occupied by

$$n_1 = \frac{100}{19.09} \text{ lb-moles of the gas}$$

is

$$V_1 = \frac{n_1 RT}{P_1} = \frac{(100)(10.73)(520)}{(19.09)(14.696)} = 1{,}988.3 \text{ ft}^3$$

(v) At P = 40 psig and T = 80°F, by Dalton's law of partial pressures, $P_i = y_i P$ for any component i in the mixture. Therefore, partial pressure of nitrogen = (0.019)(54.7) = 1.039 psia.

(vi) Applying Amagat's law at the same conditions, the pure component volume $V_i = y_i V$ for any i. For nitrogen, V_i = (0.019)(1,000) = 19.0 ft³.

Phase Rule and Phase Behavior

Gibb's Phase Rule. The phase rule derived by W. J. Gibbs applies to multiphase equilibria in multicomponent systems, in the absence of chemical reactions. It is written as

$$\mathcal{F} = \mathcal{C} - \mathcal{P} + 2$$

where $\mathcal{F}$ is number of degrees of freedom or variance of the system, $\mathcal{C}$ is minimum number of independent chemical components in terms of which the composition of each phase can be expressed, and $\mathcal{P}$ is number of phases in the system.

The term $\mathcal{F}$ denotes the number of independent phase variables that should be specified in order to establish all of the intensive properties of each phase present. The phase variables refer to the intensive properties of the system such as temperature (T), pressure (P), composition of the mixture (e.g., mole fractions, x_i), etc. As an example, consider the triple point of water at which all three phases—ice, liquid water, and water vapor—coexist in equilibrium. According to the phase rule,

$$\mathcal{F} = \mathcal{C} - \mathcal{P} + 2 = 1 - 3 + 2 = 0$$

The absence of any degrees of freedom implies that the triple point is a unique state that represents an *invariant* system, i.e., one in which any change in the state variables T or P is bound to reduce the number of coexisting phases.

Phase Behavior of a Pure Substance

It is evident from the preceding example that a pure substance can have at most three coexisting phases at equilibrium. At temperatures and pressures other than the triple point, a pure substance may exist either as a single phase (e.g., solid, liquid, or vapor) or as a two-phase system. Application of the phase rule for $\mathcal{P} = 1$ gives $\mathcal{F} = \mathcal{C} - \mathcal{P} + 2 = 1 - 1 + 2 = 2$, indicating that two intensive variables (P and T) can be varied simultaneously in the single-phase region. On the other hand, in the two-phase region $\mathcal{F} = 1 - 2 + 2 = 1$ so that either P or T can be independently varied, but not both for this monovariant system. Thus, the equilibrium phase behavior of a pure substance is represented by a three-dimensional surface with pressure (P), specific volume ($\hat{V}$),* and temperature (T) as coordinates; it is called a phase diagram or P-$\hat{V}$-T diagram (see Figure 2-78). Orthogonal projections of this surface onto the P-$\hat{V}$ plane, $\hat{V}$-T plane, and P-T plane provide convenient means of depicting phase phenomena on two-dimensional plots. Isotherms (T = constant), isobars (P = constant), and isochors ($\hat{V}$ = constant) are drawn as necessary to highlight important aspects of phase behavior.

The *vapor pressure* (P^*) of a pure liquid at a given temperature (T) is the pressure exerted by its vapor in equilibrium with the liquid phase in a closed system. All liquids and solids exhibit unique vapor pressure–temperature curves. For instance, in Figure 2-79, lines BA and AC represent the equilibrium vapor pressure curves of the solid and liquid phases, respectively.

Phase transitions refer to equilibrium processes involving a change of phase such as sublimation (solid to vapor), boiling or vaporization (liquid to vapor), freezing (liquid to solid), etc. On a P–T diagram (see Figure 2-79) the phase transformations take place on the sublimation curve BA, vaporization curve AC, and melting curve AD, which separate the single-phase regions. Two phases coexist in equilibrium on each of these phase-boundary curves, with the exception of the triple point, A. The vaporization curve terminates at the critical point C of the pure substance at which the distinction between liquid and vapor phases disappears and the latent heat of vaporization becomes zero. The single phase that exists above the critical point (P_c, T_c) is variously described as gas, dense fluid, or supercritical fluid. When $T > T_c$, it is impossible to liquefy a dense fluid by varying the pressure alone.

Because a phase change is usually accompanied by a change in volume the two-phase systems of a pure substance appear on a P-$\hat{V}$ (or a T-$\hat{V}$) diagram as regions with distinct boundaries. On a P-$\hat{V}$ plot, the triple point appears as a horizontal line, and the critical point becomes a point of inflection of the critical isotherm, $T = T_c$ (see Figure 2-78 and Figure 2-80).

*The intensive variable for volume (V) can be either the specific volume ($\hat{V}$, volume/mass) or the specific molal volume ($\tilde{V}$, volume/mole).

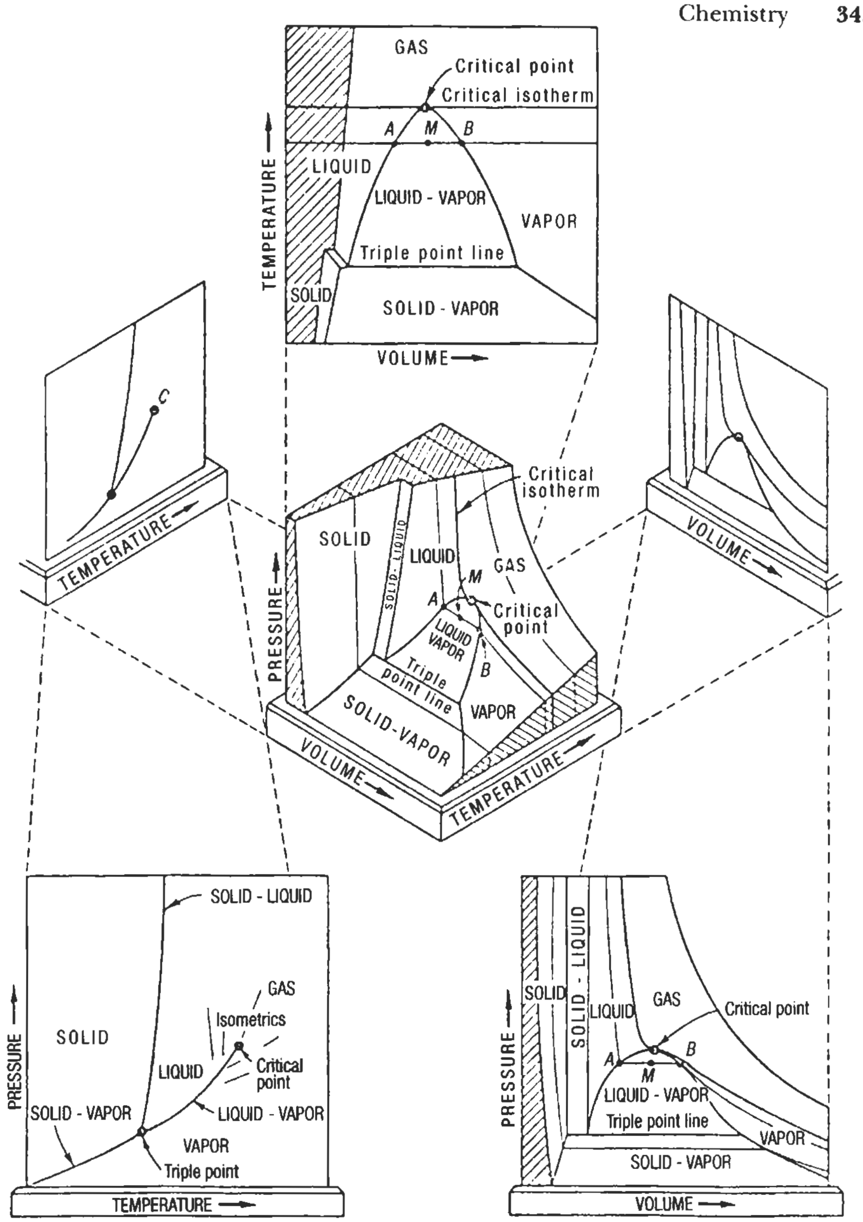


Figure 2-78. Typical phase diagram for a pure substance showing P–V–T surface and its projections [60].

Vapor–Liquid Systems. The vapor-liquid region of a pure substance is contained within the *phase* or *saturation envelope* on a P–$\hat{V}$ diagram (see Figure 2-80). A vapor, whether it exists alone or in a mixture of gases, is said to be *saturated* if its partial pressure (P_i) equals its equilibrium vapor pressure (P_i^*) at the system temperature T. This temperature is called the *saturation temperature* or *dew point* T_{dp}

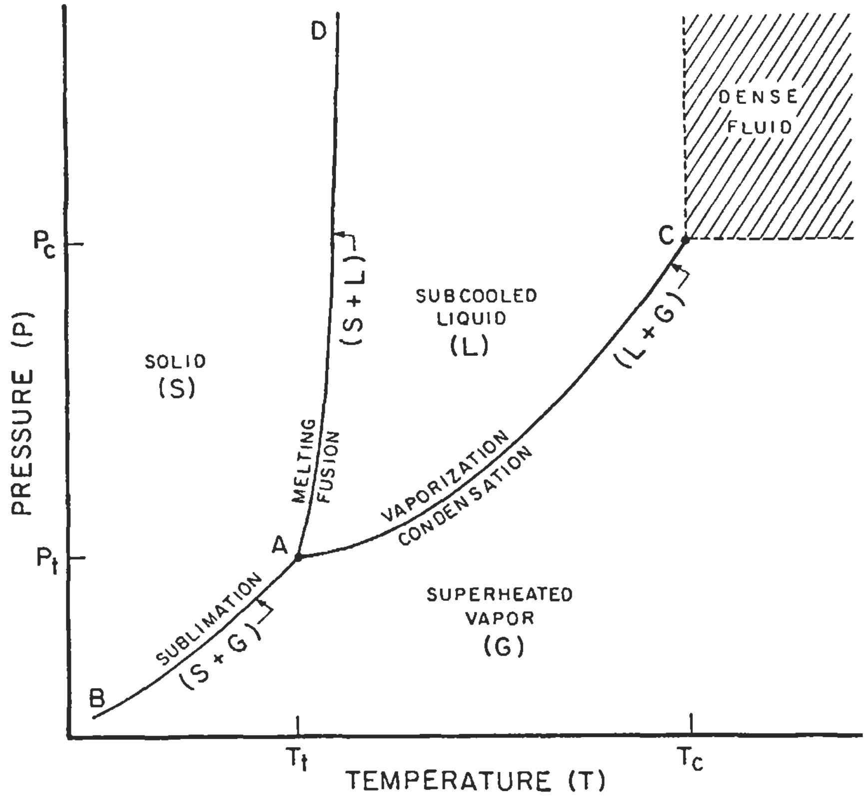


Figure 2-79. Typical P–T diagram for a pure substance (A is the triple point and C is the critical point).

of the vapor because it signals incipient condensation when the first droplet of liquid is formed from the saturated vapor at given pressure P. For a pure substance, the saturated vapor coexists in equilibrium with its saturated liquid at every point on the vaporization curve AC (see Figure 2-79) of the P-T diagram. Its coordinates are given by P (system pressure) = P* (vapor pressure at temperature T), and T (system temperature) = T_b (boiling point of the substance at pressure P). The *normal boiling point* (nbp) refers to the value of T_b attained at a total pressure of 1 standard atmosphere.

A *superheated vapor* refers to a vapor that exists under conditions such that $P_i < P_i^*(T)$. The difference between its existing temperature (T) and its dew point (T_{dp}) is termed the *degrees of superheat* of the vapor. The region of superheated vapor for a pure substance is shown on the T-P diagram (refer to Figure 2-79) by the area lying to the right of the vaporization curve AC. The term *bubble point* (T_{bp}) of a liquid refers to the temperature corresponding to incipient vaporization when the first bubble of vapor is formed from the saturated liquid at a given pressure P. For a pure component system, the bubble point and dew point temperatures are identical to the boiling point of the substance at that pressure. For binary and multicomponent systems of a given composition, $T_{dp} > T_{bp}$ at fixed P or, alternatively, P_{bp} (bubble point pressure) > P_{dp} (dew-point pressure) at fixed T. The saturated liquid curve

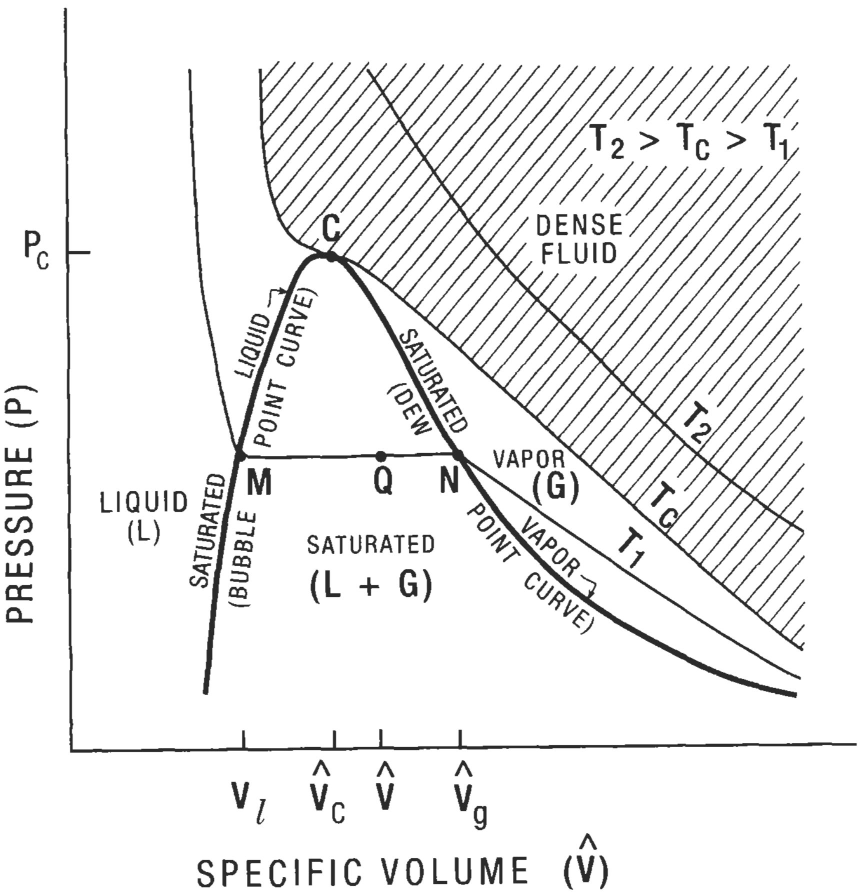


Figure 2-80. Typical P–V diagram for a pure substance showing isotherms and saturation (phases) envelope.

is thus equivalent to the bubble point curve, and the saturated vapor curve is the same as the dew point curve (see Figure 2-80). Although the isotherms on a P- $\hat{V}$ plot are horizontal lines in the vapor-liquid region of a pure substance, they become curves having negative slopes in systems containing more than one component. A liquid is termed *subcooled* if it exists such that its temperature (T) is less than the bubble-point temperature (T_{bp}) at given pressure (P). The *degrees of subcooling* are then given by the difference (T_{bp} – T). In the case of a pure substance, the area DAC lying to the left of the curve AC represents the subcooled liquid region (see Figure 2-79).

Quality of a Wet Vapor. In the vapor-liquid region of a pure substance, the composition of a two-phase system (at given T and P) varies from pure saturated liquid at the bubble point M to pure saturated vapor at the dew point N along the line MQN on the P- $\hat{V}$ diagram (Figure 2-80). For a wet vapor represented by an intermediate point Q, the *quality* (q) refers to the mass fraction of the saturated vapor present in the two-phase mixture. If m_ℓ and m_g indicate the masses of saturated liquid and saturated vapor comprising the wet vapor at point Q, then its quality may be calculated by the application of the lever rule as

$$q \equiv \frac{m_g}{m_\ell + m_g} = \frac{\hat{V} - \hat{V}_\ell}{\hat{V}_g - \hat{V}_l} = \frac{\text{length MQ}}{\text{length MN}}$$

The origin of this rule lies in the expression for the specific volume of the wet vapor, $\hat{V} = q\hat{V}_g + (1-q)\hat{V}_\ell$. The specific intensive properties of the wet vapor are obtained readily from the individual properties of its component phases by an analogous equation:

$$\mathfrak{M} = q\mathfrak{M}_g + (1 - q)\mathfrak{M}_\ell$$

where the symbols $\mathfrak{M}_g$, $\mathfrak{M}_\ell$, and $\mathfrak{M}$ refer to the specific properties of the saturated vapor, saturated liquid, and the two-phase mixture, respectively.

Clausius-Clapeyron Equation. This equation was originally derived to describe the vaporization process of a pure liquid, but it can be also applied to other two-phase transitions of a pure substance. The Clausius-Clapeyron equation relates the variation of vapor pressure (P^*) with absolute temperature (T) to the molar latent heat of vaporization, λ_v, i.e., the thermal energy required to vaporize one mole of the pure liquid:

$$\frac{dP^*}{dT} = \frac{\lambda_v}{T(\tilde{V}_g - \tilde{V}_\ell)} \doteqdot \frac{\lambda_v}{T\tilde{V}_g}$$

By neglecting the specific molar volume of the saturated liquid $\tilde{V}\ell$ in relation to that of the saturated vapor $\tilde{V}_g$, and by assuming the vapor phase to behave as an ideal gas, i.e., $P^* \tilde{V}_g = RT$, the above equation may be arranged into

$$\frac{d(\ell n P^*)}{d(1/T)} = -\frac{\lambda_v}{R}$$

This suggests that a plot of $\ell n\ P^*$ against $1/T$ should yield a line having a local slope of $(-\lambda_v/R)$. A straight line is obtained only when λ_v is nearly constant, i.e., over a narrow range of temperatures. An integrated version of the Clausius-Clapeyron equation finds use in correlation of vapor pressure data:

$$P^*(T) \doteqdot P_0^* \exp\left[-\frac{\bar{\lambda}_v}{R}\left(\frac{1}{T} - \frac{1}{T_0}\right)\right]$$

where P_0^* is the known vapor pressure at reference temperature T_0, and $\bar{\lambda}_v$ is the average value of λ_v between T_0 and T. The above equation is reasonably valid when the range $|T_0 - T|$ is small and when the two-phase region of interest is far away from the critical point.

Example 2-39

Estimate the molal latent heat of vaporization for pure benzene at its normal boiling point of 80.1°C.

Solution

Two estimates will be made using vapor pressure data from the *CRC Handbook* [63] and the integrated form of Clausius-Clapeyron equation:

$$\ell n \frac{P^*(T)}{P_0^*(T_0)} = \frac{\bar{\lambda}_v}{R}\left(\frac{1}{T_0} - \frac{1}{T}\right)$$

Estimate (i): From page D-218 of the *CRC Handbook* [63], $P_0^* = 1$ atm at $T_0 = 80.1°C = 353.3°K$ and $P^* = 2$ atm at $T = 103.8°C = 377.0°K$. With $R = 1.987$ cal/g-mole °K, we have

$$\ell n\left(\frac{2}{1}\right) = 0.6932 = \frac{\bar{\lambda}_v}{(1.987)}\left(\frac{1}{353.3} - \frac{1}{377}\right)$$

or

$$\bar{\lambda}_v = 7{,}740 \text{ cal/g-mole}$$

Estimate (ii): From page D-208 [63], $P_0^* = 400$ mmHg at $T_0 = 60.6°C = 333.8°K$ and $P^* = 760$ mmHg at $T = 80.1°C = 353.3°K$. Then, $\bar{\lambda}_v = 7{,}713$ cal/g-mole.

The second estimate compares better with the literature value of ~7,700 cal/g-mole.

Vapor–Liquid Equilibria in Binary and Multicomponent Systems

As discussed earlier, a N-component system with $N > 1$ does not exhibit a single boiling point at a given pressure as a pure substance does. Instead, at a constant pressure, a liquid mixture of fixed composition undergoes a change of phase to vapor over a range of temperatures lying between the bubble point and the dew point. Three different approaches—Raoult's law, Henry's law, and the concept of the equilibrium ratio or K factor—are available for computations involving vapor-liquid equilibria.

Raoult's Law. The molar composition of a liquid phase (ideal solution) in equilibrium with its vapor at any temperature T is given by

$$x_i = P_i/P_i^*(T) \qquad i = 1, 2, \ldots, N$$

If the vapor phase behaves as an ideal gas mixture, then by Dalton's law of partial pressures,

$$y_i = P_i/P = x_i P_i^*(T)/P \qquad i = 1, 2, \ldots, N$$

where x_i and y_i are the respective mole fractions of component i in the liquid and vapor phases, $P_i^*(T)$ is the equilibrium vapor pressure of pure liquid i at system temperature T, P_i is the partial pressure of i in the vapor phase, and P is the total pressure of the vapor phase. Each component i is distributed between the two phases to an extent dictated by the relative volatilities of the components at the system temperature. Note that $\Sigma x_i = 1$, $\Sigma y_i = 1$, and $P = \Sigma x_i P_i^*$.

Raoult's law is strictly applicable to ideal liquid solutions at all compositions, pressures, and temperatures. In an ideal or perfect solution, the components are mutually miscible in all proportions and there are no volume or thermal changes upon mixing. Solutions that approach ideality include mixtures of nonpolar hydrocarbons belonging to a homologous family (e.g., paraffins). Whereas binary mixtures of propane-butane, n-hexane-n-heptane, benzene-toluene, etc., show ideal behavior over a range of compositions, nonideal solutions comprising a solvent (1) and a solute (2) obey Raoult's law only in the limit of $x_1 \rightarrow 1$ (i.e., in dilute solutions). Raoult's law finds use in molecular weight determination of nonvolatile solutes in dilute solutions, in estimation of equilibrium solubilities of noncondensable gases in nonpolar liquids, and in vapor-liquid equilibria calculations.

Molecular Weight Determination by Application of Raoult's Law. If a small amount (m_2 in grams) of a nonvolatile, nonionized substance (solute, 2) is dissolved in m_1 grams of a volatile liquid (solvent, 1), it experiences a lowering of vapor pressure from the pure solvent value (P_1^*) to the solution value (P) at the system temperature. This is a consequence of Raoult's law because the total vapor pressure of the dilute solution ($x_2 << 1$) is given by $P^* = x_1P_1^* + x_2P_2^* \doteq x_1P_1^*$.

(a) The *relative lowering of vapor pressure* of the solvent is then

$$\frac{P_1^* - P^*}{P_1^*} = \frac{\Delta P^*}{P_1^*} \approx x_2 = \frac{(m_2/M_2)}{(m_1/M_1) + (m_2/M_2)} \doteq \frac{m_2}{M_2}\frac{M_1}{m_1}$$

Measurement of P^* and P_1^* at one temperature enables determination of unknown M_2.

(b) Equivalently, the presence of the nonvolatile solute causes an *elevation of the boiling point* of the liquid from pure solvent value T_b to solution value T. The above equation is coupled with Clausius-Clapeyron equation to yield

$$\Delta T_b = T - T_b = \frac{-RT_b^2}{\lambda_v}\ell n(1 - x_2) \doteq \frac{RT_b^2}{\lambda_v}x_2 \doteq \frac{RT_b^2}{\lambda_v}\frac{M_1}{M_2}\frac{m_2}{m_1}$$

where λ_v is the molal latent heat of vaporization of pure solvent at its boiling point (T_b) and R is the universal gas constant. By measuring ΔT_b and T_b at the same total pressure, x_2 can be computed and hence M_2.

(c) An alternative procedure is to consider the presence of the solute as responsible for a *depression in the freezing point* of the solvent from the pure liquid value T_f to the solution value T. Application of Raoult's law together with Clausius-Clapeyron equation gives a similar result:

$$\Delta T_f = T_f - T \doteq \frac{RT_f^2 x_2}{\lambda_f} \doteq \frac{RT_f^2}{\lambda_f}\frac{M_1}{M_2}\frac{m_2}{m_1}$$

where λ_f is the molal latent heat of fusion of the solvent at T_f. Techniques (b) and (c) provide more accurate data and therefore better estimates for M_2 than method (a).

Example 2-40

A solution is prepared by dissolving 0.911 g of carbon tetrachloride in 50.00 g of benzene. (i) Calculate the freezing point depression of the solution if pure benzene has a fp of 5.53°C and a latent heat of fusion of 30.45 cal/g. (ii) What will be the elevation in boiling point of the solution if pure benzene has a nbp of 80.1°C and a latent heat of vaporization of 7,700 cal/g-mole [63].

Solution

MW of solute (CCl_4) = M_2 = 153.82 m_2 = 0.911 g

MW of solvent (C_6H_6) = M_1 = 78.12 m_1 = 50.00 g

$$x_2 = \frac{(m_2/M_2)}{(m_1/M_1)+(m_2/M_2)} = 0.917\times10^{-2}$$

(i) Because $x_2 << 1$, the solution is dilute and Raoult's law may be applied

$$\Delta T_f = T_f - T \doteq \frac{RT_f^2}{\lambda_f} x_2$$

With R = 1.987 cal/g-mole, T_f = 273.2 + 5.53 = 278.73°K, and λ_f = (30.45)(78.12) cal/g-mole, we obtain ΔT_f = 0.595°C, which compares well with an experimental observation of 0.603°C.

(ii) With T_b = 273.2 + 80.1 = 353.3°K, λ_v = 7,700 cal/g-mole, we get

$$\Delta T_b = T - T_b = \frac{RT_b^2 x_2}{\lambda_v} = 0.295°C$$

Henry's Law. This is an empirical formulation that describes equilibrium solubilities of noncondensable gases in a liquid when Raoult's law fails. It states that the mole fraction of a gas (solute i) dissolved in a liquid (solvent) is proportional to the partial pressure of the gas above the liquid surface at given temperature. That is,

$$x_i = P_i/H_i(T)$$

where the constant of proportionality $H_i(T)$ is known as Henry's law constant, with units of pressure per mole fraction. It is a characteristic of the gas-liquid system and increases with T. Experimentally determined values of H_i are available in the standard references for various gas-liquid systems [61,62,65,66]. If the gas phase is assumed to be ideal, then the equilibrium mole fraction of component i in the gas phase is

$$y_i = P_i/P = x_iH_i(T)/P$$

Henry's law is a reasonable approximation in the absence of gas-liquid reactions when total pressure is low to moderate. Deviations are usually manifested in the form of H_i dependence on P and phase compositions.

Example 2-41

A gas mixture has a molal composition of 20% H_2S, 30% CO_2 and 50% N_2 at 20°C and a total pressure of 1 atm. Use Henry's law to calculate the volumes of these gases that may be dissolved in 1,000 lb of water at equilibrium.

Solution

By Henry's law, the partial pressure of solute i in the gas phase is $P_i = H_i(T)x_i$, where x_i is the mole fraction of i in solution. Data on Henry's law constant are obtained from Chapter 14 of Perry and Chilton's *Chemical Engineers' Handbook* [62] for gas-water systems at 20°C.

Gas i:	H_2S	CO_2	N_2
H_i (atm/mole fraction):	4.83×10^2	1.42×10^3	8.04×10^4

Assuming ideal-gas law to hold, $P_i = y_iP$, where P = 1 atm and y_i = mole fraction of i in the gas phase. The equilibrium mole fraction x_i of gas i in solution is then given by

$$x_i = P_i/H_i = y_iP/H_i$$

Gas i	y_i	P_i (atm)	x_i	n_i (lb-moles)	V_i (ft^3)
H_2S	0.20	0.20	4.14×10^{-4}	2.30×10^{-2}	8.86
CO_2	0.30	0.30	2.11×10^{-4}	1.17×10^{-2}	4.52
N_2	0.50	0.50	6.22×10^{-6}	3.45×10^{-4}	0.13

The above table shows the rest of the required results for the number of moles n_i of each gas i present in the aqueous phase and the corresponding gas volume V_i dissolved in it. Since $x_i << 1$, the total moles of liquid is $n \approx (1000/18.02)$ lb-moles in 1000 lb of water and so $n_i = x_in$ can be calculated. At T = 20°C = 68°F = 528°R and P = 1 atm, the molal volume of the gas mixture is

$$\tilde{V} = \frac{RT}{P} = \frac{(0.7302)(528)}{(1)} = 385.55 \text{ ft}^3/\text{lb-mole}$$

The volume of each gas dissolved in 1,000 lb of water is then $V_i = n_i\tilde{V}$

Equilibrium Distribution Ratio or K factor. This is also termed *distribution coefficient* in the literature; it is a widely accepted method of describing vapor-liquid equilibria in nonideal systems. For any component i distributed between the vapor phase and liquid phase at equilibrium, the distribution coefficient or K factor is defined by

$$K_i = y_i/x_i = K_i(T,P)$$

The dimensionless K_i is regarded as a function of system T and P only and not of phase compositions. It must be experimentally determined. Reference 64 provides charts of K_i (T,P) for a number of paraffinic hydrocarbons. K_i is found to increase with an increase in system T and decrease with an increase in P. Away from the critical point, it is invariably assumed that the K_i values of component i are independent of the other components present in the system. In the absence of experimental data, caution must be exercised in the use of K-factor charts for a given application. The term *distribution coefficient* is also used in the context of a solute (solid or liquid) distributed between two immiscible liquid phases; y_i and x_i are then the equilibrium mole fractions of solute i in each liquid phase.

Example 2-42

The distribution coefficient for n-heptane (solute i) distributed between ethylene glycol (solvent 1) and benzene (solvent 2) at 25°C is given as the ratio of mass fractions

$(\omega_{i,1}/\omega_{i,2}) = K_i = 0.30$. Suppose 60 g of n-heptane are added to a solvent mixture of 600 g of ethylene glycol and 400 g of benzene. Assuming that the solvents are immiscible, determine the amount of n-heptane dissolved in each liquid phase at equilibrium.

Solution

Consider the equilibrium state of the system assuming that m grams of n-heptane are dissolved in the benzene phase. Then the mass fraction of n-heptane in this phase is $\omega_{i,2} = m/(400 + m)$.

Because (60 - m) g of i are present in the glycol phase, the equilibrium mass fraction of i in this phase is $\omega_{i,1} = (60 - m)/(600 + 60 - m)$. Thus,

$$\frac{\omega_{i,1}}{\omega_{i,2}} = K_i = 0.3 = \frac{(60-m)}{(660-m)}\frac{(400+m)}{m}$$

which must be solved for m. A quadratic equation in m is obtained by cross-multiplication and rearrangement of terms:

$$0.3m\,(660 - m) - (60 - m)(400 + m) = 0$$

or

$$0.7m^2 + 538m - 24{,}000 = 0$$

Its roots are

$$m = \frac{-538 \pm \sqrt{(538)^2 - 4(0.7)(-24{,}000)}}{2(0.7)}$$

Only the positive root is the physically meaningful value and so

$$m = (-538 + 597.197)/1.4 = 42.28 \text{ g}$$

$$\therefore \omega_{i,2} = 42.284/442.28 = 0.0956 \text{ in benzene phase}$$

and

$$\omega_{i,1} = (60 - 42.28)/617.72 = 0.0287 \text{ in glycol phase}$$

Check: Note that $w_{i,1}/w_{i,2} = 0.0287/0.0956 = 0.3$

Thermochemistry

Thermochemistry is concerned with the study of thermal effects associated with phase changes, formation of chemical compounds or solutions, and chemical reactions in general. The amount of heat (Q) liberated (or absorbed) is usually measured either in a batch-type bomb calorimeter at fixed volume or in a steady-flow calorimeter at constant pressure. Under these operating conditions, $Q = Q_v = \Delta U$ (net change in the internal energy of the system) for the bomb calorimeter, while $Q = Q_p = \Delta H$ (net change in the enthalpy of the system) for the flow calorimeter. For a pure substance, the thermodynamic properties U and H are functions of the state variables, viz. temperature (T) and pressure (P), and its state of aggregation [e.g., liquid (ℓ), gas or vapor (g), solid (s), etc.]. Here, we will briefly review certain basic definitions and terminology employed in the area of thermochemistry and consider some applications pertaining to combustion of fuels.

Heat of Reaction (ΔH_r). The heat of a chemical reaction carried out at constant pressure (P) is given by the difference between the total enthalpies of the reactants and products.

$$\Delta H_r = H_{products} - H_{reactants} = \sum_P n_j H_j - \sum_R n_i H_i$$

where the subscripts i and j refer to reactant i and product j, n represents the number of moles, and symbols Σ_R and Σ_P imply summations over all reactants (i = 1, 2, . . .) and all products (j = 1, 2, . . .), respectively. ΔH_r has units of calories or BTUs; its value depends on the amounts and physical states of the reactants and products as well as on the reaction conditions. Note that ΔH_r is *negative* for an *exothermic* reaction in which heat is spontaneously liberated and is *positive* for an *endothermic* reaction in which heat is absorbed from the surroundings.

Standard-State Enthalpy Changes ($\Delta H°$). To expedite calculations, thermochemical data are ordinarily presented in the form of standard-state enthalpy changes of the system $\Delta H°(T,P)$, with the requirement that materials start and end at the same temperature (T) and pressure (P) and in their standard states of aggregation, i.e.,

$$\Delta H^0(T,P) = H^0_{final}(T,P) - H^0_{initial}(T,P)$$

It has been traditional to choose the reference state as P = 1 standard atmosphere and T = 25°C (77°F) in expressing ΔH^0 values. Examples include standard heats of reaction ΔH^0_r, heats of formation ΔH^0_f, heats of combustion ΔH^0_c, heats of vaporization ΔH^0_v or λ_v, heats of solution ΔH^0_s, etc. To avoid confusion, the standard state of aggregation of each substance taking part in the thermochemical process must be specified by an appropriate letter symbol adjoining its chemical formula. The standard state for a gas is the ideal gas at 1 atm and specified T. The standard state for a solid is its stable crystalline form (e.g., rhombic sulfur) or amorphous form existing at the specified P and T. In the absence of such information, the normal state of aggregation of the material at given P and T is assumed. Tabulated values of standard-state enthalpy changes (ΔH^0) are readily available from a number of sources including handbooks and textbooks [59–63,65,66].

Standard Heat of Reaction. This is the standard enthalpy change accompanying a chemical reaction under the assumptions that the reactants and products exist in their standard states of aggregation at the same T and P, and stoichiometric amounts of reactants take part in the reaction to completion at constant P. With P = 1 atm and T = 25°C as the standard state, $\Delta H^0_r(T,P)$ can be written as

$$\Delta H^0_r(25°C, 1\ \text{atm}) = \sum_P \upsilon_j H^0_j(25°C, 1\ \text{atm}) - \sum_R \upsilon_i H^0_i(25°C, 1\ \text{atm})$$

Because v represents the stoichiometric coefficient of a given species, the value of ΔH^0_r clearly depends on the way the stoichiometric equation is written for the reaction. It is conventional to express the above equation in a simplified form as

$$\Delta H_r^0(25°C, 1\text{ atm}) = \sum_P \upsilon_j H_j^0 - \sum_R \upsilon_i H_i^0$$

where it is understood that H_i^0 and H_j^0 are to be evaluated at the reaction conditions of 1 atm and 25°C. Another common practice is to use the number of moles n_i (or n_j) in place of the stoichiometric coefficient v_i (or v_j).

Relation Between Q_p and Q_v. Let the heats of reaction measured at constant pressure and at constant volume be $Q_p = \Delta H_r(T,P)$ and $Q_v = \Delta U_r(T,P)$. Since $H = U + PV$, it follows that Q_p and Q_v are related by

$$\Delta H_r = \Delta U_r + P\Delta V$$

where ΔV = total volume of products − total volume of reactants.

If all of the species are gaseous and obey the ideal gas law, then

$$\Delta H_r = \Delta U_r + RT\Delta n$$

where $\Delta n = \Sigma_P n_j - \Sigma_R n_i$ = net change in the number of moles of the system.

The *standard heat of formation* (ΔH_f^0) of a chemical compound is the standard heat of reaction corresponding to the chemical combination of its constituent elements to form one mole of the compound, each existing in its standard state at 1 atm and 25°C. It has units of cal/g-mole.

$$\Delta H_f^0(1\text{ atm}, 25°C) = H^0_{\text{compound}} - H^0_{\text{elements}}$$

By convention, the standard-state enthalpies of the elements (H_i^0) are taken to be zero at 1 atm and 25°C so that $\Delta H_f^0 = H^0_{\text{compound}}$.

The *standard heat of combustion* (ΔH_c°) of a chemical substance (usually an organic compound) is the same as the standard heat of reaction for complete oxidation of 1 mole of the substance in pure oxygen to yield $CO_2(g)$ and $H_2O(\ell)$ as products. A reference state of 25°C and 1 atm is assumed in quoting standard heats of combustion in cal/g-mole. The value of ΔH_c° is always negative because combustion is an exothermic reaction. Note that the standard heats of combustion for carbon and hydrogen are the same as the heats of formation for $CO_2(g)$ and $H_2O(\ell)$, respectively.

Laws of Thermochemistry. Lavoisier and Laplace (1780) found that the heat required to decompose a chemical compound into its elements was numerically equal to the heat generated in its formation under the same conditions of T and P. That is, $\Delta H_d = -\Delta H_f$, where the subscript d refers to decomposition reaction [52, p. 24; 61, p. 303].

An important corollary of this postulate is known as *Hess's law of constant heat summation* (1840): The overall heat of a chemical reaction is the same whether the reaction occurs in a single step or multiple steps.

The two basic principles permit the algebraic manipulation of chemical reactions (represented by their stoichiometric equations and associated enthalpy changes) in order to achieve desired thermochemical results.

Applications. (1) Heats of formation data of reactants and products can be used to calculate the standard heat of a chemical reaction by applying Hess's law. Thus,

$$\Delta H_r^0 = \sum_P \upsilon_j \Delta H^0_{f,j} - \sum_R \upsilon_i \Delta H^0_{f,i}$$

where $\Delta H^0_{f,i}$ and $\Delta H^0_{f,j}$ are the standard heats of formation of reactant i and product j, respectively.

(2) For reactions involving only organic compounds as reactants, ΔH°_r can be determined using heats of combustion data.

$$\Delta H^0_r = \sum_R \upsilon_i \Delta H^0_{c,i} - \sum_P \upsilon_j \Delta H^0_{c,j}$$

These two approaches are useful when a direct measurement of ΔH_r is not possible because of experimental difficulties.

Example 2-43

Calculate the heat of reaction at the standard reference state (1 atm, 25°C) for

(i) $n\text{-}C_4H_{10}(g) \rightarrow C_2H_4(g) + C_2H_6(g)$

and

(ii) $CO_2(g) + H_2(g) \rightarrow CO(g) + H_2O(\ell)$

using heats of formation data. Also check results using heats of combustion data.

Solution

The required data are tabulated below in units of kcal/g-mole.

	$-\Delta H^0_f$	$-\Delta H^0_c$
$n\text{-}C_4H_{10}(g)$	29.812	687.982
$C_2H_4(g)$	−12.496	337.234
$C_2H_6(g)$	20.236	372.820
$CO_2(g)$	94.052	—
$CO(g)$	26.416	67.636
$H_2(g)$	0	68.317
$H_2O(\ell)$	68.317	—

Using heats of formation data, by Hess' law

$$\text{(i)}\ \Delta H^0_r = \Delta H^0_f(C_2H_4) + \Delta H^0_f(C_2H_6) - \Delta H^0_f(C_4H_{10})$$

$$= +12.496 - 20.236 - (-29.812)$$

$$= +22.072 \text{ kcal}$$

$$\text{(ii)}\ \Delta H^0_r = \Delta H^0_f(CO) + \Delta H^0_f(H_2O) - \Delta H^0_f(CO_2) - \Delta H^0_f(H_2)$$

$$= -26.416 - 68.317 - (-94.052) - 0$$

$$= -0.681 \text{ kcal}$$

Check: Using heats of combustion data,

(i) $\Delta H_r^0 = \Delta H_c^0(C_4H_{10}) - \Delta H_c^0(C_2H_4) - \Delta H_c^0(C_2H_6)$

$= (-687.982) - (-337.234) - (-372.820)$

$= +22.072$ kcal for first reaction

(ii) $\Delta H_r^0 = \Delta H_c^0(CO_2) + \Delta H_c^0(H_2) - \Delta H_c^0(CO) - \Delta H_c^0(H_2O)$

$= 0 - 68.317 - (-67.636) - 0$

$= -0.681$ kcal for second reaction

Thus, both methods yield identical results for the heats of reaction.

(3) Very frequently ΔH_f^0 data are available for inorganic substances but not for organic compounds for which ΔH_c^0 values are more readily available. Because ΔH_f^0 of hydrocarbons are not easily measurable, they are often deduced by Hess's law from known ΔH_c^0 of the hydrocarbon and known ΔH_c° values of the products of combustion.

Example 2-44

Find the standard heat of formation of benzene (ℓ) given the following heats of combustion data (in kcal/g-mole) at 1 atm and 25°C:

(i) $C(c) + O_2(g) \rightarrow CO_2(g)$ $\qquad \Delta H_c^0 = -94.05$

(ii) $H_2(g) + 1/2\ O_2(g) \rightarrow H_2O(\ell)$ $\qquad \Delta H_c^0 = -68.32$

(iii) $C_6H_6(\ell) + 15/2\ O_2(g) \rightarrow 6CO_2(g) + 3H_2O(\ell)$ $\qquad \Delta H_c^0 = -780.98$

Solution

The desired formation reaction is

$6C(c) + 3H_2(g) \rightarrow C_6H_6(\ell) \qquad \Delta H_f^0 = ?$

which is equivalent to [6 Equation (i) + 3 Equation (ii) – Equation (iii)].
By Hess' law,

$$\Delta H_f^0 = \sum_R \upsilon_i \Delta H_{c,i}^0 - \sum_P \upsilon_j \Delta H_{c,j}^0$$

$= 6(-94.05) + 3(-68.32) - 1(-780.98)$

$= -564.30 - 204.96 + 780.98 = 11.72$ kcal/g-mole

Effect of Temperature on the Heat of Reaction. It is possible to calculate the heat of a chemical reaction $\Delta H_r(T_1)$ at any temperature T_1 and pressure P, provided we know the standard heat of reaction $\Delta H_r^0(T_0)$ at reference conditions of T_0 and P (e.g., 25°C and 1 atm).

Referring to the schematic diagram in Figure 2-81, it is clear that the thermal energy balance should include terms ΔH_R and ΔH_p, which represent the sensible heats necessary to raise the temperatures of reactants and products from T_0 to T_1. Thus,

$$\Delta H_r^0(T_0) + \Delta H_P = \Delta H_R + \Delta H_r(T_1)$$

or

$$\Delta H_r(T_1) = \Delta H_r^0(T_0) + \Delta H_P - \Delta H_R$$

$$= \Delta H_r^0(T_0) + \sum_P \int_{T_0}^{T_1} \upsilon_j C_{p,j} dT - \sum_R \int_{T_0}^{T_1} \upsilon_i C_{p,i} dT$$

The integral terms representing ΔH_p and ΔH_R can be computed if molal heat capacity data $C_p(T)$ are available for each of the reactants (i) and products (j). When phase transitions occur between T_0 and T_1 for any of the species, proper accounting must be made by including the appropriate latent heats of phase transformations for those species in the evaluation of ΔH_R and ΔH_p terms. In the absence of phase changes, let $C_p(T) = a + bT + cT^2$ describe the variation of C_p (cal/g-mole °K) with absolute temperature T (°K). Assuming that constants a, b, and c are known for each species involved in the reaction, we can write

$$\Delta H_r(T_1) = \Delta H_r^0(T_0) + \alpha(T_1 - T_0) + \beta\frac{(T_1^2 - T_0^2)}{2} + \gamma\frac{(T_1^3 - T_0^3)}{3}$$

where the coefficients, α, β and γ are defined by

$$\alpha \equiv \sum_P \upsilon_j a_j - \sum_R \upsilon_i a_i$$

$$\beta \equiv \sum_P \upsilon_j b_j - \sum_R \upsilon_i b_i$$

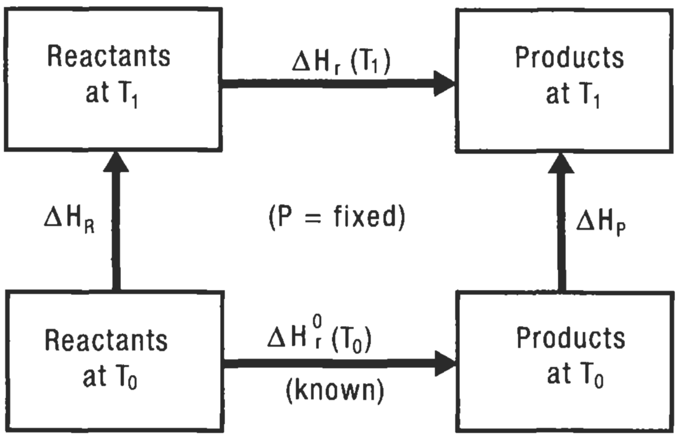


Figure 2-81. Schematic representation to calculate the heat of reaction at temperature T_1.

$$\gamma \equiv \sum_{P} \upsilon_j c_j - \sum_{R} \upsilon_i c_i$$

In a more compact notation, the same result becomes

$$\Delta H_r(T_1) = H_0 + \alpha T_1 + \frac{\beta T_1^2}{2} + \frac{\gamma T_1^3}{3}$$

where

$$H_0 \equiv \Delta H_r^0(T_0) - \left(\alpha T_0 + \frac{\beta T_0^2}{2} + \frac{\gamma T_0^3}{3} \right)$$

is a function of T_0 only.

Sometimes tabulated values of the *mean molal heat capacities* $\overline{C}_P(T)$ are more easily accessible than $C_p(T)$ data, with respect to a reference temperature of $T_0 = 25°C$ (see Table 2-45). Since $\overline{C}_P$ is defined over the range T_0 and T_1 by

$$\overline{C}_p = \frac{1}{(T_1 - T_0)} \int_{T_0}^{T_1} C_p(T)dT$$

we can rewrite the expression for $\Delta H_r(T_1)$ as

$$\Delta H_r(T_1) = \Delta H_r^0(T_0) + \left[\sum_{P} \upsilon_j \overline{C}_{p,j} - \sum_{R} \upsilon_i \overline{C}_{p,i} \right](T_1 - T_0)$$

Table 2-45
Mean Molal Heat Capacities of Gases Between 25°C and T°C
(Reference pressure = 0)
(cal/(g-mole)(°C)) [61, p. 258]

T(°C)	H_2	N_2	CO	Air	O_2	NO	H_2O	CO_2	HCl	Cl_2	CH_4	SO_2	C_2H_4	SO_3	C_2H_6
25	6.894	6.961	6.965	6.972	7.017	7.134	8.024	8.884	6.96	8.12	8.55	9.54	10.45	12.11	12.63
100	6.924	6.972	6.983	6.996	7.083	7.144	8.084	9.251	6.97	8.24	8.98	9.85	11.35	12.84	13.76
200	6.957	6.996	7.017	7.021	7.181	7.224	8.177	9.701	6.98	8.37	9.62	10.25	12.53	13.74	15.27
300	6.970	7.036	7.070	7.073	7.293	7.252	8.215	10.108	7.00	8.48	10.29	10.62	13.65	14.54	16.72
400	6.982	7.080	7.136	7.152	7.406	7.301	8.409	10.462	7.02	8.55	10.97	10.94	14.67	15.22	18.11
500	6.995	7.159	7.210	7.225	7.513	7.389	8.539	10.776	7.06	8.61	11.65	11.22	15.60	15.82	19.39
600	7.011	7.229	7.289	7.299	7.616	7.470	8.678	11.053	7.10	8.66	12.27	11.45	16.45	16.33	20.58
700	7.032	7.298	7.365	7.374	7.706	7.549	8.816	11.303	7.15	8.70	12.90	11.66	17.22	16.77	21.68
800	7.060	7.369	7.443	7.447	7.792	7.630	8.963	11.53	7.21	8.73	13.48	11.84	17.95	17.17	22.72
900	7.076	7.443	7.521	7.520	7.874	7.708	9.109	11.74	7.27	8.77	14.04	12.01	18.63	17.52	23.69
1000	7.128	7.507	7.587	7.593	7.941	7.773	9.246	11.92	7.33	8.80	14.56	12.15	19.23	17.80	24.56
1100	7.169	7.574	7.653	7.660	8.009	7.839	9.389	12.10	7.39	8.82	15.04	12.28	19.81	18.17	25.40
1200	7.209	7.635	7.714	7.719	8.068	7.898	9.524	12.25	7.45	8.94	15.49	12.39	20.33	18.44	26.15
1300	7.252	7.692	7.772	7.778	8.123	7.952	9.66	12.39							
1400	7.288	7.738	7.815	7.824	8.166	7.994	9.77	12.50							
1500	7.326	7.786	7.866	7.373	8.203	8.039	9.89	12.69							
1600	7.380	7.844	7.922	7.929	8.260	8.092	9.95	12.75							
1700	7.421	7.879	7.958	7.965	8.305	8.124	10.13	12.70							
1800	7.467	7.924	8.001	8.010	8.349	8.164	10.24	12.93							
1900	7.505	7.957	8.033	8.043	8.383	8.192	10.34	13.01							
2000	7.548	7.994	8.069	8.081	8.423	8.225	10.43	13.10							
2100	7.588	8.028	8.101	8.115	8.460	3.255	10.52	13.17							
2200	7.624	8.054	8.127	3.144	8.491	8.277	10.61	13.24							

In the event the mean molal heat capacity data are available with a reference temperature T_r other than T_0 = 25°C (see Table 2-46 for data with T_r = 0°C), the following equation can be used to calculate $\Delta H_r(T_1)$:

$$\Delta H_r(T_1) = \Delta H_r^0(T_0) + (T_1 - T_r)A - (T_0 - T_r)B$$

where

$$A, B = \sum_P \upsilon_j \overline{C}_{p,j} - \sum_R \upsilon_i \overline{C}_{p,i}$$

excepting that the mean heat capacities are evaluated between T_r and T_1 for term A and between T_r and T_0 for term B.

Effect of Pressure on ΔH_r. Consider a reaction carried out at reference temperature T = 25°C but at a constant pressure P_1 different from the initial standard-state pressure P_0 = 1 atm. The value of ΔH_r (P_1) can be found from an energy balance similar to the previous analysis:

$$\Delta H_r(P_1) = \Delta H_r^0(P_0) + \Delta H_P - \Delta H_R$$

The terms ΔH_p and ΔH_R now denote the enthalpy changes associated with the change of pressure from P_0 to P_1. Thus

Table 2-46
Mean Molal Heat Capacities of Combustion Gases Between 0°C and T°C
(Reference temperature = 0°C; pressure = 1 atm) [60, p. 273]
($\overline{C}_p$ in cal/(gm-mole)(°C)
(To convert to J/kg mole)(°K), multiply by 4184.)

T(°C)	N_2	O_2	Air	H_2	CO	CO_2	H_2O
0	6.959	6.989	6.946	6.838	6.960	8.595	8.001
18	6.960	6.998	6.949	6.858	6.961	8.706	8.009
25	6.960	7.002	6.949	6.864	6.962	8.716	8.012
100	6.965	7.057	6.965	6.926	6.973	9.122	8.061
200	6.985	7.154	7.001	6.955	7.050	9.590	8.150
300	7.023	7.275	7.054	6.967	7.057	10.003	8.256
400	7.075	7.380	7.118	6.983	7.120	10.360	8.377
500	7.138	7.489	7.190	6.998	7.196	10.680	8.507
600	7.207	7.591	7.266	7.015	7.273	10.965	8.644
700	7.277	7.684	7.340	7.036	7.351	11.221	8.785
800	7.350	7.768	7.414	7.062	7.428	11.451	8.928
900	7.420	7.845	7.485	7.093	7.501	11.68	9.070
1000	7.482	7.916	7.549	7.128	7.570	11.85	9.210
1100	7.551	7.980	7.616	7.165	7.635	12.02	9.348
1200	7.610	8.039	7.674	7.205	7.688	12.17	9.482
1300	7.665	8.094	7.729	7.227	7.752	12.32	9.613
1400	7.718	8.146	7.781	7.260	7.805	12.44	9.740
1500	7.769	8.192	7.830	7.296	7.855	12.56	9.86

$$\Delta H_r(P_1) = \Delta H_r^0(P_0) + \sum_P \int_{P_0}^{P_1} \left(\frac{\partial H_j}{\partial P}\right)_T dP - \sum_R \int_{P_0}^{P_1} \left(\frac{\partial H_i}{\partial P}\right)_T dP$$

where $\partial/\partial P$ denotes partial differentiation with respect to pressure P. For solids and liquids away from the critical point, the variation in enthalpy with pressure at constant T is quite small and, therefore, $\Delta H_r(P_1) \doteqdot \Delta H_r^0(P_0)$ is assumed under these circumstances. For gaseous reactants and products that follow ideal-gas law, H = H(T) only so that the effect of pressure is zero, i.e., $\Delta H_r(P_1) = \Delta H_r^0(P_0)$. In nonideal gas systems, the enthalpy changes are nonzero, but the effect is usually small up to moderate pressures.

Heating Values of Combustion Fuels. The calorific value or heating value (HV) of a fuel (usually a hydrocarbon) is the negative value of its standard heat of combustion at 1 atm and 25°C, expressed in cal/g or Btu/lb. It is termed *higher heating value* (HHV) if $H_2O(\ell)$ is a combustion product and is calculated as $HHV = (-\Delta H_c^0)/M$, where M is the molecular weight of the fuel. An appropriate ΔH_c° value must be used in referring to the *lower heating value* (LHV) based on $H_2O(g)$ as a combustion product. Both are related by

$$HHV = LHV + (\upsilon_W \lambda_v)/M$$

where υ_W is the stoichiometric coefficient for water in the combustion reaction of 1 mole of fuel, and λ_v is the molal latent heat of vaporization for water at 25°C and 1 atm = 10,519 cal/g-mole = 18,934 Btu/lb-mole.

For a fuel mixture composed of combustible substances i = 1, 2, . . . , the heating value is calculated as $HV = \sum \omega_i (HV)_i$, where ω_i is the mass fraction of the i^{th} substance having a heating value of $(HV)_i$.

Adiabatic Reaction Temperature (T_{ad}). The concept of adiabatic or theoretical reaction temperature (T_{ad}) plays an important role in the design of chemical reactors, gas furnaces, and other process equipment to handle highly exothermic reactions such as combustion. T_{ad} is defined as the final temperature attained by the reaction mixture at the completion of a chemical reaction carried out under adiabatic conditions in a closed system at constant pressure. Theoretically, this is the maximum temperature achieved by the products when stoichiometric quantities of reactants are completely converted into products in an adiabatic reactor. In general, T_{ad} is a function of the initial temperature (T_i) of the reactants and their relative amounts as well as the presence of any nonreactive (inert) materials. T_{ad} is also dependent on the extent of completion of the reaction. In actual experiments, it is very unlikely that the theoretical maximum values of T_{ad} can be realized, but the calculated results do provide an idealized basis for comparison of the thermal effects resulting from exothermic reactions. Lower feed temperatures (T_i), presence of inerts and excess reactants, and incomplete conversion tend to reduce the value of T_{ad}. The term *theoretical* or *adiabatic flame temperature* (T_{fl}) is preferred over T_{ad} in dealing exclusively with the combustion of fuels.

Calculation of T_{ad}. To calculate T_{ad} (or T_{fl} for a combustible fuel), we refer to Figure 2-82 and note that $Q = \Delta H = 0$ for the adiabatic reaction process. Taking 25°C (= 298.2°K) and 1 atm as the reference state, the energy balance can be expressed as

$$\Delta H = 0 = H_{products}(T_{ad}) - H_{reactants}(T_i)$$

$$= -\Delta H_R + \Delta H_r^0(25°C) + \Delta H_P$$

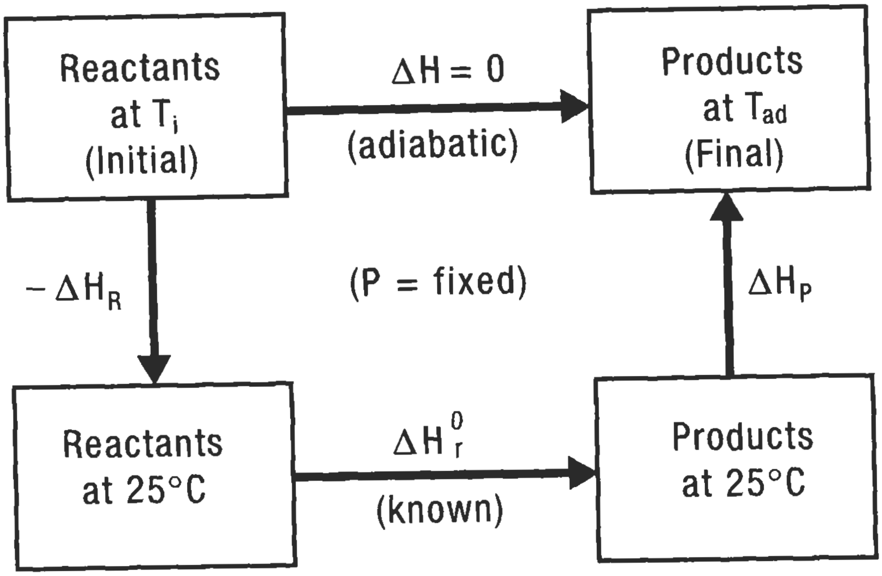


Figure 2-82. Schematic representation to calculate the adiabatic reaction temperature (T_{ad}).

or

$$\Delta H_P = -\Delta H_r^0 + \Delta H_R$$

where

$$\Delta H_R = \sum_R \int_{298}^{T_{ad}} n_i C_{p,i} dT$$

and

$$\Delta H_P = \sum_P \int_{298}^{T_{ad}} n_j C_{p,j} dT + \sum_P n_j \lambda_{v,j}$$

The second term on the right side of the expression for ΔH_p accounts for *any* phase changes that may occur between 25°C and T_{ad} for the final products; it should be deleted if not applicable. Using molal heat capacity data $C_p(T)$ for all the species present, the following equality is solved for T_{ad} by trial and error.

$$\sum_P \int_{298}^{T_{ad}} n_j C_{p,j} dT = -\Delta H_r^0 - \sum_P n_j \lambda_{v,j} + \sum_R \int_{298}^{T_i} n_i C_{p,i} dT$$

For instance, a quadratic expression for $C_p(T)$ will require the solution of a cubic equation in T_{ad}.

An alternative representation is useful when data on mean molal heat capacities $\overline{C}_p(T)$ are available with 25°C as the reference temperature (T_0). Then the equation

$$\sum_P n_j \overline{C}_{p,j}(T_{ad} - 298) = -\Delta H_r^0 - \sum_P n_j \lambda_{v,j} + \sum_R \overline{C}_{p,i}(T_i - 298)$$

can be solved for T_{ad}, though it still requires a trial-and-error procedure.

Example 2-45

A natural gas having the volumetric composition of 90% methane, 8% ethane, and 2% nitrogen at 1 atm and 25°C is used as fuel in a power plant. To ensure complete combustion 75% excess air is also supplied at 1 atm and 25°C. Calculate (i) the lower and higher heating values of the fuel at 25°C and (ii) the theoretical maximum temperature in the boiler assuming adiabatic operation and gaseous state for all the products.

Solution

Assuming ideal gas behavior at 1 atm and 25°C, the volume composition is identical with the mole composition. Choose a *basis* of 1 g-mole of natural gas (at 1 atm and 25°C) which contains 0.90 g-mole of CH_4 (MW = 16.04), 0.08 g-mole of C_2H_6 (MW = 30.07), and 0.02 g-moles of N_2 (MW = 28.01). Then the molecular weight of the natural gas is

$$M = \sum y_i M_i = 0.90(16.04) + 0.08(30.07) + 0.02\,(28.01)$$

$$= 17.40 \text{ g/g-mole}$$

(i) The combustion reactions of interest are

$$CH_4(g) + 2O_2(g) \rightarrow CO_2(g) + 2H_2O(\ell) \qquad \Delta H_C^0 = -212.80$$

$$C_2H_6(g) + \frac{7}{2}O_2(g) \rightarrow 2CO_2(g) + 3H_2O(\ell) \qquad \Delta H_C^0 = -372.82$$

where the standard heat of combustion Δh_c° at 1 atm and 25°C are stated in kcal/g-mole.

With $H_2O(\ell)$ as a product, the higher heating value (HHV) of the fuel is calculated as

$$HHV = 0.9\left[-\Delta H_C^0(CH_4)\right] + 0.08\left[-\Delta H_C^0(C_2H_6)\right]$$

$$= 0.9(212.80) + 0.08(372.82) = 221.35 \text{ kcal/g-mole}$$

$$= \frac{221.35}{17.40} = 12.72 \text{ kcal/g}$$

Note that $H_2O(\ell) \rightarrow H_2O(g)$ at 1 atm and 25°C has $\lambda_v = \Delta H_v^\circ = 10.519$ kcal/g-mole. With $H_2O(g)$ as a combustion product, the lower heating value (LHV) of the fuel is

$$LHV = HHV - n_w\lambda_v$$

$$= 221.35 - [0.9(2) + 0.08(3)](10.519) \text{ kcal/g-mole}$$

$$= 199.89 \text{ kcal/g-mole} = 11.49 \text{ kcal/g}$$

(ii) The adiabatic flame temperature T_{fl} must be calculated with all products in the gaseous state. So the appropriate standard heat of reaction at 1 atm and 25°C is the heat of combustion of the fuel with $H_2O(g)$ as a product, i.e., the negative of LHV.

$$\therefore \Delta H_r^0 = -(LHV) = -199.89 \text{ kcal/g-mole}$$

$$\text{Stoichiometric amount of } O_2 \text{ required} = 2(0.9) + \frac{7}{2}(0.08)$$

$$= 2.08 \text{ g-moles}$$

Because 75% excess air is supplied, amount of O_2 supplied = 1.75(2.08) = 3.64 g-moles.

Assuming dry air to contain 79% N_2 and 21% O_2 by moles, N_2 supplied = (3.64)(0.79)/0.21 = 13.693 g-moles.

The mole compositions of the feed gases entering (reactants i) at 25°C and of the products leaving the boiler at T_{fl} are tabulated below:

Feed at 25°C			Products at T_{fl}		
Gas i	n_i (g-moles)		Gas j	n_j (g-moles)	
CH_4	0.90		CO_2	0.9(1) + 0.08(2) =	1.06
C_2H_6	0.08		H_2O	0.9(2) + 0.08(3) =	2.04
N_2	0.02	13.713			
N_2 in air	13.693		N_2		13.373
Stoich. O_2	2.08	3.64			
Excess O_2	1.56		O_2		1.56
Total	18.333			Total	18.373

Applying the heat balance equation, $\Delta H = 0 = \Delta H_r^0 - \Delta H_R + \Delta H_P$, we realize that $\Delta H_R = 0$ because the feed gases are supplied at the reference temperature of 25°C = 298°K.

$$\therefore \Delta H_P = \sum_P \int_{298}^{T_{fl}} n_j C_{p,j} dT = -\Delta H_r^0 = +199{,}890 \text{ cal}$$

must be solved for unknown T_{fl}.

Method 1. Use $C_p(T)$ data for product gases from Himmelblau [60, pp. 494–497]. The equation is $C_p = a + bT + cT^2$, with T in °K and C_p in cal/g-mole °K.

Gas	a	b × 10²	c × 10⁵	Applicable range (°K)
CO_2(g)	6.393	1.0100	−0.3405	273–3700
H_2O(g)	6.970	0.3464	−0.0483	
N_2(g)	6.529	0.1488	−0.02271	273–3700
O_2(g)	6.732	0.1505	−0.01791	273–3700

The equation for T_{fl} now becomes

$$199{,}890 = \alpha(T_{fl} - 298) + \frac{\beta}{2}(T_{fl}^2 - 298^2) + \frac{\gamma}{3}(T_{fl}^3 - 298^3)$$

where $\alpha = \sum_P n_j a_j$

$$= 1.06(6.393) + 2.04(6.970) + 13.713(6.529) + 1.56(6.732)$$

$$= 6.777 + 14.219 + 89.532 + 10.502 = 121.029$$

Similar calculations yield

$$\beta = \sum_P n_j b_j = 4.053 \times 10^{-2}$$

and

$$\gamma = \sum_P n_j c_j = -0.799 \times 10^{-5}$$

A trial-and-error procedure is needed to solve the cubic equation in T_{fl}:

$$199{,}890 = 121.029(T_{fl} - 298) + 2.027 \times 10^{-2}(T_{fl}^2 - 298^2)$$
$$- 0.266 \times 10^{-5}(T_{fl}^3 - 298^3)$$

As a first trial let T_{fl} = 1,500°C = 1773°K. Then

RHS = 178,518 + 61,909 − 14,774 = 225,652 cal (too high)

Try T_{fl} = 1,400°C = 1,673°K as a second trial value. Then

RHS = 166,415 + 54,925 − 12,402 = 206,566 (high)

Similar calculations show that with T_{fl} = 1345°C = 1618°K, the RHS = 159,759 + 51,257 − 11,212 = 199,804 which is very close to LHS = 199,890

∴ The solution is $T_{fl} \doteq 1345°\text{C}$.

Method 2. Use mean heat capacity data from Table 2-45 with reference conditions of P = 0 and T = 25°C for the combustion gases. Then the equation for T_{fl} becomes

$$199{,}890 = \sum_P n_j \overline{C}_{p,j}(T_{fl} - 25)$$

$$= \left[1.06\overline{C}_p(CO_2) + 2.04\overline{C}_p(H_2O) + 13.713\overline{C}_p(N_2) + 1.56\overline{C}_p(O_2)\right] \times (T_{fl} - 25)$$

With a first trial value of T_{fl} = 1400°C, we have

$$RHS = [1.06(12.50) + 2.04(9.77) + 13.713(7.738) + 1.56(8.166)](1400 - 25)$$

$$= (152.031)(1375) = 209{,}043 \text{ cal} \quad \text{(high)}$$

For the second trial, try T_{fl} = 1350°C. Then

$$RHS = [1.06(12.445) + 2.04(9.715) + 13.713(7.715) + 1.56(8.145)](1325) = 200{,}754 \text{ cal.}$$

which is nearer to the LHS. Thus, using interpolated values of $\overline{C}_p$ between the temperatures 1300°C and 1400°C, we find good agreement between RHS = 199,927 and LHS = 199,890 at T_{fl} = 1345°C.

Method 3. Use mean molal heat capacity data from Table 2-46 with reference conditions of 0°C and 1 atm. Then the equation for T_{fl} is

$$199{,}890 = \sum_P n_j \overline{C}_{p,j}(T_{fl} - 0) - \sum_P n_j \overline{C}_{p,j}(25 - 0)$$

or

$$\sum_P n_j \overline{C}_{p,j} T_{fl} = 199{,}890 + [1.06(8.716) + 2.04(8.012) + 13.713(6.960) + 1.56(7.002)](25)$$

$$= 199{,}890 + 3{,}299 = 203{,}189 \text{ cal}$$

which can be written as

$$203{,}189 = [1.06\overline{C}_p(CO_2) + 2.04\overline{C}_p(H_2O) + 13.713\overline{C}_p(N_2) + 1.56\overline{C}_p(O_2)]T_{fl}$$

Again, assume a trial value of T_{fl} = 1400°C for which

$$RHS = [1.06(12.44) + 2.04(9.74) + 13.713(7.718) + 1.56(8.146)](1400)$$

$$= 212{,}240 \quad \text{(high)}$$

The results obtained with other trial values are shown below:

2nd trial: T_{fl} = 1350°C RHS = 203,855

3rd trial: T_{fl} = 1345°C RHS = 203,034

Because the last trial gives a close agreement with the LHS value of 203,189 cal, we take the solution to be $T_{fl} \doteq 1{,}345°C$. It is concluded that all of the three methods of calculations provide the same final answer.

ENGINEERING DESIGN

Introduction

The Petroleum Engineer

Petroleum engineers are traditionally involved in activities known in the oil industry as the "front end" of the petroleum fuel cycle (petroleum is either liquid or gaseous hydrocarbons derived from natural deposits–reservoirs–in the earth). These front end activities are namely exploration (locating and proving out the new geological provinces with petroleum reservoirs that may be exploited in the future), and development (the systematic drilling, well completion, and production of economically producible reservoirs). Once the raw petroleum fluids (e.g., crude oil and natural gas) have been produced from the earth, the "back end" of the fuel cycle takes the produced raw petroleum fluids and refines these fluids into useful products.

Because of the complex interdisciplinary nature of the engineering activities of exploration and development, the petroleum engineer must be conversant with fundamentals of designing devices and systems particular to the petroleum industry [67,68]. The petroleum engineer must be competent in skills related to engineering design. Design involves planning, development, assembly and implementation of plans to achieve a prescribed and specified result.

Multidisciplinary Team

The exploration activities directed at locating new petroleum-producing provinces are cost-intensive operations. The capital expended in the search for new petroleum-producing provinces is always at risk because there are no guarantee that such searches will be economically successful. There is no way to actually "know" if crude oil or natural gas is present in a particular geologic formation except to drill into the formation and physically test it. Highly sophisticated geologic and geophysical methods can be used to identify the possible subsurface geological conditions that might contain crude oil and/or natural gas. However, the final test is to drill a well to the rock formation (reservoir) in question and physically ascertain whether it contains petroleum, and if it does, ascertain if the petroleum can be produced economically. Thus, in the early part of the exploration phase geologists, geochemists, geophysicists and petroleum engineers must form teams in order to carry out effective investigations of possible petroleum producing prospects [68]. In general, such teams are initially driven by the geologic, geochemical and geophysical sciences that are to be used to infer possible subsurface locations for new deposits of petroleum. However, once the subsurface locations have been identified, the process of discovery becomes driven by the necessity to drill and complete a test well to the prospective reservoir. The skills of the exploration geologist are important for successful drilling. These exploration (wildcat) wells are usually drilled in remote locations where little or no previous subsurface engineering experience is available, thus, they are inherently risky operations.

Development of new proven petroleum resources are also cost-intensive operations, but generally lack the high risk of nearly total capital loss that exists in exploration activities. Development operations begin after the exploration test wells have proven that crude oil and/or natural gas can be economically recovered from a new reservoir. These operations require that numerous development (or production) wells be drilled and completed into proven resources. Production equipment are placed in these wells and the crude oil and/or natural gas is recovered and transported to refineries (in the back of the fuel cycle). The placement of these wells and the selection of their respective production rates within a particular reservoir or field must be planned carefully. The pattern of well locations and production rates should be designed so that the largest fraction of the in-place petroleum can eventually be recovered even though the maximum recoverable fraction of the oil or natural gas in place will not reach 100 percent. As developmental wells are drilled and completed, the production geologists ensure that wells are drilled to correct subsurface targets.

The development effort is largely a engineering activity requiring a great deal of planning. This planning requires increased interaction of multiple engineering disciplines to assure that the wells are drilled, completed and produced in a safe, efficient and economical manner.

Engineering Arts and Sciences

There are three basic engineering fields of knowledge. These are called the engineering arts and sciences. They are the mechanical arts and sciences, the chemical arts and sciences, and the electrical arts and sciences. Every specific engineering discipline, such as petroleum engineering, utilizes one of the basic engineering fields of knowledge as their scholarly foundation.

The mechanical arts and sciences are based on classical physics (or Newtonian mechanics). The chemical arts and sciences are based on classical and modern chemistry. The electrical arts and sciences are based on modern particle physics. Table 2-47 gives a listing of the engineering disciplines that usually have the mechanical arts and sciences as their foundation. Table 2-48 gives a listing of the engineering disciplines that usually have the chemical arts and sciences as their foundation. And Table 2-49 gives a listing of the engineering disciplines that usually have the electrical arts and sciences as their foundation.

Note that a few disciplines such as petroleum engineering, environmental engineering and nuclear engineering are listed under two of these general engineering fields. At one engineering university, for example, the petroleum engineering curriculum may have a mechanical arts and sciences foundation. At another university, the petroleum engineering curriculum could have a chemical arts and sciences foundation. However, most petroleum engineering curricula are based on the

Table 2-47
Mechanical Arts and Sciences Engineering Disciplines

• Mechanical Engineering	• Environmental Engineering
• Civil Engineering	• Engineering Mechanics
• Aeronautical and Aerospace Engineering	• Engineering Science
• Materials and Metallurgical Engineering	• Industrial Engineering
• Mining Engineering	• Nuclear Engineering
• Petroleum Engineering	• Fire Protection Engineering
• Marine Engineering	• Engineering Physics
• Ocean Engineering	

Table 2-48
Chemical Arts and Sciences Engineering Disciplines

• Chemical Engineering
• Petroleum Engineering
• Environmental Engineering
• Nuclear Engineering

Table 2-49
Electrical Arts and Sciences Engineering Disciplines

• Electrical Engineering
• Electronics Engineering
• Engineering Physics

mechanical arts and sciences. Petroleum engineering curricula that are based on the mechanical arts and sciences usually have a balanced emphasis regarding drilling and completions, production and reservoir engineering subjects. Curricula that are based on the chemical arts and sciences generally concentrate on reservoir engineering subjects.

Similar situations occur with curricula in environmental engineering and in nuclear engineering.

Scientific and Engineering Philosophies

Scientific Method

The foundation philosophies of science and engineering are quite different. In the historical sense, it is obvious that scientific inquiry came first. In order for humans to "engineer," they had to be able to predict certain potentially useful physical phenomena. Thus, the scientific method evolved to aid humans in systematically discovering and understanding natural phenomena. The scientific method is a set of prescribed steps that are used to assist in understanding hitherto unknown natural phenomena. Table 2-50 gives a listing of the basic steps of the scientific method. The scientific method is an iterative procedure and in modern settings requires the development of a carefully devised plan of work. Such a plan requires both experimentation and analysis. It is necessary for the scientist to keep careful records in order to assure that little will be missed and that other investigators will be able to carry out any follow-on work. It is important to understand that the act of seeking pure knowledge is mutually exclusive of any desire to apply the gained knowledge to anything useful. In essence, the scientist seeks a unique and comprehensive understanding (or solution) of phenomena.

Engineering Method

The scientific method has been successfully used throughout human history to enlighten us to our natural world and beyond. Since the earliest days there have been

Table 2-50
Scientific Method

• Observe a phenomenon
• Postulate a theory to explain the phenomenon
• Develop and conduct an experiment to test the validity of the theory
• Using the test results, draw conclusions as to the validity of the theory
• Re-postulate the theory in light of these conclusions
• Iterate the above steps and continue to refine theory

individuals who desired to use scientific knowledge for practical use within their respective societies. The modern engineer would have little trouble recognizing these individuals as our forerunners.

Modern engineering has its formal roots in France where, in 1675, Louis XIV established a school for a corps of military engineers. Shortly afterward, early in the reign of Louis XV, a similar school was created for civilian engineers [69]. These original engineering schools were created to provide trained individuals who could improve the well-being of French society by creating an economical infrastructure of water, sewer and transportation systems. By the time modern engineering was established in the United States, the idea of the dependence of engineering practice on scientific knowledge was well established. This is illustrated by the original charter of the Institution of Civil Engineers in 1828 (the first professional engineering society in the United States, know known as the American Society of Civil Engineers) by the following statement . . .

> ". . . .the art of directing the great sources of power in Nature for the use and convenience of man. . . ."

In recent years there has been a greater appreciation of the word "use." As society and engineers have become more aware of the systems or process aspects of nature, it has become clear that the use of natural resources must stop short of abuse.

In early engineering practice, products (devices or systems) evolved slowly with step-by-step improvements made after studying the results of actual usage. In those early days, engineering judgments were made by intuition (i.e., parts are strong enough, etc.), and engineering design was the same as engineering drawing. As society's needs for new products expanded and as each product become more complicated, it also became necessary to develop an engineering methodology that would assure economic and systematic development of products. In time, the engineering community developed the engineering method. Table 2-51 lists the basic steps of the engineering method.

The unique characteristics of the engineering method are:

1. the engineering method is initiated by the desire to create a product that will meet the needs of society, and
2. the created product will be economically affordable and "safe" for use by society.

However, like the scientific method, the engineering method is an interative process and embedded in the engineering method is the systematic use of the scientific method itself to predict behavior of a prototype device or process. This means both methods require the use of trial and error. The interdependency of trail and error was explained by someone who said." One cannot have trail and error without error."

Table 2-51
Engineering Method

- A problem is recognized or a need is identified.
- Establish performance specifications (based on the details of the need stated above)
- Describe the preliminary characteristics of the life-cycle that the product design** should have (i.e., relative to production, distribution, consumption and retirement)
- Create first prototype design
- Evaluate feasibility of first prototype design from both the technical and economic point-of-view and modify first prototype design appropriately
- Evaluate performance characteristics of first prototype using models (usually both analytic and physical) and using the results, modify first prototype design appropriately
- Fabricate first prototype
- Test first prototype
- Iterate the above process until society is satisfied and accepts the product
- Plan for the maintenance and retirement of the product through its life-cycle

**The product in petroleum engineering may be as large and as inclusive as an entire oil field or it may be as small as a component of a machine or a process used in production.

Since the desire of society* to improve its condition will likely always exceed its complete scientific and economic knowledge to do so, the engineer must learn to deal with ambiguity—physical as well as societal. Ambiguities in engineering will always result in several useful engineering solutions (or alternative solutions) that can meet a perceived need. Thus, the engineering method, by its very nature, will yield alternative solutions. Society is then required to select one or more of those solutions to meet its need. However, society may decline all of the alternative solutions if the long term results of applying these solutions are unknown or if they have known detrimental outcomes. This was the situation in the United States and in other countries regarding the commercial application of nuclear energy.

The "Art" of Engineering Design

Engineering design is a creative act in much the same way that the creation of fine art is a creative act. This is why the three general fields of knowledge are described as "arts and sciences." This description acknowledges that engineering activity is a combination of creative "art" and knowledge in science. The act of design is directed at the creation of a product (for society's use) which can be, in broad terms, either a device or a system.

Probably the most difficult part of engineering design is the "definition of the problem." This part of engineering design is described by the first three steps of Table 51 (i.e., the engineering method). To define a problem for which the engineer may design several alternatives solutions requires extensive study of society's need and the analysis of applicable technologies that may be brought to bear. The most important aspect of engineering design is the assessment of whether society will accept or reject the engineer's proposed designs. Modern engineering practice is

*"Society" is used in a generic sense here. Depending on context, society may mean society at large. It may mean a local community, corporation, operating division of a company or corporation, or it may mean the immediate association of employees or personnel.

littered with engineered "solutions" to the wrong problem. Such situations emphasize the need for engineers to assure the correct problem is being solved.

Device Design

A device is a product having all major parts are essentially made by a single manufacture. Thus, the device design and its manufacturing operation are under the complete control of the designer. This definition does not dictate a particular product size or complexity, but does infer that such a product will have limited size and complexity.

The creation of these device designs is somewhat abstract with regards to providing products for the "needs of society." The device designs are in support of an industry which is vital to society's operation. Thus, the designer of a device for the petroleum industry designs, in a sense, for society's needs. In the petroleum industry, suppliers of hardware and services for operating companies function more in the area of designing devices than in the design and supply of systems (or processes) that produce natural petroleum resources.

System Design

A system design is a product that is made up of a combination of devices and components. As described above the devices within a system are under the control of the designer and are designed specifically for the system. Components, on the other hand, are other devices and/or subsystems which are not made to the specification of the system designer. Usually these components are manufactured for a number of applications in various systems. Thus, the system design and the fabrication of the system are under the control of the system designer. The definition of a system infer complexity in design and operation.

In the petroleum industry, the term system and process are often considered synonymous. This is basically because engineering designers in the industry create "systems" that will carry out needed operational "processes" (e.g., waterflood project).

Role of Models

Once the problem has been defined and the decision has been made to proceed with a design effort (device or system), the conceptual designs and/or parts of the device or system are modeled, both mathematically and physically. Engineers design by creating models that they can visualize, think about, and rearrange until they believe they have an adequate design concept for practical application. Therefore, modeling is fundamental to successful engineering design efforts and is used to predict the operating characteristics of a particular design concept in many operational situations.

Mathematical Models. In modern engineering practice it is usually possible to develop mathematical models that will allow engineers to understand the operational characteristics of a design concept without actually fabricating or testing the device or system itself. Thus, analysis (using mathematical models) usually allows for great economic savings during the development of a device or system. To carry out such analyses, it is necessary to separate a particular design concept into parts—each of which being tractable to mathematical modeling. Each of these parts can be analyzed to obtain needed data regarding the operating characteristics of a design concept. Although repetitive analyses of models are exceedingly powerful design tools, there is always the risk that some important physical aspect of a design will be overlooked by the designer. This could be a hitherto unknown phenomenon, or an ignored phenomenon, or simply an interaction of physical phenomenon that the separate analysis models can not adequately treat.

Physical Models. At some point in the design process it is usually necessary to create physical models. Often these physical models are of separate parts of the design concept that have defied adequate mathematical modeling. But most often physical models are created to assist in understanding groups or parts of a design concept that are known to have important interactions. As in mathematical modeling, physical modeling usually requires that the design concept be separated into parts that are easily modeled. Physical modeling (like mathematical modeling) allows the engineer to carry out economic small scale experiments that give insight into the operational characteristics of the design concept. Physical modeling must be carried out in accordance to certain established rules. These rules are based on principals given in the technique known as dimensional analysis (also known as Lord Rayleigh's method) or in a related technique known as the Buckingham Π theorem [70,71,72].

Dimensional Analysis. In the design of rather simple devices or systems, dimensional analysis can be used in conjunction with physical model experimental investigations to gain insight into the performance of a particular design concept. It is usually possible to define the performance of a simple device or system with a certain number of well chosen geometric and performance related variables that describe the device or system. Once these variables have been selected, dimensional analysis can be used to:

1. Reduce the number of variables by combining the variables into a few dimensionless terms. These dimensionless terms can be used to assist in the design of a physical model that, in turn, can be used for experimentation (on an economic scale) that will yield insight into the prototype device or system.
2. Develop descriptive governing relationships (in equation and/or graphical form) utilizing the dimensionless terms (with the variables) and the model experimental results that describe the performance of the physical model.
3. Develop device or system designs utilizing the descriptive governing relationships between the dimensionless terms (and subsequently between the variables) that are similar to those of the tested physical model, but are the dimensional scale of the desired device or system.

Dimensional analysis techniques are especially useful for manufacturers that make families of products that vary in size and performance specifications. Often it is not economic to make full-scale prototypes of a final product (e.g., dams, bridges, communication antennas, etc.). Thus, the solution to many of these design problems is to create small scale physical models that can be tested in similar operational environments. The dimensional analysis terms combined with results of physical modeling form the basis for interpreting data and development of full-scale prototype devices or systems. Use of dimensional analysis in fluid mechanics is given in the following example.

Example 1

To describe laminar flow of a fluid, the unit shear stress τ is some function of the dynamic viscosity $\mu(\text{lb/ft}^2)$, and the velocity difference $dV(\text{ft/sec})$ between adjacent laminae that are separated by the distance $dy(\text{ft})$. Develop a relationship for τ in terms of the variables μ, dV and dy.

The functional relationship between τ and m, dV and dy can be written as

$$\tau = K(\mu^a\, dV^b\, dy^c)$$

Using Table 2-52 the dimensional form of the above is

$$(FL^{-2}) = K\,(FL^{-2}T)^a\,(LT^{-1})^b\,(L)^c$$

Table 2-52
Dimensions and Units of Common Variables

		Dimensions		Units	
Symbol	**Variable**	**MLT**	**FLT**	**USCU***	**SI**
		Geometric			
L	Length	L		ft	m
A	Area	L^2		ft^2	m^2
V	Volume	L^3		ft^3	m^3
		Kinematic			
τ	Time	T		s	s
ω f	Angular velocity Frequency	T^{-1}		s^{-1}	s^{-1}
V	Velocity	LT^{-1}		ft/s	m/s
υ	Kinematic viscosity	L^2T^{-1}		ft^2/s	m^2/s
Q	Volume flow rate	L^3T^{-1}		ft^3/s	m^3/s
α	Angular acceleration	T^{-2}		s^{-2}	s^{-2}
a	Acceleration	LT^{-2}		ft/s^{-2}	m/s^2
		Dymanic			
ρ	Density	$ML\text{–}1$	$FL^{-4}T^2$	$slug/ft^3$	kg/m^3
M	Mass	M	$FL^{-1}T^2$	slugs	kg
I	Moment of inertia	ML^2	FLT^2	$slug•ft^2$	$kg•m^2$
μ	Dynamic viscosity	$ML^{-1}T^{-1}$	$FL^{-2}T$	slug/ft/s	kg/m/s
M	Mass flow rate	MT^{-1}	$FL^{-1}T^{-1}$	slug/s	kg/s
MV Ft	Momentum Impulse	MLT^{-1}	FT	lbf•s	N•s
$M\omega$	Angualar momentum	ML^2T^{-1}	FLT	$slug•ft^2/s$	$kg•m^2/s$
γ	Specific weight	$ML^{-2}T^{-2}$	FL^{-3}	lbf/ft^3	N/m^3

Table 2-52 (continued)

p τ E	Pressure Unit shear stress Modulus of elasticity	$ML^{-1}T^{-2}$	lbf/ft^2	N/m^2	
σ	Surface tension	MT^{-2}	FL^{-1}	lbf/ft	N/m
F	Force	MLT^{-2}	F	lbf	N
E W FL	Energy Work Torque	ML^2T^{-2}	FL	lbf/ft	J
P	Power	ML^2T^{-3}	FLT^{-1}	lbf•ft/s	W
υ	Specific volume	$M^{-1}L4T^{-2}$	ft^3/lbm	m^3/kg	

* United States customary units.

The equations for determining the dimensional exponents a, b and c are

Force F: $1 = a + 0 + 0$

Length L: $-2 = -2a + b + c$

Time T: $0 = a - b + c$

Solving the above equations for a, b and c yields

$a = 1$

$b = 1$

$c = -1$

Thus, inserting the exponent values gives

$\tau = K(\mu 1\ dV^1\ dy^{-1})$

which can be written as

$\tau = K[\mu\ (dV/dy)]$

where the constant K can be determined experimentally for the particular design application.

Buckingham Π. The Buckingham Π theorem is somewhat more sophisticated than the dimensional analysis technique. Although directly related to the dimensional analysis technique, Buckingham Π is usually used in design situations where there is less understanding of exactly which performance characteristics are ultimately important for prototype design. Also, the Buckingham Π theorem is generally more applicable to the design of complex devices or systems. In particular, Buckingham Π is usually used when the number of dimensionless terms needed to describe a device or system exceeds four. The fundamental theorem is . . .

> "if an equation is dimensionally homogeneous (and contains all the essential geometry and performance variables) it can be reduced to a relationship among a complete set of dimensionless terms of the variables."

Applications of the Buckingham Π theorem results in the formulation of dimensionless terms called Π ratios. These Π ratios have no relation to the number 3.1416.

Example 2

Consider the rather complicated phenomenon of the helical buckling of drill pipe or production tubing inside a casing. This is a highly complicated situation that must be considered in many petroleum engineering drilling and production operations. In general, this involves the torsional post-buckling behavior of a thin walled elastic pipe. The mathematical model for this situation is very complicated and dependable closed-form solutions of this problem do not exist. But still the designer must obtain some insight into this phenomena before an engineering design can be completed. The problem can be approached by utilizing the principals of physical modeling based on the Buckingham Π theorem. What is sought is a set of dimensional relationships (the Π ratios) that are made up of the important variables of the problem. These Π ratios may then be used to created a small-scale experiment that will yield data that can be used to assist in the design the prototype (or full-scale pipe system). The important variables in this problem are EI (lb-in^2 or lb-ft^2) the flexural rigidity of the thin-walled elastic pipe (which is directly related to the torsional rigidity GJ through known material constants), the length of the pipe L(ft), the shortening of the pipe δ(ft) (as it helically twists in the casing), the annular distance between the outside of the pipe and the inside of the casing $(d_2 - d_1)$(ft) (where d_1 is the outside diameter of the pipe and d_2 is the inside diameter of the casing), the applied torque to the pipe T(ft-lbs), and the applied tension in the pipe P(lbs).

Using Table 52 the variables are EI(FL^2), L(L), d(L), $(d_2 - d_1)$(L), T(FL), and P(F). Note that this I is moment-area which is in the units of ft^4 (not to be confused with I given in Table 52 which is moment of inertia, see Chapter 2, Strength of Materials, for clarification). The number of Π ratios that will describe the problem is equal to the number of variables (6) minus the number of fundamental dimensions (F and L, or 2). Thus, there will be four Π ratios (i.e., 6 − 2 = 4), Π_1, Π_2, Π_3, and Π_4. The selection of the combination of variables to be included in each n ratio must be carefully done in order not to create a complicated system of ratios. This is done by recognizing which variables will have the fundamental dimensions needed to cancel with the fundamental dimensions in the other included variables to have a truly dimensionless ratio. With this in mind, Π_1 is

$$\Pi_1 = EI^a\, T^b\, L^c$$

and Π_2 is

$$\Pi_2 = T^d\, P^e\, L^f$$

and Π_3 is

$$\Pi_3 = L^g\,(d_2 - d_1)^h$$

and Π_4 is

$$\Pi_4 = (d_2 - d_1)^i \delta^j$$

In dimensional form Π_1 and each variable of Π_1 becomes

$$(F^0 L^0) = (FL^2)^a (FL)^b (L)^c$$

and for Π_2

$$(F^0 L^0) = (FL)^d (F)^e (L)^f$$

and for Π_3

$$(F^0 L^0) = (L)^g (L)^h$$

and for Π_4

$$(F^0 L^0) = (L)^i (L)^j$$

For Π_1 the equations for a, b and c are

Force F: $0 = a + b$

Length L: $0 = 2a + b + c$

This yields $b = -a$ and $c = a$, which is

$$\Pi_1 = EI^a T^{-a} L^a$$

or

$$\Pi_1 = [EI/T L]^a$$

and since Π_1 is a dimensionless ratio, then the actual value of a is not important. Therefore, the above becomes simply

$$\Pi_1 = EI/T L.$$

Similarly it is found that

$$\Pi_2 = T/P L$$

$$\Pi_3 = L/(d_2 - d_1)$$

and

$$\Pi_4 = (d_2 - d_1)/\delta$$

These Π ratios allow the following relationships to be made between the performance characteristics found for a small-scale experimental model and a full-scale prototype:

$[\Pi_1]_m = [\Pi_1]_p$

$[\Pi_2]_m = [\Pi_2]_p$

$[\Pi_3]_m = [\Pi_3]_p$

and

$[\Pi_4]_m = [\Pi_4]_p$

The above Π ratios can be used as the basis for a small-scale experiment to obtain data that might allow an investigator to determine the number of helixes N, and the shortening δ, that will occur in a full-scale tubing string under particular torsional post-buckling conditions (see Figures 2-83 and 2-84). The small-scale experiment will yield experimental data that can be related through the Π ratios to the full-scale prototype. The use of Π ratios and experimental data requires extensive statistical and deterministic analyses.

Analysis, Synthesis and the Creative Art of Design

Evolution was an early engineering design methodology. This methodology (basically trial and error) is a primitive form of physical modeling. Prior to the development of more sophisticated engineering design methodologies, the only way

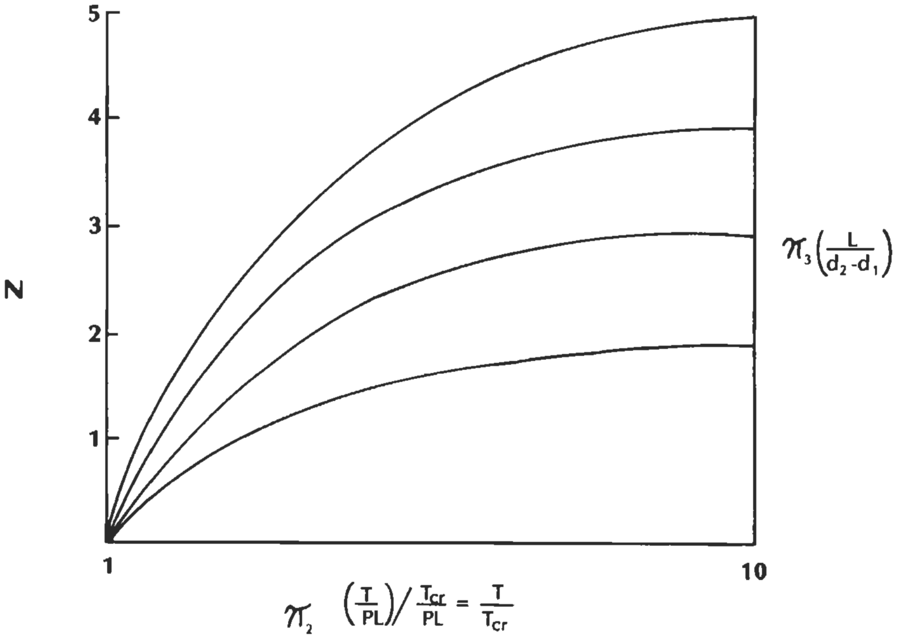


Figure 2-83. Typical Plot of N (number of helixes) versus Torque (for specific geometry ratios).

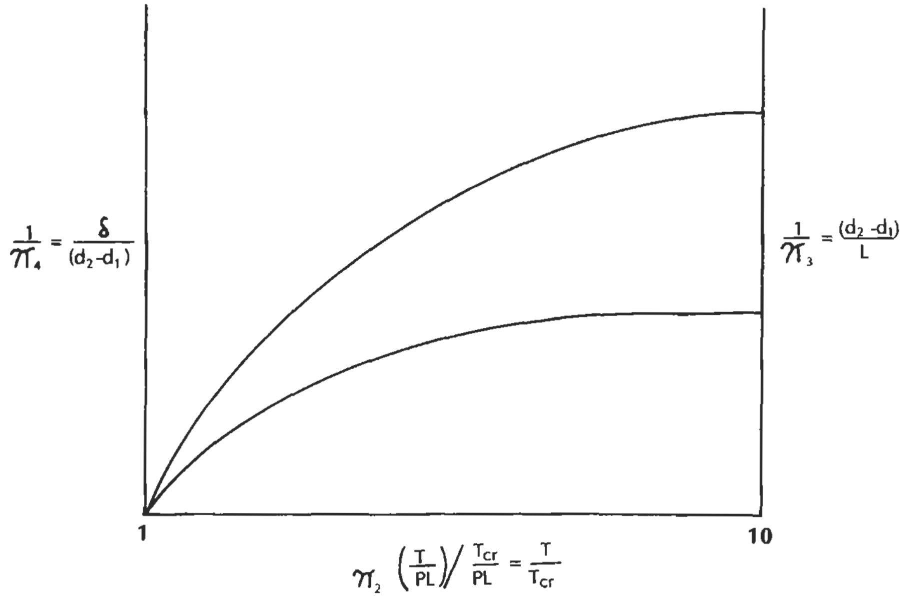


Figure 2-84. Typical Plot of d (shortening) versus Torque (for specific geometry ratios).

to develop a useful device or system was to fabricate a conceptual design and put it into operation. If the design failed to meet the desired specifications, then the failure was analyzed and the appropriate changes made to the design and another product fabricated and placed into operation. But design by evolution was not very economical and as engineering design methodologies were developed that would allow for design choices to be made without resorting to costly fabrication, design engineers were driven to accept these new methodologies. Modern engineering design methodologies are based on the design engineer's ability to appropriately utilize analysis and synthesis to create the desired device or system [73,74,75,76].

Nearly all modern engineering design situations require the use of models, both mathematical and physical. Once societal needs are defined, the designer specifies the performance characteristics required for a design. Also, at this point, the designer can often sketch or otherwise define the various initial alternative design concepts. The designer then separates each alternative design concept into parts that are tractable to either mathematical or physical modeling (or both). The designer utilizes a wide variety of analysis techniques (inherit in mathematical models but also part of the assessment of test data from physical models) to attempt to predict the performance characteristics of the various proposed alternative design concepts. This usually requires repeated analysis and the development of numerous sketches and fabrication drawings for the separate parts of each proposed alternative design concept. Thus, the design of a device or system is a lengthy iterative process.

Once the designer has developed confidence in the analysis techniques pertaining to the various parts of a design concept (whether derived from mathematical models or from physical models), the designer can begin the process of synthesis. Synthesis is basically the combining of the analyses (and any other pertinent information) to allow for the design concept to be viewed as a whole. In general, it is during the process of synthesis that the creative capabilities of the individual designer are utilized the most. The designer utilizes knowledge of the performance specifications, results from modeling, economics, experience with other similar projects, aesthetics (if needed), and intuition to create many easy-to-make choices concerning the design. Thus, the synthesis process often allows the designer to proceed to the final design of a device or system with little iteration.

Creativity in engineering is the art portion of engineering. It is a unique sum of an individual design engineer's experience, practice and intuition with a flair for the aesthetic. Creativity can be found in any part of the engineering design process, but it is usually in the synthesis portion of the process that a designer's aptitude for creativity can make the difference between a mediocre design and a truly successful design. Thus, creativity is an essential part of the engineering method.

Feasibility Studies and Economics

Once an engineering design has been identified (that will meet the performance specifications) it is necessary to ask a very important question, "Can and should the device or system be made?" In general, this question can be divided into three corollary questions [73]:

1. Can the design of the anticipated alternative devices or systems be physically realized?
2. Is the production of alternative devices or systems compatible with the capabilities and goals of the producing company?
3. Can the alternative devices or systems be economically produced and economically operated?

Only if all of the above three questions are answered in the affirmative can and should the device of system be made. The gathering of information to answer these questions is called the feasibility study.

Physical realizability is often the most difficult of the above three corollary questions to answer. In general, to answer this question it is necessary to know; 1) whether the materials and components required by the engineering design are available; and 2) whether the manufacturing (and/or fabrication) techniques and skilled craftsmen needed to fabricate the product are also available. These two assessments are difficult to make because they often involve the projection of future technological developments. Technological developments usually do not occur according to schedule.

Company compatibility requires the analysis of the engineering, manufacturing (with quality assurance), distribution, and sales capabilities of an organization to produce and realize a profit from a given engineering design or product. Often an engineering department within a company may be capable of designing a particular device or system, but the production and sales departments are not capable of carrying out their respective tasks. Also, a new product line under consideration may be beyond the scope of the overall business goals and objectives of the company.

As in all business situations, the economics of the development of a newly designed device or system is of paramount importance. In order to fully understand the economic feasibility of a new design, the engineer must make the following economic assessments:

1. Evaluate two or more manufacturing techniques.
2. If components of a new system design are available from sources outside the company, then an evaluation must be made as to whether it is more economic to buy these components from outside or to manufacture them within.

3. Engineers will constantly seek ways of reducing manufacturing costs. As these design modifications are made, economic evaluations of changes must be made.
4. An overall economic evaluation must be made to ensure that the contemplated project for petroleum development will recover sufficient capital to pay for the total cost of development, installation and assembly effort of the company. Further, the capital return must yield a financial return consistent with the overall company risk involved. Thus, it is very important that the petroleum engineer designing a recovery system understand the company's evaluation criteria that will be used by management in judging whether a new project is to go forward. Chapter 7 (Petroleum Economics) has detailed discussions of general engineering economics and product (or project) evaluation criteria.

Design in the Petroleum Industry

Producing and Service Companies

The petroleum industry is a highly complex primary industry. Its function in society is to provide a reliable supply of liquid and gaseous hydrocarbon fuels and lubricants (and well as other related products) for both industrial and private customers. The industry is composed of two basic business elements 1) producing companies (often called operating companies) and 2) service companies.

Producing companies locate the subsurface petroleum resources; recover the oil, gas or condensate; and then market the recovered resource to customers. These activities involves planning and carrying out exploration, drilling and well completion, and the production activities.

The service companies provide a supporting role to the producing companies. They supply the products and services that allow the producing companies to explore, drill and complete wells, and produce the petroleum from the earth. Once the petroleum has been brought to the surface, the service companies also assist in providing safe transportation of the petroleum to refining facilities. There is a great variety of service companies. Many service companies are dedicated only to supporting the producing companies (e.g., geophysical surveying) while other service companies supply products and services to a number of primary industries.

Service Company Product Design

In general, engineering design of devices or systems takes place throughout the petroleum industry. However, service industries are responsible for supplying (thus, designing and manufacturing/fabricating) the vast majority of the devices and systems (and often the personnel to operate them) that are utilized by the producing companies. Therefore, the service companies do far more traditional device and system engineering design than the producing companies (e.g., geophysical survey tool design, drilling fluid design, drill bit design, cement design and placement system design, production pump design, etc.). Because of the complexity of the petroleum industry, these engineering design activities cover all three of the basic engineering fields (e.g., mechanical arts and sciences, the chemical arts and sciences, and the electrical arts and sciences).

Producing Company Project Design

The producing companies are responsible for carrying out the complex operations of exploration, drilling and completion, and production. These projects vary from the seeking oil and gas deposits at 15,000 ft of depth in the continental U.S. to seeking similar deposits at 15,000 ft below some remote ocean floor. The engineering planning and design of these complex projects is a special class of systems engineering design. Although dealing with quite different technologies, production operations have similar concerns related to overall system outcome as do large mining projects or large civil engineering projects (such as the construction of a dam or large transportation system). Thus, the engineering project design and management aspects of systems engineering are of primary interest to the engineering staffs of the producing companies [77].

Engineering Ethics

The Profession of Engineering

An engineer is a professional in exactly the same context as a medical doctor, a lawyer, clergy, teacher, etc. The dictionary defines a "profession" as

"an occupation requiring advanced education."

In this context an advanced education means an academic education beyond secondary education. The term "professional" is defined as

"connected with, or engaged in a profession."

The professional should not be confused with the other occupations that make up the economic doers in modern society, namely craftsmen, tradesmen, and laborers. The difference between these other occupations and the professional is in the definition of the profession and professional. Very few occupations *require* an advanced education to be employed in that occupation. A plumber does not have to have an advanced academic education to be in the plumbing business. But the plumber does have to have certain manual skills in order to be successful in that occupation. Thus, the plumber is a craftsman. A banker does not have to have an advanced education to be in the banking business. Also, the banker does not have to have any particular manual skills to be a banker. Thus, the banker is considered to be a tradesman. It should be noted that the definition of the term "professional" does not depend upon whether the engineer (or medical doctor, etc.) is legally registered (with a state board) as a licensed professional engineer (or registered as a licensed medical doctor, etc.) [78].

There are certain unwritten accepted societal ideals that also separate professionals from the other occupations. The most important of these is that professionals, in carrying out the duties of their respective professions, usually are doing work that can affect society. This is particularly true of the engineering profession. The engineering profession designs many devices and systems that if not designed correctly and fail can have catastrophic effects on society as a whole. For example, when a civil engineer does not correctly design a community water supply and sewer system and the community becomes sick from contaminated drinking water, then society as a whole suffers from the civil engineer's incompetence. Likewise, when a petroleum engineer does not correctly design the reserve pits at the drilling location and drilling mud contaminated with crude oil is washed into a nearby stream, society as a whole suffers from the petroleum engineer's incompetence. Also, when a petroleum engineer plans and operates a drilling location that is quite near a high voltage power line and fails to get the local power company to either provide a by-pass line or rubber boot the line, and subsequently a crane operator on location is electrocuted, society as a whole suffers from the petroleum engineer's incompetence. In this case, "society" is represented by the craftsmen and laborers working at the location [79,80].

NSPE Code of Ethics

All professional groups in society have a code of ethics to guide their respective members as they carry out their professional duties for society. There is a general code of ethics of the National Society of Professional Engineers (NSPE) which basically covers all registered professional engineers regardless of specific discipline. The preamble and fundamental canons of the NSPE Code of Ethics is given below [81].

Preamble. Engineering is an important and learned profession. The members of the profession recognize that their work has a direct and vital impact on the quality of life for all people. Accordingly, the services provided by engineers require honesty, safety and welfare. In the practice of their profession, engineers must perform under a standard of professional behavior which requires adherence to the highest principles of ethical conduct on behalf of the public, clients, employers and the profession.

Fundamental Canons. Engineers, in the fulfillment of their professional duties, shall:

1. Hold paramount the safety, health and welfare of the public in the performance of their professional duties.
2. Perform services only in areas of their competence.
3. Issue public statements only in an objective and truthful manner.
4. Act in professional matters for each employer or client as faithful agents or trustees.
5. Avoid deceptive acts in the solicitation of professional employment.

SPE Code of Ethics

Each specific engineering discipline has developed its own code of ethics. These discipline specific codes of ethics generally reflect the working situations that are relevant that discipline. This can be seen in the Society of Petroleum Engineers (SPE), Guide for Professional Conduct. Because the devices and systems that petroleum engineers design are usually in operational locations that are remote from most of general public, the issue concerning the "safety, health and welfare of the public" is not as immediate to the petroleum engineer as it is to other disciplines (e.g., civil engineers). Thus, the "safety, health and welfare of the public" is found further down in this list of canons than in the NSPE Code of Ethics. The preamble, the fundamental principle, and the canons of professional conduct of the SPE Guide for Professional Conduct is given below [82].

Preamble. Engineers recognize that the practice of engineering has a direct and vital influence on the quality of life of all people. Therefore, engineers should exhibit high standards of competency, honesty, and impartiality; be fair and equitable; and accept a personal responsibility for adherence to applicable laws, the protection of the public health, and maintenance of safety in their professional actions and behavior. These principles govern professional conduct in serving the interests of the public, clients, employers, colleagues, and the profession.

The Fundamental Principle. The engineer as a professional is dedicated to improving competence, service, fairness, and the exercise of well-founded judgment in the practice of engineering for the public, employers, and clients with fundamental concern for the public health and safety in the pursuit of this practice.

Canons of Professional Conduct.

1. Engineers offer services in the areas of their competence affording full disclosure of their qualifications.
2. Engineers consider the consequences of their work and societal issues pertinent to it and seek to extend public understanding of those relationships.
3. Engineers are honest, truthful, and fair in presenting information and in making public statements reflecting on professional matters and their professional role.
4. Engineers engage in professional relationships without bias because of race, religion, sex, age, national origin, or handicap.
5. Engineers act in professional matters for each employer or client as faithful agents or trustees disclosing nothing of a proprietary nature concerning the business affairs or technical processes of any present or former client or employer without specific consent.
6. Engineers disclose to affected parties known or potential conflicts of interest or other circumstances which might influence or appear to influence judgment or impair the fairness or quality of their performance.
7. Engineers are responsible for enhancing their professional competence throughout their careers and for encouraging similar actions by their colleagues.
8. Engineers accept responsibility for their actions; seek and acknowledge criticism of their work; offer honest criticism of the work of others; properly credit the contributions of others; and do not accept credit for work not theirs.
9. Engineers, perceiving a consequence of their professional duties to adversely affect the present or future public health and safety, shall formally advise their employers of clients and, if warranted, consider further disclosure.
10. Engineers act in accordance with all applicable laws and the canons of ethics as applicable to the practice of engineering as stated in the laws and regulations governing the practice of engineering in their country, territory, or state, and lend support to others who strive to do likewise.

Intellectual Property

Since one of the main focuses of the engineer is to provide technological solutions for the needs of society, then creative invention is a major part of the engineer's design activity. Therefore, the engineer must be familiar with the current laws (and any imminent future changes in the laws) for the protections of intellectual property. Intellectual property is composed of four separate legal entities. These are trade secrets, patents, trademarks, and copyrights.

Trade Secrets

Trade secrets are creative works (usually methods, processes, or designs) that are not covered by the strict definition of "an invention" required for a legal patent. Or they are actual inventions that a company or individual does not want to expose to public scrutiny through the patent process (since the published patent must contain a written description of the invention).

Companies and individuals have the legal right to protect and maintain trade secrets. The key point, however, is that trade secrets must be something that can be kept secret. Anyone who discovers a trade secret via his or her own efforts may use it. The only legal protection for trade secrets is against persons who obtain these secrets by stealing them and then using them, or after having been entrusted with these secrets as a necessary part of their job and then using them [73,83].

The term trade secret is well understood by U.S. courts and is a legal term. In general, the U.S. courts have strongly upheld the protection of company- or individual-owned trade secrets. Cases against ex-employees or others who have obtained trade secrets by stealth are relatively easy to prosecute (compared to patent, trademark, or copyright infringement cases).

Patents

When the U.S. Constitution was framed in 1789 it contained this provision (Article 1, Section 8) [83,84,85,86]:

> "The Congress shall have power to promote the progress of science and useful arts by securing for limited times to authors and inventors the exclusive right to their respective writings and discoveries."

In 1790 the formal U.S. Patent system was founded. Since that time the U.S. Patent Office has undergone numerous changes that have generally improved legal protection for intellectual property owned by individual citizens.

A U.S. Patent is a legal document granted only to the individual inventor that allows the inventor to have the exclusive right for a limited time to exclude all others from making, using or selling his or her invention in the U.S. There are restrictions to this statement. An inventor does not have an absolute right to use his or her patented invention, if doing so would also involve making, using or selling another invention patented by someone else. The time limits for the exclusive use of an invention by the inventor are fixed by low and can only be extended in special circumstances. After this time limit expires, anyone may make and sell the invention without permission of the inventor. There are three legal categories of patents; utility patents, design patents, and plant patents.

Utility Patents. Utility patents are granted to individuals only who have invented or discovered any new and useful method, process, machine, manufacture, or matter composition. These patents must be useful to society (have utility). A utility patent has a total time limit for exclusive use by the inventor of 17 years.

Design Patents. Design patents are granted to individuals who have invented new, original and ornamental designs for an article of manufacture. A design patent has a total time limit for exclusive use by the inventor of 14 years.

Plant Patents. Plant patents are granted to individuals who have invented or discovered and a sexually reproduced a new and distinct variety of plant. This includes mutants, hybrids, and newly found seedlings. A plant patent has a total time limit for exclusive use by the inventor of 17 years.

Only the actual inventor may apply for a patent (utility, design, or plant patent). Once a patent has been obtained, the patent can be sold or mortgaged. Also, the owner of a patent may assign part or all interest in the patent to another individual or a company or other business entity. The owner of a patent may also grant licenses to others (either individuals or business entities) to make, use or sell the invention.

In addition to each of the above patent's unique characteristic (i.e., utility, ornamental or original design, be a plant), an invention, in order to be granted a U.S. Patent, must demonstrate two additional characteristics: 1) the invention must be novel; and 2) the invention must not be obvious. The steps to pursue to obtain a patent are as follows:

1. Make sure the invention is new and practical and is not already in use. This requires a preliminary search of the industrial literature to determine if the invention is practical and is not already in use in the public domain.
2. Keep records that document when and what was invented. It is important that accurate records are kept showing your original sketches with a disclosure statement describing what and how your invention works. It is useful to have someone witness this disclosure document and verify the date that this invention took place. It is often during this step that the invention concept is either modeled (mathematical or physical or both) and tested. Thus, accurate records of these analyzes or test results should also be kept. In the U.S. it is the first to invent that will obtain a patent in the event of two individuals inventing the same thing. Keep the disclosure document secret until the patent application is submitted to the patent office.
3. Search the existing patents relating to the invention. Carry out an extensive patent search of existing patents to find patents that are related to the new invention. Make sure the new invention is novel and unobvious and is not in conflict with existing patents. A search of U.S. patents can be made in the Search Room of the U.S. Patent and Trademark Office in Crystal Plaza, 2021 Jefferson Davis Highway, Arlington Virginia. An individual inventor or a patent attorney may hire agents that will carry out thorough patent searches. in addition, every state in the U.S. has a U.S. Patent Depository Library where patent searches may be conducted by private individuals.
4. Prepare and file the patent application documents. This document is called a "specification." The specification must be clear enough so that anyone skilled in the subject matter of the invention could recreate your invention and use it. In the written portion of the specifications the inventor must state the claims of the invention. These claims must show that the invention is novel and unobvious. Also, where applicable, illustrations must accompany the specification. There are filing fees for patent applications and if the application is successful there are maintenance fees to keep the patent in force. Literature describing patent applications and how they are to be submitted can be obtained from the Commissioner of Patents and Trademarks, Washington, D. C., 20231. This is also the address to submit properly prepared patent applications.

Infringement of a patent occurs when an individual or company makes, uses, or sells an invention without the permission of the inventor, assignee, owner, or licensee. The U.S. courts will award damages and place penalties on the infringer. But patent infringement cases are very costly and time consuming.

Any inventor may apply for a patent regardless of age, sex, or citizenship. Once a U.S. Patent has been granted most other countries allow the inventor up to one year to submit a patent application for a foreign patent. The exception is Japan which requires nearly immediate submittal of a patent application to the Japanese patent office at nearly the same time the inventor is submitting to the U.S. Patent office.

It should be noted that all other countries grant patents to inventors that are the first to submit patent applications to their respective patent offices. We are the only country the grants a patent to the first to invent (not the first to the patent office). However, in 2003 the U.S. may change to the "first to the patent office" method of granting patents.

Trademarks

A trademark is any work, name, symbol, or device or any combination of these adopted and used by manufacturers or merchants to identify their goods and distinguish them from those manufactured or sold by others. A service mark is a mark used in the sale or advertising of services to identify the services of one individual or company and distinguish them from the services of others [87,88].

Unlike a patent, but like a copyright, ownership of a trademark or service mark is acquired by use. An individual or company using the mark within a state may register the mark in that state. The Secretary of State within each state has the forms for the registration of a trademark or service mark. Once a mark has been used in interstate or foreign commerce, the mark may be registered in the U.S. Patent and Trademark Office. This is accomplished by filing an application to that office (see Reference 87). Prior to receiving federal registration of the mark, the symbols ™ and ℠ may be used (these symbols give notice that the marks have been filed with the state). After the mark has received federal registration the symbol ® should be used.

There are two classes of registration for a mark. The Principal Register is for unique and distinctive marks that when applied to products and services are not likely to cause confusion or deception. To be registered on the Principal Register a mark must be in continuous, exclusive interstate use. Marks not registrable on the Principal Register may be registered on the Supplemental Register.

The term of a federal registered trademark and service mark is 20 years. In order to secure the mark for the full 20 years, an affidavit must be filed with the Commissioner of Patents and Trademarks in the sixth year showing the mark is still in use. It is advisable to conduct a search in the U.S. Patent and Trademark Office to determine whether a mark under consideration might conflict with existing registered marks. Trademarks and service mark registrations may be assigned after registration.

Once federal certification of a mark has been issued, the mark is protected nationwide. Any infringement by an individual or company of a mark after federal registration can be subject to damage claims by the mark owner in U.S. District Courts.

Foreign countries require registration of a mark in compliance with local laws. The local laws vary a great deal; thus, it is necessary to consult a local attorney or a U.S. attorney familiar with foreign patent and trademark law in order to register a mark in a foreign country.

Information concerning the registration of marks in the U.S. may be obtained from the Commissioner of Patents and Trademarks, Washington, D. C., 20231.

Copyrights

The copyright protects creative works such a literary works, musical works, dramatic works, pictorial, graphic, sculptural, motion pictures and other audio-visual works, and computer software. It is the later that is of most concern to the engineer [89,90].

Registration for copyright protection may be obtained by application to the Register of Copyrights, Library of Congress, Washington, D. C., 20540. This registration must take place on or about the time of the first sale of the creative work. The forms for application are obtained from the address above (the application fee is $10)

The notice of copyright is given by "copyright" or the abbreviation "Copr" or by the symbol ©. The notice must be accompanied by the name of the copyright owner and the year of the first publication. Like a patent, only the author of the creative work can be the copyright owner. However, an assignee may file for the copyright on behalf of the author of a creative work. In general, the term of a copyright is for the life of the author (of the creative work) plus 50 years. Any infringement by an individual or company of a registered copyright can be subject to damage claims by the creative work owner or assignee in U.S. District Courts.

Under the Universal Copyright Convention, a U.S. citizen may obtain a copyright in most countries of the world by simply publishing within the U.S. using the © symbol and the appropriate author and date of publication notices (only the © symbol is recognized worldwide).

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3

Auxiliary Equipment

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Chapter 3

Auxiliary Equipment

This chapter describes the various auxiliary equipment that are important to the function of oil field activities. This equipment is used in drilling, well completion, production, and related operations. The discussions that follow will concentrate on the basic operation characteristics and specifications required by a user. The aim is to give the user the information needed to ascertain whether the equipment available for a particular field operation is adequate, or if the equipment available is not adequate, how to specify what equipment will be necessary.

PRIME MOVERS

The prime mover is the unit that first converts an energy source into a mechanical force. Typical prime movers are internal combustion motors, gas turbines, water turbines, steam engines and electrical motors. The discussion will be limited to the prime movers that are most used in modern well drilling and production operations. These are internal combustion motors, gas turbine motors and electric motors.

Internal Combustion Engines

The internal combustion engines considered in this section are piston-type engines. The combustion process in such engines are assumed to be constant volume, constant pressure, or some combination of both. Piston-type internal combustion engines are made up of a series of pistons that can move in enclosed chambers called cylinders. These engines can be designed in a variety of configurations. The most widely used configurations are the following [1]:

- An in-line type engine has all cylinders aligned and on one side of the crankshaft (see Figure 3-1). These engines are found in sizes from 4 pistons (or cylinders) to as many as 16.
- A V-type engine has an equal number of cylinders aligned on two banks of the engine. These banks form a V shape (see Figure 3-2). These engines are found in sizes from 4 cylinders to as many as 12.
- An opposed-type engine has an equal number of cylinders aligned on two banks of the engine. These banks are horizontally opposed to one another on opposite sides of the crankshaft (see Figure 3-3). These engines are found in sizes from 4 cylinders to as many as 8.

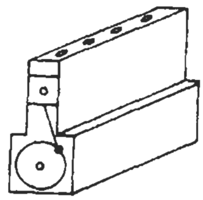


Figure 3-1. In-line type engine [1].

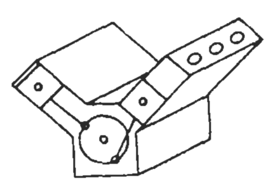


Figure 3-2. V-type engine [1].

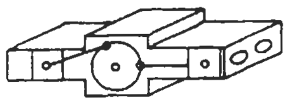


Figure 3-3. Opposed type engine [1].

One of the more important specifications of an internal combustion engine is its displacement volume. The displacement volume, $V_d(in^3)$, is (per revolution)

$$V_d\,(in^3) = \frac{\pi}{4} d^2\, l\, n \qquad (3\text{-}1)$$

where d is the outside diameter of the piston (or bore of cylinder in in.), l is the stroke (or length of movement) of the piston within the cylinder in in., and n is the number of cylinders.

The nominal compression ratio (which is usually specified) is the displacement volume plus the clearance volume divided by the clearance volume. Because of the mechanics of intake value closing, the actual compression ratio r_a is less than the nominal. Thus, the compression pressure p (psia) may be estimated by

$$p = r_a \bullet p_m \qquad (3\text{-}2)$$

where p_m is the intake manifold pressure in psia.

There are two basic combustion cycles. These are

1. Spark-ignition, or otto cycle, engine which is fueled by a gas and air mixture,
2. Compression-ignition, or diesel cycle engine, which is fueled by a diesel oil and air mixture.

There can also be a combination of the two cycles in an engine. Such an engine is called a mixed, combination, or limited pressure cycle.

The spark-ignition cycle engine uses volatile liquids or gases as fuel. Such engines have compression ratios from 6:1 to 12:1. Liquid gasoline is the most common fuel for this cycle, especially for automobile, truck, and airplane engines. Large stationery engines usually use commercial gases such as natural gas, produces gas, or coal gas. All engines operating on this cycle use carburetors, gas-mixing valves, or fuel injection systems to mix the volatile fuel with the appropriate volume of air for subsequent combustion. This mixing can be carried out prior to the mixture being placed in the cylinder of the engine, or mixed directly in the cylinder.

The compression-ignition cycle engine uses low volatile fuel. These are typically fuel oils. The compression ratios for these engine are from 11.5:1 to 22:1. These engines are used to power large trucks and buses. They are also used for large engines. The compression-ignition cycle engine usually is equipped with a fuel injection system which allows the fuel to be mixed with air directly in the cylinders of the engine.

The internal combustion engine can be operated as a two-stroke or a four-stroke engine.

The two-stroke engine requires two piston strokes (or one crackshaft revolution) for each cycle. In one upward (compression) stroke of the piston, the combustible mixture is brought into the cylinder through the intake valve and ignited near the top of the stroke as the piston is forced downward (to provide power); the exhaust valve is opened and the spent gases are allowed to escape. Near the bottom of their stroke the intake valve is opened again and the combustible mixture again is brought into the cylinder.

The four-stroke engine requires four piston strokes (or two crankshaft revolutions) for each cycle. In a downward stroke, the intake valve is opened and the combustible mixture is brought into the cylinder. In an upward stroke the fuel–air mixture is compressed and ignited near the top of the stroke. This forces the piston downward (to provide power). In the next upward stroke the exhaust valve is opened and the spent gases are forced from the cylinder. In the following downward stroke, the cycle is repeated with the opening of the intake valve.

The standards for internal combustion engines have been established by the American Petroleum Institute (API), the Diesel Engine Manufacturers Association (DEMA), and the Internal Combustion Engine Institute (ICEI). In addition, some of the engine manufacturers have their own rating procedures. It is important to know which standards have been applied for the rating of an engine. A consistent set of standards should be used by the engineer when comparing the ratings of various engines for the purpose of selecting the appropriate design for field applications.

The API standards describe the method used in rating engines and the recommended practice for engine installation, maintenance, and operation [2,3]. In oil field operations the API rating standards are most frequently used.

The important definitions are:

1. *Bare engine.* A bare engine shall be an engine less all accessories except those (built in or attached) absolutely required for running. All accessories normally required for operation of the engine, such as ignition, water pump, air cleaner, oil pump, governor, etc., shall be included.
2. *Power unit.* A power unit shall consist of a bare engine, plus other equipment such as a fan for air cooling, special water pumps, and so forth. When included, specific information must be given as to design factors such as ambient temperature and power consumption.
3. *Maximum standard brake horsepower.* At any rotational speed, maximum standard brake horsepower shall be the greatest horsepower, corrected to standard conditions, that can be sustained continuously under conditions as outlined under test procedure. The unit of horsepower is 33,000 ft-lb/min or 550 ft-lb/s. Standard conditions for the purpose of internal combustion engine testing and rating is 85°F(29.4°C) and 29.38 in. of mercury (99kPa). Note these values are different from standard conditions for gas and air volume specifications.
4. *Maximum standard torque.* The maximum standard torque at any given rotational speed shall be that corresponding to the maximum standard brake horsepower at that speed.

Test engines shall be of exactly the same design and equipped with the same components and accessories as engines delivered to the purchaser [2]. The observed brake horsepower H_0 obtained during the testing of a bare engine or a power unit is converted to standard brake horsepower using

$$H_s = H_0 \frac{29.38}{P_0 - E_0} \frac{460 + t_0}{520} \tag{3-3}$$

where P_0 is observed barometric pressure in in. Hg, E_0 is pressure of water vapor in air (from relative humidity data) in in. Hg, t_0 is observed air temperature in °F.

When an internal combustion engine is to be used at different operating conditions (altitude) other than the standard conditions that the engine was rated at, it is necessary to derate the engine specifications. The brake horsepower H at pressure and temperature conditions other than standard can be obtained from the following:

$$H = \frac{Hs(P - E)}{29.38} \frac{520}{t + 460} \tag{3-4}$$

where P is the local barometric pressure in in. Hg, E is pressure of water vapor in air (from relative humidity data) in in. Hg, t is the local air temperature in °F.

At any given set of ambient conditions, the horsepower H is related to the torque the engine produces at a given speed from the following:

$$H = \frac{TN}{5,252} \tag{3-5}$$

where T is the torque in ft-lb, N is the speed in rpms.

Thus for the various transmission speed ratios, and the engine speed, the maximum available torque may be found using the maximum brake horsepower which can be produced by the engine.

Figures 3-4 and 3-5 are examples of a typical manufacturer's report of test results.

There is a reduction in the power available from an internal combustion engine when there is a decrease in ambient air pressure, an increase in ambient temperature, or an increase in relative vapor pressure. Such changes reduce the mass of oxygen available for combustion inside the engine cylinder. This reduction in standard brake horsepower as a function of altitude increase and temperature increase (or decrease) can be approximated by Table 3-1 [4]. This table gives the approximate percentage of power reduction for naturally aspirated and turbocharger internal combustion engines.

Internal combustion engines can be operated by a number of liquid and gaseous fuels. Table 3-2 gives the high value heat of combustion for various field available liquid and gaseous fuels [5].

The gas-powered drilling and production internal combustion engines can be set up at the manufacturer to operate on gasoline, natural gas, or liquified petroleum gas (LPG). The manufacturer can also set up the engine to operate on all three types of gas fuels. This is accomplished by providing engines with conversion kits that can be used to convert the engine in the field.

Power unit manufacture also produce diesel engines that can be converted to operate on a dual fuel carburation of about 10% diesel and natural gas. Such conversions are more difficult than converting spark-ignition engines to various gas fuels.

Actual fuel consumption data are available from the bare engine or power unit manufacturer. Conversion kits and alternate fuel consumption data are also usually available. However, often the field engineer does not have the time to obtain the needed data directly from the manufacturer regarding the various power units that are to be used for a particular field operation. Table 3-3 gives the approximate liquid fuel consumption for multicylinder engines for various fuels. Table 3-4 gives the approximate gaseous fuel consumption for multicylinder engines for various fuels.

Figure 3-6 shows average vacuum-load curves for several engine models with four-cycle engines of two or more cylinders constructed by six representative engine manufacturers [3]. These curves cannot be used for supercharged or turbocharger engines. Vacuum readings are obtained with a conventional vacuum gauge, containing a dial graduated in inches of mercury. The intake manifold vacuum reading is taken first with the engine running at normal speed with no load and then with the engine running at normal speed with normal load. The curve selected is the one which, at no load, most closely corresponds to the no-load intake manifold vacuum reading. The point on the curve is then located with the ordinate corresponding to the reading taken at normal loading. The abscissa of this point gives the percentage of full load at which the engine is operating. For instance, if the no-load reading is 17 in. Hg and

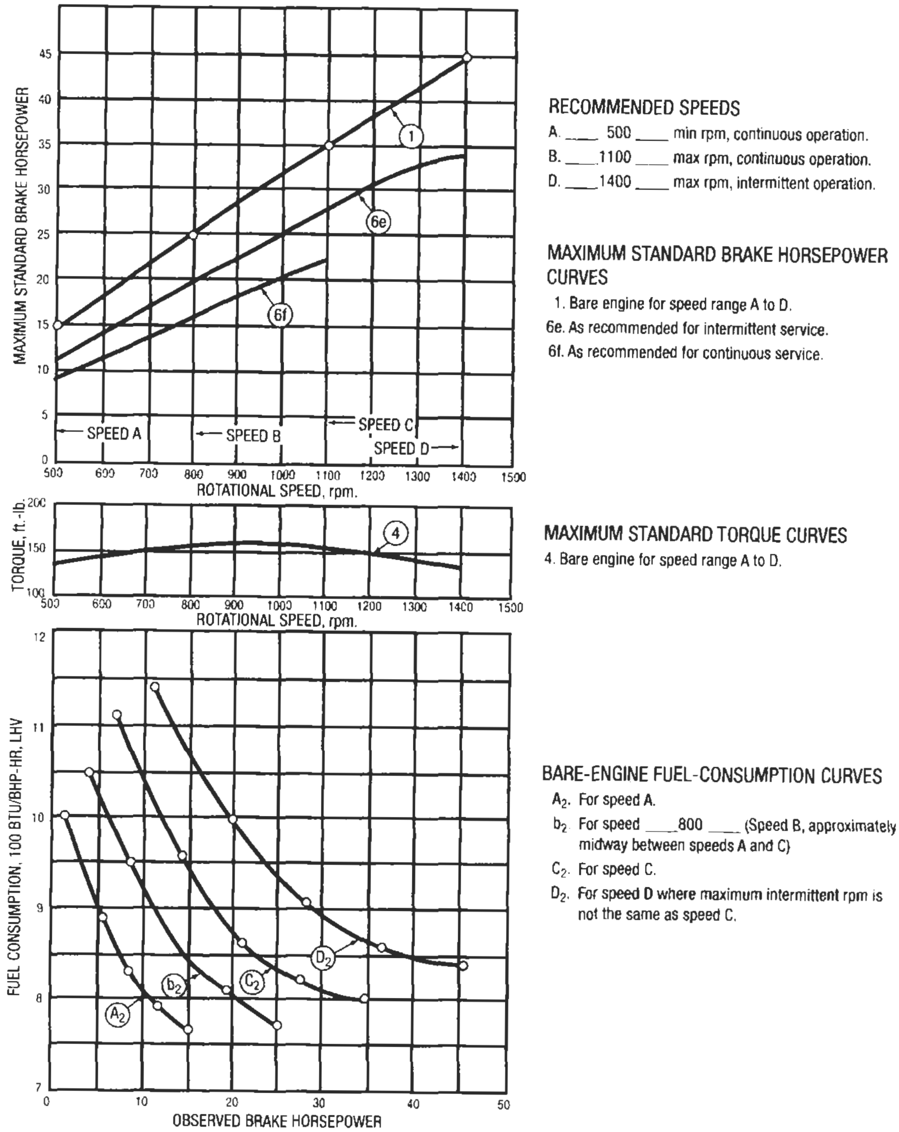


Figure 3-4. Example of test data for internal-combustion bare engines [3].

the normal reading is 10 in. Hg, the curve selected in Figure 3-4 would be the one which shows 17 in. Hg at 0% of full load. This curve would be followed down to its intersection with the horizontal line of 10 in. The abscissa of this point indicates that the engine is operating at 48% of full power.

The above method is useful in determining internal combustion engine loading conditions. Failure to duplicate former readings at no load and at normal speed indicates that the engine is in poor condition. Failure to duplicate former readings

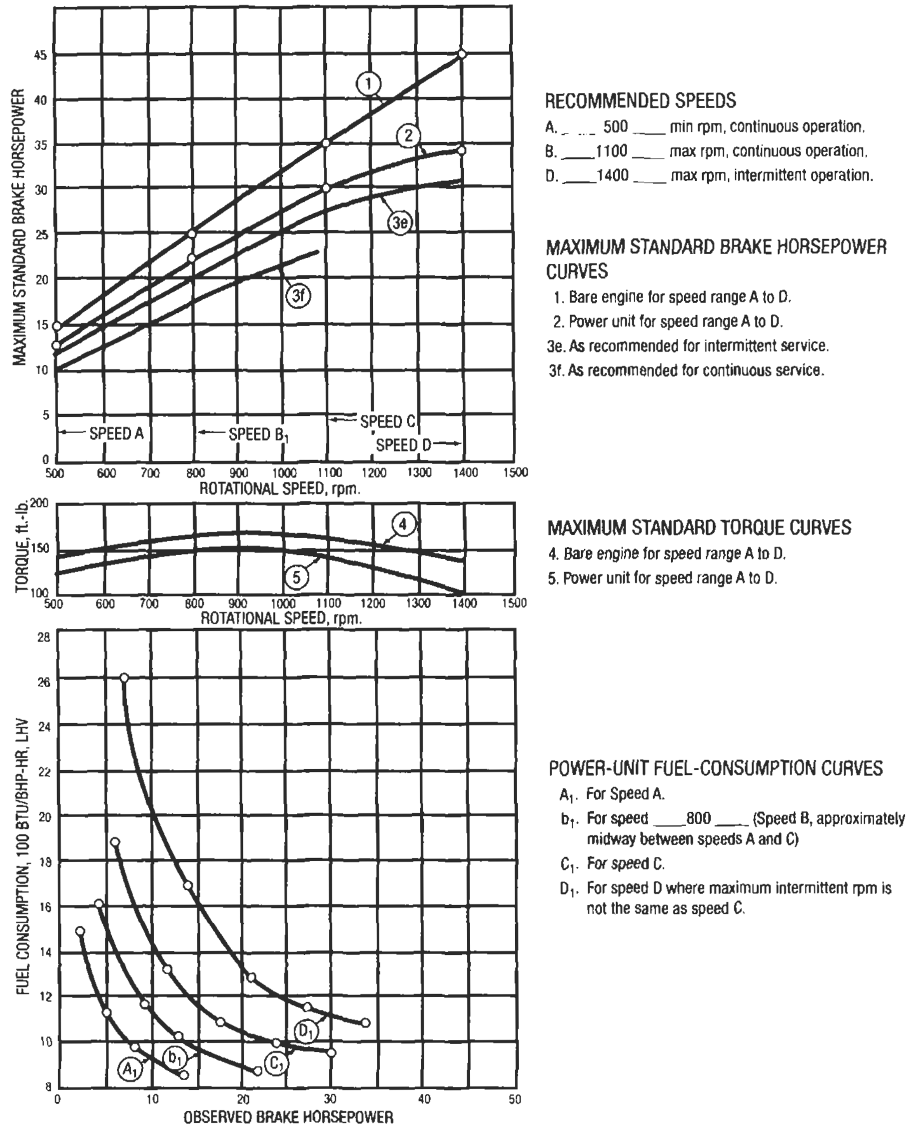


Figure 3-5. Example of test data for internal combustion power units [3].

at normal load and normal speed indicates a change in either engine efficiency or load conditions.

Gas Turbine

Gas turbines are essentially constant speed machines and, therefore, generally not suited to be used for the mechanical driving of most equipment in the oil field. Gas turbines are used in combination electrical generating systems and electrical motors

Table 3-1
Internal Combustion Engine Power Reduction (Percent) [3]

Engine	For Each 1000 ft Increase	For Each 10°F Increase or Decrease
Naturally Aspirated	3.66	0.72
Turbo-charged	2.44	1.08

Table 3-2
Net Heat of Combustion (High Heat Value) for Various Liquid and Gaseous Fuels [4]

Fuel	BTU/lb	BTU/gal	BTU/SCF[a]	Specific Gravity	Specific Weight (lb/gal)
Methane	23890		995	0.54	
Natural Gas	26411		1100	0.65	
Propane	21670	91014			4.2
Butane	21316	104448			4.9
Motor Gasoline	20750	128650			6.2
Aviation Gasoline	21000	126000			6.0
Methanol	9758	42935			4.4
Ethanol	12770	80451			6.3
Kerosene	20000	134000			6.7
JP-4	18400	123280			6.7
JP-5	18300	128100			7.0
Diesel Grade 1-D (Save on No. 1 Fuel Oil and Grade 1-GT)	19940	137586			6.9
Diesel Grade 2-D (Save on No. 2 Fuel Oil and Grade 1-GT)	19570	140904			7.2

[a] SCF is at standard mid-latitude average atmospheric conditions of 14.696 psia and 59°F.

Table 3-3
Approximate Liquid Fuel Consumption for Multicylinder Internal Combustion [4,5]

Fuel	100% Load (lb/hp-hr)	75% Load (lb/hp-hr)	50% Load (lb/hp-hr)
Motor Gasoline	0.62	0.67	0.76
Propane/Butane	0.5	0.56	0.64
Diesel	0.5	0.53	0.63

Table 3-4
Approximate Gaseous Fuel Consumption for Multicylinder Internal Combustion Engines [4,5]

Fuel	100% Load (SCF/hp-hr)	75% Load (SCF/hp-hr)	50% Load (SCF/hp-hr)
Natural Gas	9.3	10.0	11.1

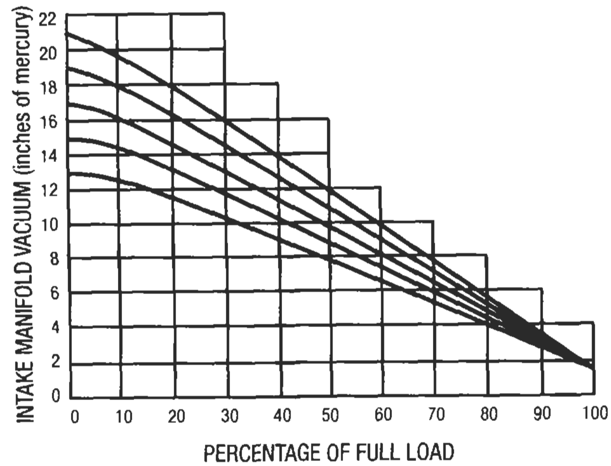


Figure 3-6. Intake vacuum vs. load curves [3].

to power drilling rigs and other auxiliary drilling and production equipment. In general, gas turbine are used where a constant speed power source is needed [1,5,6].

There are some rare situations where the gas turbine is used to power, through direct linkage, a mechanical unit, e.g., a hydraulic fracture pump.

The most common use of the gas turbine power system in the oil and gas industry is in combination with an electrical system (i.e., electric generators and electric motors). In 1965 such a system was used to power a rotary rig. This was a 3,000-hp rig developed by Continental-Ensco. The rig used three 1,100-hp Solar Saturn single-shaft gas turbines. These gas turbines operated at 22,300 rpm and were connected through double reduction gear transmissions to DC generators.

Gas turbines can be fueled by either gas or liquid hydrocarbons. The gas turbine is often used as a means of utilizing what would otherwise be waste hydrocarbons gases and liquids to generate local electric power via the gas turbine power unit. This is called cogeneration.

The basic concept of the gas turbine is to inject the fuel into a steady flow of compressed air (see Figure 3-7). The compressed air is ignited and the expanding exhaust is forced to pass through a series of stationary and rotating turbine blades. These expanding gases force the turbine shaft to rotate. This shaft power is either directly connected or connected through speed reducers to an electric generator and the compressor used to initially compress the air entering the turbine combustor.

As shown in Figure 3-7, a small electrical (or other type of motor) is needed to start the gas turbine. This is usually accomplished by disconnecting the generator load from the gas turbine (usually via a clutch mechanism on gear shaft in the

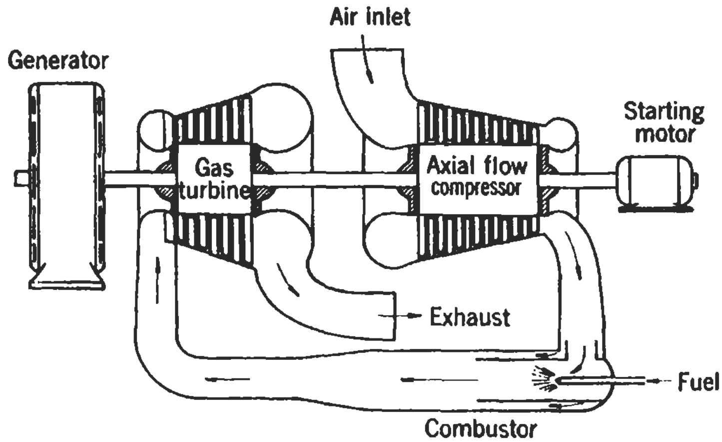


Figure 3-7. Gas turbine (open-cycle) and electric generator.

transmission). The starter motor is actuated, which brings the gas turbine shaft up to a speed where the compressor is providing compressed air at the pressure and weight rate of flow that can sustain the gas turbine operation. Once the gas turbine is up to its critical start speed (usually about one-fourth to one-third of operating speed), fuel is injected into the combustor and ignited. The fuel is then regulated to the desired operating speed for the system and the generator or other power take off re-engaged to the gas turbine shaft.

The gas turbine shown in Figure 3-7 is an open-cycle type. An open-cycle type gas turbine uses the same air that passes through the combustion process to operate the compressor. This is the type most often used for stationary power unit applications. A typical example of power requirements for an open-cycle type gas turbine would be for the unit to develop a total of 3,000 hp. However, about 2,000 hp of this would be needed to operate its compressor. This would leave 1,000 hp to operate the generator (or other systems connected to the gas turbine). Thus, such a gas turbine power unit would be rated as a 1,000-hp unit because this is the power that can be utilized to do external work.

Electric Motors

There are a number of electric motor types available. These motors are classified by the National Electrical Manufacturers Association (NEMA). These electric motor classifications are presented in the NEMA standards [7].

Figure 3-8 shows a graphic breakdown of the various electric motors available [8]. The outline in Figure 3-8 is based on an electrical classification. Besides their classification, NEMA also classified electric motors according to

- Size
  - Fractional horsepower
  - Integral horsepower
- Application
  - General purpose
  - Definite purpose (e.g., shell type)
  - Special purpose
  - Part-Winding

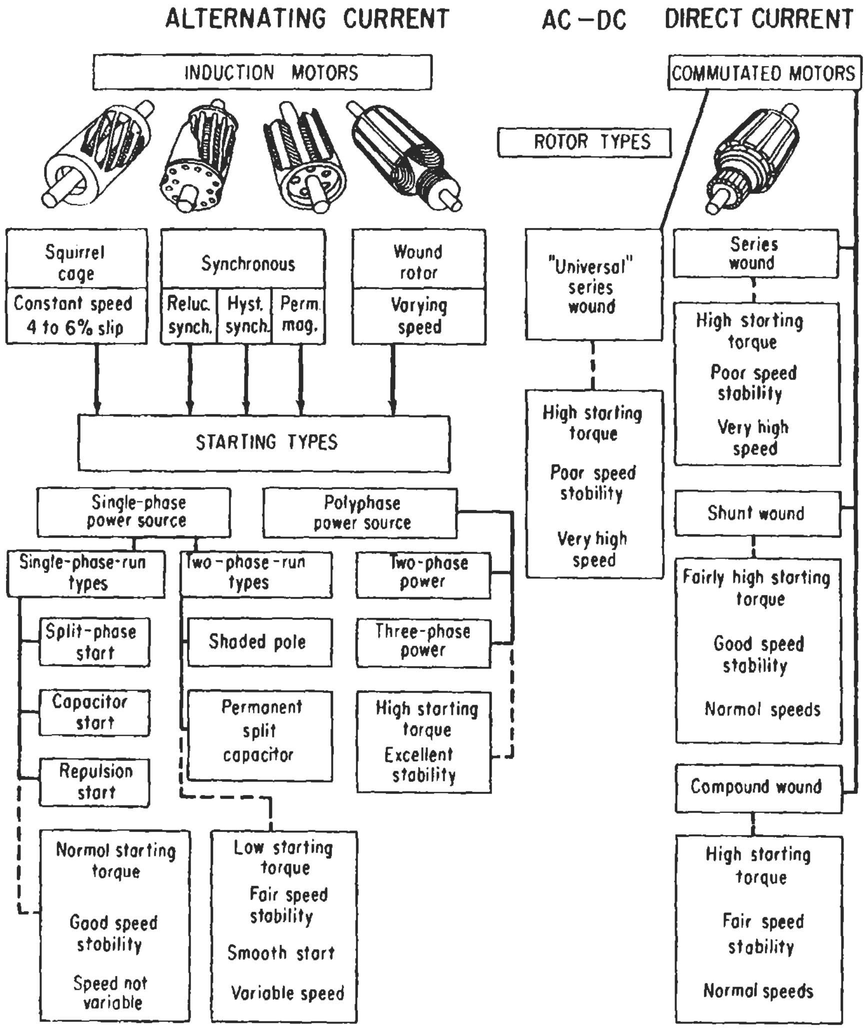


Figure 3-8. Electric motor classification.

Mechanical protection and cooling
- Open machine (with subclasses)
- Totally enclosed machine (with subclasses)

Variability of speed
- Constant speed
- Varying speed
- Adjustable speed
- Multispeed

In what follows, electric motors are described in accordance with their electric type as shown in Figure 3-8 [7,8].

Alternating-Current Motors

Alternating-current motors are of three general types, induction, synchronous, and series, and are defined as follows:

Induction Motors. An induction motor is an alternating-current motor in which a primary winding on one member (usually the stator) is connected to the power source and a polyphase secondary winding or a squirrel-cage secondary winding on the other member (usually the rotor) carries induced current. There are two types:

Squirrel-Cage Induction Motor. A squirrel-cage induction motor is one in which the secondary circuit consists of a squirrel-cage winding suitably disposed in slots in the secondary core.

Wound-Rotor Induction Motor. A wound-rotor induction motor is an induction motor in which the secondary circuit consists of a polyphase winding or coils whose terminals are either short circuited or closed through suitable circuits.

Synchronous Motor. A synchronous motor is a synchronous machine which transforms electrical power into mechanical power.

Series-Wound Motor. A series-wound motor is a motor in which the field circuit and armature circuit are connected in series.

Polyphase Motors. Alternating-current polyphase motors are of the squirrel-cage, wound-rotor, or synchronous types.

Design Letters. Polyphase squirrel-cage integral-horsepower induction motors may be one of the following:

Design A. A Design A motor is a squirrel-cage motor designed to withstand full-voltage starting and to develop locked-rotor torque as shown in MG 1-12.37, pull-up torque as shown in MG 1-12.39, breakdown torque as shown in MG 1-12.38 with locked-rotor current higher than the values shown in MG 1-12.34 for 60 Hz and MG 1–12.25 for 50 Hz and having a slip at rated load of less than 5%. Motors with 10 or more poles may have slip slightly greater than 5%.

Design B. A Design B motor is a squirrel-cage motor designed to withstand full-voltage starting and to develop locked-rotor, breakdown and pull-up torques adequate for general application as specified in MG 1-12.37, MG 1-12.38, and MG1-12.39, drawing locked-rotor current not to exceed the values shown in MG 1-12.34 for 60 Hz and MG 1-12.35 for 50 Hz, and having a slip at rated load of less than 5%. Motors with 10 and more poles may have slip slightly greater than 5%.

Design C. A Design C motor is a squirrel-cage motor designed to withstand full-voltage starting and to develop locked-rotor torque for special high-torque application up to the values shown in MG 1-12.37, pull-up torque as shown in MG 1-12.39, breakdown torque up to the values shown in MG 1-12.38, with locked-rotor current not to exceed the values shown in MG 1-12.34 for 60 Hz and MG 1-12.35 for 50 Hz, and having a slip at rated load of less than 5%.

Design D. A Design D motor is a squirrel-cage motor designed to withstand full-voltage starting and to develop high locked-rotor torque as shown in MG 1-12.37 with locked-rotor current not greater than shown in MG 1-12.34 for 60 Hz and MG 1-12.35 for 50 Hz, and having a slip at rated load of 5% or more.

Single-phase Motors. Alternating-current single-phase motors are usually induction or series motors, although single-phase synchronous motors are available in the smaller ratings.

Design Letters. Single-phase integral-horsepower motors may be one of the following:

Design L. A Design L motor is a single-phase integral-horsepower motor designed to withstand full-voltage starting and to develop a breakdown torque as shown in MG 1-10.33 with a locked-rotor current not to exceed the values shown in MG 1-12.33.

Design M. A Design M motor is a single-phase integral-horsepower motor designed to withstand full-voltage starting and to develop a breakdown torque as shown in MG 1-10.33 with a locked-rotor current not to exceed the values shown in MG 1-12.33.

Single-Phase Squirrel-cage Motors. Single-phase squirrel-cage induction motors are classified and defined as follows:

Split-Phase Motor. A split-phase motor is a single-phase induction motor equipped with an auxiliary winding, displaced in magnetic position from, and connected in parallel with, the main winding. *Note:* Unless otherwise specified, the auxiliary circuit is assumed to be opened when the motor has attained a predetermined speed. The term "split-phase motor," used without qualification, described a motor to be used without impedance other than that offered by the motor windings themselves, other types being separately defined.

Resistance-Start Motor. A resistance-start motor is a form of split-phase motor having a resistance connected in series with the auxiliary winding. The auxiliary circuit is opened when the motor has attained a predetermined speed.

Capacitor Motor. A capacitor motor is a single-phase induction motor with a main winding arranged for a direct connection to a source of power and an auxiliary winding connected in series with a capacitor. There are three types of capacitor motors, as follows:

Capacitor-start motor. A capacitor-start motor is a capacitor motor in which the capacitor phase is in the circuit only during the starting period.

Permanent-split capacitor motor. A permanent-split capacitor motor is a capacitor motor having the same value of capacitance for both starting and running conditions.

Two-value capacitor motor. A two-value capacitor motor is a capacitor motor using different values of effective capacitance for the starting and running conditions.

Shaded-Pole Motor. A shaded-pole motor is a single-phase induction motor provided with an auxiliary short-circuited winding or windings displaced in magnetic position from the main winding. This makes the motor self-starting.

Single-Phase Wound-Rotor Motors. Single wound-rotor motors are defined and classified as follows:

Repulsion Motor. A repulsion motor is a single-phase motor that has a stator winding arranged for connection to a commutator. Brushes on the commutator are short circuited and are so placed that the magnetic axis of the stator winding. This type of motor has a varying-speed characteristic.

Repulsion-Start Induction Motor. A repulsion-start induction motor is a single-phase motor having the same windings as a repulsion motor, but at a predetermined speed the rotor winding is short circuited or otherwise connected to give the equivalent of a squirrel-cage winding. This type of motor starts as a repulsion motor but operates as an induction motor with constant-speed characteristics.

Repulsion-Induction Motor. A repulsion-induction motor is a form of repulsion motor that has a squirrel-cage winding in the rotor in addition to the repulsion motor winding. A motor of this type may have either a constant-speed (see MG 1-1.30) or varying-speed (see MG 1-1.31) characteristic.

Universal Motors. A universal motor is a series-wound motor designed to operate at approximately the same speed and output on either direct current or single-phase alternating current of a frequency not greater than 60 cycles/s and approximately the same rms voltage. There are two types:

Series-Wound Motor. A series-wound motor is a commutator motor in which the field circuit and armature circuit are connected in series.

Compensated Series Motor. A compensated series motor is a series motor with a compensating field winding. (The compensating field winding and the series field winding may be combined into one field winding.)

Direct-Current Motors

Direct-current motors are of three general types, shunt wound, series wound, and compound wound, and are defined as follows:

Shunt-Wound Motor. A shunt-wound motor is a direct-current motor in which the field circuit and armature circuit are connected in parallel.

Straight Shunt-Wound Motor. A straight shunt-wound motor is a direct-current motor in which the field circuit is connected either in parallel with the armature circuit or to a separate source of excitation voltage. The shunt field is the only winding supplying field excitation.

Stabilized Shunt-Wound Motor. A stabilized shunt-wound motor is a direct-current motor in which the shunt field circuit is connected either in parallel with the armature circuit or to a separate source of excitation voltage, and which also has a light series winding added to prevent a rise in speed or to obtain a slight reduction in speed with increase in load.

Series-Wound Motor. A series-wound motor is a motor in which the field circuit and armature circuit are connected in series.

Compound-Wound Motor. A compound-wound motor is a direct-current motor which as two separate field windings. One, usually the predominating field, is connected in parallel with the armature circuit. The other is connected in series with the armature circuit.

Permanent Magnet Motor. A permanent magnet motor is a direct-current motor in which the field excitation is suppled by permanent magnets.

Rating, Performance, and Test

The following defines and describes the commonly used terms of electric motor rating, performance, and testing [7]:

Rating of a Machine. The rating of a machine shall consist of the output power together with any other characteristics, such as speed, voltage, and current, assigned to it by the manufacturer. For machines that are designed for absorbing power, the rating shall be the input power.

Continuous Rating. The continuous rating defines the load that can be carried for an indefinitely long period of time.

Short-Time Rating. The short-time rating defines the load that can be carried for a short and definitely specified time.

Efficiency. The efficiency of a motor or generator is the ratio of its useful power output to its total power input and is usually expressed in percentage.

Power Factor. The power factor of an alternating-current motor or generator is the ratio of the kilowatt input (or output) to the kVA input (or output) to the kVA input (or output) and is usually expressed as a percentage.

Service Factor of Alternating-current Motors. The service factor of an alternating-current motor is a multiplier that, when applied to the rated horsepower, indicates a permissible horsepower loading that may be carried under the conditions specified for the service factor (see MG 1-14.35).

Speed Regulation of Direct-Current Motors. The speed regulation of a direct-current motor is the difference between the steady no-load speed and the steady rated-load speed, expressed in percent of rated-load speed.

Secondary Voltage of Wound-Rotor Motors. The secondary voltage of wound-rotor motors is the open-circuit voltage at standstill, measured across the slip rings, with rated voltage applied on the primary winding.

Full-Load Torque. The full-load torque of a motor is the torque necessary to produce its rated horsepower at full-load speed. In pounds at a 1-ft radius, it is equal to the horsepower multiplied by 5,252 divided by the full-load speed.

Locked-Rotor Torque (Static Torque). The locked-rotor torque of a motor is the minimum torque that it will develop at rest for all angular positions of the rotor, with rated voltage applied at rated frequency.

Pull-Up Torque. The pull-up torque of an alternating-current motor is the minimum torque developed by the motor during the period of acceleration from rest to the speed at which breakdown torque occurs. For motors that do not have a definite breakdown torque, the pull-up torque is the minimum torque developed up to rated speed.

Breakdown Torque. The breakdown torque of a motor is the maximum torque that it will develop with rated voltage applied at rated frequency, without an abrupt drop in speed.

Pull-Out Torque. The pull-out torque of a synchronous motor is the maximum sustained torque that the motor will develop at synchronous speed with rated voltage applied at rated frequency and with normal excitation.

Pull-In Torque. The pull-in torque of a synchronous motor is the maximum constant torque under which the motor will pull its connected inertia load into synchronism, at rated voltage and frequency, when its field excitation is applied. The speed to which a motor will bring its load depends on the power required to drive it. Whether the motor can pull the load into step from this speed depends on the inertia of the revolving parts, so that the pull-in torque cannot be determined without having the Wk^2 as well as the torque of the load.

Locked-Rotor Current. The locked-rotor current of a motor is the steady-state current taken from the line with the rotor locked and with rated voltage (and rated frequency in the case of alternating-current motors) applied to the motor.

Temperature Tests. Temperature tests are tests taken to determine the temperature rise of certain parts of the machine above the ambient temperature, when running under a specified load.

Ambient Temperature. Ambient temperature is the temperature of the surrounding cooling medium, such as gas of liquid, which comes into contact with the heated parts of the apparatus. *Note:* Ambient temperature is commonly known as "room temperature" in connection with air-cooled apparatus not provided with artificial ventilation.

High-Potential Test. High-potential tests are tests that consist of the application of a voltage higher than the rated voltage for a specified time for the purpose of determining the adequacy against breakdown of insulating materials and spacings under normal conditions. (See MG 1, Part 3.)

Starting Capacitance for a Capacitor Motor. The starting capacitance for a capacitor motor is the total effective capacitance in series with the starting winding under locked-rotor conditions.

Radial Magnetic Pull and Axial Centering Force.

Radial Magnetic Pull. The radial magnetic pull of a motor or generator is the magnetic force on the rotor resulting from its radial (air gap) displacement from magnetic center.

Axial Centering Force. The axial centering force of a motor or generator is the magnetic force on the rotor resulting from its axial displacement from magnetic center. *Note:* Unless other conditions are specified, the value of radial magnetic pull and axial centering force will be for no load, with rated voltage, rated field current, and rated frequency applied, as applicable.

Induction Motor Time Constants.

General. When a polyphase induction motor is open circuited or short circuited while running at rated speed, the rotor flux linkages generate a voltage in the stator winding. The decay of the rotor flux linkages, and the resultant open-circuit terminal voltage or short-circuit current, is determined by the various motor time constants defined by the following equations:

Open-circuit AC time constant:

$$T''_{do} = \frac{X_M + X_2}{2\pi f r_2} \quad (s) \tag{3-6}$$

Short-circuit AC time constant:

$$T''d = \frac{X_S}{X_1 + X_M} T''_{do} \quad (s) \tag{3-7}$$

Short-circuit DC time constant:

$$T_a = \frac{X_S}{2\pi f r_1 \left(1 + \frac{LL_S}{kW_1}\right)} \quad (s) \tag{3-8}$$

X/R ratio:

$$X/R = \frac{X_S}{r_1 \left(1 + \frac{LL_S}{kW_1}\right)} \quad (rad) \tag{3-9}$$

Terms (see Figure 3-9):

r_1 = Stator DC resistance per phase corrected to operating temperature.
r_2 = Rotor resistance per phase at rated speed and operating temperature referred to stator.

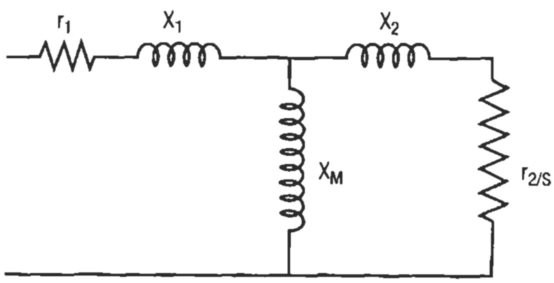


Figure 3-9. Motor circuit [7].

X_1 = Stator leakage reactance per phase at rated current.
X_2 = Rotor leakage reactance per phase at rated speed and rated current referred to stator.
X_S = Total starting reactance (stator and rotor) per phase at zero speed and locked-rotor current.
X_M = Magnetizing reactance per phase.
LL_s = Fundamental-frequency component of stray-load loss in kilowatts at rated current.
kW_1 = Stator I^2R loss in kilowatts at rated current and operating temperature.
f = Rated frequency in hertz.
s = Slip in per unit of synchronous speed.

AC Performance Examples

In general, the typical electric motor applications in the oil and gas industry are polyphase motors (either squirrel-cage or wound-rotor motors).

Squirrel-Cage Motor. This type of motor finds a broader application and a more extensive and general use than any other type of motor. This is because it is, inherently, the simplest type of electric motor and, also, has excellent characteristics and operates essentially at constant speed. It has greater reliability and low maintenance requirements and thus meets a broad range of applications.

Torque, horsepower, and speed requirements demanded in drives for most machines can be met with one of four designs of squirrel-cage polyphase induction motors. Each design offers a different combination of torque, speed, and current characteristics to meet the operating requirements of various industrial applications.

All four designs can withstand full-voltage starting directly across the power lines, that is, the motors are strong enough mechanically to withstand magnetic stresses and the locked-rotor torques developed at the time the switch is closed.

Design A produces exceptionally high breakdown torques but at the expense of high locket rotor currents that normally require provision for starting with reduced voltage. This motor is suitable for machines in which the friction and inertia loads are small.

Design B has normal starting torque adequate for a wide variety of industrial machine drives and a starting current usually acceptable on power systems. This design is suitable where slightly more than full load torque and low slip is required, also where relatively high breakdown torque is needed to sustain occasional emergency overloads, or where a low locked-rotor current is needed. These motors are for use in driving machine tools, blowers, centrifugal pumps, and textile machines.

Design C has high starting torque and a normal breakdown torque. Applications for this design are machines in which inertia loads are high at starting, but normally run at rated full load and are not subjected to high overload demands after running speed has been reached. Conveyors, plunger pumps, compressors that are not unloaded at starting, and over chain conveyors, also hoists, cranes, and machine tools where a quick start and reversal are required are typical examples of such machines.

Design D develops extremely high starting torque with moderate starting current. This design uses a high-resistance-type rotor to obtain variation of speed with load and has no sharply defined breakdown torque. This motor eases off in speed when surge loads are encountered and also develops high torque to recover speed rapidly. Typical applications for this motor are machines in which heavy loads are suddenly applied and removed at frequent intervals, such as hoists, machines with large flywheels, conventional punch presses, and centrifuges.

Slip ratings of the four designs are:

Design	Slip, %
A	Less than 5
B	Less than 5
C	Less than 5
D	5 or more

Note that motors with 10 or more poles may have slip slightly greater than 5%.

Figure 3-10 shows the typical torque-speed performance curves for various designs of polyphase squirrel-cage induction motors [9].

Tables 3-5 and 3-6 give the typical locked-rotor torque developed by Designs A, B, and C motors [8].

Table 3-7 gives the typical breakdown torque for Design B and C motors with continuous ratings [8]. Breakdown torques for Design A motors are in excess of the values given for Design B motors.

It is usually good practice to apply motors at momentary loads at least 20% below the values given for maximum torque in order to offset that much torque drop caused by an allowable 10% voltage drop.

Wound Rotor. Characteristics of wound-rotor motors are such that the slip depends almost entirely upon the load on the motor. The speed returns practically to maximum when the load is removed. This characteristic limits the use of these motors on applications where reduced speeds at light loads are described.

Wound-rotor motors are suitable for constant-speed applications that require frequent starting or reversing under load or where starting duty is severe and exceptionally high starting torque is required.

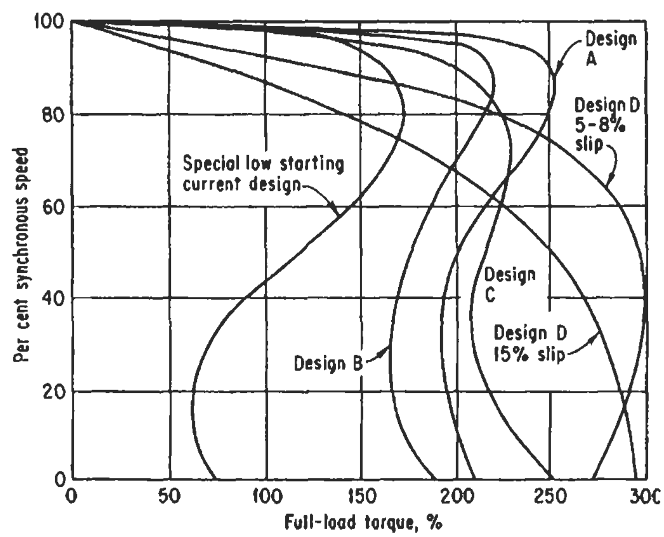


Figure 3-10. Torque-speed curves for various designs of polyphase squirrel-cage induction motors [9].

Table 3-5
Locked-Rotor Torque of Design A and B Motors with Continuous Ratings [8]

Hp	Synchronous speeds, rpm							
Frequency:								
60 cycles.....	3600	1800	1200	900	720	600	514	450
50 cycles.....	3000	1500	1000	750	600	500	428	375
½	...	...	...	150	150	115	110	105
¾	...	...	175	150	150	115	110	105
1	...	275	175	150	150	115	110	105
1½	175	265	175	150	150	115	110	105
2	175	250	175	150	145	115	110	105
3	175	250	175	150	135	115	110	105
5	150	185	160	130	130	115	110	105
7½	150	175	150	125	120	115	110	105
10	150	175	150	125	120	115	110	105
15	150	165	140	125	120	115	110	105
20	150	150	135	125	120	115	110	105
25	150	150	135	125	120	115	110	105
30	150	150	135	125	120	115	110	105
40	135	150	135	125	120	115	110	105
50	125	150	135	125	120	115	110	105

Values are expressed in per cent of full-load torque and represent the upper limit of the range of application.
Values are based on rated voltage and frequency.

Table 3-6
Locked Rotor Torque of Design C Motors with Continuous Ratings [8]

Hp	Synchronous speeds, rpm		
Frequency:			
60 cycles........	1800	1200	900
50 cycles........	1500	1000	750
3	....	250	225
5	250	250	225
7½	250	225	200
10	250	225	200
15	225	200	200
20	200	200	200
25 and larger	200	200	200

Values are expressed in per cent of full-load torque and represent the upper limit of the range of application.
Values are based on rated voltage and frequency.

Table 3-7
Breakdown Torque of Design B and C, 60 and 50-cycle Motors with Continuous Ratings [8]

Hp	Synchronous speed, rpm	Design B	Design C	Hp	Synchronous speed, rpm	Design B	Design C
½	900–750	250			1800–1500	275	
	Lower than 750	200		3	1200–1000	250	225
					900–750	225	200
	1200–1000	275			Lower than 750	200	
¾	900–750	250					
	Lower than 750	200			3600–3000	225	
					1800–1500	225	200
	1800–1500	300		5	1200–1000	225	200
1	1200–1000	275			900–750	225	200
	900–750	250			Lower than 750	200	
	Lower than 750	200					
					3600–300	215	
	3600–300	300			1800–1500	215	190
	1800–1500	300		7½	1200–1000	215	190
1½	1200–1000	275			900–750	215	190
	900–750	250			Lower than 750	200	
	Lower than 750	200					
					3600–3000	200	
	3600–3000	275			1800–1500	200	190
	1800–1500	275		10	1200–1000	200	190
2	1200–1000	250			900–750	200	190
	900–750	225			Lower than 750	200	
	Lower than 750	200					
				15 and larger	All speeds	200	190
3	3600–3000	250					

Values are expressed in per cent of full-load torque and represent the upper limit of the range of application.

Values are based on rated voltage and frequency.

A second major type of application is where speed adjustment is required. Controllers are used with these motors to obtain adjustable speed over a considerable range. But at any point of adjustment the speed will vary with a change in load. It is usually not practicable to operate at less than 50% of full speed by introducing external resistance in the secondary circuit of the motor. Horsepower output at 50% of normal speed is approximately 40% of rated horsepower.

Use of wound-rotor induction motors has been largely in continuous-duty constant-speed supplications where particularly high starting torques and low starting currents are required simultaneously, such as in reciprocating pumps and compressors. These motors are also used where only alternating current is available to drive machines that require speed adjustment, such as types of fans and conveyors.

Typical torque-speed characteristics of wound-rotor motors are shown in the curves in Figure 3-11 for full voltage and for reduced voltage that are obtained with different values of secondary resistance.

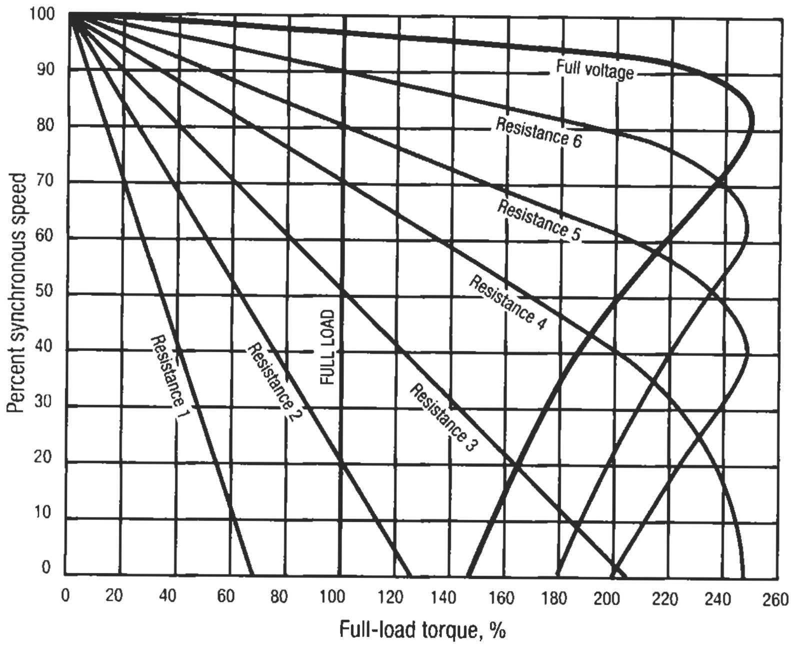


Figure 3-11. Typical torque-speed characteristics of wound-rotor motors [8].

DC Performance Examples

Important advantages of direct-current motors for machine drives are the adjustable speed over a wide range, the fact that speeds are not limited to synchronous speeds, and the variation of speed-torque characteristics [8].

Shunt-Wound Motors. These motors operate at approximately constant speed regardless of variations in load when connected to a constant supply voltage and with fixed field excitation. Maximum decrease in speed as load varies from no load to full load is about 10–12%.

Constant-speed motors are usually suited for a speed range of less than 3 to 1 by field control, but mechanical and electrical characteristics govern maximum safe speeds. With constant voltage on the armature, as the field is weakened the speed increases and the motor develops constant horsepower.

For speed ranges of 3 to 1 or greater, adjustable-voltage operation is recommended. The adjustable-voltage drive usually includes a motor-generator set to supply an adjustable voltage supply to the armature of the drive motor. With this method of operation, a speed range of 10 to 1 readily obtained.

Regenerative braking can also be obtained with the variable-voltage drive. This system permits deceleration of the driven load by causing the motor to operate as a generator to drive the motor-generator set, thereby returning power to the alternating-current power supply.

With an adjustable voltage supply to the armature, at speeds below basic, the motor is suitable for a constant-torque drive. Minimum speeds are limited by temperature rise, because the motor carries full-load current at the lower speeds and at low speeds the ventilation is reduced. Adjustable voltage drives are used for paper mills, rubber mill machinery, winders, machine-tool drives, and hoists.

Adjustable-speed shunt-wound motors with field control are designed for operation over speed ranges of 3 to 1 or greater. Standard adjustable-speed motors are rated in three ways: tapered horsepower, continuous, 40°C rise; constant horsepower, continuous, 40°C rise; and constant horsepower, 1 hr, 50°C rise, of the next larger horsepower rating than for continuous duty.

The 1 hr, 50°C open motors develop constant horsepower over the entire speed range. Semienclosing covers can be added without changing the rating.

The 40°C continuous-rated open motors develop the rated horsepower from 150% of minimum speed to the maximum speed. From minimum speed to 159% of minimum speed, the rated horsepower will be developed continuously without exceeding safe temperature limits.

Tapered horsepower motors develop the maximum rated horsepower at three times the minimum speed, the horsepower decreasing in direct proportion to the decrease in speed down to the horsepower rating at 150% of the minimum speed. Figure 3-12 plots characteristics of shunt, series, and compound-wound direct-current motors.

Series-Wound Motors. These motors are inherently varying-speed motors with charges in load. On light or no loads, the speed may become dangerously high. These motors should be employed only where the load is never entirely removed from the motor. They should never be connected to the driven machine by belt.

Series motors are used on loads that require very high starting torques or severe accelerating duty or where the high-speed characteristics may be advantageous, such as in hoists.

Compound-Wound Motors. These motors are used to drive machines that require high starting torque or in which the loads have a pulsating torque. Changes in load usually produce wide speed regulation. This motor is not suited for adjustable speed by field control.

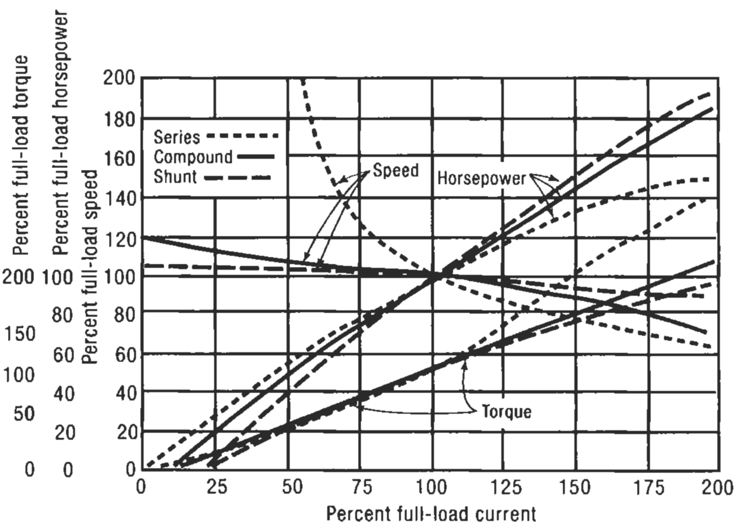


Figure 3-12. Direct-current motor performance curves [8].

The relative strength of the shunt and series fields of the motor determines to what extent the motor approaches the shunt or series characteristics.

From no load to full load, the drop in speed of compound-wound motors is approximately 25%. Compound-wound motors are used where reasonably constant speed is required and for loads where high starting torque is needed to accelerate the drive machine.

Compound-wound motors should be used on machines having flywheel or high inertia loads, wherein the dropping speed characteristics of the motor causes the flywheel to give up its energy as the shock load comes on, thereby cutting down the power peaks taken from the line and resulting in less heating of the motor.

Industry Applications

The oil and gas industry utilizes electric motors as prime movers in a number of industry operations. The major applications and the electric motors typically utilized are now described [8,10].

Well Drilling and Completions. In the past, most drilling and completion rigs utilized direct drive diesel engines as prime movers. This was especially so of the large (deeper drilling) land rigs. The smaller land rigs utilize direct drive gas (including LPG and natural gas) engines. However, over the past decade, there has been an increasing trend to utilize diesel-engine-driven generators, which in turn operate direct-current motors for large land rigs and especially offshore drilling rigs. These direct-current electric motors are used for the drawworks and rotary table drives because of high starting-torque requirements for these operations.

Production. Torque requirements vary widely during the production pumping cycle, and peaks occur when the sucker-rod string and fluid load are lifted and when the counterweight is lifted. NEMA Design D motors, although relatively expensive, are well suited to this service, since they minimize current peaks and provide adequate torque under all service conditions, including automatic operation by time control. NEMA Design C motors may be used where operating conditions are less severe. NEMA Design B motors must be used with care in this service to avoid high cyclic current peaks, which may be objectionable on a small system, particularly if several wells should "get in step." The use of Design B motors can also lead to oversizing of motors in an attempt to obtain sufficient starting torque. This results in the operation of the motor at a relatively low load factor, with consequent low power factor.

Double or Triple-Rated Motors. There are special motors developed for oil-well pumping. They are totally enclosed, fan-cooled NEMA Design D motors that can be reconnected for 2–3-hp ratings at a common speed of 1,200 r/min. Typical horsepower ratings are 20/15/10 and 50/40/30. They provide flexibility in the field since they permit the selection of the horsepower rating at which the motor may be operated most efficiently. They also permit changing the pumping speed by changing the motor pulley and reconnecting the motor.

Single-Phase Operation. If single-phase power only is available, it is advisable to consider the use of single-phase to three-phase converters and three-phase motors. This avoids the use of large single-phase capacitor start motors, which are relatively expensive and contain a starting switch that could be a source of trouble due to failure or to the presence of flammable gas in the vicinity of the well.

Oil-Well Control. A packaged control unit is available to control individual oil-well pumps. It contains, in a weatherproof enclosure, a combination magnetic starter, a time switch that can start and stop the motor according to a predetermined program, a timing relay that delays the start of the motor following a power failure, and lightning arresters. Pushbutton control is also provided.

Power-Factor Correction. The induction motors used for oil-well pumping have high starting torques with relatively low power factors. Also, the average load on these motors is fairly low. Therefore, it is advisable to consider the installation of capacitors to avoid paying the penalty imposed by most power companies for low-power factor. They will be installed at the individual motors and switched with them, if voltage drop in the distribution system is to be corrected as well as power factor. Otherwise they may be installed in large banks at the distribution center, if it is more economical to do so.

Oil Pipelines. The main pumps for an oil pipeline usually driven by 3,600 rpm induction electric motors having NEMA Design B characteristics. Full-voltage starting is used. Figure 3-13 shows a typical pumping station diagram. Such pumping stations are located at the beginning of the pipeline and at intervals along the line. Intermediate or booster stations must be capable of operating under varying conditions due to differences in liquid gravity, withdrawals at intermediate points, and the shutting down of other booster stations. Pumping stations often contain two or three pumps connected in series, with bypass arrangements using check valves across each pump. The pumps may all be of the same capacity, or one of them may be half size. By operating the pumps singly or together, a range of pumping capacities can be achieved.

Throttling of pump discharge may also be used to provide finer control and to permit operation when pump suction pressure may be inadequate for full flow operation.

Pumping stations are often unattended and may be remotely controlled by radio or telephone circuits.

Motor enclosures for outdoor use are NEMA weather-protected Type II, totally enclosed, fan-cooled, or drip-proof with weather protection. Motors of the latter type are widely used. Not only are they less expensive than the other types, but they also have a service factor of 1.15. The above enclosure types are all suitable for the Class I, Group D, Division 2 classifications usually encountered.

If the pumps are located indoors, a Division 1 classification is likely to apply. Motors must be Class I, group D, explosion-proof, or they may be separately ventilated with clean outside air brought to the motor by fans. Auxiliary devices such as alarm contacts on the motor must be suitable for the area classification. The installed costs, overall efficiencies, and service factors associated with the enclosures that are available will influence the selection.

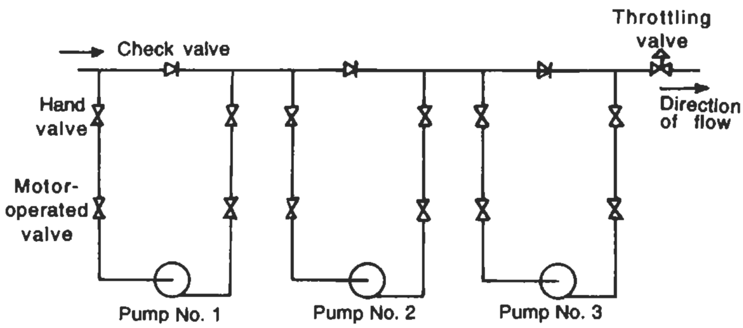


Figure 3-13. Typical oil pipeline pump station [10].

Natural Gas Pipelines. Compressor drivers are usually reciprocating gas engines or gas turbines, to make use of the energy available in the pipeline. Electric motor drives use slow-speed synchronous motors for reciprocating compressors and four or six-pole induction motors with gear increases for high-speed centrifugal compressors. Motor voltages, types, and enclosures are selected as for oil pipeline pumps. Motors used with centrifugal compressors must develop sufficient torque at the voltage available under in-rush conditions to accelerate the high inertia load. They must also have adequate thermal capacity for the long starting time required, which may be 20 or 30 s.

Motor Control

Efficient use of electric motors requires appropriate control systems. Typical control systems for AC and DC electric motor operations are now discussed [10].

Inverter-AC Motor Drives. An adjustable-frequency control of AC motors provide efficient operation with the use of brushless, high-performance induction, and synchronous motors. A typical system is shown in Figure 3-14. Such a system consists of a rectifier (which provides DC power from the AC line) and an inverter (which converts the DC power to adjustable-frequency AC power for the motor). Inverter cost per kilowatt is about twice that of controller rectifiers; thus the power convertor for an AC drive can approach three times the cost of a DC drive.

These AC drive systems require the inverters to operate with either low-slip induction motors or reluctance-type synchronous-induction motors. Such systems are used where DC commutator motors are not acceptable. Examples of such applications are motor operations in hazardous atmospheres and high motor velocities.

The power convertor must provide the AC motor with low-harmonic voltage waveform and simultaneously allow the amplitude to be adjusted. This avoids magnetic saturation of the motor as the frequency is adjusted. For constant torque, from maximum speed to base speed, the voltage is adjusted proportional to frequency. Above base speed, the motor is usually operated at constant horsepower. In this region the voltage is held constant and the flux density declines. Also, the convertor must limit the starting current, ensure operation at favorable slip, and provide a path for reverse power flow during motor slowdown.

Inverters are designed with various power semiconductor arrangements. Power semiconductor elements of the inverter operate like switches by synthesizing the motor voltage waveform from segments of the DC bus voltage. For power ranges up to about 5 hp, convertors can use power transistors to synthesize six-step (per cycle), three-phase voltage for frequency ranges from 10 to 120 Hz for standard motors and from 240 to 1,200 Hz for high-frequency motors. For the conventional drive range from 5 to 500 hp, thyristor inverters are used to develop either six-step per cycle, twelve-step per cycle, or pulse-width modulated (pwm) voltages over typical frequency ranges from 10 to 120 Hz.

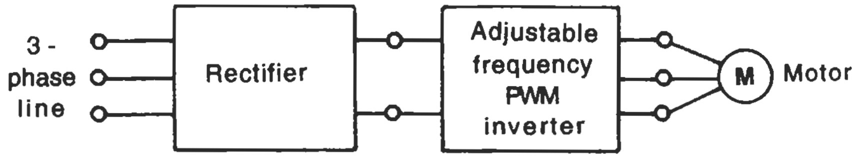


Figure 3-14. Typical invester AC motor drive consisting of rectifier-DC link, adjustable-frequency inverter, and induction of synchronous motor [10].

The voltage control of the six-step thyristor inverter can be accomplished in the rectifier, or the inverter. The output voltage of a six-step inverter is directly proportional to the DC bus voltage. Thus, control of the rectifier is usually sufficient provided that the inverter commutation capacitors are charged from a fixed-voltage bus. Another method that can be used in dealing with a fixed DC bus voltage is to operate two inverter bridges in series with adjustable relative phase shift to control the motor voltage. An additional method is to use an adjustable-ratio output transformer (Variac).

In general, the usage of the voltage control methods is being replaced by pulse-width-modulation (pwm) techniques. These techniques allow the DC bus voltage to remain fixed, and, thus, lower-cost systems with motor voltages of potentially lower harmonic content can be obtained. A variety of pwn techniques are now used in inverters. One technique requires that each half-cycle of motor voltage waveform be divided into a fixed number of pulses, (typically four or six) and the pulse width modulated to control the voltage. In another technique, the pulse number is increased with pulse period and the width modulated for voltage control. An additional technique requires that the pulse number be fixed, but the width is graded over the half-cycle as a sine function, and the relative widths are controlled.

The inverter drive system that uses a current-controlled rectifier and parallel-capacitor commutation operates to both improve reliability and reduce cost. Such systems are built commercially for the ranges from 20 to 500 hp for the typical 20:1 constant-torque speed range.

Primary-Voltage-Control-AC Motor Driver. Induction motor torque at any slip s is proportional to primary V^2. Rotor-power dissipation is equal to s times the air-gap power. These two relationships define the boundary of operation of an induction motor with primary voltage control of speed. As the speed is reduced (s increased) at constant torque, the air-gap power remains fixed, but the power divides between rotor circuit dissipation and mechanical shaft power.

Solid-state primary voltage controllers are built with phase-controlled thyristors in each AC circuit and are used with high-slip NEMA Design D motors. The thyristors are either placed in the three supply lines (as shown in Figure 3-15) in the star ends of a wye motor winding or in the arms of a delta motor winding. Commercial systems are built to 100 hp or more for pumps and related applications. Such systems have the inherent limitation of thyristor rating and rotor heating.

Wound-Rotor Motor Drives. The wound-rotor induction motor, using adjustable rotor-circuit resistance, is rarely used. However, two versions that use solid-state auxiliary equipment are finding limited application.

Just as for the primary-voltage control system, the air-gap power at slip s divides between mechanical power and rotor-circuit power in the wound-rotor drive. Except

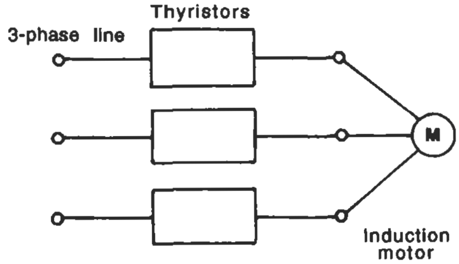


Figure 3-15. Primary voltage central system [10].

for the losses in the rotor windings, the rotor-circuit power is extracted from the slip rings and is disposed of external to the machine. One type of solid-state system is shown in Figure 3-16a. In this system the power to a resistor is controlled by thyristors. Another type of solid-state system is shown in Figure 3-16b. In this system the slip-ring power is rectified and returned to the supply line through a line-commutated convertor. The first system is wasteful of power. The second system uses power in an economical manner.

The rating of the solid-state equipment depends upon the torque requirement as a function of speed. For a constant-torque drive, the auxiliary system at starting must handle the full motor rated power. For pump drives, the auxiliary system ratings are reduced considerably. The convertor operates with phase-controlled firing signals from the supply line, such as for a DC drive rectifier and develops relatively constant-speed characteristics. The stator-rotor winding ratio is made slightly larger than unity to ensure that power flow is toward the supply line at the lowest speed (e.g., at stall).

Solid-State DC Drives. The controlled-thyristor rectifier and separate-field DC motor is the solid-state motor drive in greatest use. The combination provides control over at least a 10:1 speed range, plus an additional two to three times by field weakening. Depending upon the power level, the rectifier is operated directly from the AC supply lines, or via a transformer. Typical speed regulation of ±2% can be accomplished with a single control system. The horsepower and speed limitations are set by the DC motor, not by the semiconductor rectifiers. The DC motor and rectifier can be combined to any required power level.

Commercial solid-state DC motor drives fall into three general groups. Drives operating from single-phase lines are available in fractional horsepower sizes up to about 3 hp. Three-phase drives are available in horsepower sizes from 5 to 500 hp. Drivers above 500 hp are generally classed as special.

Speed adjustment from base speed downward is obtained by armature voltage control. The armature current and torque in this range is limited by the thyristor ratings or motor temperature rise. Control above base speed at constant horsepower is obtained by field weakening. An example system is shown in Figure 3-17a. In this

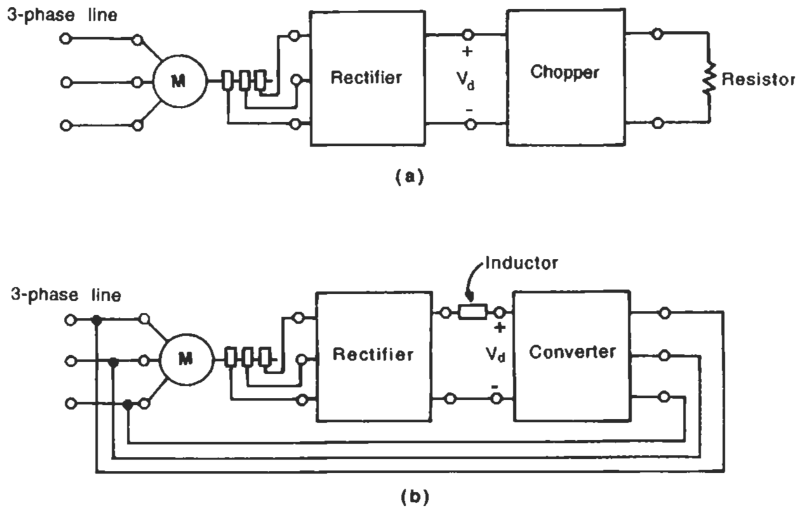


Figure 3-16. Wound-rotor systems: (a) rotor power dissipated in resistance; (b) rotor power retirement to supply line [10].

example three or six thyristors are used in the rectifier. Reversible operation requires either the use of two rectifiers connected as shown in Figure 3-17b, a reversing armature contactor, or a field-reversing technique.

Speed adjustment, current limiting, and regulation against load, temperature, and line-voltage disturbances require a control system for the armature and field rectifier thyristors. For typical ±2% regulation, the speed signal is derived from the armature voltage, corrected for IR drop. For regulation to ±0.1%, the speed signal is obtained from analog or digital tachometers and processed in solid-state digital or analog circuits. Signals that override the constant-speed control system limit armature current during speed changes, acceleration, and reversal.

POWER TRANSMISSION

In nearly all mechanical power applications in the oil and gas industry it is necessary to transmit the power generated by a prime mover to an operation (e.g., drawworks of a drilling rig, or a production pumping system). The transmission of rotary power to such operation elements is carried out by a power transmission system. Mechanical power transmission is typically carried out by power betting systems, chain systems, gear systems and by hydraulic systems, or some combination of these three [1,5].

Power Belting and V Belts

There are basically four general categories of power betting [8]. These are:

1. Flat
2. V section, or V belts

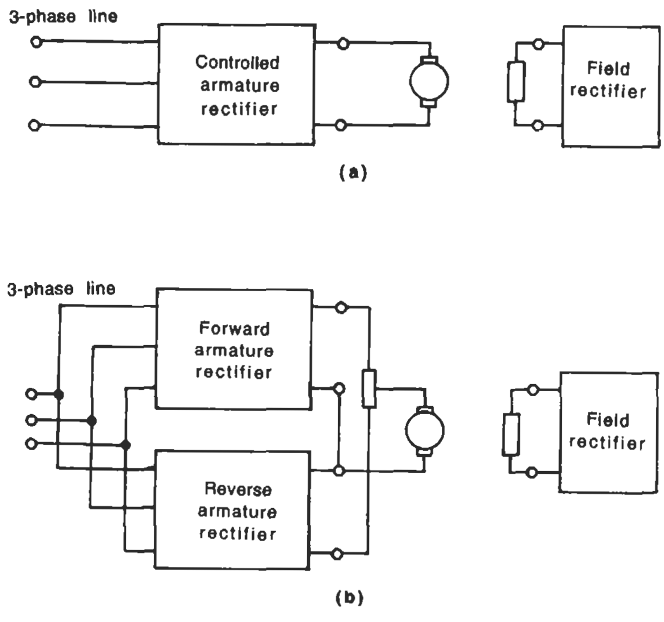


Figure 3-17. Rectifier-DC motor driver: (a) unidirectional or field-reversing; (b) bidirectional operation with armature voltage reversing [10].

3. Ribbed, or joined V Belts
4. Toothed, or timing.

These are illustrated in Figure 3-18. Flat belts are used chiefly for conveyer belt systems rather than power transmission. Toothed (or timing) belt are generally used for control of critically timed rotating elements of a mechanical systems and are not used in power transmission application. The V section and the ribbed power belting systems are used exclusively for power transmission, and thus the discussion in this section will be confined to these categories of belting.

In general, the choice of power belt drive depends upon such faster or speed, reduction ration, positive-drive requirements, and center distances, shaft relationship (i.e., magnitude of skew and load).

V belts were developed from the older rope drive systems. The grooved sheaves required for ropes became V shaped, and the belt itself was contoured to put the groove. There are several different configurations of V belts for power transmission applications. Standard sections for V belts are shown in Figure 3-19. The important difference between flat belts and V belts is that they are able to transmit higher torque at smaller widths and tensions than flat belts. The reason for this is the wedging action of the V belts is the sheave groove. Figure 3-20 shows this wedging action of the belt in its sheave groove (Figure 3-21 shows typical sheave designs for single and multiple V belts) [11].

In addition to the standard types of V belts (see Figure 3-19), many manufacturers make V belts that are specifically designed to have a higher horsepower capacity. These V belts are shown and compared to standard V-belt cross-sections. These superpower V belts allow shorter center distances and narrower sheaves without imposing any extra total bearing stresses. Such belts are able to reduce the drive by 30–50% and less for horsepower capacity. In addition, speeds can be increased up to 6000 ft/min without dynamic balancing of the sheaves because the sheaves are smaller.

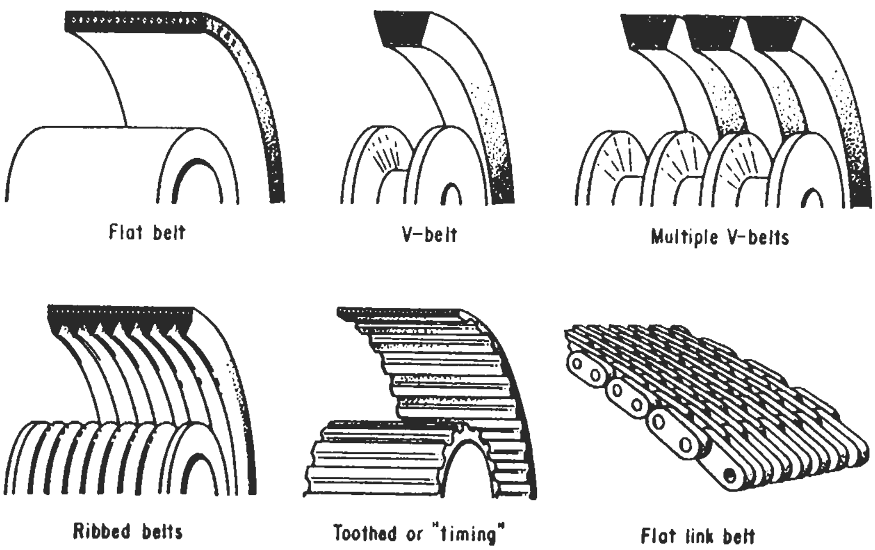


Figure 3-18. Power belt types [8].

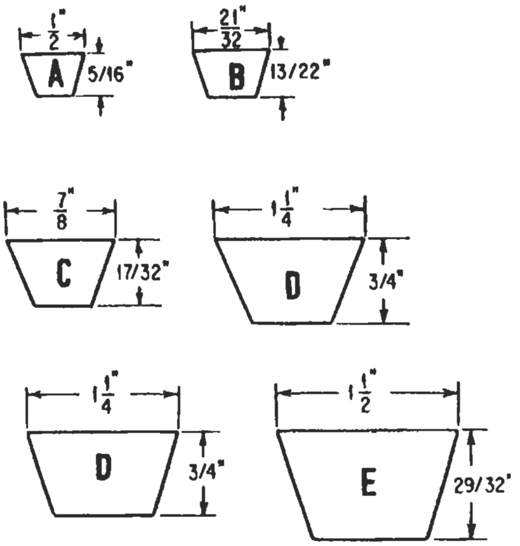


Figure 3-19. Standard V-belt coss-sections [8].

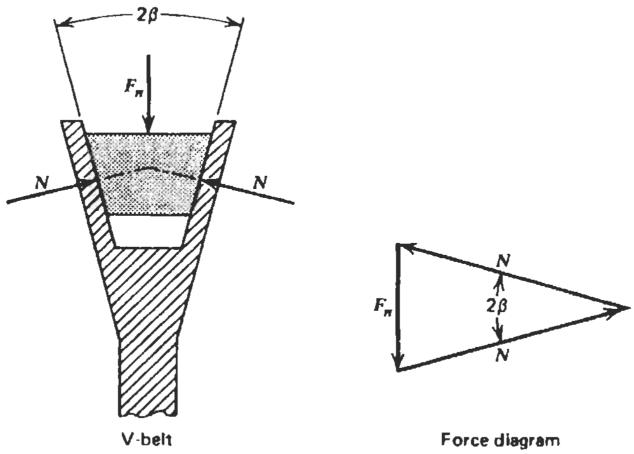


Figure 3-20. Element of V belt in sheave [11].

Figure 3-22 shows the general dimensions of an open-belt drive. For V belt drives, the pulling diameter D and d are pitch diameters [11]. The quantity C in the center distance, θ in the contact angle for the smaller pulling, and 2x is the angular deviation from a 180° angle of contact.

From Figure 3-22, θ (radians) is approximately

$$\theta \approx \pi - \frac{D - d}{c} \qquad (3\text{-}10)$$

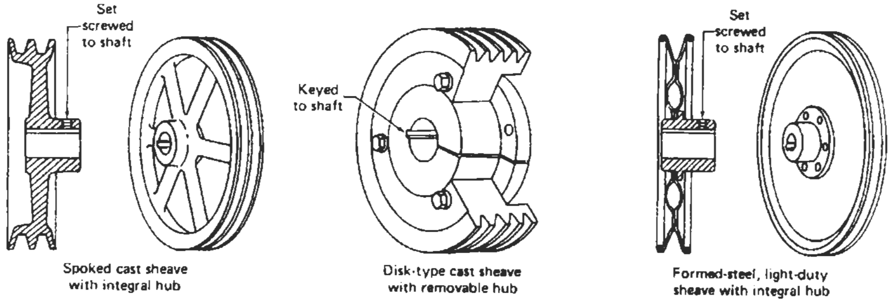


Figure 3-21. Types of sheaves for V belts [11].

For V-belt applications D and d are sheave pitch diameters. From the geometry of Figure 3-22, θ is always less than or equal to 180° or π radians. The lower guideline for θ is approximately 150°. Below this value there will be increasing tension and slip, which will result in decreased life of the V belts. This limit on θ imposes a lower limit on the center distance and thus a practical limit on the speed ratio attainable is a given V-belt design.

Shorter center distance is a very practical design objective. Such a design uses space economically and allows for a stable operation. In general, center distances are limited by the physical dimensions of the sheaves, or the minimum angle of θ, i.e., 150°. Maximum drive centers are limited only by available V-belt stocked lengths.

The center distance is

$$C = 0.0625\{b = [b^2 - 32(D - d)\ 2]^{1/2}\} \tag{3-11}$$

where D is the pitch diameter of the larger sheave (usually the driven sheave) in in., d is the pitch diameter of the smaller sheave (usually the drive sheave) in in., and the quantity b is

$$b = 4L - 6.28(D + d) \tag{3-12}$$

where L is the V-belt pitch length in in.

From the above L is

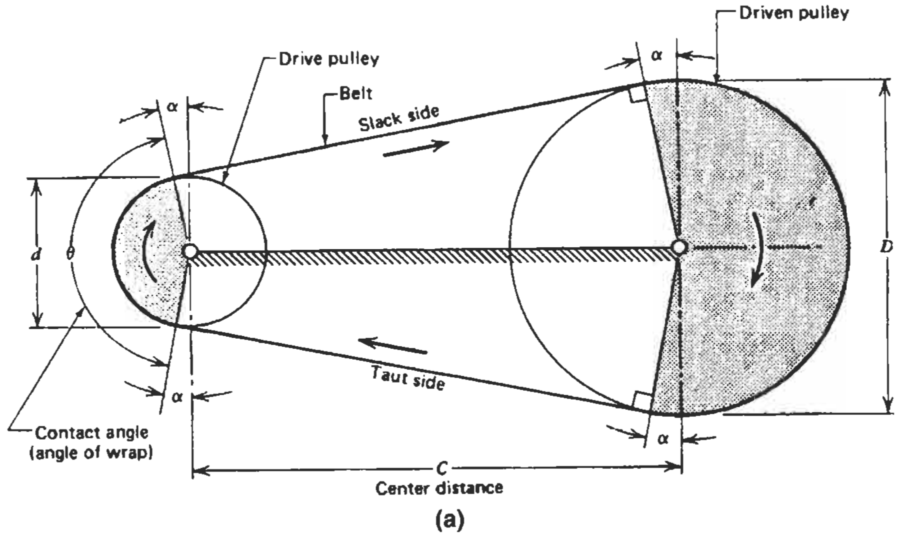
$$L = 2C + 1.57(D + d) + \frac{(D + d)^2}{4} \tag{3-13}$$

Modern V belts are nearly all of the closed-loop continuous type which come in standard lengths.

For V-belt drives with more than two sheaves, belt length is calculated from sheave coordinates and drive dimensions in layout drawings.

Most prime movers rotate at a higher speed than the operational driven equipment. Therefore, speed reduction is necessary for most belt drive systems. The speed ratio i between the prime mover drive shaft and the drive shaft is

$$i = \frac{\text{speed of drive shaft (rpm or ft/s)}}{\text{speed of driven shaft (rpm or ft/s)}} \tag{3-14}$$



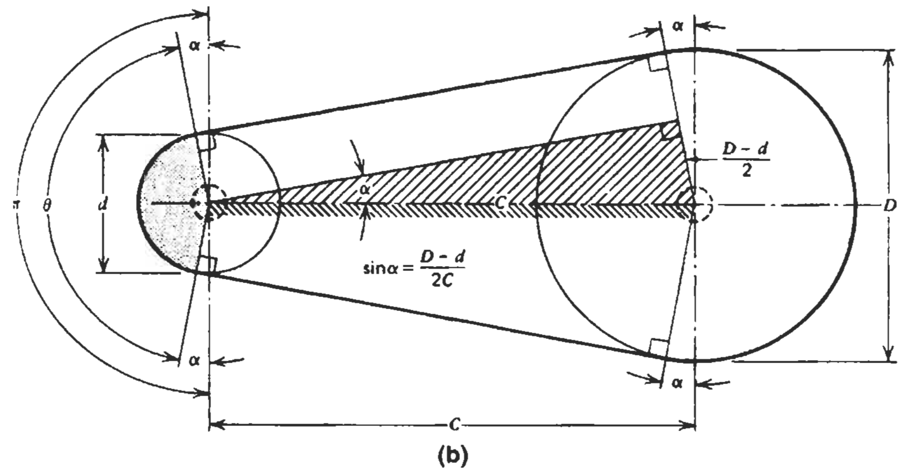


Figure 3-22. V-belt geometry [11].

This can also be expressed in terms of sheave pitch diameter. This is

$$i = \frac{\text{pitch diam. of larger sheave (in.)}}{\text{pitch diam. of smaller sheave (in.)}} \tag{3-15}$$

Under normal operating conditions speed reduction due to slippage and creep of the V-belt material is usually quite small. Neglecting these effects, the velocity v (in./s) of the V belt is

$$v = \frac{\pi D n_D}{60} = \frac{\pi d n_d}{60} \tag{3-16}$$

where n_D is the large sheave speed and n_d is the small sheave speed, both in rpm.

In general, modern V-belt speeds are limited to about 6,000 ft/min (v = 100 ft/s), and the relationship between D, d, n_D, n_d, and i is

$$n_D = \frac{d}{D} n_d = \frac{n_d}{i} \tag{3-17}$$

Slippage will lead to a reduction in V-belt speed and output speed. Therefore,

$$n_D = n_d \frac{1 - S}{i} \tag{3-18}$$

where S is the slip in percent divided by 100. When slippage occurs at both sheaves, then

$$n_D = \frac{n_d}{i}(1 - S_1)(1 - S_2) \tag{3-19}$$

where S_1 is the percent slippage in the drive sheave (i.e., d), S_2 is the percent slippage in the drive sheave (i.e., D).

From Figure 3-20 the relationship between the forces N (lb) normal to the V-belt sides and the force F_n (lb) can be obtained. This is

$$2N = \frac{F_n}{\sin \beta} \tag{3-20}$$

The total friction force, F_t (lb) acting tangent to the sheave is

$$F_t = \frac{f F_n}{\sin \beta} \tag{3-21}$$

where f is the coefficient of friction between the V belt and the sheave surface. The typical value for 2β is 32–40° (average value 36°). The typical value of f is about 0.30 (rubber against cast iron or steel).

Figure 3-23 shows the constant speed forces in the V belts with drive sheave transferring power via the V belt to the large driven sheave [11]. If centrifugal forces and slippage are neglected, then taut side tension force F_1 (lb) is

$$F_1 = F_2 e^{f\theta/\sin\beta} \tag{3-22}$$

where F_2 (lb) is the slack side tension.

The power P(hp) transmitted by the drive sheave to the driven sheave is

$$P = \frac{F_1\left(1 - e^{-(f\theta/\sin\beta)}\right)v}{550} \tag{3-23}$$

The force F_1 is the taut side of the V belt. In practice this force cannot exceed the allowable V-belt load F_{max} (lb). Equation 3-23 is valid only for rather slow speeds where centrifugal force can be neglected.

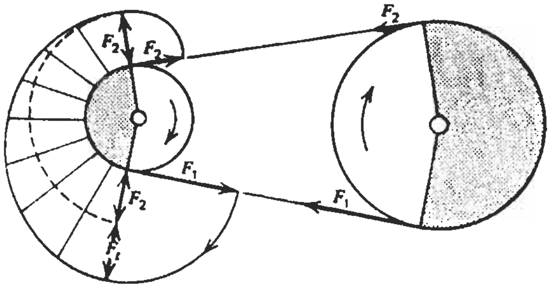


Figure 3-23. Equilibrium forces [11].

If centrifugal force is not neglected, then the power P transmitted is

$$P = \frac{(F_{max} - F_c)(1 - e^{-(f\theta/\sin\beta)})v}{550} \tag{3-24}$$

where F_c (lb) is the centrifugal force created by the high speed of the V belt. The force F_c is

$$F_c = mv^2 \tag{3-25}$$

where m is the mass per unit length of the V belt (lb-s²/ft²).

From Equations 3-24 and 3-25, an optimum velocity may be found for higher speed V-belt applications. This optimum velocity V_{op} (ft/s) is

$$V_{op} = \left(\frac{F_{max}}{3m}\right)^{1/2} \tag{3-26}$$

and the optimum power P_{op} (hp) is

$$P_{op} = \frac{(F_{max} - mV_{op2})(1 - e^{-(f\theta/\sin\beta)})V_{op}}{550} \tag{3-27}$$

Service factors for V-belt systems are typically 1.3 to 1.6.

API has developed specifications for V belting in oil-field power transmission applications [12]. Reference 12 is used to carry out detailed design calculations. Although this is an API publication, the specifications contained are consistent with specifications of other industrial groups. The basic calculation techniques above have been carried out for the stock V belts available. These data are tabulated in the API publication to simplify the design effect for V-belt power transmission systems.

Using the data contained in the API specifications for V belting, the following steps can be used to find V-belt drive system dimensions.

1. Find the design horsepower by multiplying the service factor by the name-plat horsepower of the motor.
2. Find the correct V-belt cross-section (i.e., A, B, C, D, E, 3V, 5V or 8V) from Figure 3-24 or 3-25 using the design horsepower and the speed (rpm) of the fastest shaft.
3. Calculate the speed ratio using Equation 3-14 or 3-15.

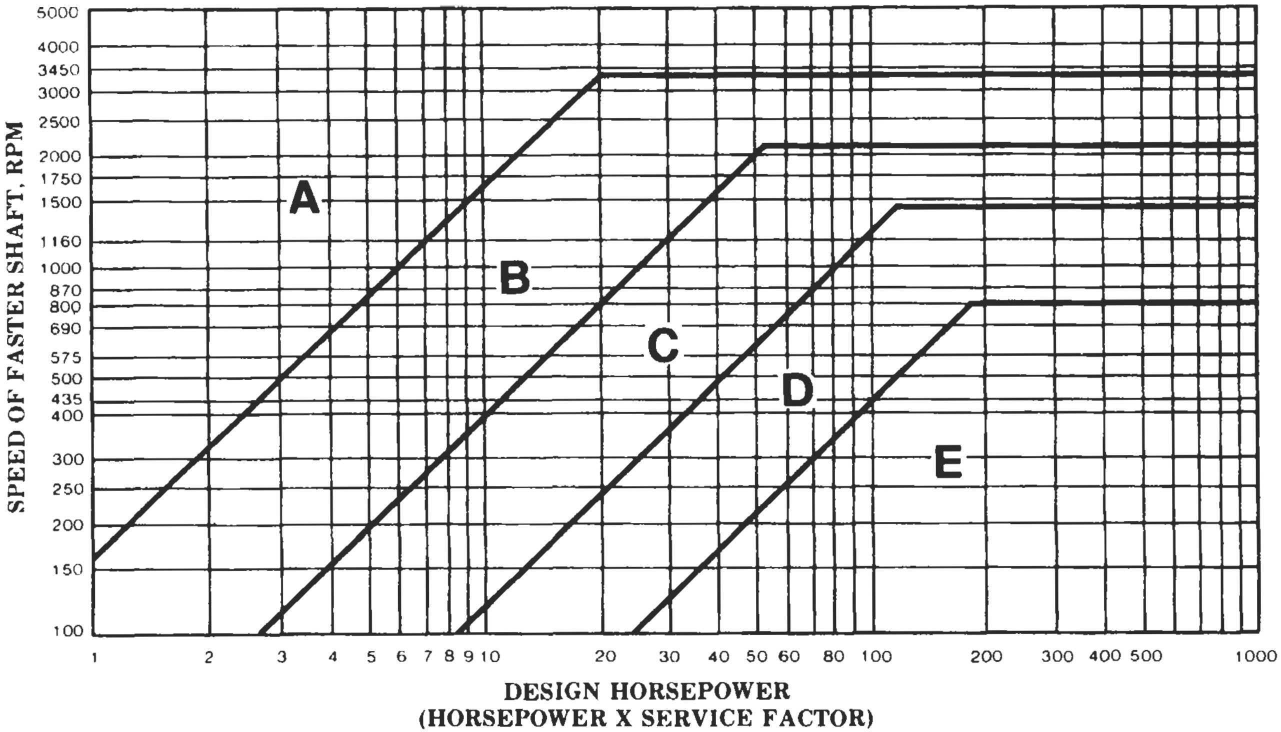


Figure 3-24. Guide for selecting V-belt cross-section [12].

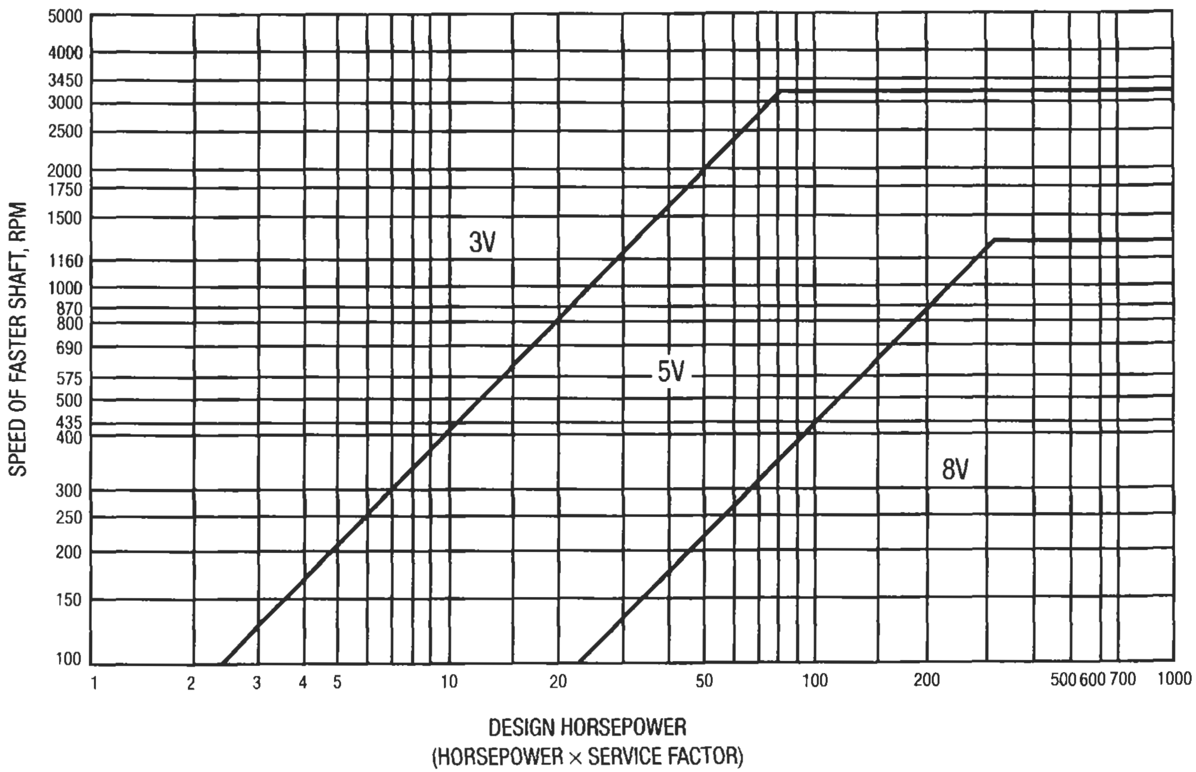


Figure 3-25. Guide for selecting narrow V-belt cross-sections [12].

4. (a) Select the drive sheave size when neither the drive sheave nor the driven sheave are fixed in size from Table 3-8. (b) Select the driven sheave size. To find the larger driven sheave size, substitute the drive sheave size obtained in a) above into D = id. Note that D and d are pitch diameters of the sheaves.
5. Find the V-belt speed using Equation 3-16 using pitch diameters of sheaves. Speed must be below 600 ft/min (or 100 ft/s).
6. Find the belt length and center distance from Equations 3-11–3-13. A center distance slightly less than, or equal to, the sum of the sheave diameters is suggested for V-belt applications. The V-belt length is calculated by using Equation 3-9 once the center distance has been estimated. This initial center distance does not result in a length for a stock V belt. It is then necessary to adjust the center distance to obtain the closest V-belt stock length. This is a trial-and-error solution.
7. Knowing the speed (in rpm) of the smallest sheave and its pitch diameter, find the horsepower capability per V belt using the appropriate table for the V-belt type to be used (see Tables 3-9–3-16).
8. Find the number of V belts required using the horsepower capacity per V belt obtained in step 7 above. This is

$$\text{Number of V belts} = \frac{\text{Design hp}}{\text{hp per belt}}$$

   If the number of V belts is a fraction, use next highest whole number.
9. The highest V-belt drive can now be tabulated. This is
   - V-belt type (step 2)
   - Number of V belts (step 8)
   - Drive sheave design (steps 4 and 8)
   - Driven sheave design (steps 4 and 8)
   - Center distance and V-belt stock length (Step 6)

Chains

Power transmission chains provide a positive drive even when operated under very adverse temperatures (−60 to 600°F) and other environmental conditions. These power transmission systems are very flexible with regards to their field applications. In general, chain drives are primarily selected for low-speed and medium-speed service. Some silent chain designs may be used in high-speed service [8,13].

There are six types of chains used for power transmission. These are roller, silent (inverted tooth), offset link (Ewart with bushing), detachable (open Ewart), pintle (closed Ewart), and bead.

(text continues on page 439)

Table 3-8
Minimum Recommended Sheave Sizes [8]

V-belt section	A	B	C	D	E
Minimum pitch dia, in	3.0	5.4	9.0	13.0	21.6
No. of grooves on stock pulleys	1–6	1–6	3–14	3–17	5–19

Table 3-9
Horsepower Ratings for A Section V Belts [12]

Basic Horsepower Per Belt — Small Sheave Pitch Diameter, Inches

1	2	3	4	5	6	7	8	9	10	11	12	13	14
RPM OF FASTER SHAFT	2.60	2.80	3.00	3.20	3.40	3.60	3.80	4.00	4.20	4.40	4.60	4.80	5.00
575	0.46	0.55	0.63	0.72	0.80	0.88	0.97	1.05	1.13	1.21	1.29	1.37	1.45
690	0.53	0.63	0.73	0.83	0.93	1.02	1.12	1.22	1.32	1.41	1.51	1.60	1.70
725	0.55	0.65	0.76	0.86	0.96	1.07	1.17	1.27	1.37	1.47	1.57	1.67	1.77
870	0.63	0.75	0.87	0.99	1.12	1.24	1.36	1.47	1.59	1.71	1.82	1.94	2.06
950	0.67	0.80	0.93	1.07	1.20	1.33	1.45	1.58	1.71	1.84	1.96	2.09	2.21
1160	0.77	0.93	1.08	1.24	1.40	1.55	1.70	1.86	2.01	2.16	2.30	2.45	2.60
1425	0.88	1.07	1.26	1.45	1.63	1.82	2.00	2.18	2.36	2.53	2.71	2.88	3.05
1750	1.01	1.23	1.46	1.68	1.90	2.11	2.33	2.54	2.75	2.96	3.16	3.36	3.56
2850	1.31	1.64	1.97	2.29	2.60	2.91	3.21	3.50	3.78	4.06	4.33	4.59	4.84
3450	1.40	1.78	2.15	2.51	2.86	3.20	3.52	3.84	4.14	4.43	4.71	4.97	5.22
200	0.20	0.23	0.27	0.30	0.33	0.36	0.40	0.43	0.46	0.49	0.52	0.55	0.59
400	0.35	0.41	0.47	0.53	0.59	0.65	0.71	0.77	0.83	0.89	0.95	1.01	1.07
600	0.48	0.56	0.65	0.74	0.83	0.91	1.00	1.09	1.17	1.26	1.34	1.42	1.51
800	0.59	0.70	0.82	0.93	1.04	1.16	1.27	1.38	1.49	1.59	1.70	1.81	1.92
1000	0.69	0.83	0.97	1.11	1.24	1.38	1.52	1.65	1.78	1.91	2.04	2.17	2.30
1200	0.78	0.95	1.11	1.27	1.43	1.59	1.75	1.91	2.06	2.21	2.37	2.52	2.67
1400	0.87	1.06	1.25	1.43	1.61	1.79	1.97	2.15	2.32	2.50	2.67	2.84	3.01
1600	0.95	1.16	1.37	1.58	1.78	1.98	2.18	2.38	2.57	2.77	2.96	3.14	3.33
1800	1.02	1.26	1.49	1.71	1.94	2.16	2.38	2.59	2.81	3.02	3.22	3.43	3.63
2000	1.09	1.34	1.59	1.84	2.08	2.32	2.56	2.79	3.02	3.25	3.47	3.69	3.91

Additional Horsepower Per Belt — Speed Ratio Range

1	15	16	17	18	19	20	21	22	23	24
RPM OF FASTER SHAFT	1.00 TO 1.01	1.02 TO 1.04	1.05 TO 1.07	1.08 TO 1.10	1.11 TO 1.14	1.15 TO 1.20	1.21 TO 1.27	1.28 TO 1.39	1.40 TO 1.64	1.65 AND OVER
575	0.00	0.01	0.02	0.03	0.03	0.04	0.05	0.06	0.07	0.08
690	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
725	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.08	0.09	0.10
870	0.00	0.01	0.03	0.04	0.05	0.06	0.08	0.09	0.10	0.12
950	0.00	0.01	0.03	0.04	0.06	0.07	0.08	0.10	0.11	0.13
1160	0.00	0.02	0.03	0.05	0.07	0.09	0.10	0.12	0.14	0.16
1425	0.00	0.02	0.04	0.06	0.08	0.11	0.13	0.15	0.17	0.19
1750	0.00	0.03	0.05	0.08	0.10	0.13	0.16	0.18	0.21	0.23
2850	0.00	0.04	0.08	0.13	0.17	0.21	0.25	0.30	0.34	0.38
3450	0.00	0.05	0.10	0.15	0.20	0.26	0.31	0.36	0.41	0.46
200	0.00	0.00	0.01	0.01	0.01	0.01	0.02	0.02	0.02	0.03
400	0.00	0.01	0.01	0.02	0.02	0.03	0.04	0.04	0.05	0.05
600	0.00	0.01	0.02	0.03	0.04	0.04	0.05	0.06	0.07	0.08
800	0.00	0.01	0.02	0.04	0.05	0.06	0.07	0.08	0.10	0.11
1000	0.00	0.01	0.03	0.04	0.06	0.07	0.09	0.10	0.12	0.13
1200	0.00	0.02	0.04	0.05	0.07	0.09	0.11	0.12	0.14	0.16
1400	0.00	0.02	0.04	0.06	0.08	0.10	0.12	0.15	0.17	0.19
1600	0.00	0.02	0.05	0.07	0.10	0.12	0.14	0.17	0.19	0.21
1800	0.00	0.03	0.05	0.08	0.11	0.13	0.16	0.19	0.21	0.24
2000	0.00	0.03	0.06	0.09	0.12	0.15	0.18	0.21	0.24	0.27

2200	1.15	1.42	1.69	1.96	2.22	2.48	2.73	2.98	3.23	3.47	3.71	3.94	4.17
2400	1.21	1.50	1.79	2.07	2.35	2.62	2.89	3.16	3.42	3.67	3.92	4.16	4.40
2600	1.25	1.57	1.87	2.17	2.47	2.76	3.04	3.32	3.59	3.86	4.12	4.37	4.61
2800	1.30	1.63	1.95	2.27	2.58	2.88	3.18	3.47	3.75	4.02	4.29	4.55	4.80
3000	1.34	1.68	2.02	2.35	2.68	2.99	3.30	3.60	3.89	4.17	4.44	4.71	4.96
3200	1.37	1.73	2.08	2.43	2.76	3.09	3.41	3.71	4.01	4.30	4.57	4.84	5.09
3400	1.40	1.77	2.14	2.50	2.84	3.18	3.50	3.82	4.12	4.41	4.68	4.95	5.20
3600	1.42	1.81	2.19	2.55	2.91	3.25	3.58	3.90	4.20	4.49	4.77	5.03	5.28
3800	1.43	1.83	2.23	2.60	2.97	3.32	3.65	3.97	4.27	4.56	4.83	5.09	5.32
4000	1.44	1.86	2.26	2.64	3.01	3.36	3.70	4.02	4.32	4.60	4.87	5.11	5.34
4200	1.44	1.87	2.28	2.67	3.04	3.40	3.74	4.05	4.35	4.63	4.88	5.11	5.32
4400	1.44	1.88	2.29	2.69	3.07	3.42	3.76	4.07	4.36	4.62	4.86	5.08	5.26
4600	1.43	1.87	2.30	2.70	3.07	3.43	3.76	4.06	4.34	4.59	4.82	5.01	5.18
4800	1.42	1.86	2.29	2.69	3.07	3.42	3.74	4.04	4.30	4.54	4.74	4.91	
5000	1.39	1.85	2.28	2.68	3.05	3.40	3.71	3.99	4.24	4.46	4.64		
5200	1.36	1.82	2.25	2.65	3.02	3.36	3.66	3.93	4.16	4.35			
5400	1.33	1.79	2.22	2.62	2.98	3.30	3.59	3.84	4.05				
5600	1.29	1.75	2.17	2.57	2.92	3.23	3.50	3.73					
5800	1.24	1.70	2.12	2.50	2.84	3.14	3.39	3.60					
6000	1.18	1.64	2.06	2.43	2.76	3.04	3.26						
6200	1.11	1.57	1.98	2.34	2.65	2.91							
6400	1.04	1.49	1.89	2.24	2.53	2.77							
6600	0.96	1.40	1.79	2.12	2.40								
6800	0.87	1.31	1.68	1.99	2.24								
7000	0.78	1.20	1.56	1.85									
7200	0.67	1.08	1.42										
7400	0.56	0.96	1.28										
7600	0.44	0.82	1.12										
7800	0.31	0.67											

2200	0.00	0.03	0.07	0.10	0.13	0.16	0.20	0.23	0.26	0.29
2400	0.00	0.04	0.07	0.11	0.14	0.18	0.21	0.25	0.29	0.32
2600	0.00	0.04	0.08	0.12	0.15	0.19	0.23	0.27	0.31	0.35
2800	0.00	0.04	0.08	0.12	0.17	0.21	0.25	0.29	0.33	0.37
3000	0.00	0.04	0.09	0.13	0.18	0.22	0.27	0.31	0.36	0.40
3200	0.00	0.05	0.09	0.14	0.19	0.24	0.29	0.33	0.38	0.43
3400	0.00	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45
3600	0.00	0.05	0.11	0.16	0.21	0.27	0.32	0.37	0.43	0.48
3800	0.00	0.06	0.11	0.17	0.23	0.28	0.34	0.40	0.45	0.51
4000	0.00	0.06	0.12	0.18	0.24	0.30	0.36	0.42	0.48	0.53
4200	0.00	0.06	0.12	0.19	0.25	0.31	0.37	0.44	0.50	0.56
4400	0.00	0.07	0.13	0.20	0.26	0.33	0.39	0.46	0.52	0.59
4600	0.00	0.07	0.14	0.21	0.27	0.34	0.41	0.48	0.55	0.61
4800	0.00	0.07	0.14	0.21	0.29	0.36	0.43	0.50	0.57	0.64
5000	0.00	0.07	0.15	0.22	0.30	0.37	0.45	0.52	0.59	0.67
5200	0.00	0.08	0.15	0.23	0.31	0.39	0.46	0.54	0.62	0.69
5400	0.00	0.08	0.16	0.24	0.32	0.40	0.48	0.56	0.64	0.72
5600	0.00	0.08	0.17	0.25	0.33	0.42	0.50	0.58	0.67	0.75
5800	0.00	0.09	0.17	0.26	0.34	0.43	0.52	0.60	0.69	0.78
6000	0.00	0.09	0.18	0.27	0.36	0.45	0.53	0.62	0.71	0.80
6200	0.00	0.09	0.18	0.28	0.37	0.46	0.55	0.64	0.74	0.83
6400	0.00	0.10	0.19	0.29	0.38	0.48	0.57	0.67	0.76	0.86
6600	0.00	0.10	0.20	0.29	0.39	0.49	0.59	0.69	0.78	0.88
6800	0.00	0.10	0.20	0.30	0.40	0.51	0.61	0.71	0.81	0.91
7000	0.00	0.10	0.21	0.31	0.42	0.52	0.62	0.73	0.83	0.94
7200	0.00	0.11	0.21	0.32	0.43	0.53	0.64	0.75	0.86	0.96
7400	0.00	0.11	0.22	0.33	0.44	0.55	0.66	0.77	0.88	0.99
7600	0.00	0.11	0.23	0.34	0.45	0.56	0.68	0.79	0.90	1.02
7800	0.00	0.12	0.23	0.35	0.46	0.58	0.69	0.81	0.93	1.04

Rim Speed of the A/B Combination Sheave is Above 6000 Feet per Minute. Special Sheaves May Be Necessary.

Rim Speed of the Regular A Section Sheave is Above 6000 Feet per Minute. Special Sheaves May Be Necessary.

Table 3-10
Horsepower Ratings for B Section V Belts [12]

Basic Horsepower Per Belt

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
RPM OF FASTER SHAFT	Small Sheave Pitch Diameter, Inches																	
	4.60	4.80	5.00	5.20	5.40	5.60	5.80	6.00	6.20	6.40	6.60	6.80	7.00	7.20	7.40	7.60	7.80	8.00
575	1.62	1.77	1.91	2.05	2.19	2.33	2.47	2.61	2.75	2.89	3.02	3.16	3.30	3.43	3.57	3.70	3.84	3.97
690	1.87	2.04	2.21	2.37	2.54	2.70	2.86	3.03	3.19	3.35	3.51	3.67	3.83	3.99	4.15	4.30	4.46	4.61
725	1.94	2.12	2.29	2.47	2.64	2.81	2.98	3.15	3.32	3.49	3.65	3.82	3.99	4.15	4.32	4.48	4.64	4.80
870	2.23	2.44	2.64	2.84	3.04	3.24	3.44	3.64	3.84	4.03	4.23	4.42	4.61	4.81	5.00	5.18	5.37	5.56
950	2.38	2.60	2.82	3.04	3.26	3.47	3.69	3.90	4.11	4.32	4.53	4.74	4.94	5.15	5.35	5.55	5.76	5.96
1160	2.75	3.01	3.27	3.53	3.78	4.04	4.29	4.54	4.79	5.03	5.27	5.52	5.76	5.99	6.23	6.46	6.70	6.93
1425	3.17	3.48	3.78	4.08	4.38	4.68	4.97	5.26	5.55	5.83	6.12	6.39	6.67	6.94	7.21	7.48	7.74	8.00
1750	3.61	3.97	4.32	4.67	5.02	5.36	5.69	6.02	6.35	6.67	6.99	7.30	7.61	7.91	8.21	8.50	8.79	9.07
2850	4.47	4.94	5.40	5.84	6.26	6.67	7.07	7.44	7.81	8.15	8.48	8.79	9.08	9.36	9.61	9.85	10.06	10.26
3450	4.50	4.97	5.43	5.86	6.27	6.65	7.00	7.33	7.63	7.90	8.14	8.35						
200	0.69	0.75	0.80	0.86	0.91	0.97	1.02	1.07	1.13	1.18	1.24	1.29	1.34	1.40	1.45	1.50	1.56	1.61
400	1.22	1.32	1.42	1.53	1.63	1.73	1.83	1.93	2.03	2.13	2.23	2.33	2.43	2.53	2.63	2.73	2.83	2.92
600	1.68	1.83	1.98	2.12	2.27	2.41	2.56	2.70	2.85	2.99	3.13	3.27	3.41	3.56	3.70	3.83	3.97	4.11
800	2.09	2.29	2.48	2.66	2.85	3.04	3.22	3.41	3.59	3.77	3.96	4.14	4.32	4.49	4.67	4.85	5.03	5.20
1000	2.47	2.70	2.93	3.16	3.39	3.61	3.84	4.06	4.28	4.50	4.71	4.93	5.14	5.36	5.57	5.78	5.99	6.20
1200	2.82	3.09	3.35	3.62	3.88	4.14	4.40	4.65	4.91	5.16	5.41	5.66	5.90	6.15	6.39	6.63	6.86	7.10
1400	3.13	3.44	3.74	4.03	4.33	4.62	4.91	5.20	5.48	5.76	6.04	6.32	6.59	6.86	7.12	7.39	7.65	7.91
1600	3.41	3.75	4.08	4.41	4.74	5.06	5.38	5.69	6.00	6.31	6.61	6.91	7.20	7.49	7.78	8.06	8.34	8.61
1800	3.67	4.03	4.40	4.75	5.10	5.45	5.79	6.13	6.46	6.79	7.11	7.43	7.74	8.04	8.34	8.64	8.93	9.21
2000	3.89	4.28	4.67	5.05	5.43	5.79	6.16	6.51	6.86	7.20	7.54	7.87	8.19	8.51	8.81	9.11	9.41	9.69
2200	4.08	4.50	4.91	5.31	5.70	6.09	6.47	6.84	7.20	7.55	7.89	8.23	8.56	8.87	9.18	9.48	9.77	10.05
2400	4.24	4.68	5.10	5.52	5.93	6.33	6.72	7.10	7.47	7.82	8.17	8.51	8.83	9.14	9.45	9.74	10.01	10.28
2600	4.36	4.82	5.26	5.69	6.11	6.52	6.91	7.29	7.66	8.02	8.36	8.69	9.01	9.31	9.60	9.87	10.12	10.36
2800	4.46	4.92	5.37	5.81	6.24	6.65	7.04	7.42	7.79	8.14	8.47	8.78	9.08	9.36	9.62	9.87	10.09	10.30
3000	4.51	4.99	5.44	5.89	6.31	6.72	7.11	7.48	7.83	8.17	8.48	8.77	9.04	9.30	9.53	9.73	9.92	
3200	4.53	5.01	5.47	5.91	6.33	6.73	7.11	7.46	7.80	8.11	8.39	8.65	8.89	9.11	9.29			
3400	4.51	4.99	5.44	5.88	6.29	6.67	7.03	7.36	7.67	7.95	8.20	8.43						
3600	4.45	4.92	5.37	5.79	6.18	6.55	6.88	7.18	7.46	7.70								
3800	4.34	4.81	5.24	5.64	6.01		6.65	6.92										
4000	4.20	4.65	5.06	5.44	5.78	6.08	6.34											
4200	4.01	4.43	4.82	5.17	5.47													
4400	3.77	4.17	4.52	4.83														
4800	3.48	3.85	4.16															
4800	3.15	3.47																
5000	2.76																	

Additional Horsepower Per Belt

1	20	21	22	23	24	25	26	27	28	29
RPM OF FASTER SHAFT	Speed Ratio Range									
	1.00 TO 1.01	1.02 TO 1.04	1.05 TO 1.07	1.08 TO 1.10	1.11 TO 1.14	1.15 TO 1.20	1.21 TO 1.27	1.28 TO 1.39	1.40 TO 1.64	1.65 AND OVER
575	0.00	0.02	0.04	0.07	0.09	0.11	0.13	0.16	0.18	0.20
690	0.00	0.03	0.05	0.08	0.11	0.13	0.16	0.19	0.22	0.24
725	0.00	0.03	0.06	0.08	0.11	0.14	0.17	0.20	0.23	0.25
870	0.00	0.03	0.07	0.10	0.14	0.17	0.20	0.24	0.27	0.30
950	0.00	0.04	0.07	0.11	0.15	0.19	0.22	0.26	0.30	0.33
1160	0.00	0.05	0.09	0.14	0.18	0.23	0.27	0.32	0.36	0.41
1425	0.00	0.06	0.11	0.17	0.22	0.28	0.33	0.39	0.44	0.50
1750	0.00	0.07	0.14	0.20	0.27	0.34	0.41	0.48	0.55	0.61
2850	0.00	0.11	0.22	0.33	0.44	0.56	0.67	0.78	0.89	1.00
3450	0.00	0.13	0.27	0.40	0.54	0.67	0.81	0.94	1.08	1.21
200	0.00	0.01	0.02	0.02	0.03	0.04	0.05	0.05	0.06	0.07
400	0.00	0.02	0.03	0.05	0.06	0.08	0.09	0.11	0.12	0.14
600	0.00	0.02	0.05	0.07	0.09	0.12	0.14	0.16	0.19	0.21
800	0.00	0.03	0.06	0.09	0.12	0.16	0.19	0.22	0.25	0.28
1000	0.00	0.04	0.08	0.12	0.16	0.19	0.23	0.27	0.31	0.35
1200	0.00	0.05	0.09	0.14	0.19	0.23	0.28	0.33	0.37	0.42
1400	0.00	0.05	0.11	0.16	0.22	0.27	0.33	0.38	0.44	0.49
1600	0.00	0.06	0.12	0.19	0.25	0.31	0.37	0.44	0.50	0.56
1800	0.00	0.07	0.14	0.21	0.28	0.35	0.42	0.49	0.56	0.63
2000	0.00	0.08	0.16	0.23	0.31	0.39	0.47	0.55	0.62	0.70
2200	0.00	0.09	0.17	0.26	0.34	0.43	0.51	0.60	0.69	0.77
2400	0.00	0.09	0.19	0.28	0.37	0.47	0.56	0.65	0.75	0.84
2600	0.00	0.10	0.20	0.30	0.40	0.51	0.61	0.71	0.81	0.91
2800	0.00	0.11	0.22	0.33	0.44	0.55	0.65	0.76	0.87	0.98
3000	0.00	0.12	0.23	0.35	0.47	0.58	0.70	0.82	0.94	1.05
3200	0.00	0.12	0.25	0.37	0.50	0.62	0.75	0.87	1.00	1.12
3400	0.00	0.13	0.26	0.40	0.53	0.66	0.79	0.93	1.06	1.19
3600	0.00	0.14	0.28	0.42	0.56	0.70	0.84	0.98	1.12	1.26
3800	0.00	0.15	0.30	0.44	0.59	0.74	0.89	1.04	1.18	1.33
4000	0.00	0.16	0.31	0.47	0.62	0.78	0.93	1.09	1.25	1.40
4200	0.00	0.16	0.33	0.49	0.65	0.82	0.98	1.15	1.31	1.47
4400	0.00	0.17	0.34	0.52	0.69	0.86	1.03	1.20	1.37	1.54
4600	0.00	0.18	0.36	0.54	0.72	0.90	1.07	1.25	1.43	1.61
4800	0.00	0.19	0.37	0.56	0.75	0.94	1.12	1.31	1.50	1.68
5000	0.00	0.19	0.39	0.59	0.78	0.97	1.17	1.36	1.56	1.75

Rim Speed Above 6000 Feet Per Minute, Special Sheaves May Be Necessary.

Table 3-11
Horsepower Ratings for C Section V Belts [12]

Basic Horsepower Per Belt — Small Sheave Pitch Diameter, Inches

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
RPM OF FASTER SHAFT	7.00	7.50	8.00	8.50	9.00	9.50	10.00	10.50	11.00	11.50	12.00	12.50	13.00	14.00	16.00	18.00
435	3.44	3.93	4.42	4.91	5.39	5.87	6.35	6.83	7.30	7.76	8.23	8.69	9.14	10.05	11.82	13.55
465	3.74	4.29	4.83	5.36	5.90	6.42	6.95	7.46	7.98	8.49	9.00	9.50	10.00	10.99	12.92	14.79
575	4.27	4.90	5.53	6.15	6.76	7.37	7.97	8.57	9.18	9.75	10.33	10.91	11.48	12.61	14.80	16.90
585	4.32	4.97	5.60	6.23	6.85	7.47	8.08	8.69	9.29	9.89	10.48	11.06	11.64	12.78	15.00	17.13
690	4.90	5.63	6.36	7.09	7.80	8.51	9.21	9.90	10.58	11.26	11.93	12.59	13.24	14.53	17.00	19.33
725	5.08	5.85	6.61	7.36	8.10	8.84	9.57	10.28	10.99	11.70	12.39	13.07	13.75	15.06	17.62	20.01
870	5.79	6.68	7.57	8.43	9.29	10.14	10.97	11.79	12.60	13.39	14.18	14.95	15.70	17.18	19.95	22.49
950	6.16	7.11	8.06	8.99	9.90	10.80	11.69	12.56	13.41	14.25	15.07	15.88	16.67	18.21	21.06	23.62
1160	7.02	8.13	9.22	10.29	11.34	12.38	13.38	14.34	15.29	16.21	17.11	17.99	18.84	20.44	23.31	25.66
1425	7.91	9.18	10.43	11.83	12.80	13.93	15.02	16.07	17.08	18.05	18.97	19.65	20.67	22.19	24.59	
1750	8.68	10.10	11.46	12.76	14.00	15.18	16.29	17.32	18.29	19.18	19.99	20.72	21.37			
100	1.03	1.16	1.29	1.42	1.55	1.68	1.81	1.93	2.06	2.19	2.31	2.44	2.56	2.81	3.31	3.79
200	1.83	2.08	2.33	2.57	2.81	3.05	3.29	3.53	3.77	4.01	4.24	4.48	4.71	5.17	6.09	6.99
300	2.55	2.91	3.26	3.62	3.96	4.31	4.66	5.00	5.34	5.68	6.01	6.35	6.68	7.34	8.65	9.93
400	3.22	3.66	4.13	4.59	5.04	5.48	5.93	6.37	6.80	7.24	7.67	8.10	8.53	9.37	11.03	12.65
500	3.83	4.39	4.95	5.50	6.04	6.58	7.12	7.65	8.18	8.71	9.23	9.74	10.26	11.27	13.25	15.16
600	4.41	5.06	5.71	6.36	6.99	7.62	8.25	8.87	9.48	10.09	10.69	11.29	11.88	13.04	15.30	17.46
700	4.95	5.70	6.43	7.17	7.89	8.60	9.31	10.01	10.70	11.38	12.06	12.73	13.39	14.08	17.18	19.53
800	5.46	6.29	7.11	7.93	8.73	9.53	10.31	11.08	11.84	12.60	13.34	14.07	14.79	16.20	18.88	21.36
900	5.93	6.85	7.75	8.65	9.52	10.39	11.24	12.08	12.91	13.72	14.52	15.31	16.08	17.57	20.38	22.94
1000	6.37	7.37	8.35	9.32	10.26	11.20	12.11	13.01	13.89	14.76	15.60	16.43	17.24	18.80	21.69	24.23
1100	6.79	7.86	8.91	9.94	10.95	11.94	12.91	13.86	14.79	15.70	16.58	17.44	18.28	19.88	22.77	25.22
1200	7.17	8.31	9.42	10.52	11.56	12.63	13.65	14.64	15.60	16.54	17.45	18.33	19.18	20.79	23.62	25.89
1300	7.52	8.72	9.90	11.04	12.18	13.25	14.30	15.33	16.32	17.28	18.20	19.09	19.94	21.53	24.22	26.20
1400	7.83	9.10	10.32	11.52	12.68	13.80	14.89	15.93	16.94	17.91	18.63	19.71	20.55	22.08	24.55	
1500	8.12	9.43	10.71	11.94	13.14	14.29	15.39	16.45	17.46	18.42	19.33	20.19	21.00	22.44	24.60	
1600	8.37	9.73	11.05	12.31	13.53	14.70	15.81	16.87	17.88	18.82	19.70	20.52	21.28	22.59		
1700	8.58	9.99	11.34	12.83	13.86	15.04	16.15	17.20	18.18	19.09	19.93	20.70	21.39	22.53		
1800	8.78	10.20	11.56	12.88	14.13	15.30	16.40	17.42	18.37	19.23	20.01	20.71	21.31			
1900	8.91	10.37	11.76	13.08	14.32	15.46	16.55	17.54	18.43	19.24	19.94	20.54				
2000	9.01	10.50	11.90	13.21	14.44	15.57	16.61	17.54	18.37	19.10	19.70					
2100	9.06	10.58	11.98	13.28	14.49	15.56	16.56	17.43	18.18							
2200	9.11	10.61	12.00	13.29	14.45	15.50	16.42	17.20								
2300	9.10	10.59	11.97	13.22	14.34	15.32	16.16									
2400	9.04	10.53	11.87	13.08	14.14	15.04										
2500	8.94	10.40	11.71	12.87	13.85	14.67										
2600	8.80	10.23	11.49	12.58	13.48											
2700	8.61	10.00	11.20	12.21												
2800	8.38	9.71	10.85													
2900	8.09	9.37	10.42													
3000	7.76	8.96														
3100	7.37	8.49														
3200	6.93															
3300	6.44															

Additional Horsepower Per Belt — Speed Ratio Range

1	18	19	20	21	22	23	24	25	26	27
RPM OF FASTER SHAFT	1.00 TO 1.01	1.02 TO 1.04	1.05 TO 1.07	1.08 TO 1.10	1.11 TO 1.14	1.15 TO 1.20	1.21 TO 1.27	1.28 TO 1.39	1.40 TO 1.64	1.65 AND OVER
435	0.00	0.05	0.09	0.14	0.19	0.24	0.28	0.33	0.38	0.42
485	0.00	0.05	0.10	0.16	0.21	0.26	0.32	0.37	0.42	0.47
575	0.00	0.06	0.12	0.19	0.25	0.31	0.37	0.44	0.50	0.56
585	0.00	0.06	0.13	0.19	0.25	0.32	0.38	0.44	0.51	0.57
890	0.00	0.07	0.15	0.22	0.30	0.37	0.45	0.52	0.60	0.67
725	0.00	0.08	0.16	0.24	0.31	0.39	0.47	0.55	0.63	0.71
670	0.00	0.09	0.19	0.28	0.38	0.47	0.57	0.86	0.75	0.85
950	0.00	0.10	0.21	0.31	0.41	0.51	0.62	0.72	0.82	0.93
1160	0.00	0.13	0.25	0.38	0.50	0.63	0.75	0.88	1.01	1.13
1425	0.00	0.15	0.31	0.46	0.62	0.77	0.93	1.08	1.24	1.39
1750	0.00	0.19	0.38	0.57	0.76	0.95	1.14	1.33	1.52	1.71
100	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.08	0.09	0.10
200	0.00	0.02	0.04	0.07	0.09	0.11	0.13	0.15	0.17	0.19
300	0.00	0.03	0.08	0.10	0.13	0.16	0.19	0.23	0.26	0.29
400	0.00	0.04	0.09	0.13	0.17	0.22	0.26	0.30	0.35	0.39
500	0.00	0.05	0.11	0.16	0.22	0.27	0.32	0.38	0.43	0.49
600	0.00	0.07	0.13	0.20	0.26	0.33	0.39	0.46	0.52	0.58
700	0.00	0.08	0.15	0.23	0.30	0.38	0.45	0.53	0.61	0.68
800	0.00	0.09	0.17	0.26	0.35	0.43	0.52	0.61	0.69	0.78
900	0.00	0.10	0.19	0.29	0.39	0.49	0.58	0.68	0.78	0.88
1000	0.00	0.11	0.22	0.33	0.43	0.54	0.65	0.76	0.87	0.97
1100	0.00	0.12	0.24	0.36	0.48	0.60	0.71	0.83	0.95	1.07
1200	0.00	0.13	0.26	0.39	0.52	0.65	0.78	0.91	1.04	1.17
1300	0.00	0.14	0.28	0.42	0.56	0.70	0.84	0.99	1.13	1.27
1400	0.00	0.15	0.30	0.46	0.61	0.76	0.91	1.06	1.21	1.36
1500	0.00	0.16	0.32	0.49	0.65	0.81	0.97	1.14	1.30	1.46
1600	0.00	0.17	0.35	0.52	0.69	0.87	1.04	1.21	1.39	1.56
1700	0.00	0.18	0.37	0.55	0.74	0.92	1.10	1.29	1.47	1.66
1800	0.00	0.20	0.39	0.59	0.78	0.98	1.17	1.37	1.58	1.75
1900	0.00	0.21	0.41	0.62	0.82	1.03	1.23	1.44	1.65	1.85
2000	0.00	0.22	0.43	0.65	0.87	1.08	1.30	1.52	1.73	1.95
2100	0.00	0.23	0.45	0.68	0.91	1.14	1.36	1.59	1.82	2.05
2200	0.00	0.24	0.48	0.72	0.95	1.19	1.43	1.67	1.91	2.14
2300	0.00	0.25	0.50	0.75	1.00	1.25	1.49	1.74	1.99	2.24
2400	0.00	0.26	0.52	0.78	1.04	1.30	1.56	1.82	2.08	2.34
2500	0.00	0.27	0.54	0.81	1.08	1.35	1.62	1.90	2.17	2.44
2600	0.00	0.28	0.56	0.85	1.13	1.41	1.69	1.97	2.25	2.53
2700	0.00	0.29	0.58	0.88	1.17	1.46	1.75	2.05	2.34	2.63
2800	0.00	0.30	0.61	0.91	1.21	1.52	1.82	2.12	2.43	2.73
2900	0.00	0.31	0.63	0.94	1.26	1.57	1.88	2.20	2.51	2.83
3000	0.00	0.33	0.65	0.98	1.30	1.63	1.95	2.28	2.60	2.92
3100	0.00	0.34	0.67	1.01	1.34	1.68	2.01	2.35	2.69	3.02
3200	0.00	0.35	0.69	1.04	1.39	1.73	2.08	2.43	2.77	3.12
3300	0.00	0.36	0.71	1.07	1.43	1.79	2.14	2.50	2.86	3.22

☐ Rim Speed Above 6000 Feet Per Minute. Special Sheaves May Be Necessary.

Table 3-12
Horsepower Ratings for D Section V Belts [12]

Basic Horsepower Per Belt — Small Sheave Pitch Diameter, Inches

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
RPM OF FASTER SHAFT	12.00	12.50	13.00	13.50	14.00	14.50	15.00	15.50	16.00	16.50	17.00	17.50	18.00	18.50	19.00	19.50	20.00	20.50	21.00	21.50	22.00	22.50	23.00	23.50	24.00
435	11.61	12.60	13.58	14.56	15.53	16.50	17.46	18.41	19.36	20.30	21.24	22.17	23.09	24.01	24.92	25.82	26.72	27.61	28.50	29.38	30.25	31.11	31.97	32.83	33.67
485	12.59	13.68	14.75	15.82	16.88	17.94	18.99	20.03	21.06	22.08	23.10	24.11	25.11	26.10	27.08	28.06	29.03	29.99	30.95	31.89	32.83	33.76	34.68	35.59	36.49
575	14.24	15.48	16.72	17.94	19.16	20.36	21.55	22.73	23.90	25.06	26.20	27.34	28.46	29.58	30.68	31.77	32.85	33.91	34.96	36.01	37.03	38.05	39.05	40.04	41.02
585	14.41	15.68	16.93	18.17	19.40	20.61	21.82	23.01	24.20	25.37	26.53	27.68	28.82	29.94	31.05	32.15	33.24	34.32	35.38	36.43	37.47	38.49	39.50	40.50	41.48
690	16.13	17.56	18.97	20.37	21.75	23.12	24.47	25.81	27.12	28.42	29.71	30.97	32.22	33.45	34.66	35.86	37.03	38.19	39.32	40.44	41.54	42.61	43.67	44.71	45.72
725	16.65	18.13	19.60	21.05	22.48	23.89	25.28	26.65	28.01	29.35	30.66	31.96	33.24	34.49	35.73	36.95	38.14	39.31	40.46	41.59	42.70	43.78	44.84	45.87	46.89
870	18.57	20.25	21.89	23.51	25.10	26.67	28.20	29.71	31.18	32.63	34.04	35.43	36.78	38.10	39.38	40.63	41.85	43.03	44.18	45.29	46.36	47.40	48.39	49.35	50.27
950	19.45	21.21	22.94	24.63	26.29	27.92	29.51	31.06	32.57	34.05	35.48	36.88	38.24	39.55	40.83	42.06	43.24	44.39	45.48	46.53	47.53	48.49	49.39	50.25	51.06
1160	21.12	23.04	24.90	26.71	28.46	30.15	31.78	33.35	34.85	36.29	37.67	38.97	40.20	41.36	42.45	43.47	44.40	45.26							
1425	21.75	23.70	25.56	27.33	28.99	30.55	32.00	33.34	34.56	35.67															
50	1.96	2.10	2.24	2.38	2.52	2.66	2.80	2.94	3.08	3.22	3.36	3.50	3.64	3.77	3.91	4.05	4.18	4.32	4.46	4.59	4.73	4.86	5.00	5.13	5.27
100	3.53	3.80	4.07	4.33	4.60	4.86	5.12	5.39	5.65	5.91	6.17	6.43	6.69	6.95	7.21	7.46	7.72	7.97	8.23	8.48	8.74	8.99	9.24	9.50	9.75
150	4.95	5.34	5.73	6.11	6.49	6.88	7.26	7.64	8.01	8.39	8.76	9.14	9.51	9.88	10.26	10.63	10.99	11.36	11.73	12.09	12.46	12.82	13.19	13.55	13.91
200	6.28	6.78	7.28	7.76	8.27	8.77	9.26	9.75	10.24	10.72	11.21	11.69	12.17	12.65	13.13	13.61	14.08	14.56	15.03	15.50	15.97	16.44	16.91	17.37	17.83
250	7.52	8.13	8.74	9.35	9.96	10.56	11.16	11.75	12.35	12.94	13.53	14.12	14.70	15.29	15.87	16.44	17.02	17.60	18.17	18.74	19.31	19.87	20.43	21.00	21.56
300	8.70	9.42	10.14	10.85	11.56	12.26	12.97	13.67	14.36	15.05	15.74	16.43	17.12	17.80	18.47	19.15	19.82	20.49	21.15	21.82	22.48	23.13	23.79	24.44	25.09
350	9.82	10.65	11.46	12.28	13.08	13.89	14.70	15.49	16.29	17.07	17.86	18.64	19.42	20.19	20.96	21.72	22.49	23.24	24.00	24.74	25.49	26.23	26.97	27.70	28.43
400	10.89	11.81	12.73	13.64	14.55	15.45	16.35	17.24	18.12	19.00	19.88	20.75	21.61	22.47	23.33	24.18	25.02	25.86	26.69	27.52	28.35	29.16	29.97	30.78	31.58
450	11.91	12.93	13.94	14.95	15.95	16.94	17.93	18.91	19.88	20.85	21.81	22.76	23.71	24.65	25.58	26.51	27.43	28.34	29.25	30.15	31.04	31.93	32.80	33.68	34.54
500	12.88	13.99	15.09	16.19	17.28	18.36	19.43	20.49	21.55	22.60	23.64	24.67	25.69	26.71	27.71	28.71	29.70	30.68	31.65	32.61	33.57	34.51	35.45	36.38	37.29
550	13.80	15.00	16.19	17.37	18.55	19.71	20.86	22.00	23.14	24.26	25.37	26.48	27.57	28.65	29.72	30.78	31.83	32.87	33.90	34.92	35.92	36.92	37.90	38.87	39.84
600	14.67	15.96	17.23	18.50	19.75	20.99	22.22	23.44	24.64	25.83	27.01	28.18	29.33	30.47	31.60	32.72	33.82	34.91	35.99	37.05	38.10	39.13	40.15	41.16	42.15
650	15.50	16.87	18.22	19.56	20.89	22.20	23.50	24.79	26.05	27.31	28.55	29.77	30.98	32.18	33.36	34.52	35.67	36.80	37.91	39.01	40.09	41.15	42.20	43.22	44.23
700	16.28	17.72	19.15	20.57	21.96	23.34	24.71	26.05	27.38	28.69	29.98	31.26	32.52	33.75	34.97	36.17	37.35	38.52	39.66	40.78	41.88	42.96	44.02	45.05	46.07
750	17.01	18.53	20.03	21.51	22.97	24.41	25.83	27.23	28.61	29.97	31.31	32.63	33.93	35.20	36.45	37.68	38.88	40.06	41.22	42.35	43.46	44.54	45.60	46.63	47.64
800	17.70	19.28	20.65	22.39	23.81	25.41	26.86	28.33	29.75	31.16	32.53	33.88	35.21	36.51	37.78	39.03	40.24	41.43	42.59	43.73	44.83	45.90	46.95	47.96	48.94
850	18.33	19.98	21.61	23.20	24.78	26.32	27.84	29.33	30.60	32.23	33.64	35.01	36.36	37.67	38.96	40.21	41.43	42.61	43.77	44.89	45.97	47.02	48.03	49.01	49.95
900	18.92	20.63	22.30	23.95	25.57	27.16	28.72	30.24	31.74	33.20	34.62	36.01	37.37	38.69	39.97	41.22	42.43	43.60	44.73	45.82	46.88	47.89	48.85	49.78	50.66
950	19.45	21.21	22.94	24.63	26.29	27.92	29.51	31.06	32.57	34.05	35.48	36.88	38.24	39.55	40.83	42.06	43.24	44.39	45.48	46.53	47.53	48.49	49.39	50.25	51.06
1000	19.93	21.74	23.51	25.25	26.94	28.59	30.20	31.77	33.30	34.78	36.22	37.61	38.96	40.26	41.51	42.71	43.86	44.96	46.00	47.00	47.93	48.82	49.64	50.41	51.12
1050	20.36	22.21	24.02	25.79	27.50	29.18	30.81	32.38	33.91	35.39	36.82	38.20	39.53	40.80	42.01	43.17	44.27	45.31	46.29	47.21	48.07	48.86	49.59		
1100	20.74	22.63	24.46	26.25	27.98	29.68	31.31	32.89	34.41	35.88	37.29	38.64	39.93	41.16	42.33	43.43	44.46	45.43	46.33	47.16					
1150	21.06	22.97	24.84	26.64	28.39	30.08	31.71	33.26	34.79	36.24	37.62	38.93	40.17	41.35	42.45	43.48	44.44	45.31							
1200	21.32	23.28	25.14	26.95	28.71	30.39	32.01	33.57	35.05	36.46	37.80	39.06	40.24	41.35	42.38	43.32	44.18								
1250	21.52	23.48	25.37	27.18	28.93	30.61	32.21	33.73	35.17	36.54	37.83	39.02	40.14	41.16	42.10										
1300	21.67	23.63	25.52	27.33	29.07	30.72	32.29	33.78	35.17	36.48	37.70	38.82	39.85												
1350	21.75	23.71	25.60	27.40	29.11	30.73	32.26	33.70	35.04	36.27	37.41	38.44													
1400	21.76	23.72	25.60	27.37	29.05	30.64	32.12	33.49	34.76	35.91	36.96														
1450	21.71	23.66	25.51	27.26	28.90	30.43	31.85	33.15	34.33	35.40															
1500	21.60	23.53	25.35	27.05	28.64	30.11	31.46	32.68																	
1550	21.42	23.32	25.10	26.75	28.28	29.68	30.94																		
1600	21.16	23.03	24.76	26.36	27.81	29.13																			
1650	20.84	22.66	24.33	25.86	27.23																				
1700	20.44	22.20	23.81	25.26	26.54																				
1750	19.98	21.66	23.20	24.56																					
1800	19.41	21.04	22.49																						
1850	18.78	20.33																							
1900	18.07																								
1950	17.28																								

Additional Horsepower Per Belt — Speed Ratio Range

1	27	28	29	30	31	32	33	34	35	36
RPM OF FASTER SHAFT	1.00 to 1.01	1.02 to 1.04	1.05 to 1.07	1.08 to 1.10	1.11 to 1.14	1.15 to 1.20	1.21 to 1.27	1.28 to 1.39	1.40 to 1.64	1.65 and over
435	0.00	0.17	0.33	0.50	0.67	0.84	1.00	1.17	1.34	1.50
485	0.00	0.19	0.37	0.56	0.75	0.93	1.12	1.30	1.49	1.68
575	0.00	0.22	0.44	0.66	0.88	1.11	1.33	1.55	1.77	1.99
585	0.00	0.22	0.45	0.66	0.90	1.12	1.35	1.57	1.80	2.02
690	0.00	0.27	0.53	0.80	1.06	1.33	1.59	1.86	2.12	2.39
725	0.00	0.28	0.56	0.84	1.11	1.39	1.67	1.95	2.23	2.51
870	0.00	0.33	0.67	1.00	1.34	1.67	2.01	2.34	2.68	3.01
950	0.00	0.37	0.73	1.10	1.46	1.83	2.19	2.56	2.92	3.28
1160	0.00	0.45	0.89	1.34	1.78	2.23	2.67	3.12	3.57	4.01
1425	0.00	0.55	1.09	1.65	2.19	2.74	3.28	3.83	4.38	4.93
50	0.00	0.02	0.04	0.06	0.08	0.10	0.12	0.13	0.15	0.17
100	0.00	0.04	0.08	0.12	0.15	0.19	0.23	0.27	0.31	0.35
150	0.00	0.06	0.12	0.17	0.23	0.29	0.35	0.40	0.46	0.52
200	0.00	0.08	0.15	0.23	0.31	0.38	0.46	0.54	0.62	0.69
250	0.00	0.10	0.19	0.29	0.38	0.48	0.58	0.67	0.77	0.86
300	0.00	0.12	0.23	0.35	0.46	0.58	0.69	0.81	0.92	1.04
350	0.00	0.13	0.27	0.40	0.54	0.67	0.81	0.94	1.08	1.21
400	0.00	0.15	0.31	0.46	0.61	0.77	0.92	1.08	1.23	1.38
450	0.00	0.17	0.35	0.52	0.69	0.86	1.04	1.21	1.38	1.58
500	0.00	0.19	0.38	0.58	0.77	0.96	1.15	1.35	1.54	1.73
550	0.00	0.21	0.42	0.64	0.85	1.06	1.27	1.48	1.69	1.90
600	0.00	0.23	0.46	0.69	0.92	1.15	1.38	1.61	1.85	2.07
650	0.00	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25
700	0.00	0.27	0.54	0.81	1.08	1.35	1.61	1.88	2.15	2.42
750	0.00	0.29	0.58	0.87	1.15	1.44	1.73	2.02	2.31	2.59
800	0.00	0.31	0.61	0.92	1.23	1.54	1.84	2.15	2.46	2.77
850	0.00	0.33	0.65	0.98	1.31	1.63	1.96	2.29	2.61	2.94
900	0.00	0.35	0.69	1.04	1.38	1.73	2.07	2.42	2.77	3.11
950	0.00	0.37	0.73	1.10	1.46	1.83	2.19	2.56	2.92	3.28
1000	0.00	0.38	0.77	1.15	1.54	1.92	2.30	2.69	3.08	3.46
1050	0.00	0.40	0.81	1.21	1.81	2.02	2.42	2.83	3.23	3.63
1100	0.00	0.42	0.84	1.27	1.89	2.11	2.54	2.96	3.38	3.80
1150	0.00	0.44	0.86	1.33	1.77	2.21	2.65	3.09	3.54	3.98
1200	0.00	0.46	0.92	1.39	1.84	2.31	2.77	3.23	3.65	4.15
1250	0.00	0.48	0.96	1.44	1.82	2.40	2.88	3.36	3.84	4.32
1300	0.00	0.50	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.49
1350	0.00	0.52	1.04	1.56	2.07	2.59	3.11	3.63	4.15	4.67
1400	0.00	0.54	1.07	1.62	2.15	2.69	3.23	3.77	4.31	4.84
1450	0.00	0.56	1.11	1.67	2.23	2.79	3.34	3.90	4.46	5.01
1500	0.00	0.58	1.15	1.73	2.30	2.88	3.46	4.04	4.61	5.19
1550	0.00	0.60	1.19	1.79	2.38	2.98	3.57	4.17	4.77	5.36
1600	0.00	0.62	1.23	1.85	2.46	3.08	3.89	4.30	4.92	5.53
1650	0.00	0.63	1.27	1.91	2.54	3.17	3.80	4.44	5.08	5.71
1700	0.00	0.65	1.30	1.96	2.61	3.27	3.92	4.57	5.23	5.88
1750	0.00	0.67	1.34	2.02	2.69	3.36	4.03	4.71	5.38	6.05
1800	0.00	0.69	1.38	2.08	2.77	3.46	4.15	4.84	5.54	6.22
1850	0.00	0.71	1.42	2.14	2.84	3.56	4.26	4.98	5.69	6.40
1900	0.00	0.73	1.46	2.19	2.92	3.65	4.36	5.11	5.84	6.57
1950	0.00	0.75	1.50	2.25	3.00	3.75	4.49	5.25	6.00	6.74

☐ Rim Speed Above 6000 Feet Per Minute, Special Sheaves May Be Necessary.

Table 3-13
Horsepower Ratings for E Section V Belts [12]

Basic Horsepower Per Belt

Small Sheave Pitch Diameter, Inches

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
RPM OF FASTER SHAFT	18.00	19.00	20.00	21.00	22.00	23.00	24.00	25.00	26.00	27.00	28.00	29.00	30.00	31.00	32.00	33.00	34.00	35.00	36.00
435	27.04	29.73	32.37	34.98	37.55	40.09	42.59	45.05	47.47	49.85	52.19	54.49	56.75	58.97	61.15	63.28	65.38	67.41	69.40
485	29.24	32.14	35.00	37.81	40.58	43.29	45.96	48.58	51.15	53.87	58.13	58.55	60.90	63.20	65.45	67.64	69.76	71.83	73.83
575	32.79	36.04	39.23	42.35	45.39	48.37	51.27	54.09	58.84	59.50	62.09	64.59	67.00	69.33	71.57	73.72	75.77	77.72	79.58
585	33.15	36.44	39.68	42.80	45.88	48.87	51.79	54.63	57.39	60.08	62.65	65.15	67.57	69.89	72.11	74.24	76.27	78.20	80.03
690	36.55	40.15	43.64	47.02	50.29	53.44	56.47	59.38	62.16	64.81	67.33	69.71	71.95	74.05	76.00	77.79	79.43	80.90	
50	4.52	4.92	5.32	5.72	6.12	6.51	6.91	7.30	7.69	8.08	8.46	8.85	9.23	9.62	10.00	10.38	10.78	11.14	11.52
100	8.21	8.96	9.71	10.46	11.20	11.94	12.68	13.41	14.15	14.87	15.60	18.32	17.04	17.76	18.47	19.19	19.90	20.60	21.31
150	11.58	12.65	13.73	14.80	15.87	18.93	17.99	19.05	20.09	21.14	22.18	23.21	24.24	25.28	28.28	27.30	28.31	29.31	30.31
200	14.68	18.08	17.47	18.86	20.24	21.61	22.97	24.32	25.66	27.00	28.33	29.85	30.96	32.27	33.57	34.68	38.14	37.42	38.88
250	17.61	19.31	21.00	22.88	24.35	28.00	27.85	29.28	30.90	32.50	34.10	35.68	37.28	38.82	40.37	41.90	43.43	44.94	46.44
300	20.37	22.38	24.33	26.29	28.22	30.15	32.05	33.94	35.81	37.67	39.50	41.32	43.13	44.91	48.88	48.43	50.17	51.88	53.58
350	22.97	25.23	27.47	29.68	31.87	34.04	36.19	38.31	40.41	42.48	44.53	46.56	48.58	50.54	52.50	54.42	56.33	58.20	60.05
400	25.42	27.93	30.42	32.87	35.29	37.89	40.05	42.38	44.88	46.95	49.18	51.38	53.55	55.68	57.78	59.84	61.87	63.86	65.82
450	27.72	30.47	33.18	35.85	38.48	41.08	43.63	48.14	48.81	51.04	53.42	55.76	58.05	60.30	62.50	64.65	66.76	68.81	70.82
500	29.86	32.83	35.75	38.82	41.43	44.20	46.92	49.58	52.18	54.74	57.23	59.67	62.05	64.38	66.62	68.61	70.94	73.00	75.00
550	31.85	35.02	38.12	41.16	44.14	47.05	49.90	52.67	55.38	58.02	60.59	63.08	65.49	67.83	70.09	72.27	74.36	76.37	78.29
600	33.88	37.02	40.28	43.47	46.58	49.61	52.55	55.41	58.19	60.87	63.46	65.96	68.36	70.67	72.87	74.97	76.97	78.85	80.63
650	35.35	38.84	42.24	45.54	48.75	51.86	54.87	57.77	60.57	63.26	65.83	68.28	70.62	72.83	74.92	76.98	78.70	80.39	81.94
700	36.84	40.46	43.97	47.36	50.64	53.80	56.83	59.74	62.51	65.15	67.65	70.01	72.22	74.28	76.18	77.93	79.51		
750	38.16	41.88	45.47	48.92	52.23	55.40	58.42	61.28	63.99	66.53	68.90	71.10	73.13	74.97	76.62				
800	39.29	43.08	46.72	50.20	53.51	56.65	59.61	62.38	64.97	67.36	69.55	71.53	73.30						
850	40.23	44.07	47.73	51.19	54.46	57.53	60.39	63.02	65.44	67.62	69.56								
900	40.97	44.83	48.47	51.69	55.08	58.03	80.73	63.18	65.36										
950	41.51	45.35	48.94	52.27	55.33	58.12	60.62	62.82											
1000	41.84	45.62	49.12	52.32	55.22	57.79	60.04												
1050	41.94	45.64	49.01	52.04	54.72														
1100	41.81	45.38	48.59	51.40															
1150	41.45	44.86	47.85																
1200	40.83	44.04																	
1250	39.97	42.93																	
1300	38.84																		

Additional Horsepower Per Belt

Speed Ratio Range

1	21	22	23	24	25	26	27	28	29	30
RPM OF FASTER SHAFT	1.00 TO 1.01	1.02 TO 1.04	1.05 TO 1.07	1.08 TO 1.10	1.11 TO 1.14	1.15 TO 1.20	1.21 TO 1.27	1.28 TO 1.39	1.40 TO 1.64	1.65 AND OVER
435	0.00	0.32	0.64	0.98	1.28	1.60	1.92	2.24	2.56	2.87
485	0.00	0.36	0.71	1.07	1.42	1.78	2.14	2.49	2.85	3.20
575	0.00	0.42	0.84	1.27	1.69	2.11	2.53	2.96	3.38	3.80
585	0.00	0.43	0.86	1.29	1.72	2.15	2.58	3.01	3.44	3.88
690	0.00	0.51	1.01	1.52	2.02	2.53	3.04	3.55	4.05	4.56
50	0.00	0.04	0.07	0.11	0.15	0.18	0.22	0.26	0.29	0.33
100	0.00	0.07	0.15	0.22	0.29	0.37	0.44	0.51	0.59	0.88
150	0.00	0.11	0.22	0.33	0.44	0.55	0.88	0.77	0.88	0.99
200	0.00	0.15	0.29	0.44	0.59	0.73	0.88	1.03	1.18	1.32
250	0.00	0.18	0.37	0.55	0.73	0.92	1.10	1.28	1.47	1.65
300	0.00	0.22	0.44	0.66	0.88	1.10	1.32	1.54	1.78	1.98
350	0.00	0.26	0.51	0.77	1.03	1.28	1.54	1.80	2.06	2.31
400	0.00	0.29	0.59	0.88	1.17	1.47	1.76	2.08	2.35	2.64
450	0.00	0.33	0.66	0.99	1.32	1.65	1.98	2.31	2.64	2.97
500	0.00	0.37	0.73	1.10	1.47	1.84	2.20	2.57	2.94	3.30
550	0.00	0.40	0.81	1.21	1.61	2.02	2.42	2.83	3.23	3.63
600	0.00	0.44	0.88	1.32	1.76	2.20	2.64	3.08	3.53	3.98
650	0.00	0.48	0.95	1.43	1.91	2.39	2.86	3.34	3.82	4.29
700	0.00	0.51	1.03	1.54	2.05	2.57	3.08	3.60	4.11	4.62
750	0.00	0.55	1.10	1.65	2.20	2.75	3.30	3.85	4.41	4.95
800	0.00	0.59	1.17	1.76	2.35	2.94	3.52	4.11	4.70	5.28
850	0.00	0.62	1.25	1.87	2.49	3.12	3.74	4.37	4.99	5.61
900	0.00	0.66	1.32	1.98	2.64	3.30	3.96	4.63	5.29	5.94
950	0.00	0.70	1.39	2.10	2.79	3.49	4.18	4.88	5.58	6.27
1000	0.00	0.73	1.47	2.21	2.93	3.67	4.40	5.14	5.88	6.60
1050	0.00	0.77	1.54	2.32	3.08	3.85	4.62	5.40	6.17	6.93
1100	0.00	0.81	1.61	2.43	3.23	4.04	4.84	5.65	6.46	7.26
1150	0.00	0.84	1.69	2.54	3.37	4.22	5.06	5.91	6.76	7.60
1200	0.00	0.88	1.76	2.65	3.52	4.41	5.28	6.17	7.05	7.93
1250	0.00	0.92	1.83	2.76	3.67	4.59	5.50	6.42	7.34	8.26
1300	0.00	0.95	1.91	2.87	3.82	4.77	5.72	6.68	7.64	8.59

Rim Speed Above 6000 Feet Per Minute. Special Sheaves May Be Necessary

Table 3-14
Horsepower Ratings for 3V Section V Belts [12]

Basic Horsepower Per Belt

Small Sheave Outside Diameter, Inches

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
RPM of Faster Shaft	2.0	2.2	2.4	2.6	2.8	3.0	3.2	3.4	3.6	3.8	4.0	4.2	4.4	4.6	4.8	5.0	5.2	5.4	5.6	5.8	6.0	6.2	6.4	6.6	6.8	7.0
575	0.19	0.32	0.44	0.56	0.68	0.80	0.92	1.04	1.16	1.27	1.39	1.51	1.62	1.74	1.85	1.97	2.08	2.20	2.31	2.42	2.54	2.65	2.76	2.87	2.98	3.10
690	0.21	0.36	0.50	0.65	0.79	0.93	1.07	1.21	1.35	1.49	1.63	1.77	1.90	2.04	2.18	2.31	2.45	2.58	2.72	2.85	2.98	3.11	3.25	3.38	3.51	3.64
725	0.22	0.37	0.52	0.67	0.82	0.97	1.12	1.26	1.41	1.56	1.70	1.84	1.99	2.13	2.27	2.41	2.56	2.70	2.84	2.98	3.12	3.25	3.39	3.53	3.67	3.80
870	0.24	0.42	0.60	0.77	0.95	1.13	1.30	1.47	1.65	1.82	1.99	2.16	2.33	2.50	2.67	2.83	3.00	3.16	3.33	3.49	3.66	3.82	3.98	4.14	4.31	4.47
950	0.25	0.44	0.64	0.83	1.02	1.21	1.40	1.59	1.77	1.96	2.15	2.33	2.51	2.70	2.88	3.06	3.24	3.42	3.60	3.77	3.95	4.13	4.30	4.48	4.65	4.82
1160	0.27	0.50	0.73	0.97	1.20	1.42	1.65	1.88	2.10	2.32	2.54	2.76	2.98	3.20	3.42	3.63	3.85	4.06	4.27	4.48	4.69	4.90	5.11	5.32	5.52	5.73
1425	0.28	0.57	0.85	1.13	1.40	1.68	1.95	2.22	2.49	2.76	3.02	3.29	3.55	3.81	4.07	4.33	4.58	4.84	5.09	5.34	5.59	5.84	6.09	6.33	6.57	6.82
1750	0.29	0.63	0.97	1.31	1.64	1.97	2.30	2.62	2.94	3.26	3.58	3.90	4.21	4.52	4.83	5.13	5.44	5.74	6.03	6.33	6.62	6.92	7.20	7.49	7.77	8.05
2850	0.24	0.77	1.29	1.81	2.31	2.82	3.31	3.80	4.29	4.76	5.23	5.69	6.15	6.59	7.04	7.47	7.90	8.32	8.73	9.13	9.53	9.91	10.29	10.66	11.02	11.37
3450	0.18	0.80	1.41	2.01	2.61	3.19	3.77	4.33	4.88	5.43	5.96	6.49	7.00	7.50	7.99	8.47	8.93	9.38	9.82	10.25	10.66	11.06	11.45	11.82	12.17	12.51
100	0.06	0.08	0.11	0.12	0.16	0.18	0.20	0.23	0.25	0.27	0.30	0.32	0.34	0.36	0.39	0.41	0.43	0.45	0.48	0.50	0.52	0.54	0.57	0.59	0.61	0.63
200	0.10	0.14	0.19	0.24	0.28	0.33	0.37	0.42	0.46	0.50	0.55	0.59	0.64	0.68	0.72	0.77	0.81	0.85	0.90	0.94	0.98	1.02	1.07	1.11	1.15	1.19
300	0.13	0.20	0.26	0.33	0.40	0.46	0.53	0.59	0.66	0.72	0.79	0.85	0.91	0.98	1.04	1.10	1.17	1.23	1.29	1.35	1.42	1.48	1.54	1.60	1.66	1.72
400	0.16	0.24	0.33	0.42	0.50	0.59	0.67	0.76	0.84	0.93	1.01	1.10	1.18	1.26	1.34	1.43	1.51	1.59	1.67	1.75	1.83	1.91	2.00	2.08	2.16	2.24
500	0.18	0.29	0.39	0.50	0.61	0.71	0.82	0.92	1.02	1.13	1.23	1.33	1.44	1.54	1.64	1.74	1.84	1.94	2.04	2.14	2.24	2.34	2.44	2.54	2.63	2.73
600	0.20	0.33	0.45	0.58	0.70	0.83	0.95	1.08	1.20	1.32	1.44	1.56	1.68	1.80	1.92	2.04	2.16	2.28	2.40	2.52	2.63	2.75	2.87	2.98	3.10	3.22
700	0.21	0.36	0.51	0.65	0.80	0.94	1.08	1.23	1.37	1.51	1.65	1.79	1.93	2.07	2.20	2.34	2.48	2.61	2.75	2.89	3.02	3.15	3.29	3.42	3.56	3.69
800	0.23	0.40	0.56	0.73	0.89	1.05	1.21	1.37	1.53	1.69	1.85	2.01	2.17	2.32	2.48	2.63	2.79	2.94	3.09	3.25	3.40	3.55	3.70	3.85	4.00	4.15
900	0.24	0.43	0.61	0.80	0.98	1.16	1.34	1.52	1.70	1.87	2.05	2.22	2.40	2.57	2.75	2.92	3.09	3.26	3.43	3.60	3.77	3.94	4.10	4.27	4.44	4.60
1000	0.25	0.46	0.66	0.86	1.06	1.26	1.46	1.66	1.85	2.05	2.24	2.43	2.63	2.82	3.01	3.20	3.39	3.57	3.76	3.95	4.13	4.31	4.50	4.68	4.86	5.04
1100	0.26	0.49	0.71	0.93	1.15	1.36	1.58	1.79	2.01	2.22	2.43	2.64	2.85	3.06	3.27	3.47	3.68	3.88	4.08	4.28	4.49	4.69	4.88	5.08	5.28	5.48
1200	0.27	0.51	0.75	0.99	1.22	1.46	1.70	1.93	2.16	2.39	2.62	2.84	3.07	3.29	3.52	3.74	3.96	4.18	4.40	4.62	4.83	5.05	5.26	5.48	5.69	5.90
1300	0.28	0.54	0.80	1.05	1.31	1.56	1.81	2.06	2.31	2.55	2.80	3.04	3.29	3.53	3.77	4.00	4.24	4.48	4.71	4.94	5.17	5.40	5.63	5.86	6.09	6.31
1400	0.28	0.56	0.84	1.11	1.38	1.65	1.92	2.19	2.45	2.72	2.98	3.24	3.50	3.76	4.01	4.26	4.52	4.77	5.02	5.26	5.51	5.75	6.00	6.24	6.48	6.72
1500	0.29	0.58	0.88	1.17	1.46	1.75	2.03	2.32	2.60	2.88	3.16	3.43	3.71	3.98	4.25	4.52	4.78	5.05	5.31	5.58	5.84	6.09	6.35	6.61	6.86	7.11
1600	0.29	0.60	0.92	1.22	1.53	1.84	2.14	2.44	2.74	3.03	3.33	3.62	3.91	4.20	4.48	4.77	5.05	5.33	5.61	5.88	6.16	6.43	6.70	6.97	7.23	7.50
1700	0.29	0.62	0.95	1.28	1.60	1.92	2.24	2.56	2.88	3.19	3.50	3.80	4.11	4.41	4.71	5.01	5.31	5.60	5.89	6.18	6.47	6.75	7.04	7.32	7.59	7.87
1800	0.29	0.64	0.99	1.33	1.67	2.01	2.35	2.68	3.01	3.34	3.66	3.99	4.31	4.62	4.94	5.25	5.56	5.87	6.17	6.48	6.78	7.07	7.37	7.66	7.95	8.23
1900	0.29	0.66	1.02	1.38	1.74	2.10	2.45	2.80	3.14	3.49	3.83	4.17	4.50	4.83	5.16	5.49	5.81	6.13	6.45	6.76	7.08	7.38	7.69	7.99	8.29	8.59
2000	0.29	0.67	1.06	1.43	1.81	2.18	2.55	2.91	3.27	3.63	3.99	4.34	4.69	5.04	5.38	5.72	6.05	6.39	6.72	7.04	7.37	7.69	8.00	8.32	8.63	8.93
2100	0.29	0.69	1.09	1.48	1.87	2.26	2.64	3.03	3.40	3.78	4.15	4.51	4.88	5.23	5.59	5.94	6.29	6.64	6.98	7.32	7.65	7.98	8.31	8.63	8.95	9.26
2200	0.28	0.70	1.12	1.53	1.94	2.34	2.74	3.14	3.53	3.92	4.30	4.68	5.06	5.43	5.80	6.16	6.52	6.88	7.23	7.58	7.93	8.27	8.60	8.94	9.26	9.59
2300	0.28	0.72	1.15	1.58	2.00	2.42	2.83	3.24	3.65	4.05	4.45	4.85	5.24	5.62	6.00	6.38	6.75	7.12	7.48	7.84	8.20	8.55	8.89	9.23	9.56	9.89
2400	0.28	0.73	1.18	1.62	2.06	2.49	2.92	3.35	3.77	4.19	4.60	5.01	5.41	5.81	6.20	6.59	6.97	7.35	7.73	8.09	8.46	8.82	9.17	9.51	9.86	10.19
2500	0.27	0.74	1.20	1.66	2.12	2.57	3.01	3.45	3.89	4.32	4.74	5.16	5.58	5.99	6.40	6.79	7.19	7.58	7.96	8.34	8.71	9.08	9.44	9.79	10.14	10.48
2600	0.26	0.75	1.23	1.71	2.18	2.64	3.10	3.56	4.00	4.45	4.89	5.32	5.75	6.17	6.58	6.99	7.40	7.80	8.19	8.57	8.95	9.33	9.69	10.05	10.40	10.75
2700	0.26	0.76	1.26	1.75	2.23	2.71	3.19	3.66	4.12	4.57	5.03	5.47	5.91	6.34	6.77	7.19	7.60	8.01	8.41	8.80	9.19	9.57	9.94	10.30	10.66	11.01
2800	0.25	0.77	1.28	1.79	2.29	2.78	3.27	3.75	4.23	4.70	5.16	5.62	6.07	6.51	6.95	7.38	7.80	8.22	8.62	9.02	9.42	9.80	10.18	10.55	10.91	11.26
2900	0.24	0.77	1.30	1.83	2.34	2.85	3.35	3.85	4.34	4.82	5.29	5.76	6.22	6.68	7.12	7.56	7.99	8.42	8.83	9.24	9.63	10.02	10.40	10.78	11.14	11.49
3000	0.23	0.78	1.33	1.86	2.39	2.92	3.43	3.94	4.44	4.94	5.42	5.90	6.37	6.84	7.29	7.74	8.18	8.61	9.03	9.44	9.84	10.24	10.62	10.99	11.36	11.71
3100	0.22	0.79	1.35	1.90	2.44	2.98	3.51	4.03	4.54	5.05	5.55	6.04	6.52	6.99	7.46	7.91	8.36	8.79	9.22	9.64	10.04	10.44	10.82	11.20	11.56	11.91
3200	0.21	0.79	1.37	1.93	2.49	3.04	3.58	4.12	4.65	5.16	5.67	6.17	6.66	7.14	7.62	8.08	8.53	8.97	9.40	9.82	10.23	10.63	11.02	11.39	11.75	12.10
3300	0.20	0.79	1.38	1.97	2.54	3.10	3.66	4.21	4.74	5.27	5.79	6.30	6.80	7.29	7.77	8.24	8.70	9.14	9.58	10.00	10.41	10.81	11.20	11.57	11.93	12.28
3400	0.18	0.80	1.40	2.00	2.58	3.16	3.73	4.29	4.84	5.38	5.91	6.43	6.93	7.43	7.92	8.39	8.85	9.31	9.74	10.17	10.58	10.98	11.37	11.74	12.10	12.44
3500	0.17	0.80	1.42	2.03	2.63	3.22	3.80	4.37	4.93	5.48	6.02	6.55	7.06	7.57	8.06	8.54	9.01	9.46	9.90	10.33	10.74	11.14	11.53	11.89	12.25	12.58
3600	0.16	0.80	1.43	2.06	2.67	3.27	3.87	4.45	5.02	5.58	6.13	6.66	7.19	7.70	8.20	8.68	9.15	9.61	10.05	10.48	10.89	11.29	11.67	12.03	12.38	12.71
3700	0.14	0.80	1.45	2.09	2.71	3.33	3.93	4.52	5.11	5.68	6.23	6.78	7.31	7.82	8.33	8.82	9.29	9.75	10.19	10.62	11.02	11.42	11.80	12.16	12.50	12.82
3800	0.13	0.80	1.46	2.11	2.75	3.38	4.00	4.60	5.19	5.77	6.33	6.89	7.42	7.94	8.45	8.94	9.42	9.88	10.32	10.75	11.16	11.55	11.92	12.27	12.61	12.92
3900	0.11	0.80	1.47	2.14	2.79	3.43	4.06	4.67	5.27	5.86	6.43	6.99	7.53	8.06	8.57	9.07	9.54	10.00	10.41	10.87	11.28	11.64	12.02	12.37	12.69	
4000	0.09	0.79	1.48	2.16	2.83	3.48	4.12	4.74	5.35	5.95	6.53	7.09	7.64	8.17	8.68	9.18	9.66	10.12	10.56	10.98	11.38	11.74	12.13	12.48		
4100	0.07	0.79	1.49	2.19	2.86	3.53	4.17	4.81	5.43	6.03	6.62	7.19	7.74	8.27	8.79	9.29	9.77	10.23	10.66	11.08	11.47	11.85	12.20			
4200	0.05	0.79	1.50	2.21	2.90	3.57	4.23	4.87	5.50	6.11	6.70	7.28	7.83	8.37	8.89	9.39	9.87	10.32	10.76	11.17	11.56	11.92				
4300	0.04	0.78	1.51	2.23	2.93	3.61	4.28	4.93	5.57	6.19	6.78	7.36	7.93	8.47	8.99	9.48	9.96	10.41	10.84	11.26	11.63	11.98				
4400	0.02	0.77	1.52	2.25	2.96	3.65	4.33	4.99	5.64	6.26	6.86	7.45	8.01	8.55	9.07	9.57	10.04	10.49	10.91	11.31	11.68					
4500	0.00	0.77	1.52	2.26	2.99	3.69	4.38	5.05	5.70	6.35	6.94	7.53	8.09	8.63	9.15	9.65	10.12	10.56	10.98	11.37						
4600	0.00	0.76	1.53	2.28	3.01	3.73	4.43	5.10	5.76	6.39	7.01	7.60	8.17	8.71	9.23	9.72	10.18	10.62	11.03	11.41						
4700	0.00	0.75	1.53	2.29	3.04	3.76	4.47	5.15	5.82	6.46	7.07	7.67	8.24	8.78	9.29	9.78	10.24	10.67	11.07							
4800	0.00	0.74	1.53	2.31	3.06	3.80	4.51	5.20	5.87	6.52	7.14	7.73	8.30	8.84	9.35	9.84	10.29	10.71								
4900	0.00	0.73	1.53	2.32	3.09	3.83	4.55	5.25	5.92	6.57	7.19	7.79	8.36	8.90	9.41	9.89	10.33	10.75								
5000	0.00	0.72	1.53	2.33	3.11	3.86	4.59	5.29	5.97	6.62	7.25	7.84	8.41	8.95	9.45	9.92	10.36									

Additional Horsepower Per Belt

Speed Ratio Range

1	28	29	30	31	32	33	34	35	36	37
RPM of Faster Shaft	1.00 to 1.01	1.02 to 1.05	1.06 to 1.11	1.12 to 1.18	1.19 to 1.26	1.27 to 1.38	1.39 to 1.57	1.58 to 1.94	1.95 to 3.38	3.39 and over
575	0.00	0.01	0.03	0.05	0.07	0.08	0.10	0.11	0.12	0.12
690	0.00	0.01	0.03	0.06	0.08	0.10	0.11	0.13	0.14	0.15
725	0.00	0.01	0.04	0.06	0.08	0.10	0.12	0.14	0.15	0.16
870	0.00	0.02	0.04	0.07	0.10	0.12	0.14	0.16	0.18	0.19
950	0.00	0.02	0.05	0.08	0.11	0.13	0.16	0.18	0.19	0.20
1160	0.00	0.02	0.06	0.10	0.14	0.18	0.19	0.22	0.24	0.25
1425	0.00	0.03	0.07	0.12	0.17	0.20	0.24	0.27	0.29	0.31
1750	0.00	0.03	0.08	0.15	0.20	0.25	0.29	0.33	0.36	0.38
2850	0.00	0.05	0.14	0.25	0.33	0.40	0.47	0.53	0.58	0.61
3450	0.00	0.06	0.17	0.30	0.40	0.48	0.57	0.65	0.70	0.74
100	0.00	0.00	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
200	0.00	0.00	0.01	0.02	0.02	0.03	0.03	0.04	0.04	0.04
300	0.00	0.01	0.01	0.03	0.04	0.04	0.05	0.06	0.06	0.06
400	0.00	0.01	0.02	0.03	0.05	0.06	0.07	0.07	0.08	0.08
500	0.00	0.01	0.02	0.04	0.06	0.07	0.08	0.09	0.10	0.11
600	0.00	0.01	0.03	0.05	0.07	0.08	0.10	0.11	0.12	0.13
700	0.00	0.01	0.03	0.06	0.08	0.10	0.12	0.13	0.14	0.15
800	0.00	0.01	0.04	0.07	0.09	0.11	0.13	0.15	0.16	0.17
900	0.00	0.02	0.04	0.08	0.11	0.13	0.15	0.17	0.18	0.19
1000	0.00	0.02	0.05	0.08	0.12	0.14	0.17	0.18	0.20	0.22
1100	0.00	0.02	0.05	0.09	0.13	0.16	0.18	0.21	0.22	0.24
1200	0.00	0.02	0.06	0.10	0.14	0.17	0.20	0.22	0.24	0.26
1300	0.00	0.02	0.06	0.11	0.15	0.18	0.22	0.24	0.26	0.28
1400	0.00	0.03	0.07	0.12	0.16	0.20	0.23	0.26	0.29	0.30
1500	0.00	0.03	0.07	0.13	0.18	0.21	0.25	0.28	0.31	0.32
1600	0.00	0.03	0.08	0.14	0.19	0.23	0.27	0.30	0.33	0.35
1700	0.00	0.03	0.08	0.15	0.20	0.24	0.29	0.32	0.35	0.37
1800	0.00	0.03	0.08	0.15	0.21	0.26	0.30	0.34	0.37	0.38
1900	0.00	0.03	0.09	0.16	0.22	0.27	0.32	0.36	0.39	0.41
2000	0.00	0.04	0.10	0.17	0.23	0.29	0.33	0.37	0.41	0.43
2100	0.00	0.04	0.10	0.18	0.25	0.30	0.35	0.39	0.43	0.45
2200	0.00	0.04	0.11	0.19	0.26	0.31	0.37	0.41	0.45	0.47
2300	0.00	0.04	0.11	0.20	0.27	0.33	0.38	0.43	0.47	0.50
2400	0.00	0.04	0.12	0.21	0.28	0.34	0.40	0.45	0.49	0.52
2500	0.00	0.05	0.12	0.21	0.29	0.35	0.42	0.47	0.51	0.54
2600	0.00	0.05	0.13	0.22	0.30	0.37	0.43	0.49	0.53	0.56
2700	0.00	0.05	0.13	0.23	0.32	0.38	0.45	0.50	0.55	0.58
2800	0.00	0.06	0.14	0.24	0.33	0.40	0.47	0.52	0.57	0.60
2900	0.00	0.06	0.14	0.25	0.34	0.41	0.48	0.54	0.59	0.63
3000	0.00	0.06	0.15	0.26	0.35	0.43	0.50	0.56	0.61	0.65
3100	0.00	0.06	0.15	0.27	0.36	0.44	0.52	0.58	0.63	0.67
3200	0.00	0.06	0.16	0.28	0.37	0.45	0.53	0.60	0.65	0.69
3300	0.00	0.06	0.16	0.28	0.39	0.47	0.55	0.62	0.67	0.71
3400	0.00	0.06	0.17	0.29	0.40	0.48	0.56	0.64	0.69	0.73
3500	0.00	0.06	0.17	0.30	0.41	0.50	0.58	0.65	0.71	0.76
3600	0.00	0.07	0.18	0.31	0.42	0.51	0.60	0.67	0.73	0.78
3700	0.00	0.07	0.18	0.32	0.43	0.52	0.61	0.69	0.75	0.80
3800	0.00	0.07	0.19	0.33	0.44	0.54	0.63	0.71	0.77	0.82
3900	0.00	0.07	0.19	0.34	0.46	0.55	0.65	0.73	0.79	0.84
4000	0.00	0.07	0.20	0.34	0.47	0.57	0.66	0.75	0.81	0.86
4100	0.00	0.07	0.20	0.35	0.47	0.58	0.68	0.76	0.83	0.88
4200	0.00	0.07	0.20	0.36	0.49	0.59	0.69	0.78	0.85	0.90
4300	0.00	0.07	0.21	0.36	0.50	0.60	0.71	0.80	0.87	0.92
4400	0.00	0.07	0.21	0.37	0.51	0.62	0.73	0.82	0.89	0.94
4500	0.00	0.08	0.22	0.38	0.52	0.63	0.74	0.84	0.91	0.97
4600	0.00	0.08	0.22	0.39	0.53	0.65	0.76	0.86	0.93	0.99
4700	0.00	0.08	0.23	0.40	0.55	0.66	0.78	0.87	0.95	1.01
4800	0.00	0.08	0.23	0.41	0.56	0.68	0.79	0.89	0.97	1.03
4900	0.00	0.08	0.24	0.42	0.57	0.69	0.81	0.91	0.99	1.05
5000	0.00	0.09	0.24	0.42	0.58	0.70	0.83	0.93	1.01	1.07

☐ Belt Speed over 6500 feet per minute. Special sheaves may be necessary.

Table 3-15
Horsepower Ratings for 5V Section V Belts [12]

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	1	27	28	29	30	31	32	33	34	35	36
	Basic Horsepower Per Belt																										Additional Horsepower Per Belt									
	Small Sheave Outside Diameter, Inches																										Speed Ratio Range									
RPM of Faster Shaft	7.0	7.2	7.4	7.6	7.8	8.0	8.2	8.4	8.6	8.8	9.0	9.5	10.0	10.5	11.0	11.5	12.0	12.5	13.0	13.5	14.0	14.5	15.0	15.5	16.0	RPM of Faster Shaft	1.00 to 1.01	1.02 to 1.05	1.06 to 1.11	1.12 to 1.18	1.19 to 1.26	1.27 to 1.38	1.39 to 1.57	1.58 to 1.94	1.95 to 3.38	3.39 and over
435	4.51	4.74	4.98	5.22	5.46	5.70	5.93	6.17	6.40	6.64	6.87	7.46	8.04	8.62	9.19	9.77	10.34	10.90	11.47	12.03	12.59	13.14	13.70	14.25	14.80	435	0.00	0.04	0.11	0.20	0.27	0.33	0.38	0.43	0.47	0.50
485	4.96	5.21	5.47	5.74	6.00	6.26	6.52	6.78	7.04	7.30	7.56	8.20	8.85	9.48	10.12	10.75	11.37	12.00	12.62	13.24	13.85	14.46	15.07	15.68	16.28	485	0.00	0.05	0.13	0.22	0.30	0.37	0.43	0.48	0.53	0.56
575	5.72	6.02	6.33	6.64	6.94	7.25	7.55	7.86	8.16	8.46	8.76	9.51	10.26	11.00	11.74	12.47	13.20	13.92	14.64	15.35	16.06	16.77	17.47	18.16	18.85	575	0.00	0.06	0.15	0.26	0.36	0.45	0.51	0.57	0.62	0.66
585	5.80	6.11	6.43	6.74	7.05	7.36	7.67	7.98	8.28	8.59	8.90	9.66	10.41	11.17	11.91	12.66	13.39	14.13	14.86	15.58	16.30	17.02	17.73	18.43	19.13	585	0.00	0.06	0.15	0.26	0.36	0.43	0.52	0.58	0.63	0.67
690	6.66	7.03	7.39	7.75	8.11	8.47	8.83	9.18	9.54	9.90	10.25	11.13	12.00	12.87	13.73	14.58	15.43	16.27	17.11	17.94	18.76	19.58	20.39	21.19	21.98	690	0.00	0.07	0.18	0.32	0.43	0.52	0.61	0.68	0.75	0.79
725	6.94	7.32	7.70	8.08	8.46	8.83	9.21	9.58	9.95	10.32	10.69	11.61	12.52	13.42	14.32	15.21	16.09	16.97	17.84	18.70	19.55	20.40	21.24	22.07	22.90	725	0.00	0.07	0.19	0.33	0.45	0.55	0.64	0.72	0.79	0.83
870	8.07	8.52	8.96	9.41	9.85	10.29	10.72	11.16	11.60	12.03	12.46	13.53	14.59	15.64	16.68	17.71	18.73	19.74	20.74	21.73	22.70	23.67	24.62	25.56	26.49	870	0.00	0.08	0.23	0.40	0.54	0.66	0.77	0.86	0.94	1.00
950	8.67	9.15	9.63	10.11	10.59	11.06	11.53	12.00	12.47	12.93	13.40	14.55	15.69	16.81	17.93	19.03	20.12	21.19	22.25	23.30	24.34	25.36	26.37	27.36	28.33	950	0.00	0.09	0.25	0.43	0.59	0.72	0.84	0.94	1.03	1.09
1160	10.17	10.74	11.30	11.87	12.43	12.99	13.54	14.09	14.64	15.18	15.73	17.07	18.40	19.70	20.99	22.25	23.50	24.73	25.93	27.10	28.26	29.39	30.50	31.58	32.64	1160	0.00	0.11	0.30	0.53	0.72	0.87	1.02	1.15	1.26	1.33
1425	11.89	12.56	13.23	13.89	14.55	15.20	15.85	16.49	17.13	17.76	18.39	19.96	21.46	22.96	24.39	25.81	27.16	28.54	29.85	31.13	32.35	33.54	34.68	35.79	36.85	1425	0.00	0.14	0.37	0.65	0.88	1.07	1.26	1.42	1.54	1.63
1750	13.76	14.53	15.31	16.07	16.82	17.57	18.31	19.04	19.77	20.48	21.19	22.92	24.60	26.23	27.79	29.30	30.74	32.13	33.46	34.69	35.87	36.97	38.00			1750	0.00	0.17	0.46	0.80	1.08	1.32	1.55	1.74	1.89	2.01
2850	17.68	18.64	19.58	20.49	21.37	22.22	23.04	23.84	24.60	25.34	26.04															2850	0.00	0.27	0.75	1.30	1.77	2.15	2.52	2.83	3.08	3.27
3450	17.97	18.89	19.79	20.67																						3450	0.00	0.33	0.90	1.58	2.15	2.60	3.05	3.43	3.74	3.96
100	1.26	1.31	1.37	1.42	1.48	1.55	1.61	1.67	1.73	1.79	1.85	2.00	2.15	2.30	2.45	2.60	2.75	2.89	3.04	3.19	3.33	3.48	3.63	3.77	3.92	100	0.00	0.01	0.03	0.04	0.06	0.08	0.09	0.10	0.11	0.11
200	2.29	2.41	2.52	2.64	2.76	2.88	2.99	3.11	3.22	3.34	3.45	3.74	4.02	4.31	4.59	4.87	5.16	5.44	5.71	5.99	6.27	6.55	6.82	7.10	7.37	200	0.00	0.02	0.05	0.09	0.12	0.15	0.18	0.20	0.22	0.23
300	3.27	3.44	3.61	3.78	3.95	4.12	4.29	4.45	4.62	4.79	4.95	5.37	5.78	6.20	6.61	7.02	7.43	7.83	8.24	8.64	9.04	9.44	9.84	10.24	10.63	300	0.00	0.02	0.08	0.14	0.18	0.23	0.27	0.30	0.32	0.34
400	4.19	4.41	4.63	4.86	5.08	5.29	5.51	5.73	5.95	6.17	6.39	6.93	7.47	8.00	8.54	9.07	9.60	10.12	10.65	11.17	11.69	12.20	12.72	13.23	13.74	400	0.00	0.04	0.10	0.18	0.25	0.30	0.35	0.40	0.43	0.46
500	5.08	5.35	5.63	5.89	6.16	6.43	6.70	6.96	7.23	7.50	7.76	8.43	9.08	9.74	10.39	11.04	11.68	12.32	12.96	13.59	14.22	14.85	15.48	16.10	16.71	500	0.00	0.05	0.13	0.23	0.31	0.38	0.44	0.50	0.54	0.57
600	5.93	6.25	6.57	6.88	7.20	7.52	7.84	8.15	8.47	8.78	9.09	9.87	10.64	11.41	12.18	12.94	13.69	14.44	15.18	15.92	16.66	17.39	18.11	18.83	19.53	600	0.00	0.06	0.16	0.27	0.37	0.45	0.53	0.60	0.65	0.69
700	6.74	7.11	7.48	7.85	8.21	8.57	8.94	9.30	9.66	10.02	10.38	11.27	12.15	13.03	13.90	14.76	15.62	16.47	17.32	18.16	18.99	19.81	20.63	21.44	22.25	700	0.00	0.07	0.18	0.32	0.44	0.53	0.62	0.70	0.76	0.80
800	7.54	7.95	8.36	8.77	9.18	9.59	10.00	10.41	10.81	11.21	11.62	12.61	13.60	14.59	15.56	16.52	17.48	18.43	19.36	20.29	21.21	22.12	23.02	23.91	24.80	800	0.00	0.08	0.21	0.37	0.50	0.60	0.71	0.80	0.87	0.92
900	8.30	8.76	9.22	9.67	10.13	10.58	11.03	11.48	11.93	12.37	12.81	13.92	15.01	16.09	17.15	18.21	19.26	20.29	21.31	22.33	23.32	24.31	25.29	26.23	27.20	900	0.00	0.09	0.24	0.41	0.56	0.68	0.80	0.90	0.97	1.03
1000	9.06	9.56	10.06	10.56	11.06	11.55	12.04	12.53	13.02	13.50	13.97	15.17	16.35	17.53	18.68	19.83	20.96	22.07	23.17	24.25	25.32	26.37	27.41	28.43	29.43	1000	0.00	0.10	0.26	0.46	0.62	0.75	0.89	0.99	1.08	1.15
1100	9.75	10.30	10.84	11.38	11.92	12.45	12.98	13.51	14.04	14.56	15.08	16.37	17.65	18.91	20.16	21.39	22.57	23.75	24.93	26.07	27.19	28.30	29.39	30.45	31.49	1100	0.00	0.11	0.29	0.50	0.68	0.83	0.97	1.09	1.19	1.26
1200	10.44	11.03	11.61	12.19	12.76	13.34	13.91	14.47	15.03	15.59	16.15	17.53	18.89	20.23	21.56	22.85	24.10	25.36	26.57	27.76	28.94	30.08	31.21	32.30	33.37	1200	0.00	0.12	0.31	0.55	0.75	0.91	1.06	1.19	1.30	1.38
1300	11.10	11.73	12.35	12.96	13.58	14.19	14.79	15.39	15.99	16.59	17.18	18.63	20.07	21.47	22.85	24.21	25.53	26.83	28.10	29.34	30.54	31.72	32.86	33.97	35.03	1300	0.00	0.13	0.34	0.59	0.81	0.98	1.15	1.29	1.41	1.49
1400	11.74	12.40	13.06	13.71	14.36	15.00	15.64	16.28	16.91	17.53	18.15	19.69	21.19	22.66	24.10	25.50	26.87	28.21	29.51	30.78	32.01	33.19	34.34	35.45	36.52	1400	0.00	0.13	0.37	0.64	0.87	1.06	1.24	1.39	1.53	1.61
1500	12.35	13.05	13.74	14.43	15.11	15.78	16.45	17.12	17.78	18.44	19.08	20.69	22.25	23.77	25.26	26.71	28.11	29.48	30.80	32.08	33.31	34.50	35.66	36.73	37.77	1500	0.00	0.14	0.39	0.68	0.93	1.13	1.33	1.49	1.62	1.72
1600	12.92	13.66	14.39	15.11	15.82	16.53	17.23	17.92	18.61	19.29	19.97	21.62	23.24	24.81	26.36	27.82	29.25	30.63	31.96	33.24	34.46	35.63	36.74	37.79	38.78	1600	0.00	0.15	0.42	0.73	1.00	1.21	1.41	1.59	1.73	1.84
1700	13.49	14.25	15.01	15.76	16.50	17.23	17.96	18.68	19.39	20.10	20.80	22.51	24.17	25.77	27.33	28.83	30.27	31.66	32.99	34.24	35.44	36.57	37.64	38.63		1700	0.00	0.16	0.45	0.78	1.06	1.28	1.50	1.69	1.84	1.95
1800	14.02	14.81	15.59	16.37	17.16	17.90	18.65	19.39	20.13	20.86	21.57	23.33	25.02	26.66	28.23	29.74	31.16	32.56	33.86	35.09	36.24	37.32				1800	0.00	0.17	0.47	0.82	1.12	1.36	1.59	1.78	1.95	2.06
1900	14.52	15.34	16.15	16.95	17.74	18.52	19.30	20.06	20.82	21.56	22.30	24.08	25.81	27.46	29.04	30.55	31.97	33.32	34.59	35.77	36.86					1900	0.00	0.18	0.50	0.87	1.18	1.43	1.68	1.88	2.05	2.18
2000	14.99	15.83	16.67	17.49	18.31	19.11	19.90	20.69	21.46	22.21	22.96	24.78	26.52	28.17	29.73	31.24	32.64	33.95	35.16							2000	0.00	0.19	0.52	0.91	1.24	1.51	1.77	1.99	2.17	2.29
2100	15.43	16.30	17.15	18.00	18.83	19.65	20.46	21.26	22.04	22.81	23.57	25.40	27.15	28.80	30.36	31.81	33.17	34.42								2100	0.00	0.20	0.55	0.96	1.31	1.58	1.86	2.09	2.27	2.41
2200	15.86	16.75	17.60	18.47	19.32	20.15	20.97	21.78	22.57	23.35	24.13	25.97	27.69	29.33	30.86	32.27	33.57									2200	0.00	0.21	0.58	1.01	1.37	1.66	1.94	2.19	2.38	2.52
2300	16.22	17.12	18.02	18.90	19.76	20.61	21.44	22.25	23.05	23.83	24.60	26.44	28.15	29.76	31.25	32.60										2300	0.00	0.22	0.60	1.05	1.43	1.73	2.03	2.29	2.49	2.64
2400	16.56	17.46	18.39	19.28	20.16	21.01	21.85	22.67	23.47	24.25	25.02	26.86	28.54	30.10	31.52											2400	0.00	0.23	0.63	1.10	1.49	1.81	2.12	2.39	2.60	2.75
2500	16.87	17.81	18.73	19.63	20.51	21.37	22.21	23.02	23.83	24.61	25.37	27.17	28.82	30.33												2500	0.00	0.24	0.66	1.14	1.55	1.88	2.21	2.49	2.71	2.87
2600	17.15	18.10	19.02	19.93	20.82	21.68	22.52	23.34	24.13	24.90	25.65	27.41	29.02													2600	0.00	0.25	0.68	1.19	1.62	1.96	2.30	2.59	2.82	2.98
2700	17.39	18.35	19.28	20.19	21.07	21.93	22.77	23.58	24.37	25.13	25.86	27.58														2700	0.00	0.26	0.71	1.23	1.68	2.04	2.39	2.69	2.92	3.10
2800	17.59	18.55	19.49	20.40	21.28	22.14	22.97	23.77	24.54	25.29	26.00	27.65														2800	0.00	0.27	0.73	1.28	1.74	2.11	2.47	2.78	3.03	3.21
2900	17.76	18.72	19.66	20.56	21.44	22.29	23.11	23.89	24.65	25.37	26.06															2900	0.00	0.28	0.76	1.33	1.80	2.19	2.56	2.88	3.14	3.33
3000	17.89	18.85	19.78	20.66	21.55	22.38	23.18	23.95	24.68	25.38																3000	0.00	0.29	0.79	1.37	1.87	2.26	2.65	2.98	3.25	3.44
3100	17.98	18.94	19.86	20.75	21.60	22.42	23.20	23.94	24.65																	3100	0.00	0.30	0.81	1.42	1.93	2.34	2.74	3.08	3.36	3.56
3200	18.02	18.98	19.89	20.76	21.60	22.40	23.15																			3200	0.00	0.31	0.84	1.46	1.99	2.41	2.83	3.18	3.47	3.67
3300	18.04	18.97	19.87	20.73	21.54	22.31																				3300	0.00	0.32	0.87	1.51	2.05	2.49	2.92	3.28	3.57	3.79
3400	18.00	18.92	19.80	20.64	21.42																					3400	0.00	0.33	0.89	1.55	2.12	2.56	3.00	3.38	3.68	3.90
3500	17.92	18.83	19.68	20.49																						3500	0.00	0.34	0.92	1.60	2.18	2.64	3.09	3.48	3.79	4.02
3600	17.80	18.68	19.51																							3600	0.00	0.35	0.94	1.65	2.24	2.72	3.18	3.58	3.90	4.13
3700	17.64	18.49																								3700	0.00	0.36	0.97	1.69	2.30	2.79	3.27	3.68	4.01	4.24
3800	17.43																									3800	0.00	0.37	1.00	1.74	2.36	2.87	3.36	3.78	4.12	4.36

Belt Speed over 6500 feet per minute. Special sheaves may be necessary.

Table 3-16

Horsepower Ratings for 8V Section V Belts [12]

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
	Basic Horsepower Per Belt																								
	Small Sheave Outside Diameter, Inches																								
RPM OF FASTER SHAFT	13.0	13.5	14.0	14.5	15.0	15.5	16.0	16.5	17.0	17.5	18.0	18.5	19.0	19.5	20.0	20.5	21.0	21.5	22.0	22.5	23.0	23.5	24.0	24.5	25.5
435	20.00	21.48	22.96	24.43	25.90	27.35	28.80	30.25	31.69	33.12	34.54	35.96	37.37	38.70	40.18	41.57	42.95	44.83	45.70	47.07	48.43	49.78	51.12	52.46	55.11
485	21.88	23.51	25.14	26.75	28.36	29.96	31.55	33.14	34.71	36.28	37.84	39.39	40.93	42.47	43.99	45.51	47.02	48.52	50.01	51.49	52.96	54.43	55.89	57.33	60.20
575	25.12	27.01	28.89	30.76	32.61	34.45	36.78	38.10	39.91	41.70	43.49	45.26	47.02	48.77	50.50	52.22	53.93	55.63	57.31	58.98	60.63	62.28	63.91	65.52	68.71
585	25.47	27.39	29.99	31.19	33.07	34.93	36.79	38.63	40.47	42.29	44.09	45.89	47.67	49.44	51.19	52.94	54.66	56.38	58.08	59.77	61.45	63.11	64.75	66.38	69.60
690	29.00	31.19	33.37	35.53	37.67	39.80	41.91	44.00	46.07	48.12	50.16	52.17	54.17	56.15	58.11	60.04	61.96	63.86	65.74	67.39	69.43	71.24	73.03	74.80	78.27
725	30.12	32.40	34.66	36.91	39.13	41.34	43.52	45.69	47.83	49.95	52.06	54.14	56.20	58.24	60.25	62.25	64.22	66.16	68.09	69.99	71.86	73.71	75.54	77.34	80.86
870	34.45	37.06	39.65	42.21	44.74	47.23	49.60	52.14	54.54	56.91	59.24	61.55	63.81	66.05	68.24	70.40	72.53	74.61	76.66	78.67	80.63	82.56	84.45	86.29	89.85
950	36.61	39.39	42.14	44.85	47.52	50.15	52.74	55.29	57.80	60.28	62.70	65.09	67.43	69.73	71.98	74.19	76.35	78.46	80.52	82.53	84.49	86.40	88.26	90.06	93.50
1160	41.50	44.63	47.70	50.70	53.65	56.53	59.34	62.08	64.76	67.36	69.89	72.35	74.73	77.04	79.26	81.41	83.47	85.45	87.34	89.15	90.86				
1425	45.85	49.22	52.49	55.65	58.70	61.64	64.46	67.16	69.74	72.19	74.52	76.61													
50	3.93	3.23	3.43	3.64	3.84	4.04	4.24	4.43	4.63	4.83	5.03	5.23	5.42	5.62	5.82	6.01	7.21	6.41	6.80	6.80	6.99	7.19	7.38	7.57	7.96
100	5.61	6.00	6.38	6.76	7.15	7.53	7.91	8.29	8.67	9.05	9.43	9.80	10.18	10.56	10.93	11.31	11.68	12.05	12.42	12.80	13.17	13.54	13.91	14.28	15.01
150	8.01	8.58	9.14	9.70	10.26	10.81	11.37	11.92	12.47	13.02	13.57	14.12	14.67	15.22	15.76	16.31	16.85	17.39	17.93	18.47	19.01	19.55	20.09	20.62	21.69
200	10.30	11.04	11.77	12.50	13.22	13.95	14.67	15.40	16.17	16.83	17.55	18.26	18.98	19.69	20.40	21.11	21.81	22.52	23.22	23.92	24.62	25.32	26.01	26.71	28.10
250	12.50	13.40	14.30	15.19	16.08	16.97	17.86	18.74	19.63	20.51	21.38	22.26	23.13	24.00	24.87	25.73	26.60	27.46	28.31	29.17	30.02	30.86	31.73	32.37	34.26
300	14.61	15.68	16.74	17.79	18.85	19.90	20.94	21.99	23.02	24.06	25.09	26.12	27.15	28.17	29.19	30.21	21.22	32.23	33.24	34.24	35.24	36.24	37.24	38.23	40.20
350	16.66	17.88	19.10	20.32	21.53	22.73	23.93	25.13	26.32	27.50	28.69	29.87	31.04	32.21	33.38	34.54	35.69	36.85	38.00	39.14	40.28	41.42	42.55	43.68	45.92
400	18.65	20.02	21.40	22.76	24.12	25.48	26.83	28.17	29.51	30.84	32.17	33.49	34.81	36.12	37.42	38.72	40.02	41.31	42.59	43.87	45.14	46.40	47.66	48.92	51.41
450	20.57	22.10	23.62	25.14	26.64	28.14	29.64	31.12	32.60	34.08	35.54	37.00	38.45	39.90	41.34	42.77	44.19	45.60	47.01	48.41	49.81	51.19	52.57	53.94	56.66
500	22.43	24.11	25.78	27.44	29.09	30.73	32.36	33.99	35.60	37.21	38.81	40.39	41.97	43.55	45.11	46.66	48.21	49.74	51.27	52.78	54.29	55.79	57.27	58.75	61.68
550	24.24	26.06	27.87	29.67	31.45	33.23	35.00	36.75	38.50	40.73	41.96	43.67	43.37	47.06	48.74	50.41	52.06	53.71	55.34	56.96	58.57	60.17	61.75	63.33	66.43
600	25.99	27.95	29.89	31.83	33.75	35.65	37.55	39.43	41.29	43.15	44.99	46.82	48.63	50.44	52.22	54.00	55.75	57.50	59.23	60.95	62.65	64.33	66.00	67.66	70.92
650	27.69	29.78	31.85	33.91	35.96	37.99	40.01	42.00	43.99	45.96	47.91	49.85	51.76	53.67	55.35	57.42	59.27	61.11	62.93	64.72	66.30	68.27	70.01	72.73	75.12
700	29.32	31.54	33.74	35.93	38.09	40.24	42.37	44.48	46.58	48.65	50.71	52.74	54.76	56.75	58.73	60.68	62.62	84.53	66.47	68.29	70.14	71.96	73.76	75.54	79.03
750	30.90	33.23	35.57	37.87	40.15	42.41	44.65	46.86	49.06	51.23	53.38	55.50	57.60	59.68	61.74	63.76	65.77	67.75	69.70	71.63	73.53	75.40	77.25	79.07	82.62
800	32.42	34.88	37.32	39.73	42.12	44.48	46.82	49.13	51.42	53.68	55.92	58.12	60.30	62.45	64.57	66.66	68.73	70.76	72.76	74.74	76.67	78.58	80.45	82.30	85.88
850	33.88	36.45	39.00	41.52	44.01	46.47	48.90	51.30	53.67	56.01	58.32	60.60	62.84	65.05	67.23	69.37	71.48	73.56	75.59	77.59	79.56	81.48	83.37	85.21	88.79
900	35.28	37.96	40.61	43.22	45.81	48.36	50.87	53.35	55.80	58.72	60.59	62.92	65.22	67.48	69.90	71.88	74.03	76.12	78.18	80.20	82.17	84.09	85.97	87.81	91.34
950	36.61	39.39	42.14	44.85	47.52	50.15	52.74	55.79	57.80	60.78	62.70	65.09	67.43	69.73	71.98	74.19	76.35	78.46	80.52	82.53	84.49	86.40	88.26	90.06	93.50
1000	37.89	40.76	45.39	46.38	49.13	51.84	54.50	57.11	59.68	62.20	64.67	67.10	69.47	71.79	74.06	76.28	78.44	80.54	82.39	84.59	86.52	88.39	90.21	91.96	95.27
1050	39.09	42.05	44.97	47.63	50.65	53.42	56.14	58.81	61.42	63.98	66.49	68.93	71.32	73.66	75.93	78.14	80.29	82.37	84.39	86.35	88.24	90.06	91.81	93.48	96.62
1100	40.23	43.27	46.26	49.19	52.08	54.90	57.67	60.38	63.02	63.61	68.14	70.60	72.99	75.32	77.58	79.77	81.89	83.94	85.91	87.81	89.63	91.38	93.04		
1150	41.30	44.41	47.47	50.46	53.40	56.27	59.07	61.81	64.48	67.09	69.62	72.08	74.46	76.77	79.01	81.16	83.23	85.23	87.13	88.96	90.69				
1200	42.29	45.47	48.59	51.63	54.61	57.52	60.35	63.11	65.80	68.40	70.93	73.37	75.73	78.01	80.20	82.30	84.31	86.23	88.05						
1250	43.22	46.46	49.62	52.71	55.72	58.65	61.50	64.27	66.96	69.55	72.06	74.47	76.79	79.02	81.15	83.18	85.11								
1300	44.07	47.35	50.56	53.68	56.72	59.67	62.52	65.29	67.96	70.53	73.00	75.37	77.64	79.80	81.85	83.79									
1350	44.84	48.17	51.40	54.55	57.60	60.55	63.40	66.15	68.80	71.33	73.76	76.07	78.26	80.34											
1400	45.53	48.89	52.15	55.31	58.37	61.31	64.15	66.87	69.47	71.95	74.31	76.55	78.65												
1450	46.14	49.53	52.80	55.96	59.01	61.94	64.74	67.42	69.97	72.38	74.67														
1500	46.67	50.07	53.35	56.50	59.53	62.43	65.19	67.81	70.29	72.62															
1550	47.11	50.52	53.79	56.92	59.92	62.77	65.48	68.03	70.43																
1600	47.47	50.87	54.12	57.23	60.18	62.98	65.61	68.08																	
1650	47.73	51.12	54.34	57.41	60.31	63.04	65.59																		
1700	47.90	51.26	54.45	57.47	60.30	62.94																			
1750	47.98	51.31	54.45	57.39	60.14																				
1800	47.97	51.74	54.32	57.19																					
1850	47.85	51.07	54.07																						
1900	47.63	50.78	53.70																						
1950	47.32	50.38																							
2000	46.89																								

1	27	28	29	30	31	32	33	34	35	36
	Additional Horsepower Per Belt									
	Speed Ratio Range									
RPM OF FASTER SHAFT	1.00 to 1.01	1.02 to 1.05	1.06 to 1.11	1.12 to 1.18	1.19 to 1.26	1.27 to 1.38	1.39 to 1.57	1.58 to 1.94	1.95 to 3.38	3.39 and over
435	0.00	0.20	0.56	0.97	1.32	1.60	1.67	2.11	2.30	2.43
485	0.00	0.23	0.62	1.08	1.47	1.78	2.09	2.35	2.56	2.71
575	0.00	0.27	0.73	1.28	1.74	2.11	2.47	2.79	3.03	3.21
585	0.00	0.27	0.75	1.30	1.77	2.15	2.52	2.83	3.09	3.27
690	0.00	0.32	0.88	1.54	2.09	2.54	2.97	3.34	3.64	3.86
725	0.00	0.34	0.93	1.62	2.20	2.66	3.12	3.51	3.83	4.05
870	0.00	0.41	1.11	1.94	2.64	3.20	3.74	4.21	4.59	4.86
950	0.00	0.45	1.21	2.12	2.88	3.49	4.09	4.60	5.01	5.31
1160	0.00	0.54	1.48	2.58	3.52	4.26	4.99	5.62	6.12	6.48
1425	0.00	0.67	1.82	3.17	4.32	5.24	6.13	6.90	7.52	7.96
50	0.00	0.02	0.06	0.11	0.15	0.18	0.22	0.24	0.26	0.28
100	0.00	0.05	0.13	0.22	0.30	0.37	0.43	0.48	0.53	0.56
150	0.00	0.07	0.19	0.33	0.45	0.55	0.65	0.73	0.79	0.84
200	0.00	0.09	0.26	0.45	0.61	0.73	0.86	0.97	1.06	1.12
250	0.00	0.12	0.32	0.56	0.76	0.92	1.06	1.21	1.32	1.40
300	0.00	0.14	0.38	0.67	0.91	1.10	1.29	1.45	1.58	1.68
350	0.00	0.16	0.45	0.78	1.06	1.29	1.51	1.70	1.85	1.96
400	0.00	0.19	0.51	0.89	1.21	1.47	1.72	1.94	2.11	2.24
450	0.00	0.21	0.58	1.00	1.36	1.65	1.94	2.18	2.37	2.52
500	0.00	0.23	0.64	1.11	1.52	1.84	2.15	2.42	2.64	2.79
550	0.00	0.26	0.70	1.23	1.67	2.02	2.37	2.66	2.90	3.07
600	0.00	0.28	0.77	1.34	1.82	2.20	2.58	2.91	3.17	3.35
650	0.00	0.30	0.83	1.45	1.97	2.39	2.80	3.15	3.43	3.63
700	0.00	0.33	0.89	1.56	2.12	2.57	3.01	3.39	3.69	3.91
750	0.00	0.35	0.96	1.67	2.27	2.76	3.23	3.63	3.96	4.19
800	0.00	0.38	1.02	1.78	2.43	2.94	3.44	3.88	4.22	4.42
850	0.00	0.40	1.09	1.89	2.58	3.12	3.66	4.12	4.49	4.75
900	0.00	0.42	1.15	2.00	2.73	3.31	3.87	4.36	4.75	5.03
950	0.00	0.45	1.21	2.12	2.88	3.49	4.09	4.60	5.01	5.31
1000	0.00	0.47	1.28	2.23	3.03	3.67	4.30	4.84	5.28	5.59
1050	0.00	0.49	1.34	2.34	3.18	3.86	4.52	5.09	5.54	5.97
1100	0.00	0.52	1.41	2.45	3.34	4.04	4.73	5.33	5.80	6.15
1150	0.00	0.54	1.47	2.56	3.49	4.23	4.95	5.57	6.07	6.43
1200	0.00	0.56	1.53	2.67	3.64	4.41	5.16	5.81	6.33	6.71
1250	0.00	0.59	1.60	2.78	3.79	4.59	5.38	6.05	6.60	6.99
1300	0.00	0.61	1.66	2.90	3.94	4.78	5.60	6.30	6.86	7.27
1350	0.00	0.63	1.73	3.01	4.09	4.95	5.81	6.54	7.12	7.55
1400	0.00	0.66	1.79	3.12	4.24	5.14	6.03	6.78	7.39	7.82
1450	0.00	0.68	1.85	3.23	4.40	5.33	6.24	7.02	7.65	8.10
1500	0.00	0.70	1.92	3.34	4.55	5.51	6.46	7.27	7.92	8.38
1550	0.00	0.73	1.98	3.45	4.70	5.70	6.67	7.51	8.18	8.66
1600	0.00	0.75	2.04	3.56	4.85	5.88	6.89	7.75	8.44	8.94
1650	0.00	0.77	2.11	3.68	5.00	6.06	7.10	7.99	8.71	9.22
1700	0.00	0.80	2.17	3.79	5.15	6.25	7.32	8.23	8.97	9.50
1750	0.00	0.72	2.24	3.90	5.31	6.43	7.53	8.48	9.23	8.78
1800	0.00	0.84	2.30	4.01	5.46	6.61	7.75	8.72	9.50	10.06
1850	0.00	0.87	2.36	4.12	5.61	6.80	7.95	8.96	9.76	10.31
1900	0.00	0.89	2.41	4.23	5.76	6.96	8.18	8.96	9.76	10.34
1950	0.00	0.91	2.49	4.34	5.91	7.17	8.39	9.45	10.29	10.90
2000	0.00	0.94	2.56	4.86	6.06	7.35	8.61	9.99	10.55	11.18

[ ] Belt Speed over 6500 feet per minute. Special sheaves may be necessary.

(text continued from page 429)

Some of these are shown in Figure 3-26. Because most modern power transmissions use roller chains, silent chains or the offset link (Ewart) chains, these will be the only chain types discussed since they are quite important in oil field applications.

The term chain drive denotes a combination of chain and sprockets, with the sprockets mounted on rotating shafts.

Standardization of chains is under the jurisdiction of the American National Standards Institute [14,15].

Chain Terminology

There are basic terminology terms that aid in the description of chains. These are (see Figure 3-27; [11]) the following:

- *Pitch* is the distance between any point on a link and the same point on the next link in a straight (unarticulated) chain.
- *Drive sprocket* is usually the sprocket that is provided with the shaft import power (usually the smaller diameter sprocket).
- *Driven sprocket* is the sprocket and shaft that the chain is transferring power to (usually the larger diameter sprocket).
- *Center distance* is the length between the centers of the drive and drive sprocket shafts.

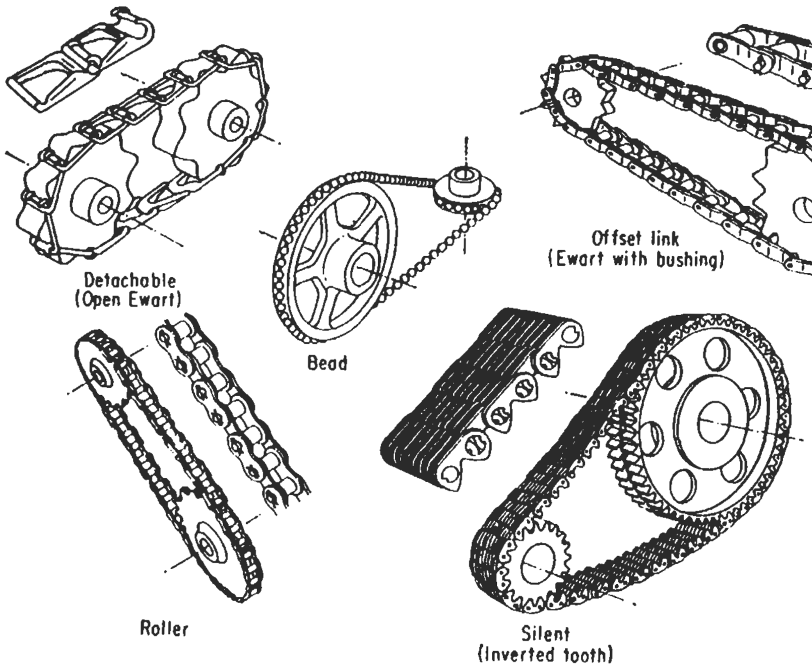


Figure 3-26. Commonly used chain types [8].

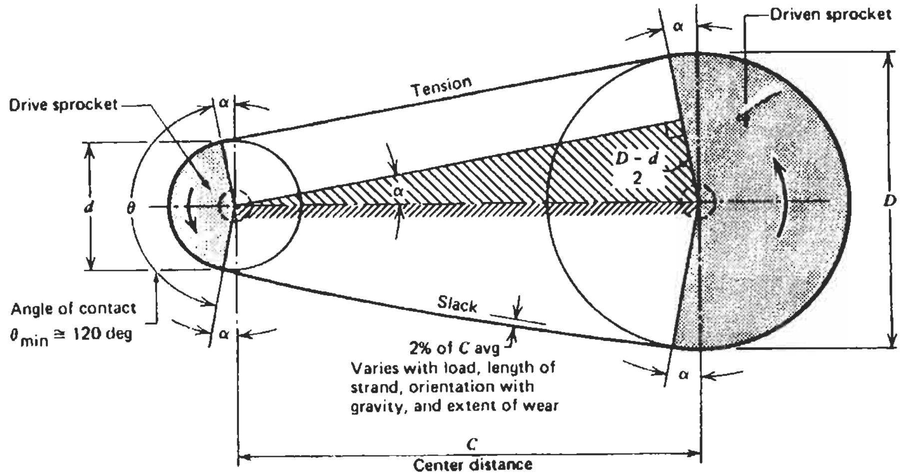


Figure 3-27. Chain drive terminology and geometry [11].

- *Chain length* is the total length of the chain (usually measured in pitches).
- *Pitch diameter* (of either the drive sprocket or driven sprocket) is generally on a theoretical circle described by the centerline of the chain as it passes over the sprocket.
- *Chain rating* is the load that a chain will satisfactorily handle over extended periods of time.
- *Angle of wrap* or *angle of contact* is the angular section of a sprocket that is in contact with the chain.

Roller Chain

Where accurate and higher-speed chain drives are required, roller chains are used. Roller chains are fundamentally a development of the block chain in which rollers have been inserted. The addition of the rollers increases the overall efficiency of the chain and permits it, in its ordinary form, to be operated at high speeds; a rate of 600–800 ft/min is usually recommended. The links can be furnished in a multitude of shapes, one common form being the "offset" shown in Fig. 3-28. This type of chain is furnished in pitches, i.e., center distance between rollers, from $\frac{3}{8}$ up to $2\frac{1}{2}$ in. and in breaking strengths, dependent upon the materials and the construction used, from about 10,000 to 200,00 lb [8,11].

Precision tools are used in the manufacture of roller chains, and a high degree of accuracy is maintained. The chain is made up of links, pins, and bushings, all fabricated from a high grade of steel, with the pins and the bushings ground to ensure accuracy of pitch.

Extreme care is used in the manufacture of some of the highest grades of roller chains, with the result that they can be operated at speeds as high as 4,000 ft/min. In general use, however, unless the chain is fully enclosed and well lubricated, the upper speed limit should be kept at about 1,400 ft/min. The chains operated at these high speeds must be fabricated from accurately ground rollers, bushings, and pins and be run over cast-iron or steel sprockets having accurately machined teeth. The general structure of a roller chain for this type of high-speed service is illustrated in Figures 3-29 and 3-30.

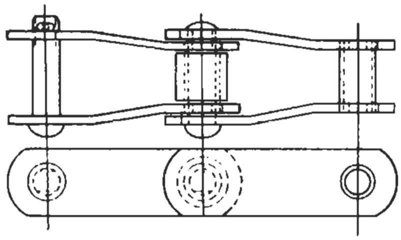


Figure 3-28. Steel offset link [8].

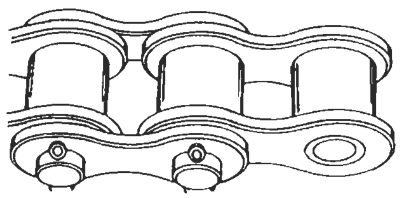


Figure 3-29. Standard roller chain configuration [8].

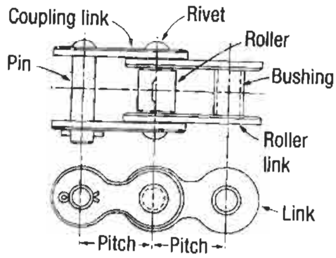


Figure 3-30. Roller chain details.

Roller chains can generally be furnished not only in single but in multiple widths as well. The use of the multiple roller chain makes possible a shorter pitch without sacrificing high-power capacity. Because the shorter pitch permits a greater number of teeth for an allowable sprocket diameter and a reduction in drive weight, quieter and smoother operation are obtained. The multiple chains may be operated at the same speed as a single chain of the same pitch. Theoretically, the power that can be transmitted by them is equal to the capacity of the single chain multiplied by the number of stands, but actually this quantity is usually reduced about 10%.

Sprockets. To secure full advantage of the modern roller chain, it should be operated on sprockets having accurately machined teeth, the profile of which has been specified or approved by the ANSI [14]. This profile, which is made up of circular arcs, is designed to compensate for the increase in pitch due to natural wear and thereby provides maximum efficiency throughout the life of the sprocket.

The shape (see Figure 3-31) of the standard form of tooth used on the roller-chain sprockets permits the rollers to ride farther out on the teeth as the chain is stretched under load or the pitch is increased by wear. A further effect of the particular design is an apparent slight increase in pitch diameter as the chain and sprockets wear, thus distributing the load over a larger number of teeth and the wear over a larger portion of the tooth surfaces.

Although as few as five or six teeth and as many as 250 have been used on sprockets with roller chains, experience and research have proved that, if speed and efficiency are to be considered, the number of teeth should be neither too few nor too many. It has been generally conceded by all chain manufacturers that 17-tooth sprockets are the minimum to be used at high speeds and that 19 or 21-tooth sprockets are even better. The upper limit should be kept at about 125. It is considered good practice to use an odd number of teeth on the smaller sprocket. For a speed ratio of 7:1 or larger, a double-reduction drive often may be actually cheaper and undoubtedly will give longer life.

If the number of teeth is reduced below the recommended minimum, the increased shock or hammer action of the rollers engaging the teeth will increase wear and materially shorten the life of the chain. If more than 125 teeth are used, a small amount of pitch elongation will cause the chain to ride the sprocket long before it is actually worn out. Figure 3-32 shows a typical double-strand roller chain and its sprockets [11].

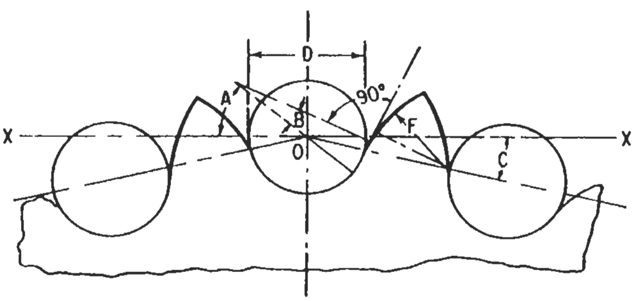


Figure 3-31. ANSI roller chain sprocket tooth [8].

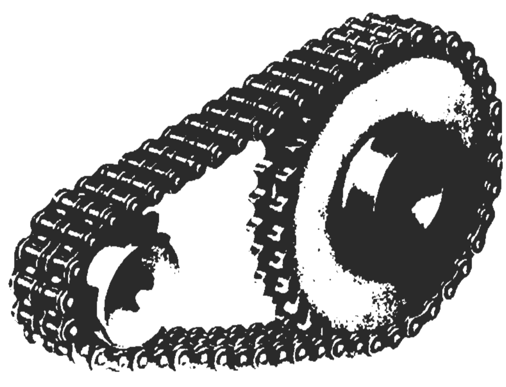


Figure 3-32. Double-strand roller chain (courtesy Borg Warner Corp.).

In general, four types of sprockets are available for roller chains. They are steel plate without hubs, cast iron or steel with hubs on one or both sides, split type, double-duty type.

The type of sprocket to be used depends entirely upon local or load conditions. The steel plate without hubs is the cheapest and is furnished for bolting to suitable hubs or flanges. The cast-iron or steel type fitted with hubs is made for direct mounting upon shaft and is fastened in place by either keys or setscrews or a combination of both. The split type is almost a necessity when the hub is mounted on a shaft with other pulleys or sheaves. Its construction facilitates installation and removal, but because of its extra cost it is usually not recommended except when solid hubs cannot be installed. The double-duty sprockets are made with steel rims or plates that may be removed or replaced without disturbing the hub, shaft, bearing, etc. Plates and hubs can be obtained either solid or split. They are particularly adapted for jobs requiring changing of drive ratios or where replacements must be made quickly.

Shaft Centers. It can be readily seen that on any chain drive the minimum center distances must be more than one-half the sum of the diameters of the two sprocket wheels. Experience has shown that best results are usually obtained when the center distance between shafts is 30 to 50 times the chain pitch. Forty times the chain pitch is about normal, and 80 times the pitch is maximum. In highly pulsating loads, 20 to 30 times the pitch is more nearly the correct center distance. Center distances of 10–12 ft are permissible with finished steel roller chains operated at moderate speeds without the use of idlers. On distances greater than this, an idler should be used to eliminate the possibility of swaying or flopping, which may cause the chain to jump the sprocket. When idlers are used, they should be placed on the slack strand of the chain. The number of teeth in the idler should be the largest possible and preferably not less than the number in the smaller sprocket of the drive.

Silent Chain

The expression "silent chain" may be somewhat misleading, for the type of chain which it is used to describe is not exactly silent, but it is much quieter in operation than the roller chain. The essential features of the silent chain are the straight-sided working jaws of the links, meshing with the straight-sided teeth of the sprocket, and the method of joining the links to form the chain. Figure 3-33 illustrates the principles of this chain, and shows that as the chain rides on the sprocket a rolling rather than a surface contact occurs. The construction tends to prevent undue vibration of the parts of the chain between sprockets [8,11].

In the silent chain the greatest wear occurs at the joint, and a number of developments and improvements have been made to reduce this to a minimum. If the first chain of this type (Figure 3-34a), the joint was made by means of a solid round pin that passed through circular holes punched in the links. The individual links bore directly upon the pins, and considerable wear resulted both in the links and the pins from the rubbing action produced by the wrapping of the chain around the sprockets. The first major improvement of this joint consisted of the placing of a bushing over the solid pin (Fig. 3-34b), thus producing a larger surface on which to distribute the load and a consequent reduction of the wear. This improvement was followed by still another one in which the solid bushing was replaced with a split or segmented one (Figure 3-34c). In still another design of joint (Figure 3-34d) the single round pin was replaced with two hardened alloy-steel specially shaped pins forming what is known as the "rocker-pin" joint. In this design the pin at one end of the link is the "seat pin" and that at the other is the "rocker," each securely held in their respective holes in the link and prevented from turning. Facing these pins in an individual link are the

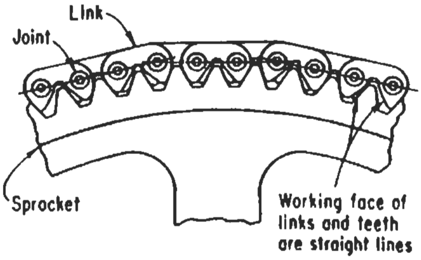


Figure 3-33. Silent chain and differential type sprocket teeth [8].

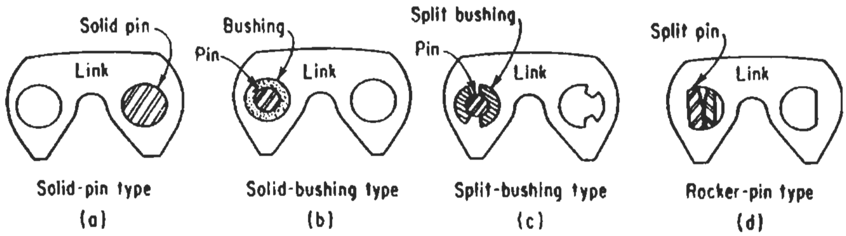


Figure 3-34. Types of silent chain joints [8].

pins in the adjoining link, with the seat pin in contact with the rocker and the rocker turning on a seat pin. The rocker is shaped and located to provide a flat surface against the seat, while the chain is pulling in a straight line and yet rolls over its cylindrical section as the chain flexes and passes on the sprocket. In this way a rolling contact occurs as the joint parts turn; the wearing qualities are improved, and vibration is reduced to a minimum.

To guide silent chains on their sprockets, a number of arrangements are used. They can, however, all be grouped under one of two heads: flanged links as part of the chain, or flanged jaws of the sprockets.

In the flanged-link type of guide, there are again two general classes: the "middle guide" and the "side flange" (Figure 3-35a,b). This method of guiding is particularly satisfactory when the drive is an integral part of the machine, and it is very reasonable at first cost. The choice between the two classes depends almost entirely upon the structural details of the sprockets. The middle guide is generally considered standard and is the one usually carried in stock by the chain manufacturer.

The "wire-flange" sprockets (Figure 3-35c) are assembled by simply inserting a preformed, crimped wire of the proper number of teeth and pitch along the edges of the teeth of the sprocket. This type of guide is recommended particularly in the case of independently mounted electric motors that are not likely to stay in accurate alignment.

The plate flange is used on the drive sprocket in those cases where heavy guiding action is expected such as is sometimes encountered in high-power devices.

Silent chains are furnished in pitches varying from $\frac{3}{16}$ to 3 in. and in weights capable of transmitting from a fraction of a horsepower up to several thousand. They can be operated at speeds up to 3,500 ft/min, but, where long life and low maintenance are desired, should be held between 1,200 and 1,500 ft/min. Although they can be

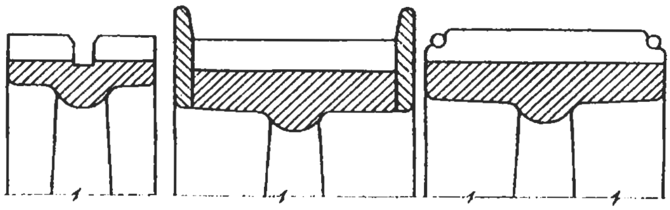


Figure 3-35. Guides for silent chain sprockets [8].

successfully operated at speeds below 1,200 ft/min, when extremely low speeds are required, it is usually more economical to use the roller chain.

Sprockets. Since the silent chain has not had the same wide usage as roller chains and because of the different types of construction that have been employed, there has been but little standardization of specializations. Each manufacturer has more or less carried out its own design and ratings, with the result that the sprockets to be used with this type of chain very accordingly.

Practically all sprockets used with silent chain are made with cut teeth. Those with up to 25 teeth are usually cut from a steel forging and are furnished with a solid hub. For sprockets with more than 25 teeth, a semisteel is used, and the sprockets are available in either solid or split form.

Differential-Cut Wheels. When a wheel has 81 or more teeth, a patented differential-cut construction is available that assures maximum drive life. The object is to cut the teeth of a wheel having a relatively large number of teeth so that two or more chain pitches, which are pivotally connected, act as a single unit to secure a greater angle of bend at the points of articulation and a corresponding larger component force acting toward the center of the wheel. For example, a 141-tooth $\frac{1}{2}$-in. pitch wheel cut to a differential of 3 will have the same operating characteristics as a 47-tooth $1\frac{1}{2}$-in. pitch wheel and will provide better chain action and longer chain life than a 141-tooth single-cut wheel. Figure 3-33 shows this type of wheel.

As a protection for equipment against overload, sprockets are sometimes provided for silent chains with a "shearing pin" or a "break-pin hub." A pin of a known strength is used, and an overload of sufficient magnitude shears the pin and leaves the plate free to turn on the hub. After the overload condition is remedied, a new pin may be easily inserted and the drive again made ready to operate.

Shaft Centers. As in roller chains, the minimum distance between shaft centers for silent chains must, in order to provide tooth clearance, be greater than half the sum of the two sprocket diameters. On large-speed reductions, experience has shown that the center distance should not be less than eight-tenths the difference in diameters of the two sprockets.

Offset Link (Ewart with Bushing), API Chain

The offset link is used principally in conveying and elevating equipment, although it is also frequently employed to transmit very little loads at comparatively low speeds.

This chain was originally patented over 60 years ago and has given a good account of itself ever since. It is furnished in either the open or closed type of link and is so designed as to permit ease of assembly and disassembly. The open-type link is illustrated in Figure 3-36 and the closed type in Figure 3-37. The use of the closed link results in greater strength and at the same time overcomes some objections by the exclusion of dust and grit.

The Ewart links are usually cast in one piece with no separate bushings or pins. The material used is generally malleable iron, although steel is also extensively employed. As a general rule the maximum speed at which this type of chain is operated is about 400 ft/min, and even at that speed it is apt to be quite noisy. Because the links used in this chain are not machined and the pitch is not very uniform, the teeth of the sprockets used with them are, in turn, generally not machined.

The offset link type of chain can be further improved by the use of a bushing, as shown in Figure 3-38. The improved or modified offset link can be used for more severe service conditions, but its safe maximum speed should, however, not exceed 400 ft/min. The offset link chain with bushing is known as the API chain [16]. These

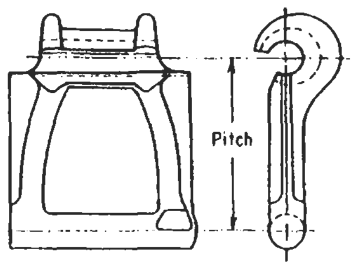


Figure 3-36. Open Ewart link [8].

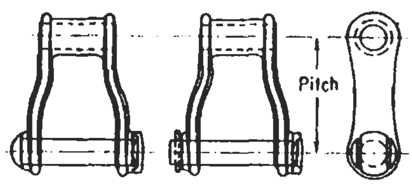


Figure 3-37. Closed Ewart link or pintle link [8].

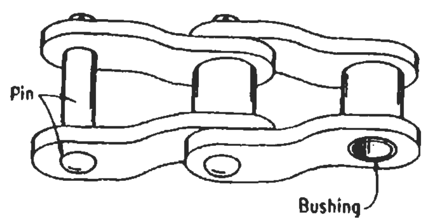


Figure 3-38. Ewart link with bushing (API chain) [6,16].

chains are in only three nominal sizes, 3 in., $3\frac{1}{8}$ in., and 4 in. In general, the use of this older type of chain is not recommended.

Design Consideration

There are design considerations for the various chain power transmission applications that are common to nearly all chain types [8]. These are discussed in the following.

Roller Chains. These chains are primarily selected for low and medium-speed service and for conveyor work, although roller chains are sometimes used for higher speeds because they operate more smoothly and quietly.

The factors that will be discussed are often overlooked or misunderstood when applying chain. Other factors to be considered are found in chain manufacturers' catalogs along with chain sizes, sprocket types, and engineering examples.

Number of Teeth in Wheel. For roller chains, pinions should have 17 teeth or more for moderate-speed drives and 21 teeth or more for high-speed drives. Fewer teeth may be used for low-speed drives, with 12 teeth a recommended minimum.

For silent chain drives, pinions with 21 teeth or more are recommended for general applications and at least 25 teeth are recommended for high-speed applications. The recommended minimum is 17 teeth.

When space limits the diameter of the larger wheel, it may be necessary to select a wider chain with a smaller pitch to obtain a desirable number of teeth in the pinion.

Small roller-chain wheels should be hardened when used on moderate and high-speed drives, very low-speed heavily loaded drives, and when operating under abrasive conditions or when the drive ratio is greater than 4:1. Steel silent chain pinions should be hardened.

For 1:1 and 2:1 ratio drives, wheels of relatively large diameter should be selected. Large diameters assure that the distance between the two spans of chain is great enough to prevent them from striking after slack from normal joint wear has accumulated. This is of particular importance for drives operating on long fixed centers with the slack span on the chain on top.

Chordal Action. The chordal rise and fall of each chain pitch as it contacts a sprocket tooth is known as chordal action and results in repeated variations in linear chain speed. As shown in Figure 3-39, the amount of chordal movement and chain-speed variation becomes progressively smaller as the number of teeth in the pinion is increased. Smoother operation and longer chain life may be obtained by selecting pinions with 21 teeth or more because chain-joint articulation is reduced. Chordal action becomes negligible when a sprocket has 25 teeth or more.

Prime Ratio. Ratios in excess of 7:1 are generally not recommended for roller chains. If greater speed reduction is required, it is desirable and usually more economical to compound two or more drives.

Properly engineered silent chain drives having ratios as great as 12:1 will perform satisfactorily. However, it might be more economical to consider a compound drive where the ratio is 8:1 or larger.

Large reduction drives on minimum wheel centers are more economical if small-pitch, wide chains are being considered. Small reduction drives on long wheel centers are cheaper when larger pitch, narrow chains are used.

Wheel Centers. To avoid interference, wheel centers must be more than one-half the sum of the wheel outside diameters. Where ratios are 2:1 to 7:1, a center distance

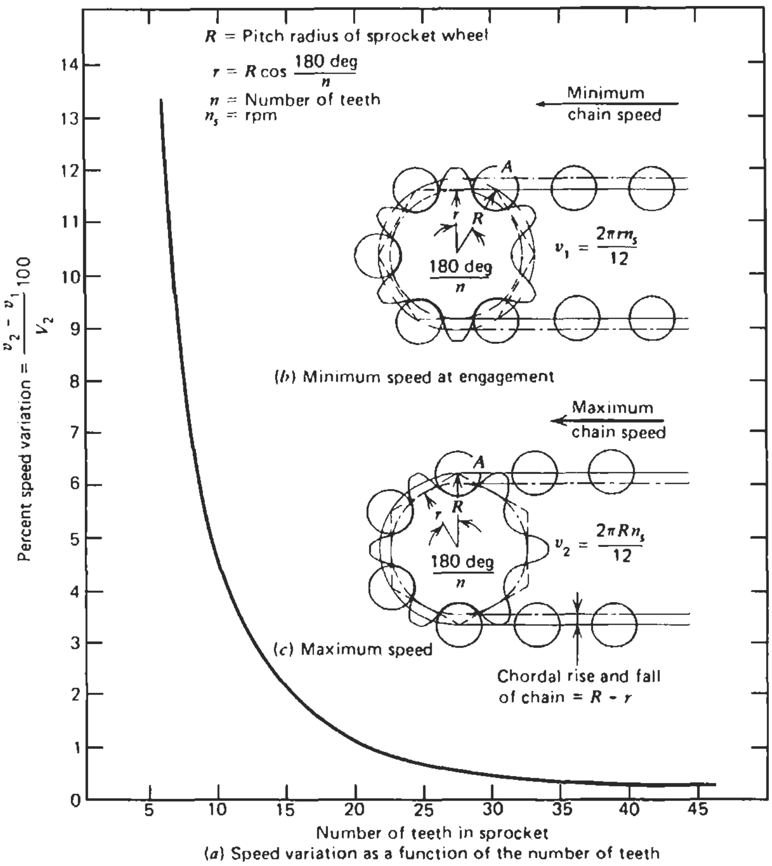


Figure 3-39. Chordal action of roller chain (courtesy of Link-Belt Co.).

equal to the diameter of the large wheel plus one-half the diameter of the small wheel is recommended. Drives so proportioned will have ample chain wrap on the small wheel. A chain wrap of 90° is regarded as an absolute minimum for load-carrying sprocket wheels, and 120° or more of wrap is considered desirable.

Chain Tension. All chain drives should have some means of controlling the chain sag caused by normal joint wear. This is of utmost importance when the drive is subject to shock or pulsating loads or to reversals in direction of rotation. The most common methods taking up chain slack are (1) drive units mounted on adjustable base plat, slide rails, or similar units; these are used extensively in motor-driven applications; and (2) the use of adjustable idlers (Figure 3-40) and chain tensioners.

An adjustable idler is recommended for drives having fixed centers, particularly if the line of centers is vertical or near vertical. With such idlers, the required chain tension can be maintained for correct chain and sprocket-wheel contact.

An adjustable idler on relatively long-center drives that are subject to pulsating loads will eliminate whipping or thrashing of the chain. Such whipping action results

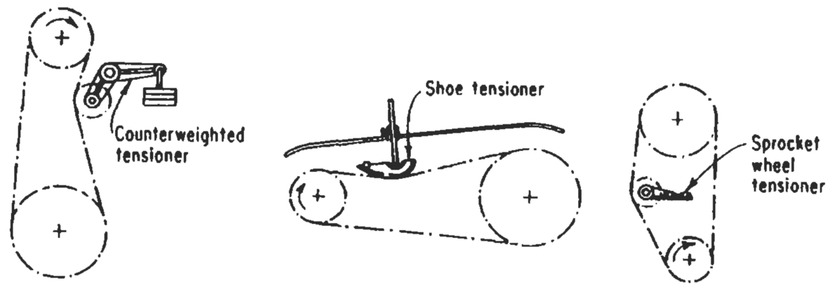


Figure 3-40. Typical roller chain tensioning methods [8].

in additional chain loading and joint wear, thereby reducing chain life. Also, an adjustable idler will provide sufficient chain wrap on the smaller sprocket wheel of large ratio, short-center drives.

Where maintenance service is infrequent, a counterweighted, spring-loaded, or automatic idler adjustment is best suited to promote long chain life. Manually adjusted idlers, if not periodically checked for proper chain tensioning, may become a destructive source in the drive system.

Adjustable-idler sprocket wheels must be securely mounted and should engage the slack or nonload carrying side of the chain. At least three teeth of the idler must be in full engagement with the chain. To take up the slack of accumulative wear of the chain, an idler needs only to be adjustable for slightly more than two chain pitches. Finally, idler sprocket wheels should have at least 17 teeth.

A hardened-steel or hardwood shoe bearing against the back of the chain is another method of controlling chain tension. The method is satisfactory for small horsepower drives operating on fixed centers at slow or moderate speed with ample lubrication.

Offset couplers may be used to adjust chain tension or chain length when other methods are not feasible. This is done by removing a section of chain having one more pitch than the offset coupler and inserting the coupler in its place.

Drive Types. Variable-speed drives may be selected on a chain-strength basis when operating with infrequent high chain loads such as from a torque converter, variable-speed motor, or multispeed transmission.

Speed-increasing drives should have at least 23 teeth in the smaller, faster running wheel. If possible, the taut span of the chain should be on top.

Vertical center drives require some form of chain tensioning or means for center adjustment to assure satisfactory operation and normal life. This is particularly important when the small wheel is in the lower position.

Fixed center drives should be selected on a conservative basis.

If an adjustable idler is not used, chain elongation may be retarded by using (1) a chain pitch larger than speed and horsepower indicate; (2) a larger service factor, thereby reducing the rate of wear in the chain joint; (3) as many teeth in the small sprocket wheel as the ratio will permit without exceeding 120 teeth in the larger wheel; and (4) good lubrication.

Lubrication. The primary purpose of chain lubrication is to maintain a film of oil between the bearing surfaces in the joints, thus assuring maximum operating efficiency. This clean oil film must be maintained at all load-carrying points where relative motion occurs, that is, between the pin and bushing on the chain, and the chain and wheel sprocket teeth. Table 3-17 gives some recommended methods for various speeds.

The method of application is primarily dictated by chain speed and the relative shaft positions. Some recommended methods are given below.

Manual Lubrication. This method is recommended for small horsepower drives with low chain speeds. Open running chains should not be exposed to abrasive dirt.

Drip-Cup Lubrication. This semiautomatic method is also suitable for small horsepower drives with low chain speeds. Cups should be located so that oil will drop to the center of the lower span at about 4 to 10 drops per minute.

Splash Lubrication. This is the simplest method of lubricating enclosed drives and is highly satisfactory for low and moderate-speed drives. In Figure 3-41 are shown a few arrangements for splash lubrication of different types of drives.

Oil-Disk Lubrication. This method is frequently used when a drive is not suitable for splash lubrication. It is highly satisfactory for moderate and semihigh-speed drives. The chain is kept above the oil level, and a circular disk, mounted to the lower wheel or shaft, dips into the oil about $\frac{1}{2}$ in. Figure 3-42 shows relative shaft positions best suited for oil-disk lubrication.

There must be sufficient disk speed to throw the oil. Figure 3-43 indicates the amount of oil delivered at various shaft speeds.

Forced lubrication is recommended for large-horsepower drives, heavily loaded drives, high-speed drives, or where splash and disk lubrication cannot be used. An oil pump—capacity about 1 gal/min—supplies a continuous spray of oil to the inside of the lower span of chain. The circulation of the lubricant aids in the dissipation of heat and results in a well-lubricated, cooler operating device.

Trial Chain Sizes. To aid in the selection of an approximate chain size, Figure 3-44 gives chains of various pitches for horsepower from 1 to 300. More precise values for chain capacity can be obtained in the various manufacturers catalogs.

Table 3-17
Recommended Methods of Chain Lubrication [8]

Chain speed, ft/min	Method
0–600	Manual: brush, oil can Slow drip: 4–10 drops/min Continuous: wick, wheel
600–1500	Rapid drip: 20 drops/min Shallow bath, disk
Over 1500	Force-feed systems

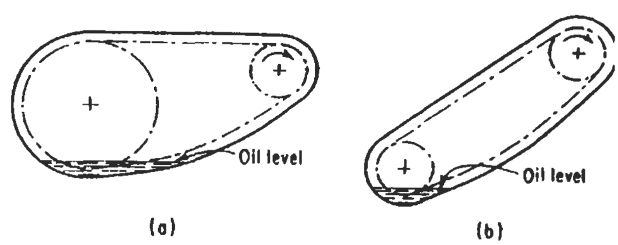


Figure 3-41. Splash-lubrication drive arrangements [8].

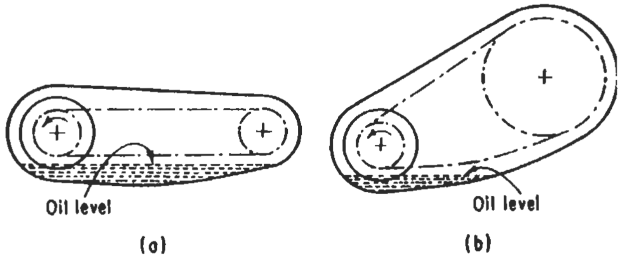
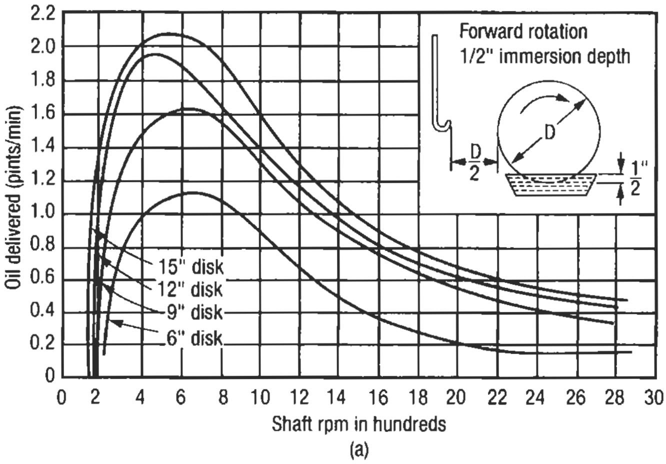


Figure 3-42. Disk lubrication (courtesy Link-Belt Co.).



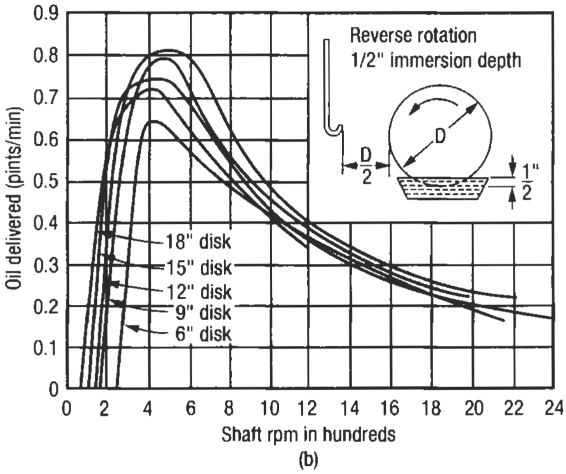


Figure 3-43. Oil delivered to a chain by forward (a) and reverse (b) rotating shafts [8].

Calculations. For a simple two-sprocket drive, the following guidelines for the initial design may be used [11]:

1. Minimum center distance is limited to the distance at which the two sprockets contact each other, or slightly more than half the sum of the outer diameters of the sprockets (see Figure 3.45a)
2. Based on experience, a center distance equal to the diameter of the large sprocket plus half the diameter of the small sprocket is a good compromise with regard to angle of wrap, wear, and initial cost (Figure 3-45). The center distance C(n) is

$$C = D + 0.5d \text{ for } D >> d \tag{3-28}$$

   where d is the driven sprocket diameter in inches D is the driven sprocket diameter in inches.
3. With large ratios, the angle of contact becomes smaller and the number of teeth engaged with the chain decreases. For angles less than 120°, θ increasingly becomes a critical factor in the design of chain drives.
4. For maximum life and minimum wear, the center distance should be chosen so as to provide an even number of links in the chain. This arrangement, coupled with an odd number of teeth in each sprocket, will minimize wear.
5. A short center distance provides a compact design (desirable) and allows for a shorter, less expensive chain. But wear is more rapid on a drive with a short center distance because the chain has fewer links and each joint must therefore articulate more often.
6. When the center distance exceeds 60 pitches, a long chain will be needed and a manufacturer's representative should be consulted.

The angle of wrap, θ (rad), is approximately

$$\theta = \pi - 2a = \frac{D - d}{C} \tag{3-29}$$

The chain length is generally measured in pitches because it consists of a whole number of links, each being of length p. Chain length is a function of the number of teeth in the two sprockets and the distance between sprocket centers. Thus, the chain length L (pitches) is

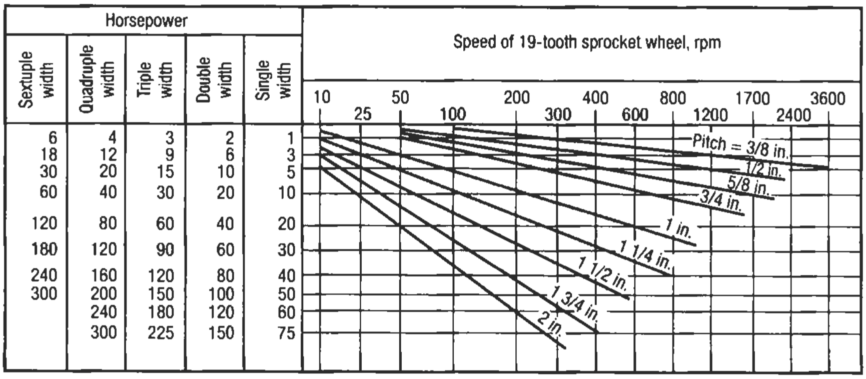


Figure 3-44. Trial-chain chart lets approximate chain be chosen (courtesy of Link-Belt Co.).

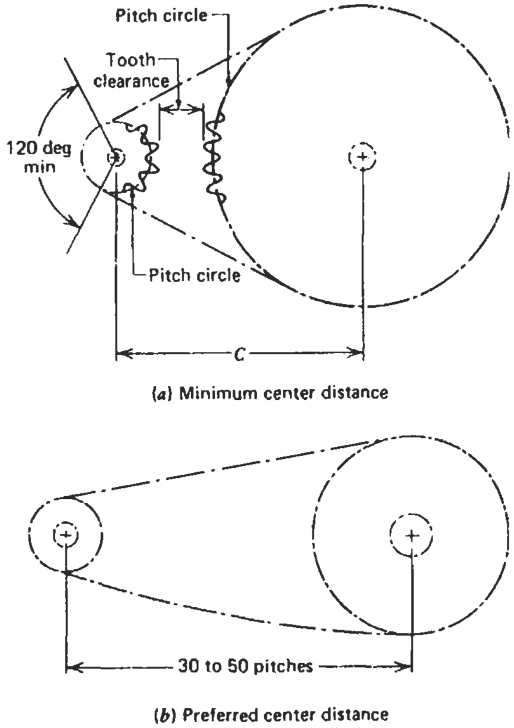


Figure 3-45. Shaft center distance (courtesy American Chair Association).

$$L = \frac{N + n}{2}\frac{2C}{p} + \frac{p(N - n)^2}{39.5C} \tag{3-30}$$

where n is the number of teeth in the drive sprocket, N is the number of teeth in the driven sprocket, C is the center distance in inches, p is pitch in inches.

Equation 3-30 does not often result in an even number of pitches. Thus, the results must be rounded off to a whole number, preferably an even whole number to avoid the specification of an offset link chain.

When several sprockets are used, center distances and chain lengths are best found by means of accurate engineering layouts and calculations.

The chain speed v (Hmin) in

$$v = \pi n_s \frac{d}{12} \tag{3-31}$$

where n_s is the drive shaft speed in rpm.

Design Data

In what follows, the pertinent design data are given for the roller chain, silent chain, and the offset link chain (API chain).

Roller Chain. Table 3-18 gives the typical service factor for roller chain drives [5].

Table 3-19 gives the basic roller chain design dimensions for ANSI standard roller chains by ANSI chain number [14].

Table 3-20 gives the horsepower ratings for single-strand roller chain drives. These data are given for each ANSI chain number [14].

Silent Chain. Table 3-21 gives the typical service factors for silent chain drives [5].

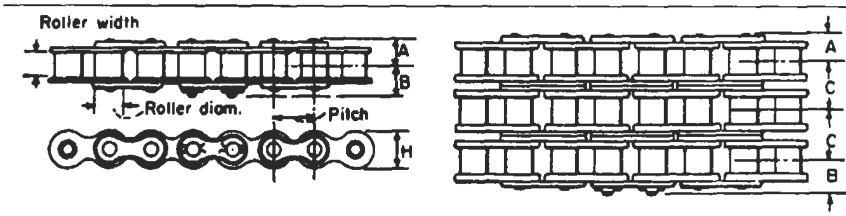
Tables 3-22 and 3-23 give the horsepower ratings per inch width for silent chain drives [5,15].

(text continues on page 3-66)

Table 3-18
Service Factors for Roller Chain Drives [5]

Type load	Service factor 10-hr day	24-day
Uniform	1.0	1.2
Moderate shock	1.2	1.4
Heavy shock	1.4	1.7

Table 3-19
Roller Chain Data and Dimensions [5]



ANSI chain number	Pitch, in.	Roller Width, in.	Roller Diam, in.	Pin diam, in.	Roller link plate Thickness, in.	Roller link plate Height H, in.	Dimensions, in. A	Dimensions, in. B	Dimensions, in. C	Tensile strength per strand, lb	Recommended max rpm, Teeth 12	18	24
25	1/4	1/8	0.130	0.091	0.030	0.230	0.150	0.190	0.260	875	5,000	7,000	7,000
35	3/8	3/16	0.200	0.141	0.050	0.344	0.224	0.290	0.400	2,100	2,380	3,780	4,200
41	1/2	1/4	0.306	0.141	0.050	0.383	0.256	0.315	......	2,000	1,750	2,725	2,850
40	1/2	5/16	0.312	0.156	0.060	0.452	0.313	0.358	0.563	3,700	1,800	2,830	3,000
50	5/8	3/8	0.400	0.200	0.080	0.594	0.384	0.462	......	6,100	1,300	2,030	2,200
50	5/8	3/8	0.400	0.200	0.080	0.545	0.384	0.462	0.707	6,600	1,300	2,030	2,200
60	3/4	1/2	0.469	0.234	0.094	0.679	0.493	0.567	0.892	8,500	1,025	1,615	1,700
80	1	5/8	0.625	0.312	0.125	0.903	0.643	0.762	1.160	14,500	650	1,015	1,100
100	1 1/4	3/4	0.750	0.375	0.156	1.128	0.780	0.910	1.411	24,000	450	730	850
120	1 1/2	1	0.875	0.437	0.187	1.354	0.977	1.123	1.796	34,000	350	565	650
140	1 3/4	1	1.000	0.500	0.220	1.647	1.054	1.219	1.929	46,000	260	415	500
160	2	1 1/4	1.125	0.562	0.250	1.900	1.250	1.433	2.301	58,000	225	360	420
180	2 1/4	1 13/32	1.406	0.687	0.281	2.140	1.421	1.770	2.530	76,000	180	290	330
200	2 1/2	1 1/2	1.562	0.781	0.312	2.275	1.533	1.850	2.800	95,000	170	260	300
240	3	1 7/8	1.875	0.937	0.375	2.850	1.722	2.200	3.375	135,000	120	190	210

Table 3-20

Horsepower Ratings for Single-Strand Roller Chain Driver [5]

Teeth	ANSI No. 25					¼-in. pitch						
	Rpm of sprocket											
	200	400	800	1,200	1,600	2,000	2,400	3,000	4,000	5,000	6,000	7,000
12	0.09	0.18	0.33	0.45	0.54	0.62	0.67	0.73	0.78	0.73		
15	0.12	0.24	0.44	0.60	0.73	0.85	0.94	1.03	1.14	1.20	1.14	
18	0.15	0.29	0.54	0.74	0.92	1.05	1.16	1.31	1.48	1.56	1.55	1.48
21	0.18	0.34	0.63	0.88	1.09	1.26	1.39	1.57	1.77	1.86	1.88	1.81
24	0.21	0.39	0.72	0.99	1.21	1.40	1.57	1.77	2.01	2.11	2.14	2.07

Teeth	ANSI No. 35					⅜-in. pitch						
	Rpm of sprocket											
	200	400	800	1,200	1,600	2,000	2,400	2,800	3,200	3,600	4,000	4,500
12	0.34	0.60	1.01	1.31	1.53	1.66	1.72	1.73				
15	0.43	0.78	1.35	1.78	2.12	2.37	2.54	2.65	2.70	2.69		
18	0.52	0.96	1.65	2.21	2.65	2.98	3.24	3.43	3.52	3.57	3.55	
21	0.61	1.12	1.95	2.61	3.14	3.53	3.86	4.08	4.22	4.28	4.28	
24	0.70	1.28	2.22	2.98	3.57	4.04	4.38	4.65	4.81	4.86	4.87	4.75

Teeth	ANSI No. 40					½-in. pitch						
	Rpm of sprocket											
	200	400	600	800	1,000	1,200	1,600	1,800	2,000	2,400	2,800	3,200
12	0.77	1.34	1.81	2.16	2.46	2.71	2.99	3.07	3.10			
15	0.99	1.76	2.40	2.93	3.38	3.77	4.32	4.52	4.67	4.81		
18	1.20	2.15	2.94	3.63	4.21	4.71	5.48	5.76	5.97	6.27	6.35	
21	1.41	2.52	3.47	4.27	4.97	5.57	6.50	6.86	7.13	7.50	7.63	
24	1.60	2.88	3.96	4.87	5.67	6.35	7.40	7.80	8.12	8.51	8.68	8.57

Teeth	ANSI No. 50					⅝-in. pitch						
	Rpm of sprocket											
	100	200	300	400	600	800	1,000	1,200	1,400	1,600	1,800	2,200
12	0.80	1.44	1.99	2.48	3.26	3.86	4.3	4.6	4.8			
15	1.02	1.87	2.61	3.27	4.39	5.31	6.0	6.8	7.0	7.3	7.5	
18	1.23	2.27	3.19	4.01	5.41	6.58	7.5	8.3	8.9	9.4	9.7	
21	1.45	2.66	3.75	4.70	6.38	7.77	8.9	9.8	10.6	11.1	11.6	11.9
24	1.65	3.05	4.27	5.37	7.28	8.85	10.2	11.2	12.1	12.6	12.1	13.6

Teeth	ANSI No. 60					¾-in. pitch						
	Rpm of sprocket											
	50	100	200	300	400	600	800	1,000	1,200	1,400	1,600	1,800
12	0.73	1.34	2.41	3.30	4.05	5.2	6.1	6.6	6.9			
15	0.92	1.72	3.14	4.34	5.39	7.1	8.5	9.5	10.2	10.6		
18	1.12	2.10	3.82	5.31	6.63	8.9	10.6	12.0	13.0	13.7	14.1	
21	1.31	2.46	4.49	6.24	7.80	10.4	12.5	14.1	15.4	16.3	16.9	
24	1.50	2.80	5.11	7.12	8.90	11.9	14.3	16.1	17.6	18.6	19.2	19.5

Teeth	ANSI No. 80					1-in. pitch						
	Rpm of sprocket											
	50	100	150	200	300	400	500	600	700	800	1,000	1,160
12	1.68	3.07	4.28	5.3	7.2	8.7	9.8	10.7	11.4			
15	2.14	3.95	5.57	7.0	9.6	11.8	13.6	15.1	16.3	17.3		
18	2.59	4.81	6.79	8.6	11.8	14.5	16.9	18.9	20.5	21.9	24.0	
21	3.03	5.62	7.96	10.1	13.9	17.1	19.9	22.3	24.3	26.0	28.5	
24	3.46	6.43	9.10	11.5	15.8	19.5	22.7	25.4	27.7	29.6	32.5	33.9

Table 3-20 (continued)

Teeth	ANSI No. 100					1¼-in. pitch						
	Rpm of sprocket											
	25	50	100	200	300	400	500	650	700	750	800	870
12	1.72	3.19	5.8	9.9	13.0	15.6	17.2					
15	2.19	4.10	7.5	13.1	17.5	21.3	24.0	27.2	28.1			
18	2.55	4.97	9.1	16.0	21.6	26.6	30.2	34.5	35.7	36.8		
21	3.08	5.80	10.7	18.9	25.5	31.4	35.7	40.9	42.3	43.5	44.6	
24	3.52	6.62	12.2	21.5	29.2	35.4	40.5	46.5	48.1	49.5	50.6	52.0

Teeth	ANSI No. 120					1½-in. pitch						
	Rpm of sprocket											
	25	50	75	100	150	200	250	300	350	400	500	600
12	2.90	5.4	7.6	9.6	13.2	16.2	18.7	21.0	22.8	24.3		
15	3.71	6.9	9.8	12.5	17.3	21.6	25.3	28.6	31.4	33.9	38.0	
18	4.74	8.4	12.0	15.3	21.3	26.6	31.3	35.4	39.2	42.4	47.9	
21	5.24	9.9	14.0	17.9	24.9	31.2	36.8	41.7	46.2	50.0	56.7	61.7
24	5.99	11.3	16.0	20.4	28.5	35.7	41.9	47.6	52.6	57.1	64.6	70.3

Teeth	ANSI No. 140					1¾-in. pitch						
	Rpm of sprocket											
	20	30	50	100	150	200	250	300	350	400	450	475
12	3.72	5.4	8.4	14.8	20.1	24.5	28.1	31.0				
15	4.73	6.9	10.8	19.3	26.6	32.8	38.2	42.8	46.7			
18	5.73	8.3	13.1	23.7	32.7	40.5	47.3	53.2	58.4	62.9		
21	6.70	9.7	15.3	27.7	38.4	47.6	55.7	62.8	69.0	74.5	79.0	
24	7.65	11.1	17.5	31.7	43.7	54.3	63.6	71.6	78.7	84.8	89.9	92.4

Teeth	ANSI No. 160					2-in. pitch						
	Rpm of sprocket											
	10	20	40	80	120	160	200	240	280	320	360	400
12	2.9	5.5	10.1	18.0	24.6	30.1	34.8	38.6				
15	3.7	7.0	13.0	23.5	32.4	40.2	47.0	52.9	58.0	62.4		
18	4.4	8.5	15.8	28.7	39.7	49.5	58.1	65.7	72.4	78.3		
21	5.2	9.9	18.5	33.6	46.7	58.1	68.3	77.5	85.5	92.5	99	
24	5.9	11.3	21.1	38.4	53.5	66.5	78.0	88.3	97.4	105.4	112	118

Teeth	ANSI No. 180					2¼-in. pitch						
	Rpm of sprocket											
	10	20	40	60	80	100	140	180	220	260	300	330
12	4.09	7.72	14.2	19.9	25.0	29.7	33.7	44.4				
15	5.19	9.84	18.3	25.8	32.7	39.0	50.2	59.8	68.1			
18	6.27	11.9	22.1	31.4	39.9	47.7	61.8	73.9	84.5	93.7		
21	7.33	14.0	26.0	36.9	47.0	56.3	73.0	87.6	100.3	111.4	121.1	
24	8.38	15.9	29.7	42.1	53.6	64.7	83.2	99.7	114.2	126.9	137.7	145.0

Teeth	ANSI No. 220					2½-in. pitch						
	Rpm of sprocket											
	10	20	40	60	80	100	120	160	200	240	260	280
12	5.6	10.5	19.1	26.8	33.6	39.6	45.1	54.4				
15	7.1	13.4	24.7	34.8	43.9	52.2	59.8	73.4	85			
18	8.6	16.2	30.0	42.4	53.7	64.1	73.7	90.7	105	118		
21	10.0	18.9	35.1	49.7	63.1	75.3	86.6	106.9	124	139	146	
24	11.4	21.6	40.2	56.8	71.9	86.0	98.8	121.8	142	159	166	173

Table 3-20 (continued)

Teeth	ANSI No. 240				3-in. pitch							
	Rpm of sprocket											
	10	20	30	40	60	80	100	120	140	160	180	200
12	9.5	17.7	25.1	32.0	44.4	55.1	64.7	73.0				
15	12.2	22.6	32.4	41.5	58.0	72.7	86.9	97.9	108.0	118.5		
18	14.6	27.7	39.3	50.5	70.8	87.1	105.7	120.7	134.5	147.0	158.0	
21	17.0	32.1	46.0	59.1	83.1	104.7	124.2	142.2	158.7	173.5	187.2	199.4
24	19.4	36.7	52.5	67.5	94.8	119.4	141.8	162.3	180.8	197.6	212.9	226.9

Table 3-21
Service Factors for Width-Silent Chain Driver [5]

Load type	Service factor 10-h day	24-h day
Uniform	1.0	1.3
Moderate shock	1.4	1.7
Heavy shock	1.7	2.0

Table 3-22
Horsepower Ratings per Inch Width Silent Chain Driver [5]

No. of teeth in smaller sprocket	Pitch, in.	Rpm of small sprocket						
		500	1,000	1,500	2,000	3,000	4,000	5,000
21	3/8	2.2	4.1	5.8	7.2	9.1	9.9	9.5
25	3/8	2.6	4.9	7.0	8.8	11.0	12.0	12.0
29	3/8	3.0	5.8	.2	10.3	13.0	15.0	15.0
33	3/8	3.5	6.6	9.4	12.0	15.0	17.0	17.0
37	3/8	3.9	7.3	11.0	13.0	17.0	19.0	19.0
21	1/2	4	7	....	11	13		
25	1/2	4	8	....	14	17	16	
29	1/2	5	10	....	17	20	20	
33	1/2	6	11	....	19	23	23	
37	1/2	7	13	....	21	26		

		Rpm of small sprocket							
		500	1,000	1,200	1,800	2,000	2,500	3,000	3,500
21	5/8	6	10	12	15	16	16	16	
25	5/8	7	13	15	19	20	21	21	19
29	5/8	8	15	17	22	24	25	25	23
33	5/8	9	17	20	26	27	29	29	27
37	5/8	10	19	22	29	31	34	33	
21	3/4	8	14	16	19	20	19		
25	3/4	10	17	20	25	25	24		
29	3/4	12	21	24	29	30	30		
33	3/4	13	24	27	34	35	35		
37	3/4	15	27	31	38	39	39		

Table 3-23
Horsepower Ratings per Inch Width-Silent Chain Driver [5]

No. of teeth in smaller sprocket	Pitch, in.	Rpm of small sprocket							
		300	500	700	1,000	1,200	1,500	1,800	2,000
21	1	9	14	18	23	25	26	26	
25	1	11	17	22	28	31	33	33	33
29	1	13	20	26	33	37	40	41	40
33	1	14	23	30	39	43	47	47	46
37	1	16	26	34	43	48	52	53	
21	1¼	14	21	26	32	33			
25	1¼	16	25	32	40	42	42		
29	1¼	19	30	38	47	50	51		
33	1¼	22	34	44	55	58	59		
37	1¼	24	38	50	61	65			

		Rpm of small sprocket								
		200	300	400	500	600	700	800	900	1,000
21	1½	13	19	24	29	32	35	37	39	39
25	1½	16	23	30	35	40	44	47	49	52
29	1½	19	27	35	41	47	52	56	59	60
33	1½	22	31	40	47	54	60	64	68	70
37	1½	24	35	47	53	61	67	72	77	79
45	1½	30	43	54	65	74	81	86	90	
21	2	23	32	40	42	50	52			
25	2	28	39	49	56	62	66	68	68	
29	2	33	46	58	67	74	79	82	82	
33	2	37	53	66	77	85	91	94	94	
37	2	42	60	74	88	99	102	105		
45	2	51	72	90	105	115	121			

Drilling Applications. Drilling equipment utilizes chain drives in various applications on the drilling rig itself and its auxiliary equipment. The drives for this type of equipment are called compound drives. Such drive equipment are subject to vary severe operational loads. Table 3-24 gives the typical service factors for these chain drives [16].

PUMPS

Pumps are a mechanical device that forces a fluid to move from one position to another. Usually a pump refers to the mechanical means to move incompressible (or nearly incompressible) fluid or liquid. Pumps are our earliest machine and are to this day one of our most numerous mechanical devices.

Pumps are a very essential part of the oil and gas industry. They are used throughout the industry, from drilling operations through to final delivery to the customer.

Classifications

Pumps are classified into two basic classes, displacement and dynamics. Figures 3-46 and 3-47 show the subclasses of pumps under each of these basic classes [17].

The most widely used pumps in the oil and gas industry are reciprocating displacement pumps (in particular piston plunger type), the rotary displacement pump, and the centrifugal dynamic pump. Only these pumps will be discussed in detail.

The reciprocating and rotary positive displacement pumps primary characteristic is that they have a nearly direct relationship between the motion of the pumping

Table 3-24
Drilling Equipment Service Factor [16]

1	2		3
	Typical Service Factor		
Compound Drives			
For Hoisting Service	12	—	16
For Pump Driving Service	16	—	21
Pump Final Drive	20	—	25
Pump Countershaft Drive[1]	20	—	25
Drawworks Input Drive Type A[2]	9	—	12
Drawworks Input Drive Type B[2]	5	—	7
Drawworks Transmission Drives[3]	—	—	—
Drawworks Low Drum Drive	2	—	3
Drawworks High Drum Drive	3.5	—	5
Rotary Countershaft Drive[4]	5	—	10
Rotary Final Drive[4]	5	—	10
Auxiliary Brake Drive[5]	—	—	—

[1]Pump countershaft drives are frequently very short centered which reduces the heat liberating surface and for high horsepower they may require supplementary cooling.

[2]Drawworks input drives are of two types. The type A drive is located between the prime movers and drawworks transmission. The type B drive is located between the transmission and drawworks. It is to be noted that drawworks input drives have smaller service factors than compound drives, based on experience. Among the reasons for this are the fewer horsepower hours logged in a given period and the fact that they are somewhat removed from the influence of engine impulses.

[3]Service factors on drawworks transmission drives have been ommitted from the Recommended Practices for two basic reasons. First, the Recommended Practices are basically intended for field design guides and it is not expected that such designs will be made in the field. Second, transmission designs cover a wide range of conditions such as shaft speeds, number of ratios, and methods of obtaining ratios, so that no one family of service factors could be made to apply.

[4]Rotary countershaft and final drives have benefited only slightly from experience as too little has been known about rotary horsepowers through the years.

[5]Auxiliary brake chain drives have been omitted because of the indefinite nature of the load. Each manufacturer has established successful drives, but these are suited to specific conditions which cannot be reduced to simple terms. Most such drives violate the rules of chain speeds and loads and are successful only because of the short duration of use and deviation from calculated loads. It is recommended that the drawworks manufacturer be consulted.

elements and the quantity of liquid pumped. Thus, in positive displacement pumps liquid displacement (or discharge from the device) is theoretically equal to the swept volume of the pumping element. Figure 3-48 shows the typical positive displacement plot of discharge rate Q (ft^3/s) versus pressure P (lbs/ft^2) [18]. The discharge rate remains the same (assuming a constant rate of rotation for the system) regardless of the pressure in the flow. The pressure in the flow is, of course, the result of resistance in the flow system the pump discharges to. If the resistance increases, rotation can be maintained and more force applied to each stroke of the pump (i.e., power). This is why the reciprocating piston plunger pump is also called a power pump. In practice,

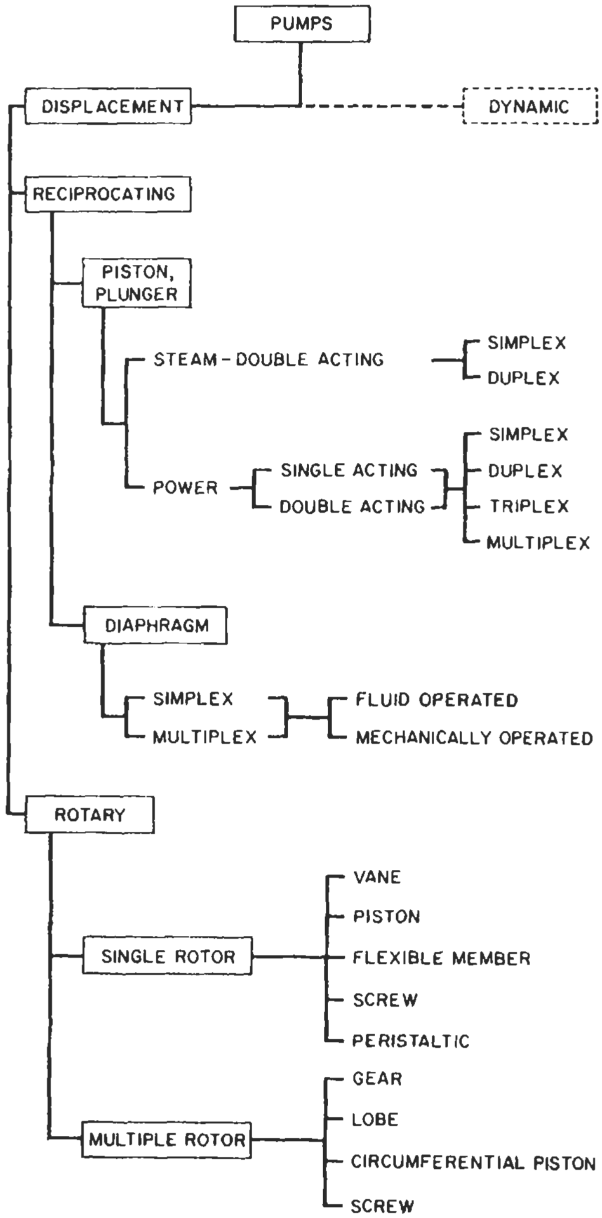


Figure 3-46. Classification of displacement pumps.

pressure does have some influence on the capacity of these pumps. This is because as the pressure increases, there is some leakage of the seals in the system. This leakage is somewhat proportional to the pressure, particularly beyond some characteristic

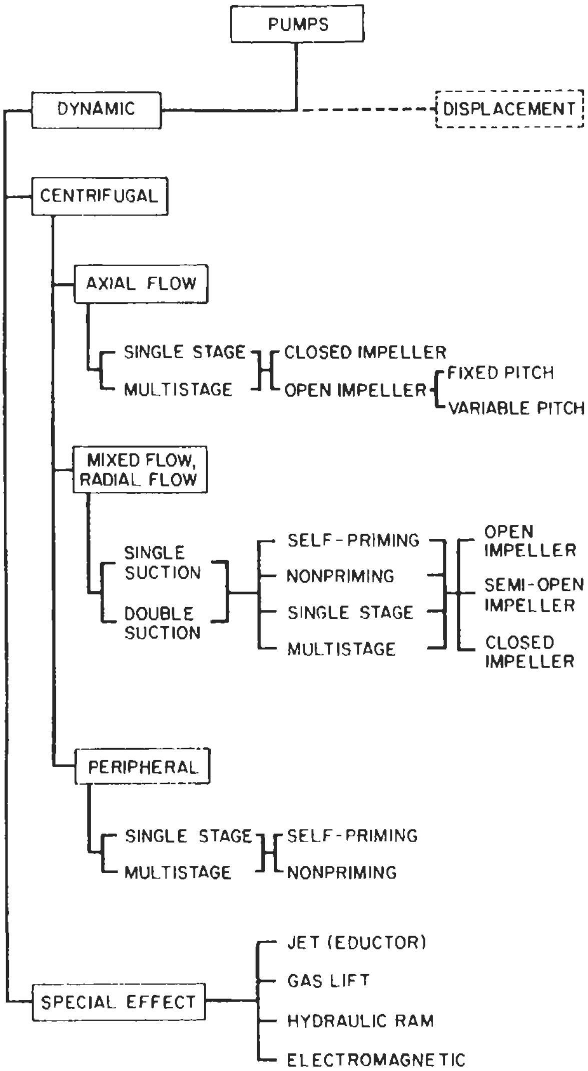


Figure 3-47. Classification of dynamic pumps [17].

pressure related to the seals. The difference between theoretical flow and the actual flow of a pump is often referred to as slip. This slip is shown in Figure 3-48.

In the dynamic pump, in particular, the centrifugal pump, the discharge rate Q is determined by the resistance pressure P in the flow system the pump discharges to (assuming some given speed of the pump). This is illustrated in Figure 3-49.

General Calculations

There are several important calculations that are needed to properly evaluate and select the appropriate positive displacement pump [17,18,19,20,21].

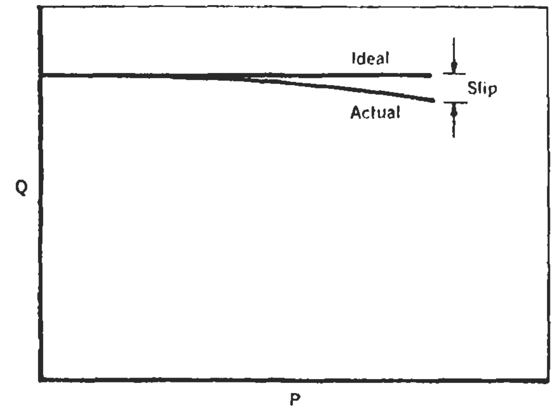


Figure 3-48. Positive displacement pump [17].

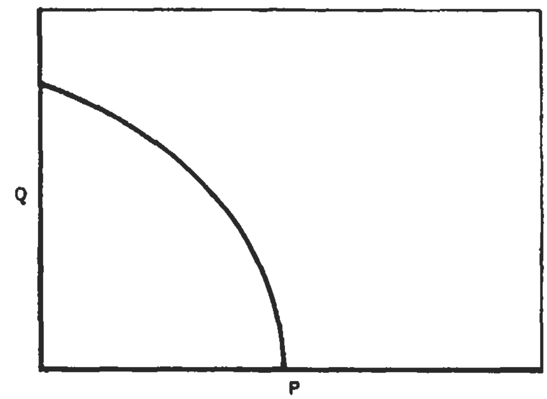


Figure 3-49. Dynamic (centrifugal) pumps [17].

The theoretical power **P** (hp), which is the power actually imported to the fluid by the pump, is

$$\mathbf{P} = \frac{qp}{1{,}714} \tag{3-32}$$

where q is the discharge rate in gpm and p is the differential pressure in psi (usually the actual gauge discharge pressure).

The total pump power input **Ppi** is the net power delivered to the pump drive shaft by the prime mover system. This is

$$\mathbf{Ppi} = \mathbf{P} + \mathbf{Ppl} \tag{3-33}$$

where **Ppl** is the pump power loss.

The total power input **P**ti is the net power delivered by the prime mover. This is

$$\mathbf{P}\mathrm{ti} = \mathbf{P}\mathrm{pi} + \mathbf{P}\mathrm{pml} \tag{3-34}$$

where **P**pml is the prime mover loss.

Overall efficiency e_0 is

$$e_0 = \frac{\mathbf{P}}{\mathbf{P}\mathrm{ti}} \tag{3-35}$$

usually multiplied by 100 and expressed as a percentage.

Pump efficiency e_p is

$$e_p = \frac{\mathbf{P}}{\mathbf{P}\mathrm{pi}} \tag{3-36}$$

(usually expressed as a percentage).

Mechanical efficiency e_m (which usually refers to the efficiency of the prime mover to supply power to the pump) is

$$e_m = \frac{\mathbf{P}\mathrm{pi}}{\mathbf{P}\mathrm{ti}} \tag{3-37}$$

(usually expressed as a percentage).

Reciprocating Pumps

The piston plunger pump is the simplest form of a positive displacement pump. These pumps can be powered by a variety of prime movers, internal combustion engines, and electric motors (and in some cases, powered by a gas turbine motor). In such applications, the separate pump unit is connected to the prime mover by a power transmission.

The capacity of a pump is determined by the number of plungers or pistons and the size of these elements (bore and stroke). A reciprocating pump is usually designed for a specific volumetric rate capacity and pressure capability. These factors are set by the application. Once the volumetric rate capacity and pressure capability are known, a designer can determine the plunger piston bore and stroke the rotation speed range and the power of the prime mover needed to complete the system.

Reciprocating pumps are fabricated in both horizontal and vertical configurations.

Single-Acting Pump

A single-acting pump has only one power (and discharge) stroke for its pistons. Such a pump brings fluid into its chamber through the inlet or suction value or the piston is drawn backward to open the chamber. To discharge the fluid, the inlet valve is closed and the outlet valve opened as the piston is forced forward to push the fluid from the chamber into the discharge line. The piston motion is accomplished by a rotating crankshaft that is connected to the piston by a piston rod much like an internal combustion piston engine. The rotating crankshaft of the pump is rotated by the rotational power of the prime mover (through a transmission) [21].

The single-action pump is usually available with three, five and even seven pistons. The odd number of pistons allows the pump to be rotationally balanced, and the use of at least three pistons reduces the discharge pulsation of these single-acting pumps. A three piston pump single-action pump is called a triplex pump. A five piston, or seven piston single-acting pump is called a multiplex pump.

Double-Acting Pump

Double-acting pumps have two power strokes. As a piston of the pump is pushed forward, the fluid is discharged from the forward chamber into the discharge line (much like a single-action piston). But during this same stroke, the chamber behind the piston (which contains the connecting rod) is being filled via that chamber's inlet valve (see Figure 3-50). When the forward power stroke is complete and the fluid discharged from the chamber in front of the piston, the chamber behind the piston is filled. The crankshaft continues to rotate, requiring the piston to begin a rearward stroke. During this stroke the fluid behind the piston is forced from its chamber into the discharge line via the outlet valve and the chamber in front of the piston refills via its inlet valve [21].

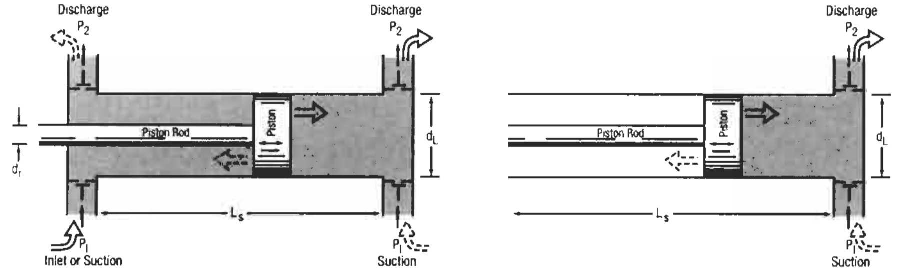
Double-acting pumps are usually available with one or two pistons.

A one-piston double-action pump is called a *double-acting simplex* (since there are older single-action steam and pneumatic driven simplex pumps).

A two piston double-action pump is called a *duplex pump*.

Flow Characteristics

All reciprocating pumps have a pulsating discharge. This is the result of the piston motion as it stops and reverses. At this moment, the flow from that piston theoretically drops to zero. Thus, the discharge curves as a function of time are those illustrated in Figure 3-51. By having two or more pistons the pulsation of the discharge from the pump can be smoothed out and the magnitude of the pulsation reduced if the pistons motions are timed for proper dynamic balancing of the pump (see Figure 3-52). For those pumps that have large pulsations, a cushion change (or accumulator) may be used in the discharge line to reduce or eliminate the pulsations (see Figure 3-53).



(a) Double-acting (duplex) design. (b) Single-acting (triplex) design.

Figure 3-50. Schematic of valve operation of single and double-acting pumps [21].

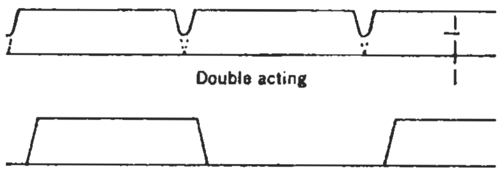


Figure 3-51. Pressure–time curves for single-acting and double-acting pumps (one piston) [21].

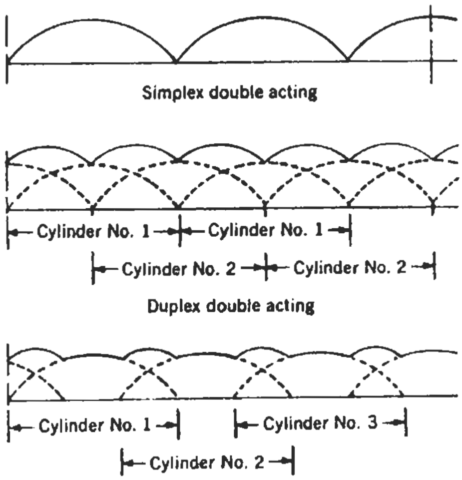


Figure 3-52. Simplex (double-acting), duplex (double-acting), triplex (single-acting), pressure–time curves [21].

Calculations

There are several important calculations that are needed in order to properly evaluate and select the appropriate reciprocating piston pump [17]. These calculations are used in conjunction with Equations 3-32–3-37.

The capacity of a reciprocating piston pump, q (in^3/min), is

$$q = V_d(1-S)N \tag{3-38}$$

where V_d is the displacement of the pumps piston in in.^{3}, S is the slip fraction, and N is the speed of the pump in rpm.

The displacement V_d (gals) of a single-acting pump is

$$V_d = \frac{nA_pL_s}{231} \tag{3-39}$$

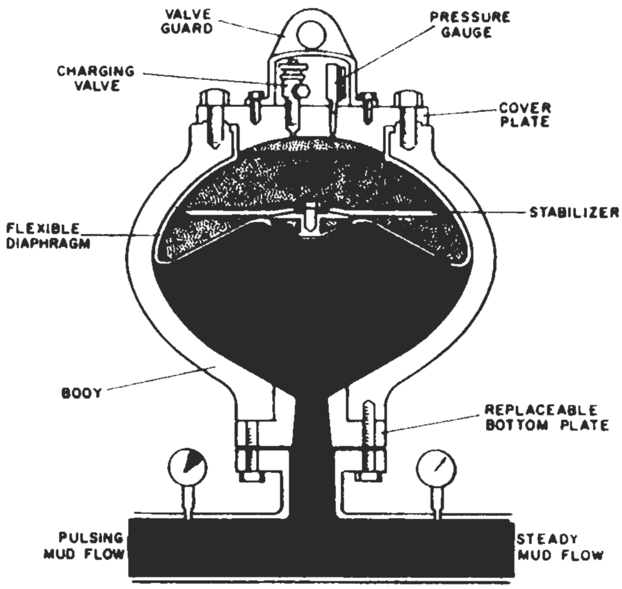


Figure 3-53. Example pulsation dampener [21].

where n is the number of pistons, A_p is the cross-sectioned area of the piston in in.^{2}, and L_s is the length of the piston stroke in in.

The displacement V_d of a double-acting pump is

$$V_d = \frac{n(2A_p - a)L_s}{231} \tag{3-40}$$

where a is the cross-sectioned area of the piston rod in in.^{2}.

The pressure p used in Equation 3-32 is the differential developed pressure (across the pump inlet and outlet). Since the inlet suction pressure is usually small compared to the discharge pressure, the discharge pressure is used. Thus, this is the application resistance pressure in most cases. Figure 3-54 shows a typical reciprocating pump performance.

The slip S is the fraction of suction capacity loss. It consists of the volumetric efficiency loss fraction l_e, the stuffing box loss fraction l_s, and the valve loss fraction l_v. The slip S is

$$S = l_e + l_b + l_v \tag{3-41}$$

The volumetric efficiency loss l_e is

$$l_e = 1 - e_v \tag{3-42}$$

where e_v is the volumetric efficiency. The volumetric efficiency is the ratio of discharge volume to suction volume and is expressed as a percentage. It is proportional to the ratio r and the developed pressure (see Figure 3-55). The ratio r is the ratio of internal volume of fluid between valves when the piston is at the top of its back stroke to the

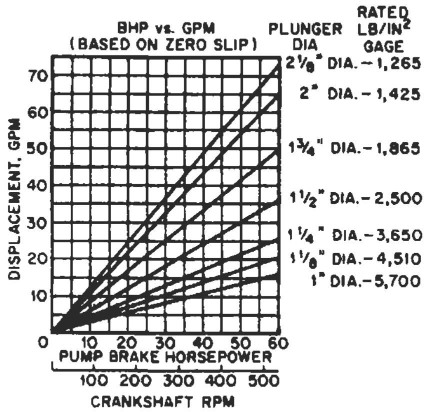


Figure 3-54. Typical power-pump performance (courtesy Ingersoll-Rand Co.).

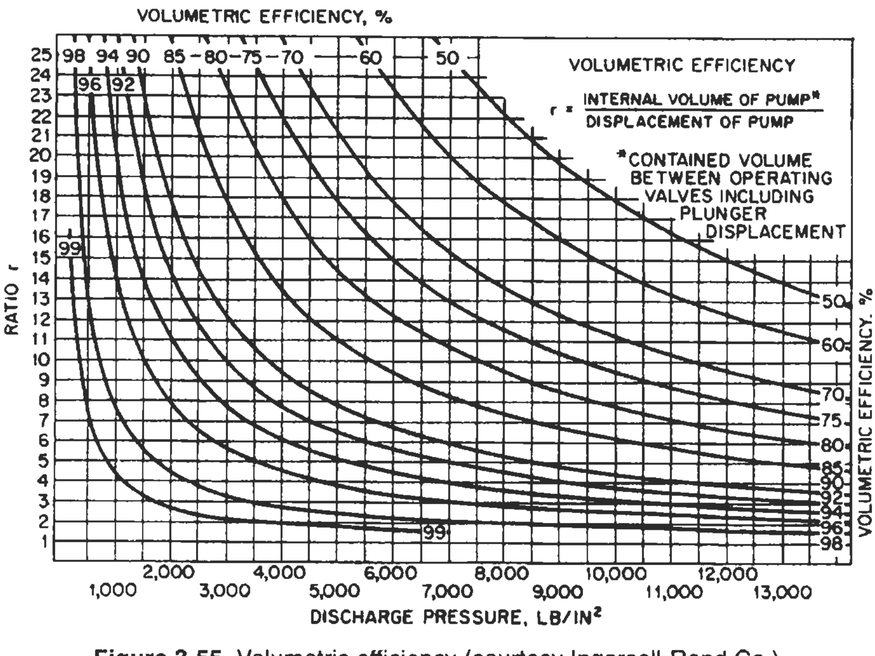


Figure 3-55. Volumetric efficiency (courtesy Ingersoll-Rand Co.).

piston displacement (see Figure 3-56). Because volume cannot readily be measured at discharge pressure, it is taken at suction pressure. This will result in a higher e_v due to fluid compressibility which is neglected.

The stuffing box loss l_b is usually negligible.

The valve loss l_v is the loss due to the flow of fluid back through the valve during closing. This is of the order of 0.02 to 0.10 depending on the valve design.

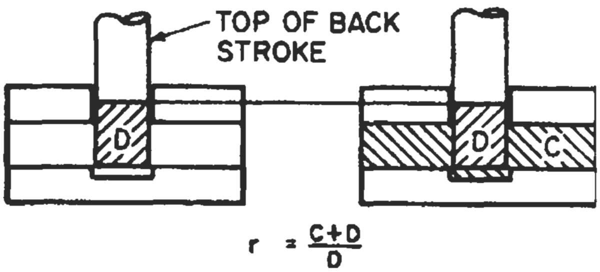


Figure 3-56. Ratio r (courtesy Ingersoll-Rand Co.).

The mechanical efficiency e_m of a power pump at full load of pressure and speed is 0.90 to 0.95. When the pump has a single built in gear as part of the power frame, the mechanical efficiency is 0.80 to 0.95. Table 3-25 shows the typical effects of speed and pressure on the mechanical efficiency of a power pump [17].

Rotary Pumps

Another important positive displacement pump is the rotary pump. This type of pump is usually of rather simple construction, having no valves and being lightweight. Such pumps can be constructed to handle small fluid capacities (i.e., less than a gal/min) to very large fluid capacities (i.e., 5,000 gal/min or greater). Rotary pumps are designed to operate at 1,000 psi discharge pressure, but the normal rotary pump design is for pressure of 25 to about 500 psi with mechanical efficiencies of 0.80 to 0.85.

Table 3-25
Effects of Speed and Pressure on Mechanical Efficiency [17,21,22]

Constant Speed	
% full-load developed pressure	**Mechanical efficiency**
20	82
40	88
60	90.5
80	92
100	92.5
Constant Developed Pressure	
% speed	**Mechanical efficiency**
44	93.3
50	92.5
73	92.5
100	92.5

Rotary pumps are classified in two basic groups and several subgroups. These are:

Single-rotor
- -vane
- -piston
- -flexible vane
- -screw

Multiple-rotor
- -lobe
- -gear
- -circumferential piston
- -screw

These pumps require the maintenance of very close clearances between rubbing surfaces for continual volumetric efficiency. Some of the important pumps are discussed.

In general, rotary pumps with discharge pressure of up to 100 psi are considered low-pressure pumps. Rotary pumps with pressure between 100 and 500 psi are considered moderate-pressure pumps. Pumps with pressure beyond 500 psi are considered to be high-pressure pumps.

Rotary pumps with volume capacities up to 50 gal/min are considered to be small-volume capacity pumps. Pumps with volume capacities from 50 to 500 gal/min are moderate-volume capacity pumps. And pumps with volume capacities beyond 500 gal/min are large-volume capacity pumps.

Gear Pumps

Gear pumps are rotary pumps in which two or more gears mesh to provide the pumping action. Usually one of the gears is capable of driving the other(s). The simplest form of this type of rotary pump is the gerotor pump. Figure 3-57 shows a typical gerotor pump configuration. These pumps are often used for small-volume capacity applications where space is quite limited [18]. Such gear pumps can be used on a rotating shaft such as a crankshaft to provide oil lubrication of critical moving machine parts.

Figure 3-58 shows an external gear pump. This is an example of a double- (multiple) rotor pump. Such a gear pump has a drive gear and a driven gear that are encased in housing with minimum clearance between the housing and the tips of the gears. The

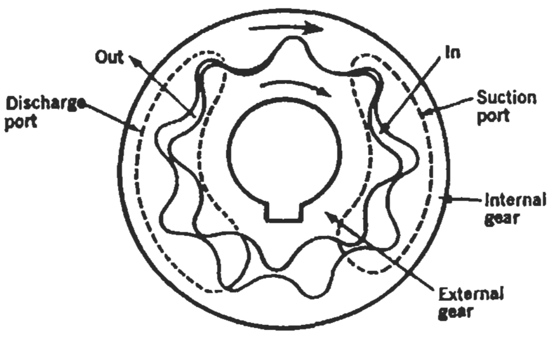


Figure 3-57. Gerotor type pump [17].

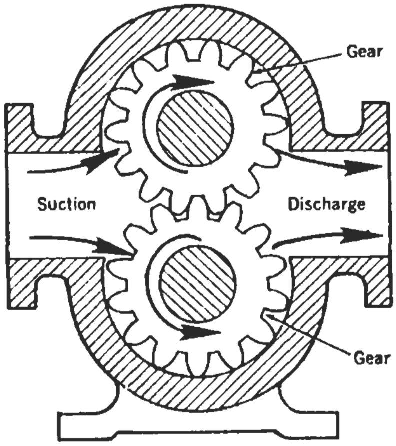


Figure 3-58. Gear pump (double rotor type) [17].

simplest type of this pump uses spur gears. The large number of gear teeth in contact with the casing minimizes leakage around the periphery. The spur-gear type of pump is limited by its characteristics of trapping liquid. This occurs on the discharge side at the point of gear intermesh. This results in a noisy operation and low mechanical efficiency, particularly at high rotary speeds. Gear pumps can also be constructed with single or double-helical gears with 15–30° angles (0.26–0.52 rad). Such a helical (or even a herringbone) gear construction will nearly eliminate the problems of liquid trapping but increase the leakage.

One-Tooth Difference. These pumps are commonly known as gerotor pumps (see Figure 3-57). An outlet gear is mounted eccentrically with the outlet casing actuated by an internal gear rotating under the action of the central shaft (keyway connected). The internal gear has one less tooth than the outside ring gear. There is clearance between the outside impeller gear and its casing and the inner gear fixed to the shaft. As the shaft rotates, the inner gear forces the outer gear to progress around the shaft at a rate slower than the shaft rotation. The liquid trapped between the two gears is forced from the space between the two gears as the rotation takes place.

Two-Teeth Difference. In this type of pump an abutment on one side plate is used to fill the clearance between the external and internal gear. This construction reduces leakage but involves the use of an overhung internal gear. Such a gear arrangement limits the application of these pumps to small and moderate-volume capacity pumps.

Circumferential Piston

Figure 3-59 shows the typical cross-section of this type of pump. In this pump liquid is pumped by the action of the rotation of the two eccentrically located piston surfaces. There is no contact between the piston surfaces.

Vane Pump

This example of a single-rotor type pump makes use of an eccentric shaft on which are mounted several sliding vanes (see Figure 3-60). The vanes are forced against the

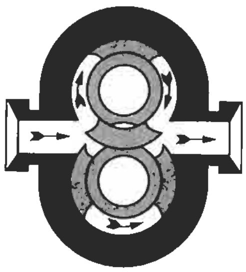


Figure 3-59. Circumferential-piston pump (double rotor) [17].

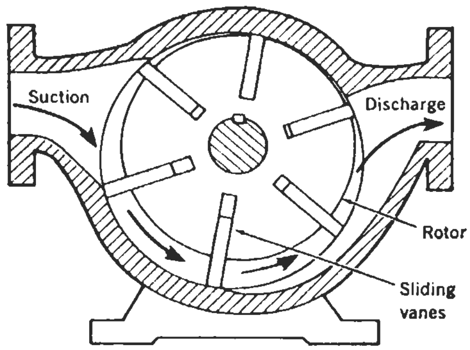


Figure 3-60. Sliding vane pump (single rotor) [17].

bore of the housing by springs and the centrifugal force of the rotor rotation. The vanes are usually made of materials that will not damage the surface of the bore (e.g., bronze and bakelite). This type of pump is useful for small and moderate-volume capacities and low pressure. This is due to the rather low speeds such pumps must be operated at. High speeds result in rapid wear of the vanes.

Screw Pumps

Screw pumps are available in single or double-rotor (multiple) types.

Figure 3-61 shows the cross-section of the typical single-rotor screw pump. In this pump there is a single helical threaded rotating element. As the screw rotates the liquid progresses axially down the pump. This pump produces continuous flow with relatively little pulsation or agitation of the fluid. Screw pumps of this type are quiet and efficient. They are available with high-speed, high-pressure, and large-volume capacities.

Figure 3-62 shows the cross-section of the typical multiple-rotor screw pump. This screw pump incorporates right-hand and left-hand intermeshing helices on parallel shafts with timing gears. Such pumps are available with high-speed, high-pressure, and large-volume capacities.

For all screw pumps, flow is continuous.

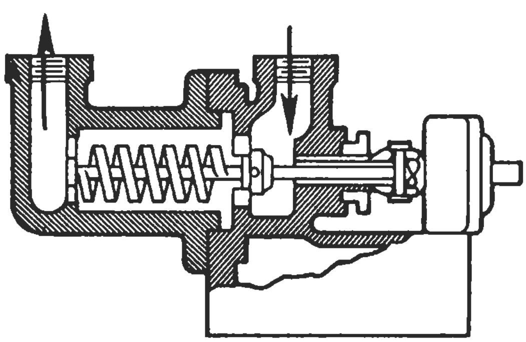


Figure 3-61. Single screen pump (single rotor) [17].

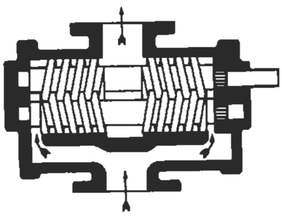


Figure 3-62. Two-screw pump (double rotor) [17].

Calculations

There are several important calculations which are needed to properly evaluate and select the appropriate rotary pump [17]. These calculations are used in conjunction with Equations 3-32–3-37.

The capacity of a rotary pump operating with zero slip is called displacement capacity, q_d (gpm). Thus, the actual capacity of a rotary pump q (gpm) is

$$q = q_d - q_l \tag{3-43}$$

or

$$q = \frac{V_d N}{231} - q_l \tag{3-44}$$

where V_d is the pump displacement per revolution in in.^{3}, N is the pump speed in rpm, and q_l is the pump loss in gpm.

Pump horsepower P is the power actually imparted to the fluid by the rotary pump in Equation 3-32.

The definitions for the various power terms and efficiencies for the rotary pumps are the same as those discussed above for the reciprocating pump; namely, Equations 3-33–3-37.

The volumetric efficiency e_v for the rotary pump is

$$e_v = \frac{q}{q_d} \tag{3-45}$$

or

$$e_v = 1 - \frac{231 q_e}{V_d N} \tag{3-46}$$

(usually expressed as a percentage).

Centrifugal Pumps

The centrifugal pump is the most important pump of the dynamic class of pumps for the oil and gas industry. Other pumps in this class are covered in other references [17,18].

In its simplest form, a centrifugal pump consists of a rotating impeller (with radical vanes) rotating at a rather high speed. The rotating impeller is encased in a rigid housing that directs the liquid within the pump (see Figure 3-63) [17]. Liquid is supplied to the inlet that feeds the liquid to the center section of the rotating impeller. The rotational motion of the impeller forces the liquid, via the centrifugal forces, to move radically outward with the aid of the stationary diffuser. The rigid housing around the impeller guides the high-velocity fluid around the inside of the housing and out of the outlet of the pump.

The capacity of this type of pump depends on the pressure head against which the pump must act (see Figure 3-49).

When the liquid within the impeller is forced radially outward to the diffuser, a major portion of the velocity energy is converted into pressure energy by the stationary diffuser vanes (see Figure 3-63). This can also be accomplished by means of a volute, which is a part of the casing design (see Figure 3-64) [17].

Centrifugal Pump Classifications

Centrifugal pumps with diffusion vanes are called *diffusion pumps* or, more recently, *vertical turbine pumps*. Those pumps with volute casings are called *volute pumps*.

Centrifugal pumps can also be classified by the design of the impeller. Centrifugal pumps may have radial-flow impellers, axial-flow impellers, and mixed-flow impellers (both radial-flow and axial-flow).

Pump impellers are further classified as to the inlet flow arrangement such as single suction (which has a single inlet on one side) and double suction (which has a double inlet on each side of the impeller).

Impellers can be further classified with regard to their physical design: a closed impeller has shrouds or sidewalls enclosing the fluid flow, an open impeller has no shrouds or sidewalls, and a semiopen impeller is a mix of the closed and open design.

Another centrifugal pump classification is whether the pump is a single-stage pump (the pressure head is developed by a single impeller) or a multistage pump (the pressure head is developed by two or more impellers).

Centrifugal pumps can be further classified by physical design or axially split, radially split and whether the axis of rotation of the impeller(s) is vertical or horizontal.

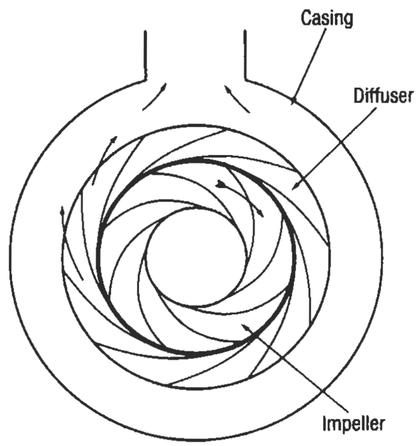


Figure 3-63. Typical diffuser-type pump [17].

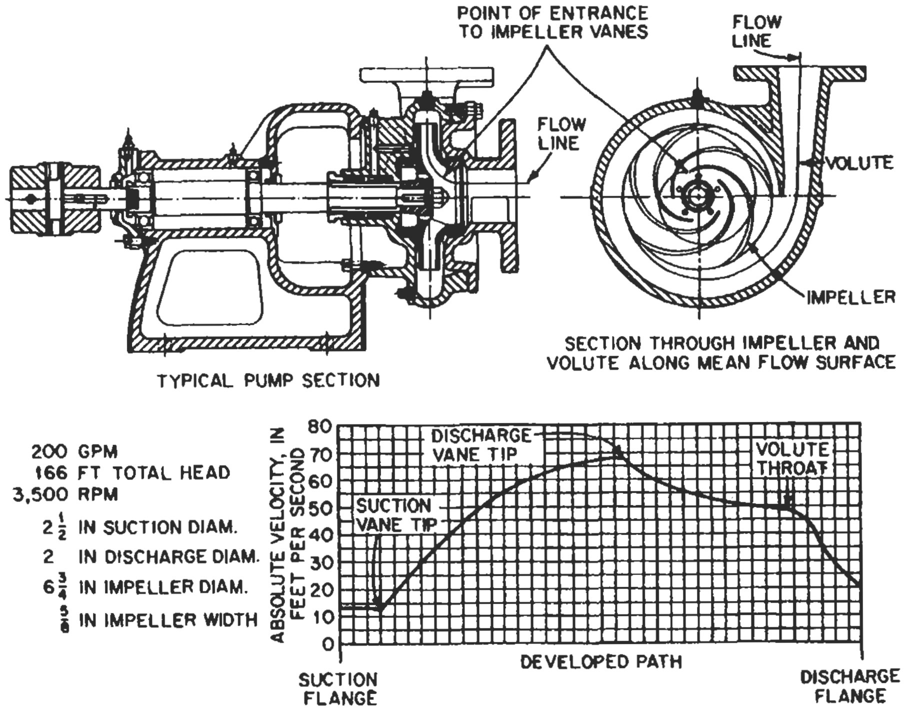


Figure 3-64. Typical volute-type pump [17].

Horizontal pumps can be classified according as end suction, side suction, bottom suction, and top suction.

In applications, centrifugal pumps can be supplied with liquid via piping, or the pump may be submerged. Vertical pumps are called dry-pit or wet-pit types. The wet-pit pump (submerged) discharges up through a pipe system to some point above the pump.

Calculations

There are important calculations that are needed to properly evaluate and select the appropriate centrifugal pump [20].

There are several similar relationships for centrifugal pumps that can be used if the effects of viscosity of the pumped fluid can be neglected. These relate the operating performance of any centrifugal pump for one set of operating conditions to those of another set of operating conditions, say conditions, and conditions 2.

The volumetric flow rate q (gpm) is related to rate q_2 (gpm) and the impeller speeds N_1 (rpm) and N_2 (rpm) by

$$\frac{q_1}{q_2} = \frac{N_1}{N_2} \tag{3-47}$$

The pressure head H_1 (ft) is related to the head H_2 (ft) and impeller speeds N_1 and N_2 by

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^{1/2} \tag{3-48}$$

The pump input power P_1 (Lp) is related to the pump input power P_2 (hp) and impeller speed N_1 and N_2 by

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^{3} \tag{3-49}$$

For a constant impeller speed, the relationship between q_1, q_2, and impeller diameter D_1 (ft) and D_2 (ft) is

$$\frac{q_1}{q_2} = \frac{D_1}{D_2} \tag{3-50}$$

The heads are related by

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^{2} \tag{3-51}$$

and the power related by

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^{3} \tag{3-52}$$

The centrifugal pump specific speed N_{ds} (rpm) (or discharge specific speed) is

$$N_{ds} = \frac{Nq^{0.5}}{H^{0.75}} \tag{3-53}$$

The centrifugal pump suction specific speed N_{ss} (rpm) is

$$N_{ss} = \frac{Nq^{0.5}}{H_{nps}^{0.75}} \tag{3-54}$$

where H_{nps} is the net positive suction head (ft).

Figure 3-65 gives the upper limits of specific speeds of single-stage, single and double-suction centrifugal pumps handling clear water at 85°F at sea level [22].

Figure 3-66 gives the upper limits of specific speeds of single-suction mixed-flow and axial-flow centrifugal pumps handling clear water at 85°F at sea level [22].

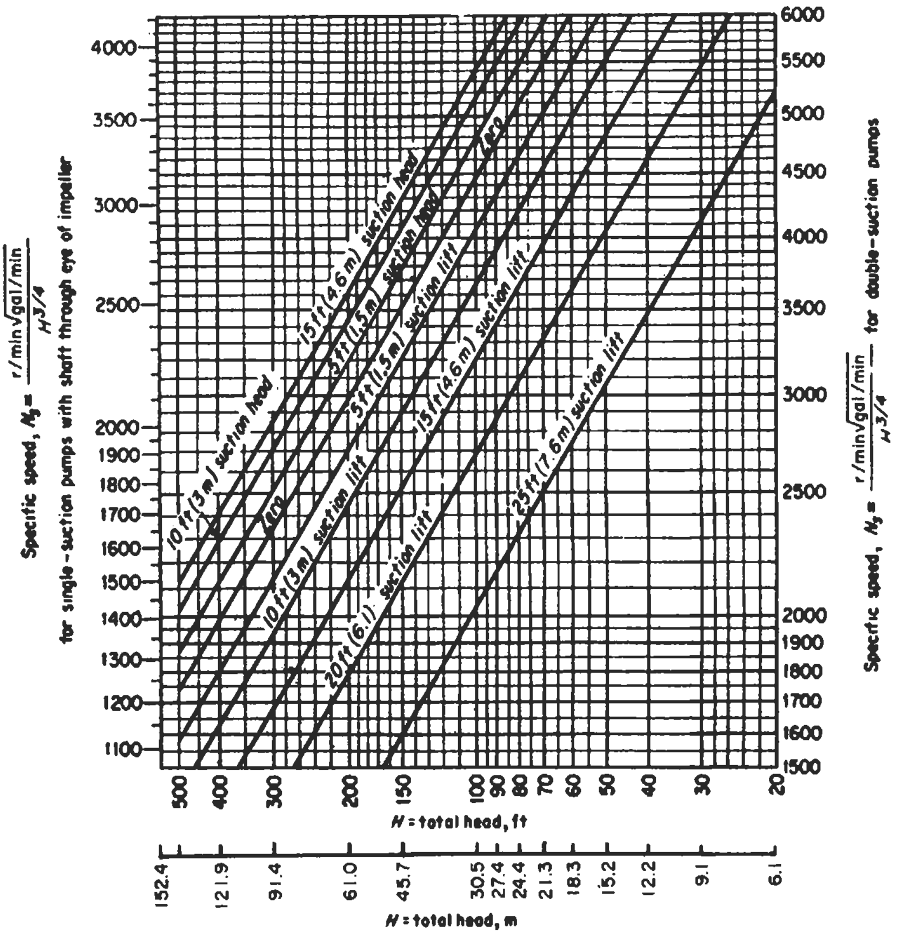


Figure 3-65. Upper limits of specific speeds of single-stage, single and double-suction centrifugal pumps handling clear water at 85°F at sea level (courtesy Hydraulic Institute).

Figure 3-67 gives the approximate relative impeller shapes and efficiency variations for various specific speeds of centrifugal pumps.

Table 3-26 gives the specific speeds for various centrifugal pump types. Table 3-27 gives the suction specific speed ratings for single-suction and double-suction centrifugal pumps. These tables are for pumps handling clear water.

COMPRESSORS

Compressors of various designs and manufacturers are used in many operations throughout the oil and gas industry. Compressors are used in some drilling operations, in many production operations, and extensively used in surface transportation of oil and gas via pipelines.

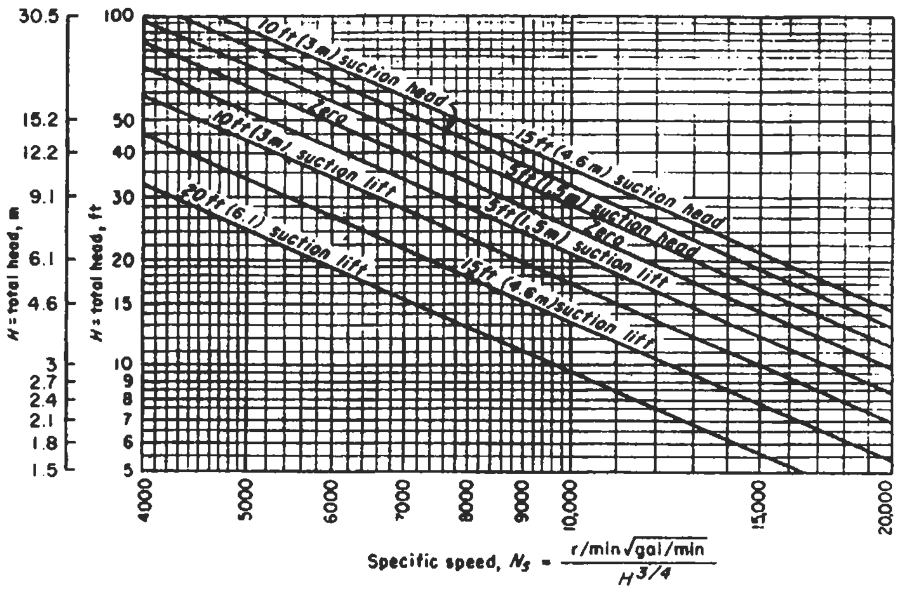


Figure 3-66. Upper limits of specific speeds of single-suction, mixed-flow and axial-flow pumps handling clear water at 85°F at sea level (courtesy Hydraulic Institute).

Air or gas compressors are very similar in design and operation to liquid pumps discussed earlier. The air and gas compressor is a mover of compressed fluids; the pumps are movers of basically incompressible fluids (i.e., liquids).

Classifications

In much the same manner as pumps, compressors are classified as one of two general classes: positive displacement or dynamic (see Figure 3-68) [23]. These two general classes of compressors are the same as that for pumps. The positive displacement class of compressors is an intermittent flow device, which is usually a reciprocating piston compressor or a rotary compressor (e.g., sliding vane, screws, etc.). The dynamic class of compressor is a continuous flow device, which is usually an axial-flow or centrifugal compressor (or mix of the two).

Each of the two general classes of compressors and their subclass types have certain advantages and disadvantages regarding their respective volumetric flow capabilities and the pressure ratios they can obtain. Figure 3-69 shows the typical application range in volumetric flowrate (actual cfm) and pressure ratio several of the most important compressor types can obtain [23].

In general, positive displacement compressors are best suited for handling high-pressure ratios (i.e., about 200), but with only moderate volumetric flowrates (i.e., up to about 10^3 actual cfm). Dynamic compressors are best suited for handling large volumetric flowrates (i.e., up to about 10^6 actual cfm), but with only moderate pressure ratios (i.e., about 20).

This is not the complete view of these two compressor classes. Figure 3-70 gives the general performance curves for various positive displacement and dynamic compressors. The positive displacement compressors, particularly the multistage

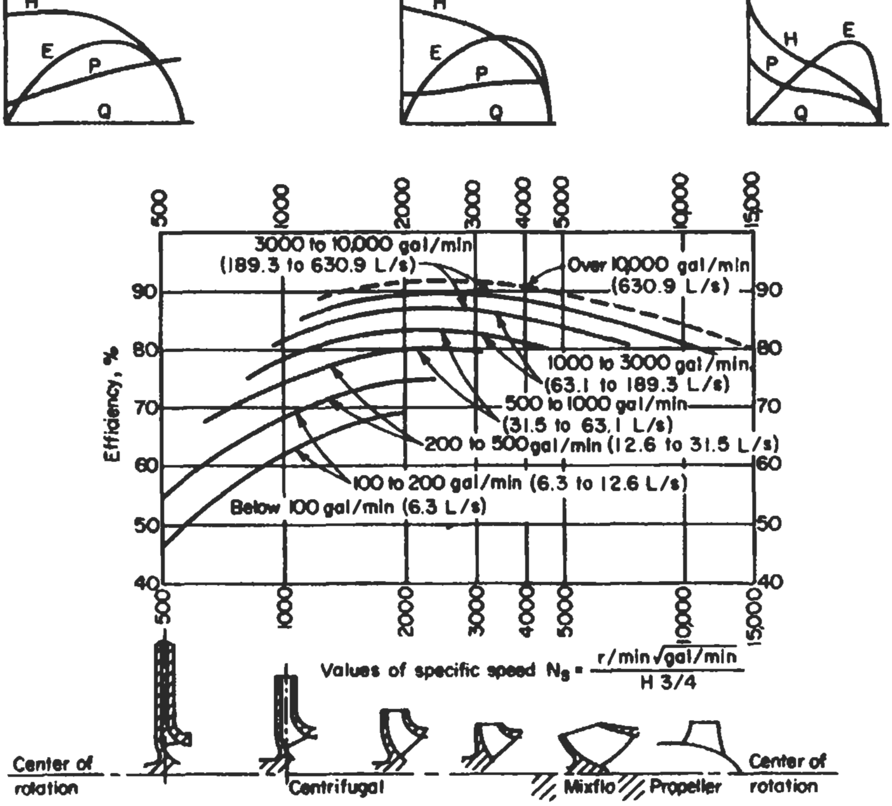


Figure 3-67. Approximate relative impeller shapes and efficiency variations for various specific speeds of centrifugal pumps (courtesy Worthington Corporation).

Table 3-26
Pump Types Listed by Specific Speed

Specific speed range	Type of pump
Below 2,000	Volute, diffuser
2,000–5,000	Turbine
4,000–10,000	Mixed-flow
9,000–15,000	Axial-flow

Courtesy of FMC Corporation.

reciprocating compressors, are very insensitive to pressure ratio changes. These compressors will produce their rated volumetric flowrate even when the pressure ratio approaches the design limit of the machine. This is less so for the rotary compressor. The dynamic compressors, however, are quite sensitive to pressure ratio changes. The volumetric rate of flow will change drastically with changes in the pressure ratio around the pressure ratio the machine has been designed.

Table 3-27
Suction Specific-Speed Ratings

Single-suction pump	Double-suction pump	Rating
Above 11,000	Above 14,000	Excellent
9,000–11,000	11,000–14,000	Good
7,000–9,000	9,000–11,000	Average
5,000–7,000	7,000–9,000	Poor
Below 5,000	Below 7,000	Very poor

Courtesy of FMC Corporation.

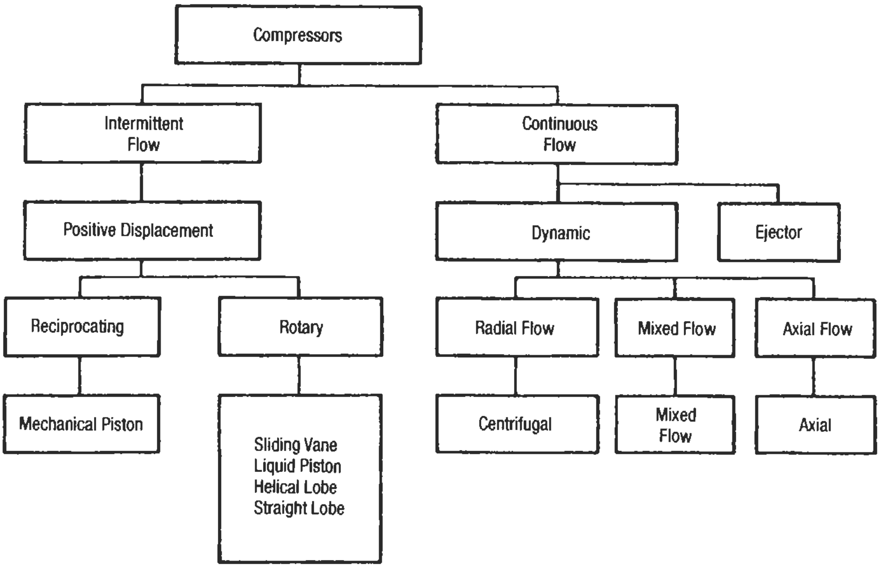


Figure 3-68. Classification of compressor [23].

Thus, the positive displacement compressors are normally applied to industrial operations where the volumetric flowrate is critical and the pressure ratio is a variable. The dynamic compressors are generally applied to industrial operations where both the volumetric flowrate and pressure ratio requirements are relatively constant.

In general, only the reciprocating compressor allows for reliable flexibility in applying variable volumetric flowrate and variable pressure ratio in an operation. The rotary compressor does not allow for variation in either (except that of pressure through the decompression of the air or gas if the system back pressure is below the design pressure of the machine). The dynamic compressors are designed for specific volumetric flowrates and pressure ratios and are not very useful when these design limits are altered.

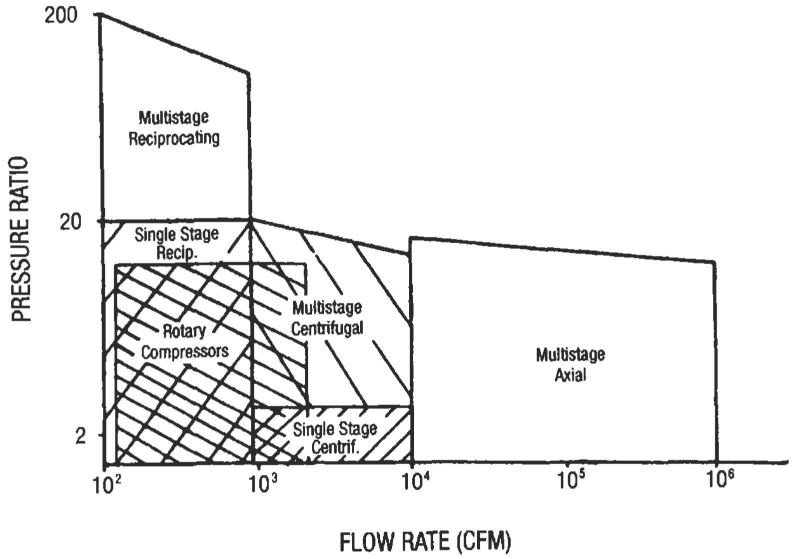


Figure 3-69. Typical application ranges of various compressor types [23].

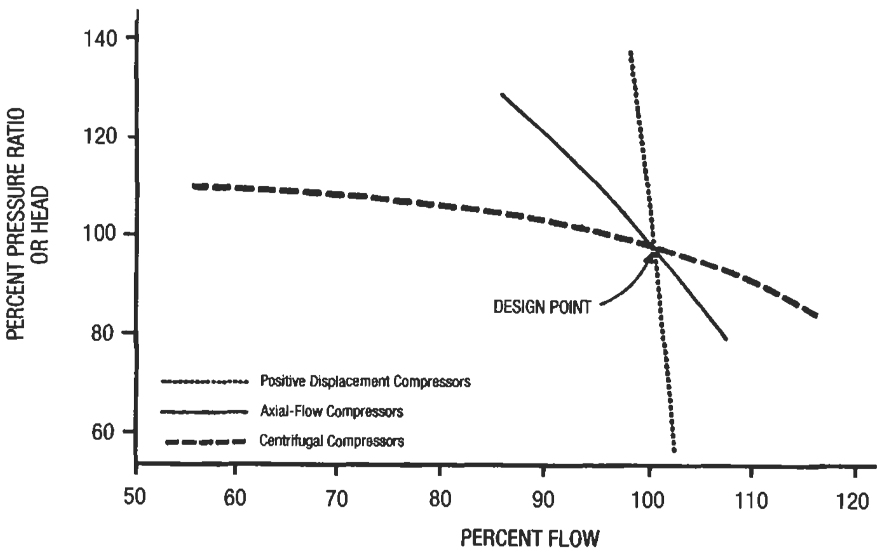


Figure 3-70. General performance curve for various compressor types [23].

Standard Units

In the United States the unit of air or any gas is referenced to the standard cubic foot of dry air. The API Mechanical Equipment Standards standard atmospheric conditions for dry air is fixed at a temperature of 60°F (which is 459.67 + 60 + 519.67°R) and a pressure of 14.696 psia (760 mm, or 29.92 in. Hg). The equation of state for the perfect gas can be written as

$$\frac{P}{\gamma} = \frac{\mathbf{R}T}{m_w} \tag{3-55}$$

where P is the pressure in lb/ft^2 absolute, γ is the specific weight in lb/ft^3, **R** is the universal gas constant in 1,545.4 ft-lb/lb-mole-°R, T is the temperature in °R, and m_w is the molecular weight of the gas in lb/lb-mole.

Thus, the specific weight γ or the weight of 1 ft^3 of dry air will be

$$\gamma = \frac{14.696(144)(28.96)}{1{,}545.4(519.67)} = 0.0763\,\text{lb/ft}^3$$

where for dry air m_w = 28.96 lb/lb-mole. Thus, a dry cubic foot of air at the API Mechanical Equipment Standards standard atmosphere condition weighs 0.0763 lb and has a specific gravity of 1.000 [5,23].

There are other organizations within the United States and regions around the world that have established different standards. The ASME standard atmosphere is at a temperature of 68°F, a pressure of 14.7 psia, and a relative humidity of 36%. The British use a standard atmosphere with a temperature of 60°F and a pressure of 30.00 in.Hg. The Europeans use a standard atmosphere with a temperature of 15°C (59°F) and pressure of 750 mmHg (14.5 psia) [24,25].

When selecting and sizing compressors, care should be taken in determining which standard has been used to rate a compressor under consideration, particularly if the compressor has been produced abroad. All further discussions in this section will utilize only the API Mechanical Equipment Standards standard atmosphere.

Compressors are rated as to their maximum volumetric flowrate they can operate at, and the maximum pressure they can maintain. These ratings are usually specified as *standard cubic feet per minute* (scfm) and psig. The scfm of volumetric flow rate refers to the compressor intake. The pressure rating refers to the output pressure capability.

General Calculations

There are several important calculations needed to properly evaluate and select the appropriate compressors [23–25].

To optimize the compressor design to minimize the overall power consumption, the compressor should have nearly equal ratios of compression in each of the stages. Thus, the total pressure ratio r_t across the compressor (i.e., input pressure to output pressure prior to after cooling) is

$$r_t = \frac{p_{out}}{p_{in}} \tag{3-56}$$

where p_{out} is the output pressure in psia, p_{in} is the input pressure in psia, and the pressure ratio for each stage r_s is

$$r_s = r_t^{1/n} \tag{3-57}$$

where n is the number of nearly equal compression stages in the compressor.

Thus, for a four-state compressor, if $p_1 = p_{in}$, then p_2 would be the pressure exiting the first stage compression. The pressure p_2 would be

$$p_2 = r_s p_1 \tag{3-58}$$

The pressure entering the second stage would be p_2 and the pressure exiting would be p_3, which is

$$p_3 = r_s p_2 \tag{3-59}$$

or

$$p_3 = r_s^2 p_1 \tag{3-60}$$

The pressure entering the third stage would p_3 and the pressure exiting would be p_4, which is

$$p_4 = r_s p_3 \tag{3-61}$$

or

$$p_4 = r_3^3 p_1 \tag{3-62}$$

The pressure entering the fourth stage would be p_4 and the pressure exiting would be p_5, which is

$$p_5 = r_s p_4 \tag{3-63}$$

or

$$p_5 = r_5^4 p_1 \tag{3-64}$$

The above pressure calculations assume intercooling between the first and second stages of compression, between the second and third stages of compression, and between the third and fourth stages of compression. The intercooling system in a multistage compressor ideally reduced the temperature of the gas leaving first stage (or the other stages) to ambient temperature, or at least the input temperature. This cooling of the gas moving from one stage to another is necessary for an efficient and economical design of the compressors. Further, if the gas moving to progressive stages were to get too hot, the machine could be severely damaged. This intercooling is normally accomplished using either a water-jacket, oil-jacket or air-cooled finned pipes between the stages.

Assuming efficient intercooling between stages, then $T_1 = T_2 = T_3$. The temperature T_4 is

$$T_4 = T_1 \frac{p_4^{k-1/k}}{p_3} \tag{3-65}$$

where k is ratio of specific heats.

The temperature T_y at the exit of the compressor is cooled with an adjustable cooling system called an aftercooler. Such a system is useful in adjusting the output air for specific application purposes. The after cooling process is assumed to be a constant pressure process. The volumetric flowrate from the fourth stage is q_4 (cfm), then the volumetric flowrate after the flow or passed through the after cooler q'_4 (cfm) will be

$$q'_4 = q_4 \frac{T_4}{T'_4} \tag{3-66}$$

where T'_4 is the final temperature the after-cooler is to cool the output (°R).

If the compressor has a capability to compress a volumetric flowrate of q_{in} (scfm), then for the multistage compressor

$$q_1 = q_{in} \tag{3-67}$$

This compressor design capability is usually stated in scfm, which means that this is the capability of the compressor when at a sea level location. If the compressor is moved to a higher elevation, then the compressor will have to be derated and the scfm capability of the machine reduced, or the actual cfm term used to describe capability of the compressor. This derating can be carried using Table 3.28.

If a compressor is rated as having a volumetric flow capability of q_{in} (scfm) and is to be used at surface location of 6,000 ft, then the compressor still has the capability of its rated volumetric flow rate, but it is now written as q_{in} (actual cfm). Thus the actual weight rate of flow through the compressor $\dot{w}$(lb/min), will be (see Table 3-28)

$$\dot{w} = 0.0639 q_{in} \tag{3-68}$$

The equivalent derated standard volumetric flowrate $q_{in\,dr}$ (scfm) is

$$q_{in\,dr} = \frac{\dot{w}}{0.0763} \tag{3-69}$$

The theoretical power **P** (hp), which is actual power imparted to the gas by the multistage compressor, is approximated as

Table 3-28
Atmosphere at Elevation (Mid-Latitudes) above Sea Level

Surface Location Above Sea Level (ft)	Pressure (psi)	Temperature (°F)	Specific Weight (lb/ft³)
0	14.696	60.00	0.0763
2,000	13.662	51.87	0.0721
4,000	12.685	44.74	0.0679
6,000	11.769	37.60	0.0639
8,000	10.911	30.47	0.0601
10,000	10.108	23.36	0.0565

$$P = n_s \frac{\dot{w}RT_{in}}{33{,}000m_w}\left(\frac{k}{k-1}\right)r_t^{k-1/kn} - 1 \tag{3-70}$$

where T_{in} is the actual average temperature of the input air (or gas) (°R).

The total compressor power input P_{ci} is the net power delivered to the compressor drive shaft by the prime mover system. This is

$$\mathbf{P}_{ci} = \mathbf{P} + \mathbf{P}_{ce} \tag{3-71}$$

where $\mathbf{P}_{ce}$ is the compressor power loss.

The total power input $\mathbf{P}_{ti}$ is the net power delivered by the prime mover. This is

$$\mathbf{P}_{ti} = \mathbf{P}_{ci} + \mathbf{P}_{pml} \tag{3-72}$$

where $\mathbf{P}_{pml}$ is the prime mover loss.

Overall efficiency e_o is

$$e_c = \frac{\mathbf{P}}{\mathbf{P}_{ti}} \tag{3-73}$$

usually multiplied by 100 and expressed as a percentage.

Compressor efficiency e_c is

$$e_c = \frac{\mathbf{P}}{\mathbf{P}_{ci}} \tag{3-74}$$

usually expressed as a percentage.

Mechanical efficiency e_m (which usually refers to the efficiency of the prime mover to supply power to the compressor) is

$$e_m = \frac{\mathbf{P}_{ci}}{\mathbf{P}_{ti}} \tag{3-75}$$

usually expressed as a percentage.

Reciprocating Compressors

The reciprocating compressor is the simplest example of the positive displacement class of compressors. This type of compressor is also the oldest. Like reciprocating pumps, reciprocating compressors can also be either single acting or double acting. Single-acting compressors are usually of the trunk type (see Figure 3-71). Double-acting compressors are usually of the crosshead type (see Figure 3-72) [4,25].

Reciprocating compressors are available in both lubricated and nonlubricated versions. The lubricated versions provide lubrication for the piston, moving pistons either through an oil lubricated intake air or gas stream, or via an oil pump and injection of oil to the piston sleeve. There are some applications where oil must be completely omitted from the compressed air or gas exiting the machine. For such applications where a reciprocating piston type of compressor is required, there are nonlubricated compressors. These compressors have piston rings and wear bands

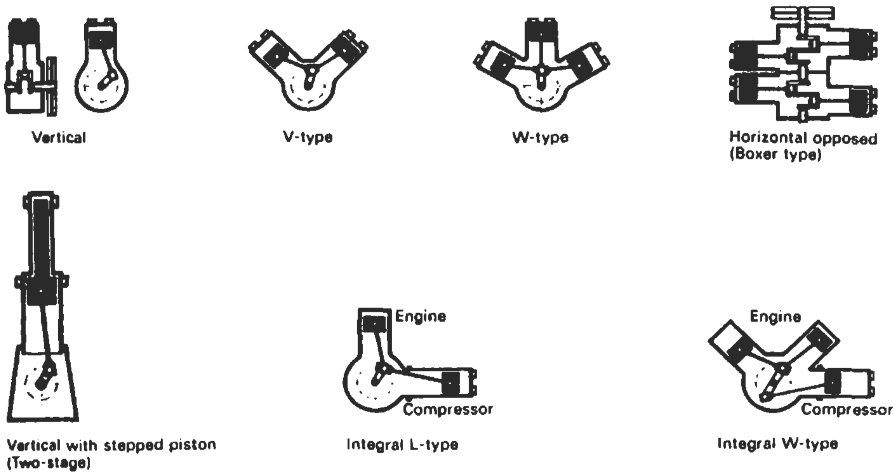


Figure 3-71. Single-acting (trunk-type) reciprocating piston compressor [23].

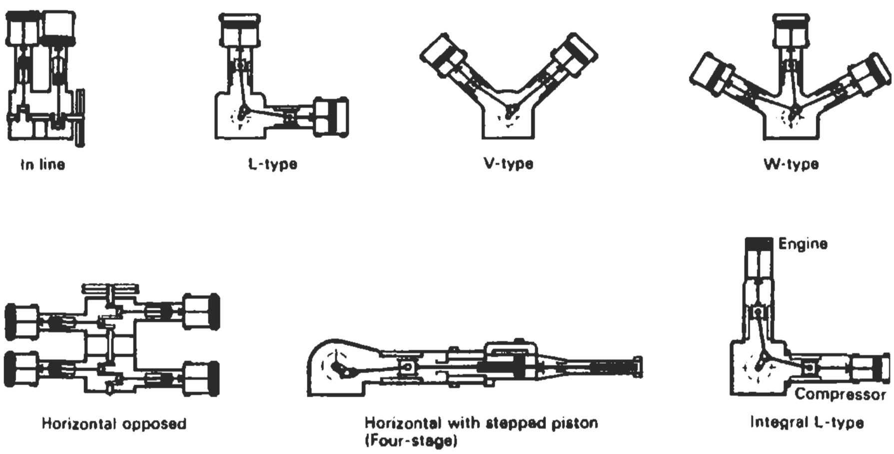


Figure 3-72. Double-acting (crosshead-type) reciprocating piston compressor [4].

around the periphery of each piston. These wear bands are made of special wear-resistant dry lubricating materials such as polytetrafluorethylene. Trunk type nonlubricated compressors have dry crankcases with permanently lubricated bearings. Crosshead type compressors usually have lengthened piston rods to ensure that no oil wet parts enter the compression space [4,25].

Most reciprocating compressors have inlet and outlet valves (on the piston heads) that are actuated by a pressure difference. These are called *self-acting valves.* There are some larger multistage reciprocating piston compressors that do have camshaft-controlled valves with rotary slide valves.

The main advantage of the multistage reciprocating piston compressor is that there is nearly total positive control of the volumetric flowrate which can be put through the machine and the pressure of the output. Many reciprocative piston compressors allow for the rotation to be adjusted, thus, changing the throughput of air or gas. Also, providing adequate power from the prime mover, reciprocating piston compressors will automatically adjust to back pressure changes and maintain proper rotation speed. These compressors are capable of extremely high output pressure (see Figure 3-68).

The main disadvantages to multistage reciprocating piston compressors is that they cannot be practically constructed in machines capable of volumetric flowrates much beyond 1,000 actual cfm. Also, the higher-capacity compressors are rather large and bulky and generally require more maintenance than similar capacity rotary compressors.

In a compressor, like a liquid pump, the real volume flowrate is smaller than the displacement volume. This is due to several factors. These are:

- pressure drop on the suction side
- heating up of the intake air
- internal and external leakage
- expansion of the gas trapped in the clearance volume (reciprocating piston compressors only)

The first three factors are present in compressors, but they are small and on the whole can be neglected. The clearance volume problem, however, is unique to reciprocating piston compressors. The volumetric efficiency e_v estimates the effect of clearance. The volumetric efficiency can be approximated as

$$e_v = 0.96\left[1 - \varepsilon\left(r_t^{1/k} - 1\right)\right] \tag{3-76}$$

where $\varepsilon = 0.04–0.12$. Figure 3-73 gives values of the term in the brackets for various values of ε and the r_t.

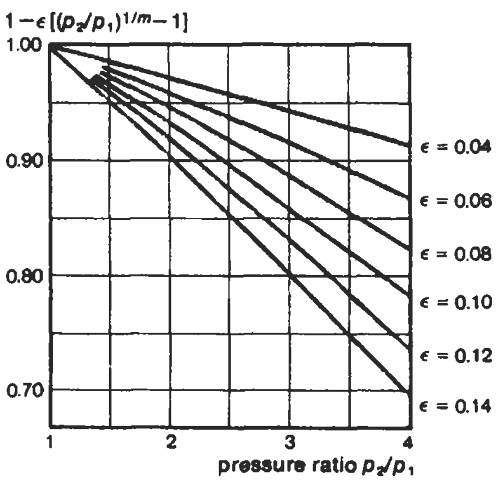


Figure 3-73. Volumetric efficiency for reciprocating piston compressors (with clearance) [4].

For a reciprocating piston compressor, Equation 3-70 becomes

$$P = n\frac{\dot{w}RT_{in}e_v}{33,000m_w}\left(\frac{k}{k-1}\right)\left(r_t^{k-1/kn}-1\right) \tag{3-77}$$

Rotary Compressors

Another important positive displacement compressor is the rotary compressor. This type of compressor is usually of rather simple construction, having no valves and being lightweight. These compressors are constructed to handle volumetric flowrates up to around 2,000 actual cfm and pressure ratios up to around 15 (see Figure 3-69). Rotary compressors are available in a variety of designs. The most widely used rotary compressors are sliding vane, rotary screw, rotary lobe, and liquid-piston.

The most important characteristic of this type of compressors is that all have a fixed built-in pressure compression ratio for each stage of compression (as well as a fixed built-in volume displacement) [25]. Thus, at a given rate of rotational speed provided by the prime mover, there will be a predetermined volumetric flowrate through the compressor, and the pressure exiting the machine at the outlet will be equal to the design pressure ratio times the inlet pressure.

If the back pressure on the outlet side of the compressor is below the fixed output pressure, the compressed gas will simply expand in an expansion tank or in the initial portion of the pipeline attached to the outlet side of the compressor. Figure 3-74 shows the pressure versus volume plot for a typical rotary compressor operating against a back pressure below the design pressure of the compressor.

If the back pressure on the outlet side of the compressor is equal to the fixed output pressure, then there is no expansion of the output gas in the initial portion of the expansion tank or the initial portion of the pipeline.

Figure 3-75 shows the pressure versus volume plot for a typical rotary compressor operating against a back pressure equal to the design's pressure of the compressor.

If the back pressure in the outlet side of the compressor is above the fixed output pressure, then the compressor must match this higher pressure at the outlet. In so doing the compressor cannot expel the compressed volume within the compressor efficiently. Thus, the fixed volumetric flowrate (at a given rotation speed) will be reduced from what it would be if the back pressure were equal to or less than the fixed output pressure. Figure 3-76 shows the pressure versus volume plot for a typical rotary compressor operating against a back pressure greater than the design pressure of the compressor.

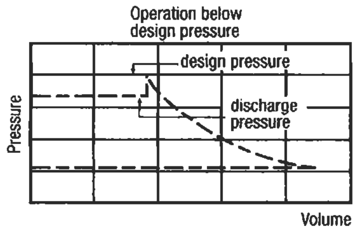


Figure 3-74. Rotary compressor with back pressure less than fixed pressure output [4].

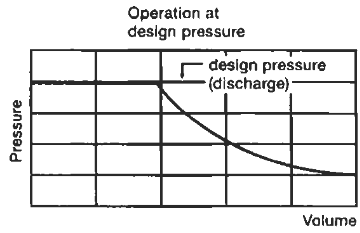


Figure 3-75. Rotary compressor with back pressure equal to fixed pressure output [4].

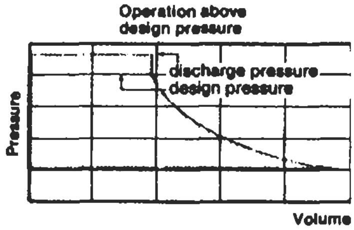


Figure 3-76. Rotary compressor with back pressure greater than fixed pressure output.

Nearly all rotary compressors can be designed with multiple stages. Such multistage compressors are designed with nearly equal compression ratios for each stage. Thus, since the volumetric flowrate (in actual cfm) is smaller from one stage to the next, the volume displacement of each stage is progressively smaller.

Sliding Vane Compressor

The typical sliding vane compressor stage is a rotating cylinder located eccentrically in the bore of a cylindrical housing (see Figure 3-77). The vanes are in slots in the rotating cylinder, and are allowed to move in and out of these slots to adjust to the changing clearance between the outside surface of the rotating cylinder and the inside bore surface of the housing. The vanes are always in contact with the inside bore due to either air pressure under the vane, or spring force under the vane. The

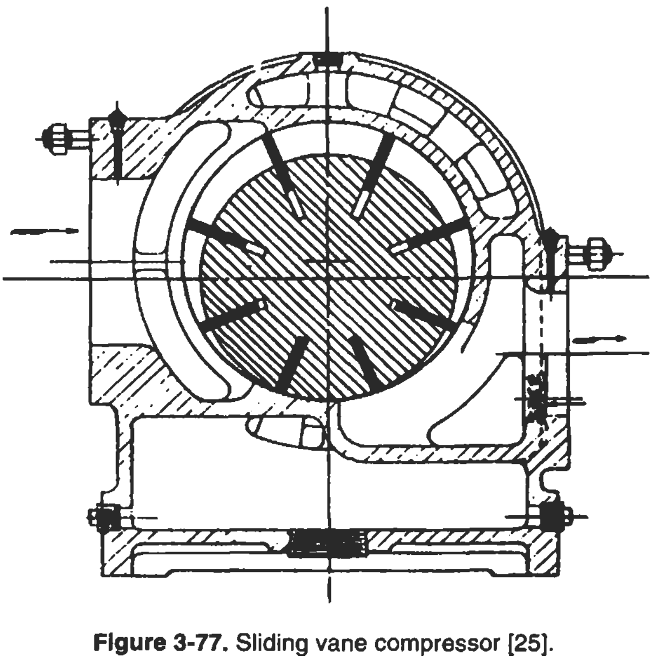


Figure 3-77. Sliding vane compressor [25].

top of the vanes slide over the inside surface of the bore of the housing as the inside cylinder rotates. Gas is brought into the compression stage through the inlet suction port. The gas is then trapped between the vanes, and as the inside cylinder rotates the gas is compressed to a smaller volume as the clearance is reduced. When the clearance is the smallest, the gas has rotated to the outlet port. The compressed gas is discharged to the pipeline system connected to the outlet side of the compressor. As each set of vanes reaches the outlet port, the gas trapped between the vanes is discharged. The clearance between the rotating cylinder and the housing is fixed, and thus the pressure ratio of compression for the stage is fixed, or built-in. The geometry, e.g., cylinder length, diameter, etc., of the inside of each compressor stage determines the displacement volume and compression ratio of the compressor.

The principal seals within the sliding vane compressor are provided by the interface between the end of the vane and the inside surface of the cylindrical housing. The sliding vanes must be made of a material that will not damage the inside surface of the housing. Therefore, most vane material is phenolic resin-impregnated laminated fabrics (such as asbestos or cotton cloth). Also, some metals other than one that would gall with the housing can be used such as aluminum. Usually, vane compressors utilize oil lubricants in the compression cavity to allow for smooth action of the sliding vanes against the inside of the housing. There are, however, some sliding vane compressors that may be operated oil-free. These utilize bronze, or carbon/graphite vanes [25].

The volumetric flowrate for a sliding vane compression stage q_s (ft^3/min) is approximately

$$q_s = 2al\,(d_2 - mt)N \tag{3-78}$$

where a is the eccentricity in ft, l is the length of the cylinder in ft, d_1 is the outer diameter of the rotary cylinder in ft, d_2 is the inside diameter of the cylindrical housing in ft, t is the vane thickness in ft, m is the number of vanes, and N is the speed of the rotating cylinder in rpm.

The eccentricity a is

$$a = \frac{d_2 - d_1}{2} \tag{3-79}$$

Some typical values of a vane compressor stage geometry are $d_1/d_2 = 0.88$, $a = 0.06d_2$, $a = 0.06d_2$, and $l/d_2 = 2.00$ to 3.00. Typical vane up speed usually does not exceed 50 ft/s.

There is no clearance in a rotary compressor. However, there is leakage of air within the internal seal system and around the vanes. Thus, the typical volumetric efficiency for the sliding vane compression is of the order of 0.82 to 0.90. The heavier the gas, the greater the volumetric efficiency. The higher the pressure ratio through the stage, the lower the volumetric efficiency.

Rotary Screw Compressor

The typical rotary screw compressor stage is made up of two rotating shafts, or screws. One is a female rotor and the other a male rotor. These two rotating components turn counter to one another (counterrotating). The two rotating elements are designed so that as they rotate opposite to one another; their respective helix forms intermesh (see Figure 3-78). As with all rotary compressors, there are no valves. The gas is sucked into the inlet post and is squeezed between the male and female

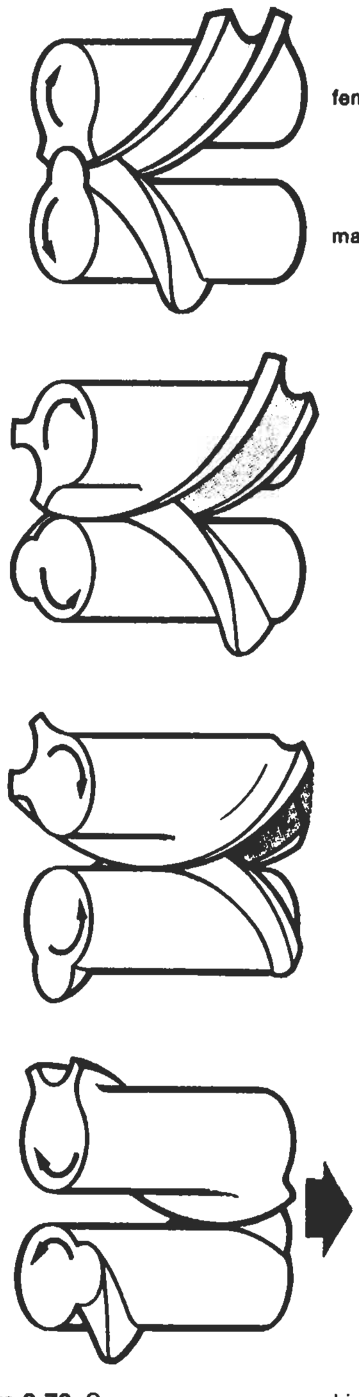


Figure 3-78. Screw compressor working principle [4].

portion of the rotating intermeshing screw elements and their housing. The compression ratio of the stage and its volumetric flowrate are determined by the geometry of the two rotating screw elements and the speed at which they are rotated.

Screw compressors operate at rather high speeds. Thus, they are rather high volumetric flowrate compressors with relatively small exterior dimensions.

Most rotary screw compressors use lubricating oil within the compression space. This oil is injected into the compression space and recovered, cooled, and recirculated. The lubricating oil has several functions

- seal the internal clearances
- cool the gas (usually air) during compression
- lubricate the rotors
- eliminate the need for timing gears

There are versions of the rotary screw compressor that utilize water injection (rather than oil). The water accomplishes the same purposes as the oil, but the air delivered in these machines is oil-free.

Some screw compressors have been designed to operate with an entirely oil-free compression space. Since the rotating elements of the compressor need not touch each other or the housing, lubrication can be eliminated. However, such rotary screw compressor designs require timing gears. These machines can deliver totally oil-free, water-free dry air (or gas).

The screw compressor can be staged. Often screw compressors are utilized in three- or four-stage versions.

Detailed calculations regarding the design of the rotary screw compressor are beyond the scope of this handbook. Additional details can be found in other references [4,25,26,27].

Rotary Lobe Compressor

The rotary lobe compressor stage is a rather low-pressure machine. These compressors do not compress gas internally in a fixed sealed volume as in other rotaries. The straight lobe compressor uses two rotors that intermesh as they rotate (see Figure 3-79). The rotors are timed by a set of timing gears. The lobe shapes may be involute or cycloidal in form. The rotors may also have two or three lobes. As the rotors turn and pass the intake port, a volume of gas is trapped and carried between the lobes and the housing of the compressor. When the lobe pushes the gas toward the outlet port, the gas is compressed by the back pressure in the gas discharge line.

Volumetric efficiency is determined by the leakage at tips of the lobes. The leakage is referred to as slip. Slippage is a function of rotor diameter, differential pressure, and the gas being compressed.

For details concerning this low pressure compressor see other references [4,25,26,27].

Liquid Piston Compressor

The liquid piston compressor utilizes a liquid ring as a piston to perform gas compression within the compression space. The liquid piston compressor stage uses a single rotating element that is located eccentrically inside a housing (see Figure 3-80). The rotor has a series of vanes extending radially from it with a slight curvature toward the direction of rotation. A liquid, such as oil, partially fills the compression space between the rotor and the housing walls. As rotation takes place, the liquid forms a ring as centrifugal forces and the vanes force the liquid to the outer boundary of the housing. Since the element is located eccentrically in the

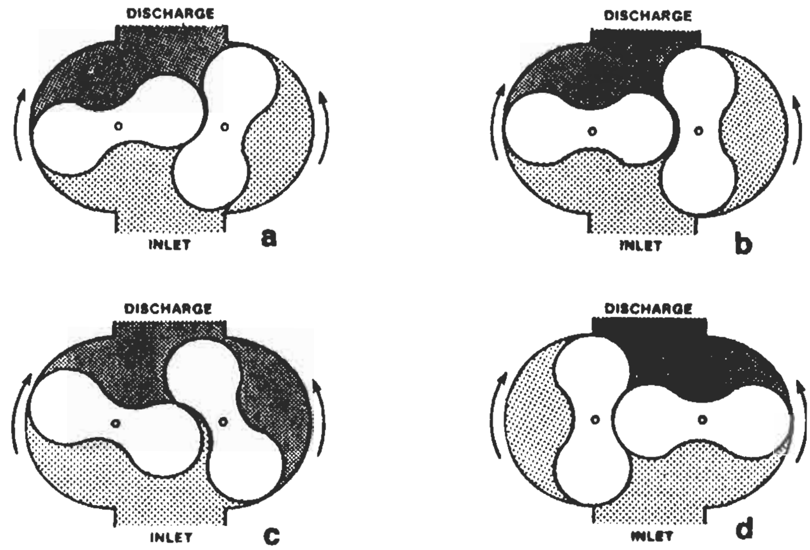


Figure 3-79. Straight lobe rotary compressor operating cycle [4,23].

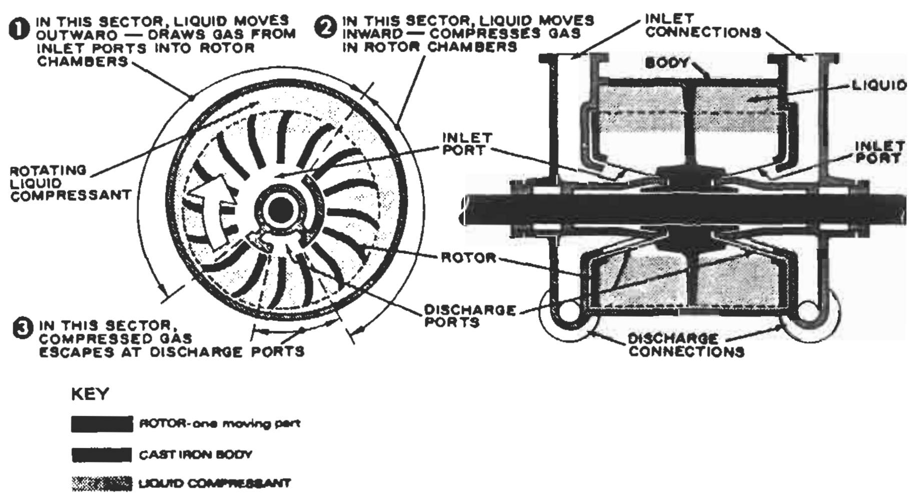


Figure 3-80. Liquid piston compressor [4,23].

housing, the liquid ring (or piston) moves in an oscillatory manner. The compression space in the center of the stage communicates with the gas inlet and outlet parts and allows a gas pocket. The liquid ring alternately uncovers the inlet part and the outlet part. As the system rotates, gas is brought into the pocket, compressed, and released to the outlet port.

The liquid compressor has rather low efficiency, about 50%. The liquid piston compressor may be staged. The main advantage to this type of compressor is that it can be used to compress gases with significant liquid content in the stream.

Summary of Rotary Compressors

The main advantage of rotary compressors is that most are easy to maintain in field conditions and in industrial settings. Also, they can be constructed to be rather portable since they have rather small exterior dimensions. Also, many versions of the rotary compressor can produce oil-free compressed gases.

The main disadvantage are that these machines operate at a fixed pressure ratio. Thus, the cost of operating the compressor does not basically change with reduced back pressure in the discharge line. As long as the back pressure is less than the pressure output of the rotary, the rotary will continue to operate at a fixed power level. Also, since the pressure ratio is built into the rotary compressor, discharging the compressor into a back pressure near or greater than the pressure output of the machine will significantly reduce the volumetic flowrate produced by the machine.

Centrifugal Compressors

The centrifugal compressor is the earliest developed dynamic, or continuous flow, compressor. This type of compressor has no distinct volume in which compression takes place. The main concept of the centrifugal compressor is use of centrifugal force to convert kinetic energy into pressure energy. Figure 3-81 shows a diagram of a single-stage centrifugal compressor. The gas to be compressed is sucked into the center of the rotating impeller. The impeller throws the gas out to the periphery by means of its radial blades and high-speed rotation. The gas is then guided through the diffuser where the high-velocity gas is slowed, which results in a high pressure. In multistage centrifugal compressors, the gas is passed to the next impeller after the diffuser of the previous impeller. In this manner, the compressor may be staged to

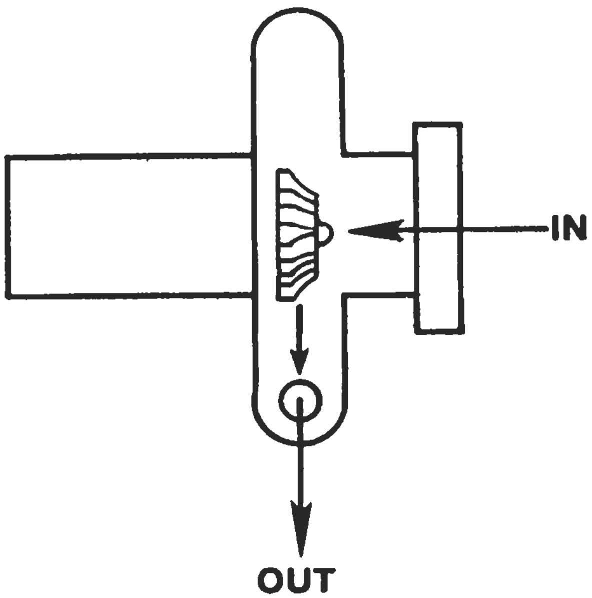


Figure 3-81. Single-stage centrifugal compressor [23].

increase the pressure of the ultimate discharge (see Figure 3-82). Since the compression pressure ratio at each stage is usually rather low, of the order 2 or so, the need for intercooling is not important after each stage. Figure 3-82 shows a typical multistage centrifugal compressor configuration with an intercooler after the first three stages of compression.

The centrifugal compressor must operate at rather high rotation speeds to be effective. Most commercial centrifugal compressors operate at speeds of the order of 20,000–30,000 rpm. With such rotation speeds very large volumes of gas can be compressed with equipment having rather modest external dimensions. Commercial centrifugal compressors can operate with volumetric flowrates up to around 10.4 actual cfm and with overall compression ratios up to about 20.

Centrifugal compressors are usually used in large processing plants and in some pipeline applications. They can be operated with any lubricant or other contaminant in the gas stream, or they can be operated with some small percentage of liquid in the gas stream.

These machines are used principally to compress large volumetric flowrates to rather modest pressures. Thus, their use is more applicable to the petroleum refining and chemical processing industries.

More details regarding the centrifugal compressor may be found in other references [4,23].

Axial-Flow Compressors

The axial compressor is a very high-speed, large volumetric flowrate machine. This is another dynamics, or continuous flow machine. This type of compressor sucks in gas at the intake port and propels the gas axially through the compression space via a series of radially arranged rotor blades and stator (diffuser) blades (see Figure 3-83). As in the centrifugal compressor, the kinetic energy of the high-velocity flow exiting each rotor stage is converted to pressure energy in the follow-on stator (diffuser) stage. Axial-flow compressors have a volumetric flowrate range of about 3×10^4–10^6 actual cfm. Their compression ratio is typically around 10 to 20. Because

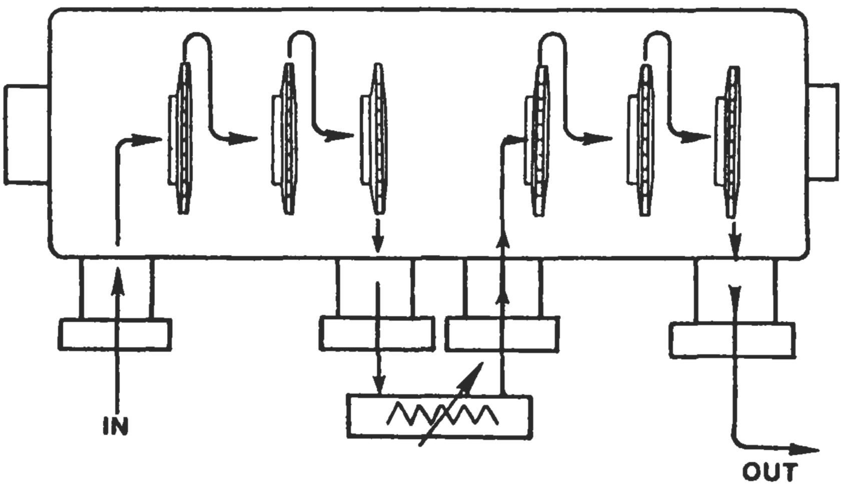


Figure 3-82. Multistage centrifugal compressor with intercooling [23].

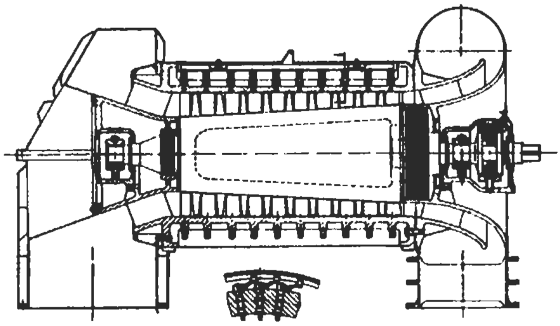


Figure 3-83. Multistage axial-flow compressor [26].

of their small diameter, their machines are principal compressor design for jet engine applications. There are some applications for axial-flow compressors for large process plant operations where very large constant volumetric flowrates are needed.

More detail regarding axial flow compressors may be found in other references [22,26].

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4

Drilling and Well Completions

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Chapter 4

Drilling and Well Completions

DERRICKS AND PORTABLE MASTS

Derricks and portable masts provide the clearance and structural support necessary for raising and lowering drill pipe, casing, rod strings, etc., during drilling and servicing operations. Standard derricks are bolted together at the well site, and are considered nonportable. Portable derricks, which do not require full disassembly for transport, are termed masts.

The derrick or mast must be designed to safely carry all loads that are likely to be used during the structure's life [1]. The largest vertical dead load that will likely be imposed on the structure is the heaviest casing string run into the borehole. However, the largest vertical load imposed on the structure will result from pulling equipment (i.e., drill string or casing string) stuck in the borehole. The most accepted method is to design a derrick or mast that can carry a dead load well beyond the maximum casing load expected. This can be accomplished by utilizing the safety factor.

The derrick or mast must also be designed to withstand wind loads. Wind loads are imposed by the wind acting on the outer and inner surfaces of the open structure. When designing for wind loads, the designer must consider that the drill pipe or other tubulars may be out of the hole and stacked in the structure. This means that there will be loads imposed on the structure by the pipe weight (i.e., setback load) in addition to the additional loads imposed by the wind. The horizontal forces due to wind are counteracted by the lattice structure that is firmly secured to the structure's foundation. Additional support to the structure can be accomplished by the guy lines attached to the structure and to a dead man anchor some distance away from it. The dead man anchor is buried in the ground to firmly support the tension loads in the guy line. The guy lines are pretensioned when attached to the dead man anchor.

The API Standard 4F, First Edition, May 1, 1985, "API Specifications for Drilling and Well Servicing Structures," was written to provide suitable steel structures for drilling and well servicing operations and to provide a uniform method of rating the structures for the petroleum industry. API Standard 4F supersedes API Standards 4A, 4D, and 4E; thus, many structures in service today may not satisfy all of the requirements of API Standard 4F [2–5].

For modern derrick and mast designs, API Standard 4F is the authoritative source of information, and much of this section is extracted directly from this standard. Drilling and well servicing structures that meet the requirements of API Standard 4F are identified by a nameplate securely affixed to the structure in a conspicuous place. The nameplate markings convey at least the following information:

Mast and Derrick Nameplate Information

a. Manufacturer's name
b. Manufacturer's address
c. Specification 4F
d. Serial number

e. Height in feet
f. Maximum rated static hook load in pounds, with guy lines if applicable, for stated number of lines to traveling block
g. Maximum rated wind velocity in knots, with guy lines if applicable, with rated capacity of pipe racked
h. The API specification and edition of the API specification under which the structure was designed and manufactured
i. Manufacturer's guying diagram—for structures as applicable
j. Caution: Acceleration or impact, also setback and wind loads will reduce the maximum rated static hook load capacity
k. Manufacturer's load distribution diagram (which may be placed in mast instructions)
l. Graph of maximum allowable static hook load versus wind velocity
m. Mast setup distance for mast with guy lines.

Substructure Nameplate Information

a. Manufacturer's name
b. Manufacture's address
c. Specification 4F
d. Serial number
e. Maximum rated static rotary capacity
f. Maximum rated pipe setback capacity
g. Maximum combined rated static rotary and rated setback capacity
h. API specification and edition under which the structure was designed and manufactured.

The manufacturer of structures that satisfy API Standard 4F must also furnish the purchaser with one set of instructions that covers operational features, block reeving diagram, and lubrication points for each drilling or well servicing structure. Instructions should include the raising and lowering of the mast and a facsimile of the API nameplate.

Definitions and Abbreviations

Definitions

The following terms are commonly used in discussing derricks and masts:

Crown block assembly: The stationary sheave or block assembly installed at the top of a derrick or mast.
Derrick: A semipermanent structure of square or rectangular cross-section having members that are latticed or trussed on all four sides. This unit must be assembled in the vertical or operation position, as it includes no erection mechanism. It may or may not be guyed.
Design load: The force or combination of forces that a structure is designed to withstand without exceeding the allowable stress in any member.
Dynamic loading: The loading imposed upon a structure as a result of motion as opposed to static loading.
Dynamic stress: The varying or fluctuating stress occurring in a structural member as a result of dynamic loading.
Erection load: The load produced in the mast and its supporting structure during the raising and lowering operation.

Guy line: A wire rope with one end attached to the derrick or mast assembly and the other end attached to a suitable anchor.
Guying pattern: A plane view showing the manufacturer's recommended loca-tions and distance to the anchors with respect to the wellhead.
Height of derrick and mast without guy lines: The minimum clear vertical distance from the top of the working floor to the bottom of the crown block support beams.
Height of mast with guy lines: The minimum vertical distance from the ground to the bottom of the crown block support beams.
Impact loading: The loading resulting from sudden changes in the motion state of rig components.
Mast: A structural tower comprising one or more sections assembled in a horizontal position near the ground and then raised to the operating position. If the unit contains two or more sections, it may be telescoped or unfolded during the erection.
Mast setup distance: The distance from the centerline of the well to a designated point on the mast structure defined by a manufacturer to assist in the setup of the rig.
Maximum rated static hook load: The sum of the weight applied at the hook and the traveling equipment for the designated location of the dead line anchor and the specified number of drilling lines without any pipe setback, sucker rod, or wind loadings.
Pipe lean: The angle between the vertical and a typical stand of pipe with the setback.
Racking platform: A platform located at a distance above the working floor for laterally supporting the upper end of racked pipe.
Rated static rotary load: The maximum weight being supported by the rotary table support beams.
Rated setback load: The maximum weight of tubular goods that the substructure can withstand in the setback area.
Rod board: A platform located at a distance above the working floor for supporting rods.
Static hook load: see *Maximum Rated Static Hook Load.*

Abbreviations

The following standard abbreviations are used throughout this section.

ABS—American Bureau of Shipping
AISC—American Institute of Steel Construction
AISI—American Iron and Steel Institute
ANSI—American National Standard Institute
API—American Petroleum Institute
ASA—American Standards Association
ASTM—American Society for Testing and Materials
AWS—American Welding Society
IADC—International Association of Drilling Contractors
SAE—Society of Automotive Engineers
USAS—United States of America Standard (ANSI)
RP—Recommended Practice

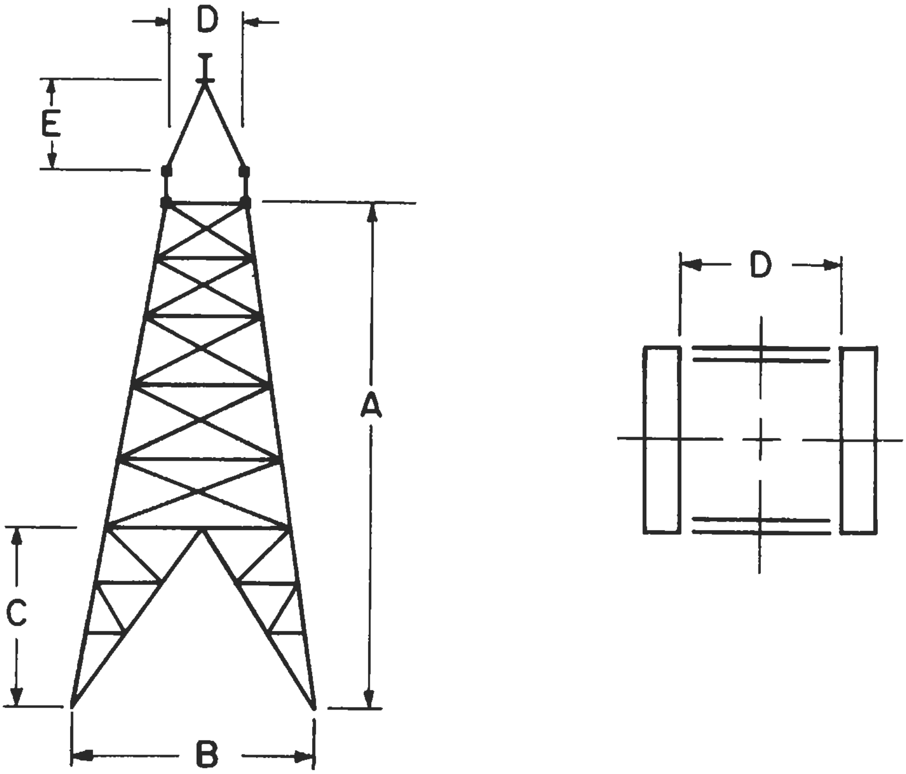
Standard Derricks

A standard derrick is a structure of square cross-section that dimensionally agrees with a derrick size shown in Table 4-1 with dimensions as designated in Figure 4-1.

Table 4-1
Derrick Sizes and General Dimensions [2]

1	2		3		4		5		6		7	
Derrick Size No.	Height A		Nominal Base Square B		Drawworks Window Opening C		V Window Opening C		Opening D		Gin Pole Clearance E	
	ft	in.	ft	in.	ft	in.	ft	in.	ft	in.	ft	in.
10	80	0	20	0	7	6	23	8	5	6	8	0
11	87	0	20	0	7	6	23	8	5	6	8	0
12	94	0	24	0	7	6	23	8	5	6	8	0
16	122	0	24	0	7	6	23	8	5	6	8	0
18	136	0	26	0	7	6	23	8	5	6	12	0
18A	136	0	30	0	7	6	23	8	5	6	12	0
19	140	0	30	0	7	6	26	6	7	6	17	0
20	147	0	30	0	7	6	26	6	6	6	17	0
25	189	0	37	6	7	6	26	6	7	6	17	0

Tolerances: A, ± 6 in.; B, ± 5 in.; C, + 3 ft., 6 in.; D, ± 2 in.; E, ± 6 in.



A — The vertical distance from the top of the base plate to the bottom of the crown block support beam.
B — The distance between heel to heel of adjacent legs at the top of the base plate.
C — The window opening measured in the clear and parallel to the center line of the derrick side from top of base plate.
D — The smallest clear dimension at the top of the derrick that would restrict passage of crown block.
E — The clearance between the horizontal header of the gin pole and the top of the crown support beam.

Figure 4-1. Derrick dimensions [2].

Derrick Window

The derrick window arrangement types A, C, D, and E, shown in Figure 4-2, shall be interchangeable. The sizes and general dimensions of the V window opening and drawworks window opening are given in Tables 4-1 and 4-2.

Foundation Bolt Settings

Foundation bolt sizes and patterns are shown in Figure 4-3. Minimum bolt sizes are used and should be increased if stresses dictate larger diameter. The

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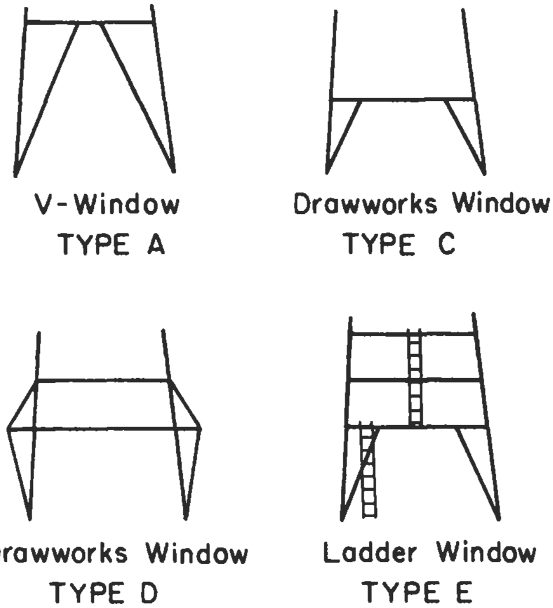

Figure 4-2. Derrick windows [2].

Table 4-2
Conversion Values
(For 0–50 Ft. Height) [2]

Pressure P Lb./Sq. Ft.	Wind Velocity V_k Knots	Wind Velocity Miles Per Hour
10	49	56
15	60	69
20	69	79
25	77	89
30	84	97
35	91	105
40	97	112
45	103	119
50	109	125
55	114	131

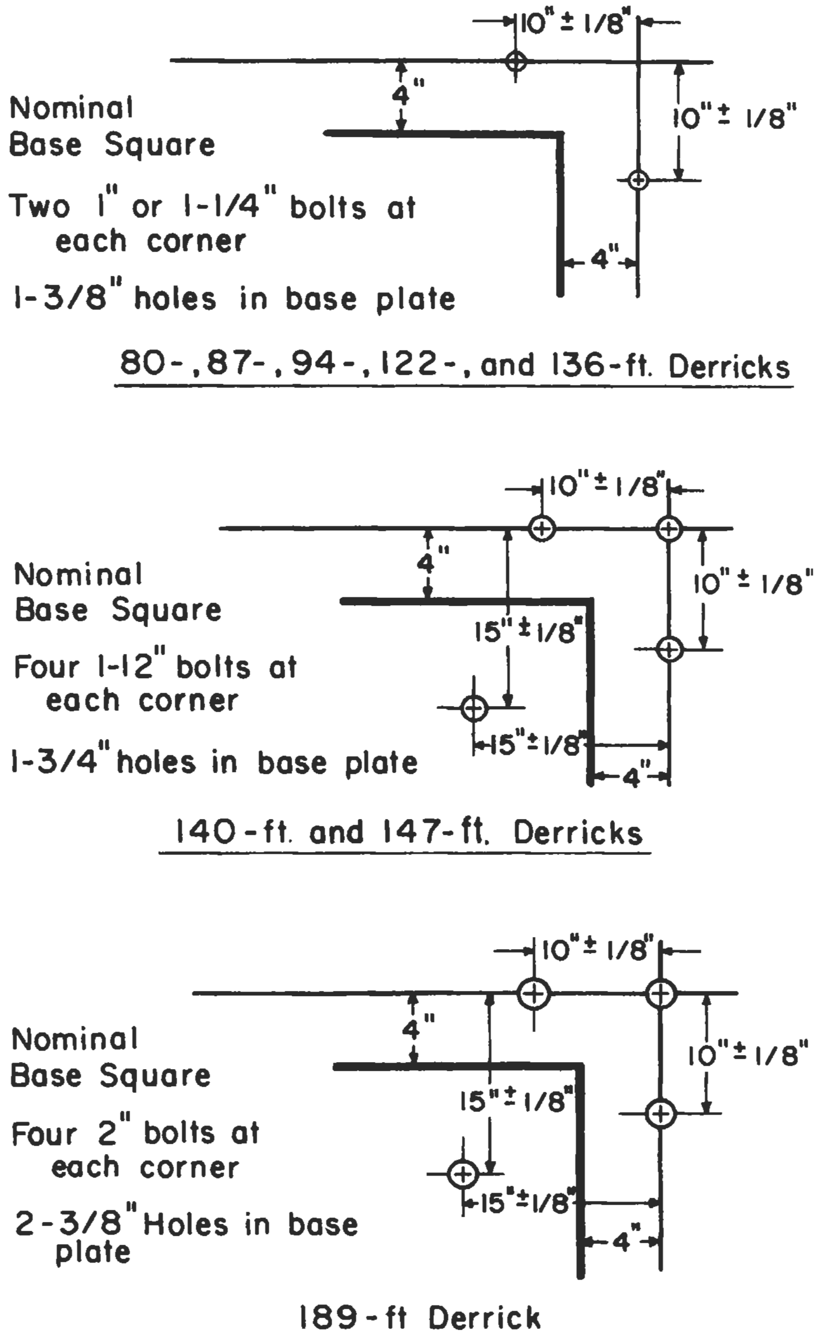

Figure 4-3. Foundation bolt pattern for derrick leg [2].

(text continued from page 503)

maximum reaction (uplift, compression, and shear) produced by the standard derrick loading foundation bolt size and setting plan should be furnished to the original user.

Load Capacities

All derricks and masts will fail under an excessively large load. Thus API makes it a practice to provide standard ratings for derricks and masts that meet its specifications. The method for specifying standard ratings has changed over the years; therefore, old structures may fail under one rating scheme and new structures may fail under another.

API Standard 4A (superseded by Standard 4F) provides rating of derrick capacities in terms of maximum safe load. This is simply the load capacity of a single leg multiplied by four. It does not account for pipe setback, wind loads, the number of lines between the crown block and the traveling block, the location of the dead line, or vibratory and impact loads. Thus, it is recommended that the maximum safe static load of derricks designed under Standard 4A exceed the derrick load as follows:

$$\text{Derrick load}^{*} = 1.5(W_h + W_c + W_t + 4F_L) \tag{4-1}$$

where W_h = weight of the traveling block plus the weight of the drillstring suspended in the hole, corrected for buoyancy effects
W_c = weight of the crown block
W_t = weight of tools suspended in the derrick
F_L = extra leg load produced by the placement of the dead and fast lines

In general, $F_L = W_h/n$ if the deadline is attached to one of the derrick legs, and $F_L = W_h/2n$ if the deadline is attached between two derrick legs. n is the number of lines between the crown block and traveling block. The formula for F_L assumes that no single leg shares the deadline and fastline loads.

The value of 1.5 is a safety factor to accommodate impact and vibration loads. Equation 4-1 does not account for wind and setback loads, thus, it may provide too low an estimate of the derrick load in extreme cases.

API Standard 4D (also superseded by Standard 4F) provides rating of portable masts as follows:

For each mast, the manufacturer shall designate a maximum rated static hook load for each of the designated line reevings to the traveling block. Each load shall be the maximum static load applied at the hook, for the designated location of deadline anchor and in the absence of any pipe-setback, sucker-rod, or wind loadings. The rated static hook load includes the weight of the traveling block and hook. The angle of mast lean and the specified minimum load guy line pattern shall be considered for guyed masts.

Under the rigging conditions given on the nameplate, and in the absence of setback or wind loads, the static hook load under which failure may occur in masts conforming to this specification can be given as only approximately twice the maximum rated static hook load capacity.

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*This is an API Standard 4A rating capacity and should not be confused with the actual derrick load that will be discussed in the section titled "Derricks and Portable Masts."

The manufacturer shall establish the reduced rated static hook loads for the same conditions under which the maximum rated static hook loads apply, but with the addition of the pipe-setback and sucker-rod loadings. The reduced rated static hook loads shall be expressed as percentages of the maximum rated static hook loads. Thus, the portable mast ratings in Standard 4D include a safety factor of 2 to allow for wind and impact loads, and require the manufacturer to specify further capacity reductions due to setback.

The policy of Standard 4D, that the manufacturer specify the structure load capacity for various loading configurations, has been applied in detail in Standards 4E (superseded by Standard 4F) and 4F. Standard 4F calls for detailed capacity ratings that allow the user to look up the rating for a specific loading configuration. These required ratings are as follows.

Standard Ratings

Each structure shall be rated for the following applicable loading conditions. The structures shall be designed to meet or exceed these conditions in accordance with the applicable specifications set forth herein. The following ratings do not include any allowance for impact. Acceleration, impact, setback, and wind loads will reduce the rated static hook load capacity.

Derrick—Stationary Base

1. Maximum rated static hook load for a specified number of lines to the traveling block.
2. Maximum rated wind velocity (knots) without pipe setback.
3. Maximum rated wind velocity (knots) with full pipe setback.
4. Maximum number of stands and size of pipe in full setback.
5. Maximum rated gin pole capacity.
6. Rated static hook load for wind velocities varying from zero to maximum rated wind velocity with full rated setback and with maximum number of lines to the traveling block.

Mast with Guy Lines

1. Maximum rated static hook load capacity for a specified number of lines strung to the traveling block and the manufacturer's specified guying.
2. Maximum rated wind velocity (knots) without pipe setback.
3. Maximum rated wind velocity (knots) with full pipe setback.
4. Maximum number of stands and size of pipe in full setback.

Mast without Guy Lines

1. Maximum rated static hook load for a specified number of lines to the traveling block.
2. Maximum rated wind velocity (knots) without pipe setback.
3. Maximum rated wind velocity (knots) with full pipe setback.
4. Maximum number of stands and size of pipe in full setback.
5. Rated static hook load for wind velocities varying from zero to maximum rated wind velocity with full rated setback and with maximum number of lines to the traveling block.

Mast and Derricks under Dynamic Conditions

1. Maximum rated static hook load for a specified number of lines to the traveling block.
2. Hook load, wind load, vessel motions, and pipe setback in combination with each other for the following:
   a. Operating with partial setback.
   b. Running casing.
   c. Waiting on weather.
   d. Survival.
   e. Transit.

Substructures

1. Maximum rated static hook load, if applicable.
2. Maximum rated pipe setback load.
3. Maximum rated static load on rotary table beams.
4. Maximum rated combined load of setback and rotary table beams.

Substructure under Dynamic Conditions

1. Maximum rated static hook load.
2. Maximum rated pipe setback load.
3. Maximum rated load on rotary table beams.
4. Maximum rated combined load of setback and rotary table beams.
5. All ratings in the section titled "Mast and Derricks under Dynamic Conditions."

Design Loadings

Derricks and masts are designed to withstand some minimum loads or set of loads without failure. Each structure shall be designed for the following applicable loading conditions. The structure shall be designed to meet or exceed these conditions in accordance with the applicable specifications set forth herein.

Derrick—Stationary Base

1. Operating loads (no wind loads) composed of the following loads in combination:
   a. Maximum rated static hook load for each applicable string up condition.
   b. Dead load of derrick assembly.
2. Wind load without pipe setback composed of the following loads in combination:
   a. Wind load on derrick, derived from maximum rated wind velocity without setback (minimum wind velocity for API standard derrick sizes 10 through 18A is 93 knots, and for sizes 19 through 25 is 107 knots).
   b. Dead load of derrick assembly.
3. Wind load with rated pipe setback composed of the following loads in combination:
   a. Wind load on derrick derived from maximum rated wind velocity with setback of not less than 93 knots.
   b. Dead load of derrick assembly.

c. Horizontal load at racking platform, derived from maximum rated wind velocity with setback of not less than 93 knots acting on full pipe setback.
d. Horizontal load at racking platform from pipe lean.

Mast with Guy Lines

1. Operating loads (no wind load) composed of the following loads in combination:
   a. Maximum rated static hook load for each applicable string up condition.
   b. Dead load of mast assembly.
   c. Horizontal and vertical components of guy line loading.
2. Wind loads composed of the following loads in combination:
   a. Wind load on mast, derived from a maximum rated wind velocity with setback of not less than 60 knots.
   b. Dead load of mast assembly.
   c. Horizontal loading at racking board, derived from a maximum rated wind velocity with setback of not less than 60 knots, acting on full pipe setback.
   d. Horizontal and vertical components of guy line loading.
   e. Horizontal and vertical loading at rod board, derived from a maximum rated wind velocity with setback of not less than 60 knots, acting on rods in conjunction with dead weight of rods.
3. Wind loads composed of the following loads in combination:
   a. Wind load on mast, derived from a maximum rated wind velocity with setback of not less than 60 knots.
   b. Dead load of mast assembly.
   c. Horizontal loading at racking platform, derived from a maximum rated wind velocity with setback of not less than 60 knots, acting on full pipe setback.
   d. Horizontal and vertical components of guy line loading.
4. Wind loads composed of the following loads in combination:
   a. Wind load on mast derived from a maximum rated wind velocity without setback of not less than 60 knots.
   b. Dead load of mast assembly.
   c. Horizontal and vertical components of guy line loading.
5. Erection loads (zero wind load) composed of the following loads in combination:
   a. Forces applied to mast and supporting structure created by raising or lowering mast.
   b. Dead load of mast assembly.
6. Guy line loading (assume ground anchor pattern consistent with manufacturer's guying diagram shown on the nameplate).
   a. Maximum horizontal and vertical reactions from conditions of loading applied to guy line.
   b. Dead load of guy line.
   c. Initial tension in guy line specified by mast manufacturer.

Mast without Guy Lines

1. Operating loads composed of the following loads in combination:
   a. Maximum rated static hook load for each applicable string up condition.
   b. Dead load of mast assembly.

2. Wind load without pipe setback composed of the following loads in combination:
   a. Wind loading on mast, derived from a maximum rated wind velocity without setback of not less than 93 knots.
   b. Dead load of mast assembly.
3. Wind load with pipe setback composed of the following loads in combination:
   a. Wind loading on mast, derived from a maximum rated wind velocity with setback of not less than 70 knots.
   b. Dead load of mast assembly.
   c. Horizontal load at racking platform derived from a maximum rated wind velocity with setback of not less than 70 knots acting on pipe setback.
   d. Horizontal load at racking platform from pipe lean.
4. Mast erection loads (zero wind load) composed of the following loads in combination:
   a. Forces applied to mast and supporting structure created by raising or lowering mast.
   b. Dead load of mast assembly.
5. Mast handling loads (mast assembly supported at its extreme ends).

Derricks and Mast under Dynamic Conditions

All conditions listed in the section titled "Load Capacities," subsection titled "Mast and Derricks under Dynamic Conditions," are to be specified by the user. Forces resulting from wind and vessel motion are to be calculated in accordance with the formulas presented in the section titled "Design Specifications," paragraphs titled "Wind," "Dynamic Loading (Induced by Floating Hull Motion)."

Substructures

1. Erection of mast, if applicable.
2. Moving or skidding, if applicable.
3. Substructure shall be designed for the following conditions:
   a. Maximum rated static rotary load.
   b. Maximum rated setback load.
   c. Maximum rated static hook load (where applicable).
   d. Maximum combined rated static hook and rated setback loads (where applicable).
   e. Maximum combined rated static rotary and rated setback loads.
   f. Wind loads resulting from maximum rated wind velocity acting from any direction on all exposed elements. Wind pressures and resultant forces are to be calculated in accordance with the equations and tables in the section titled "Design Specifications," paragraph titled "Wind." When a substructure is utilized to react guy lines to the mast, these reactions from the guy lines must be designed into the substructure.
   g. Dead load of all components in combination with all of the above.

Substructure under Dynamic Conditions

All conditions listed in the section titled "Load Capacities," paragraph titled "Structure under Dynamic Conditions," are to be specified by the user. Forces resulting from wind and vessel motion are to be calculated in accordance with formulas from the section titled "Design Specifications," paragraphs titled "Wind" and "Dynamic Loading (Induced by Floating Hull Motion)."

Design Specifications

In addition to withstanding some minimum load or loads (sections titled "Load Capacities" and "Design Specifications"), derricks and masts that satisfy API standards must also satisfy certain requirements regarding materials, allowable stresses, wind, dynamic loading, earthquakes and extremes of temperature.

Materials

The unrestricted material acceptance is not intended since physical properties are not the sole measure of acceptability. Metallurgical properties, which affect fabrication and serviceability, must also be considered.

Steel. Steel shall conform to one of the applicable ASTM specifications referred to by applicable AISC specifications. Other steels not covered by these specifications may be used provided that the chemical and physical properties conform to the limits guaranteed by the steel manufacturer. Structural steel shapes having specified minimum yield less than 33,000 psi shall not be used. Certified mill test report or certified reports of tests made in accordance with ASTM A6 and the governing specification shall constitute evidence of conformity with one of the specifications listed.

Bolts. Bolts shall conform to one of the applicable SAE, ASTM, or AISC specifications. Other bolts not covered by these specifications may be used provided the chemical, mechanical, and physical properties conform to the limits guaranteed by the bolt manufacturer. Certified reports shall constitute sufficient evidence of conformity with the specification. Bolts of different mechanical properties and of the same diameter shall not be mixed on the same drilling or servicing structure to avoid the possibility of bolts of relatively low strength being used where bolts of relatively high strength are required.

Welding Electrodes. Welding electrodes shall conform to applicable AWS and ASTM specifications or other governing codes. Newly developed welding processes shall use welding electrodes conforming to applicable AWS or other governing publications. Certified reports shall constitute sufficient evidence of conformity with the specifications.

Wire Rope. Wire rope for guy lines or erection purposes shall conform to API Specification 9A: "Specification for Wire Rope."

Nonferrous Materials. Nonferrous materials must conform to appropriate governing codes. Certified reports shall constitute sufficient evidence of conformity with such codes.

Allowable Stresses

AISC specifications for the design fabrication and erection of structural steel for buildings shall govern the design of these steel structures (for AISC specifications, see the current edition of Steel Construction Manual of the American Institute of Steel Construction). Only Part I of the AISC manual, the portion commonly referred to as elastic design, shall be used in determining allowable unit stresses; use of Part II, which is commonly referred to as plastic design, is not allowed. The AISC shall be the final authority for determination of allowable unit stresses, except that current practice and experience do not dictate the need to follow the AISC for members and connections subject to repeated variations of stress, and for the consideration of secondary stresses.

For purposes of this specification, stresses in the individual members of a latticed or trussed structure resulting from elastic deformation and rigidity of joints are defined as secondary stresses. These secondary stresses may be taken to be the difference between stresses from an analysis assuming fully rigid joints, with loads applied only at the joints, and stresses from a similar analysis with pinned joints. Stresses arising from eccentric joint connections, or from transverse loading of members between joints, or from applied moments, must be considered primary stresses.

Allowable unit stresses may be increased 20% from the basic allowable stress when secondary stresses are computed and added to the primary stresses in individual members. However, primary stresses shall not exceed the basic allowable stresses.

Wind and Dynamic Stresses (Induced by Floating Hull Motion). Allowable unit stresses may be increased one-third over basic allowable stresses when produced by wind or dynamic loading, acting alone, or in combination with the design dead load and live loads, provided the required section computed on this basis is not less than required for the design dead and live loads and impact (if any), computed without the one-third increase.

Wire Rope. The size and type of wire rope shall be as specified in API Specification 9A and by API RP 9B (see section titled "Hoisting System").

1. A mast raised and lowered by wire rope shall have the wire rope sized to have a nominal strength of at least $2\frac{1}{2}$ times the maximum load on the line during erection.
2. A mast or derrick guyed by means of a wire rope shall have the wire rope sized so as to have a nominal strength of at least $2\frac{1}{2}$ times the maximum guy load resulting from a loading condition.

Crown Shafting. Crown shafts, including fastline and deadline sheave support shafts, shall be designed to AISC specifications except that the safety factor in bending shall be a minimum of 1.67 to yield. Wire rope sheaves and bearings shall be designed in accordance with "API Specification 8A: Drilling and Production Hoisting Equipment."

Wind

Wind forces shall be applied to the entire structure. The wind directions that result in the highest stresses for each component of the structure must be determined and considered. Wind forces for the various wind speeds shall be calculated according to

$$F = (P)(A) \tag{4-2}$$

where F = Force in lb
P = Pressure in lb/ft^2
A = Total area, in ft^2, projected on a plane, perpendicular to the direction of the wind, except that the exposed areas of two opposite sides of the mast or derrick shall be used.

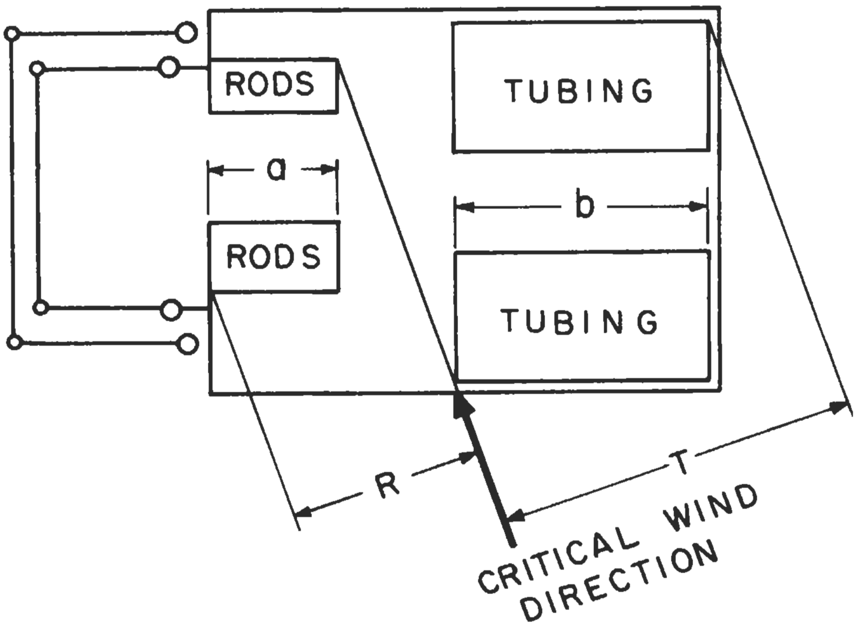
When pipe or tubing is racked in more than one area, the minimum area of setback shall be no less than 120% of the area on one side; when rods are racked on more than one area, the minimum area of rods shall be no less than 150% of the area of one side to account for the effect of wind on the leeward area (Figure 4-4).

The pressure due to wind is

$$P = 0.00338\,(V_k^2)(C_h)(C_s) \tag{4-3}$$

where P = pressure in lb/ft^2
V_k = wind velocity in knots
C_h = height coefficient

Height (ft)	C_h
0– 50	1.0
50–100	1.1
100–150	1.2
150–200	1.3
200–250	1.4



NOTE: In calculating the value of A,
If R is greater than 1.5a, use R. If not, use 1.5a.
If T is greater than 1.2b, use T. If not, use 1.2b.

Figure 4-4. Diagram of projected area [9].

Height is the vertical distance from ground or water surface to the center of area. The shape coefficient C_s for a derrick is assumed as 1.25. C_s and C_h were obtained from ABS, "Rules for Building and Classing Offshore Drilling Units, 1968."

Dynamic Loading (Induced by Floating Hull Motion)

Forces shall be calculated according to the following [6]:

$$FP = \left(\frac{WL_1}{32.2}\right)\left(\frac{4\pi^2}{T_p^2}\right)\left(\frac{\pi\phi}{180}\right) + W\sin\phi \tag{4-4}$$

$$FP = \left(\frac{W}{32.2}\right)\left(\frac{4\pi^2}{T_r^2}\right)\left(\frac{\pi\phi L}{180}\right) + W\sin\theta \tag{4-5}$$

$$FH = W + \frac{W2\pi^2 H}{T_h^2 g} \tag{4-6}$$

where W = dead weight of the point under consideration
L_1 = distance from pitch axis to the gravity center of the point under consideration in feet
L = distance from roll axis to the gravity center of the point under consideration in feet
H = heave (total displacement)
T_p = period of pitch in seconds
T_r = period of roll in seconds
T_h = period of heave in seconds
ϕ = angle of pitch in degrees
θ = angle of roll in degrees
g = gravity in 32.2 ft/s/s

Unless specified, the force due to combined roll, pitch, and heave shall be considered to be the largest of the following:

1. Force due to roll plus force due to heave.
2. Force due to pitch plus force due to heave.
3. Force due to roll and pitch determined as the square root of the sum of squares plus force due to heave.

Angle of roll or pitch is the angle to one side from vertical. The period is for a complete cycle.

Earthquake

Earthquake is a special loading condition to be addressed when requested by the user. The user is responsible for furnishing the design criteria that includes design loading, design analysis method, and allowable response.

The design criteria for land units may be in accordance with local building codes using equivalent static design methods.

For fixed offshore platform units, the design method should follow the strength level analysis guidelines in API RP 2A. The drilling and well servicing units should be able to resist the deck movement, i.e., the response of the deck to the ground motion prescribed for the design of the offshore platform. The allowable stresses for the combination of earthquake, gravity and operational loading should be limited to those basic allowables with the one-third increase as specified in AISC Part I. The computed stresses should include the primary and the secondary stress components.

Extreme Temperature

Because of the effect of low temperatures on structural steel, it will be no use to change (decrease) the allowable unit stresses mentioned in the preceding paragraphs titled "Allowable Stresses." Low temperature phenomena in steel are well established in principle. Structures to be used under extreme conditions should use special materials that have been, and are being, developed for this application.

Miscellaneous

Structural Steels. Structures shall conform to sections of the AISC "Specifications for the Design, Fabrication and Erection of Structural Steel Buildings."

Castings. All castings shall be thoroughly cleaned, and all cored holes shall be drifted to ensure free passage of proper size bolt.

Protection. Forged parts, rolled structural steel shapes and plates, and castings shall be cleaned, primed, and painted with a good commercial paint or other specified coating before shipment. Machined surfaces shall be protected with a suitable lubricant or compound.

Socketing. Socketing of raising, erecting, or telescoping mast wire ropes shall be performed in accordance with practices outlined by API RP 9B.

Recommended Practice for Maintenance and Use of Drilling and Well Servicing Structures

These general recommendations, if followed, should result in longer satisfactory service from the equipment. These recommendations should in every case be considered as supplemental to, and not as a substitute for, the manufacturer's instructions.

The safe operation of the drilling and well servicing structure and the success of the drilling operation depend on whether the foundation is adequate for the load imposed. The design load for foundation should be the sum of the weight of the drilling or well servicing structure, the weight of the machinery and equipment on it, the maximum hook load of the structure, and the maximum setback load.

Consultation with the manufacturer for approval of materials and methods is required before proceeding with repairs. Any bent or otherwise damaged member should be repaired or replaced. Any damaged compression member should be replaced rather than repaired by straightening. Drilling and well servicing structures use high-strength steels that require specific welding electrodes and welding techniques.

Fixtures and accessories are preferably attached to a structure by suitable clamps. Do not drill or burn holes in any members or perform any welding without obtaining approval of the manufacturer.

Wire line slings or tag lines should have suitable fittings to prevent the rope from being bent over sharp edges and damaged.

Loads due to impact, acceleration, and deceleration may be indicated by fluctuation of the weight indicator readings and the operator should keep the indicator readings within the required hook load capacity.

In the erecting and lowering operation, the slowest practical line speed should be used.

Girts, braces, and other members should not, under any circumstances, be removed from the derrick while it is under load.

The drilling and well servicing structure manufacturer has carefully designed and selected materials for his or her portable mast. The mast should perform satisfactorily within the stipulated load capacities and in accordance with the instructions. Every operator should study the instructions and be prepared for erecting, lowering, and using the mast.

The substructure should be restrained against up-lift, if necessary, by a suitable dead weight or a hold-down anchor. The weight of the hoist and vehicle, where applicable, may be considered as part or all of the required anchorage.

Each part of a bolted structure is designed to carry its share of the load; therefore, parts omitted or improperly placed may contribute to the structure failure. In the erection of bolted structures, the bolts should be tightened only slightly tighter than finger-tight. After the erection of the structure is completed, all bolts should be drawn tight. This procedure permits correct alignment of the structure and results in proper load distribution.

Sling Line Inspection and Replacement

One or more of the three principal factors, including wear due to operation, corrosion and incidental damage, may limit the life of a sling. The first may be a function of the times the mast is raised, and the second will be related to time and atmospheric conditions. The third will bear no relation to either, since incidental damage may occur at the first location as well as any other.

Charting of sling line replacement shows an erratic pattern. Some require replacement at a relatively early date and others last several years longer. Early replacements generally show incidental damage, and it is possible that some of the longer lived ones are used beyond the time when they should be replaced.

There is no way of judging the remaining strength of a rusty rope; therefore, rusty sling lines should be replaced. Areas adjacent to end connections should be examined closely for any evidence of corrosion.

It would no doubt be possible to establish a normal sling line life expectancy in terms of the number of locations used, as long as a set number of months was not exceeded. However, this would not preclude the necessity for careful inspection to guard against incidental damage. A line with any broken wires should be replaced. A line showing any material reduction of metal area from abrasion should be replaced. A line showing kinking, crushing, or other damage should be replaced.

Replacement of lines based on normal life expectancy will provide some degree of safety, but it is important that such provisions do not cause any degree of laxity in sling line inspection.

Sling lines should be well lubricated. The field lubricant should be compatible with the original lubricant, and to this end the rope manufacturer should be consulted. The object of rope lubrication is to reduce internal friction and to prevent corrosion.

The following routine checks, as applicable, should be made at appropriate intervals:

1. Inspect welds in erecting mechanism for cracks and other signs of deformity.
2. Follow the manufacturer's instructions in checking hydraulic circuits before lowering operation. Make sure of adequate supply of hydraulic fluid.
3. Wire rope, including operating lines, raising lines, and guy lines, should be inspected for kinks, broken wires, or other damage. Make certain that guy lines are not fouled and that other lines are in place in sheave grooves before raising or lowering operation.
4. Check safety latches and guides in telescoping mast for free operation before lowering operation. Keep latches and guides clean and properly lubricated.
5. Check unit for level and check foundation and supports for correct placement before erecting operation.
6. Check lubrication of crown sheaves.
7. Check lubrication and condition of bearings in all sheaves, sprockets, etc.
8. Check folding ladders for free operation before lowering operation.
9. During drilling operations, it is advisable to make scheduled inspections of all bolted connections to ensure that they are tight.
10. The visual field inspection of derrick or mast and substructure procedure is recommended for use by operating personnel (or a designated representative) to the extent that its use satisfies conditions for which an inspection is intended. A sample report form for this inspection procedure can be found in API Standard 4F. Forms are also available from International Association of Drilling Contractors (IADC).

Splicing locks should be checked frequently for locking position or tightness, preferably on each tour during drilling operations. To develop its rated load capacity, the axis of the structure must be in alignment throughout its length. It is important that any splice mechanism or locks be maintained in such condition as to ensure structure alignment.

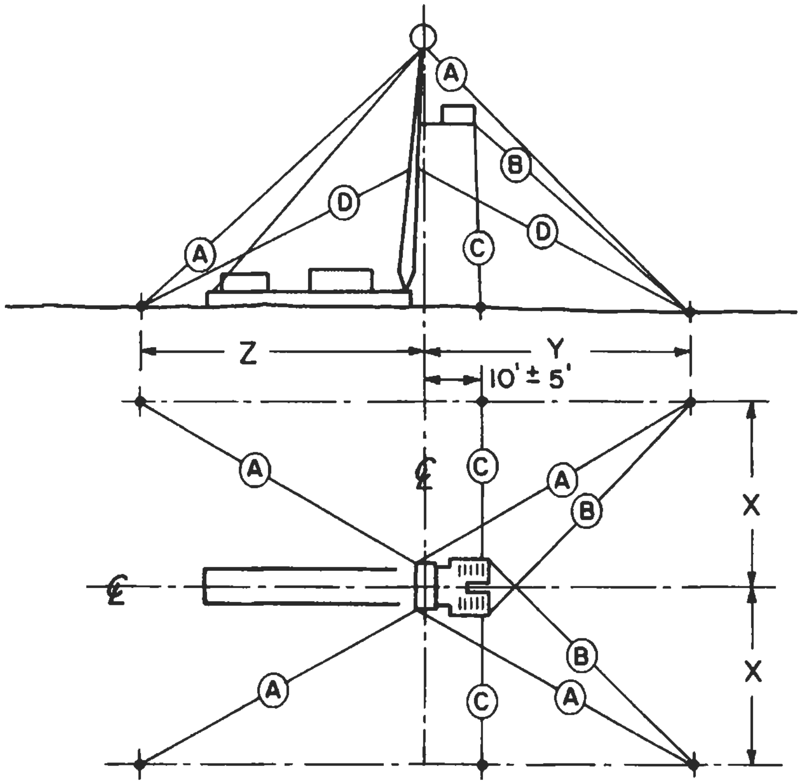
Guying for Portable Masts with Guy Lines

This recommendation is applicable for most conditions encountered in the use of this type mast. There will be exceptions where location clearance, ground conditions, or other unusual circumstances require special considerations. Figure 4-5 shows a recommended guying pattern that may be used under general conditions in the absence of an authorized API manufacturer's recommendations. Guy lines should be maintained in good condition, free from rust, corrosion, frays, and kinks. Old sand line is not recommended for guy lines.

All chains, boomers, clamps, and tensioning devices used in the guy lines shall satisfy the mast manufacturer's recommendations. In the absence of mast manufacturer's recommendations, the following minimum breaking strengths should be maintained: load guy lines—18 tons; external guy lines—12 tons; racking board guy lines—10 tons.

Guy Line Anchors for Portable Masts with Guy Lines

Guy line anchors including expanding anchors, concrete deadmen, or any other approved techniques are acceptable. The soil condition may determine the most applicable type. Recommendations for anchor design and testing are as follows:



A = Four crown to ground guys. Minimum guyline size recommended as 5/8" unless otherwise specified by mast manufacturer. Tensioning may be judged by catenary (sag). 6" catenary (approximately 1,000 lb tension) recommended on initial tensioning.

B = Two racking board to board guys. Minimum guyline size recommended is 9/16" unless otherwise specified by mast manufacturer. 12"–18" catenary (approximately 500 lb tension) recommended on initial tensioning.

C = Two additional racking board guys to ground. Recommended when winds are in excess of design magnitude (name plate rating) or when pipe set back exceeds rated racking capacity or when weather protection is used on board. Minimum guyline size recommended is 9/16" unless otherwise specified by mast manufacturer. 6"–12" catenary (approximately 1,000 lb tension) recommended on initial tensioning.

D = Two or four intermediate mast to ground guys. Recommended at option of mast manufacturer only. Minimum guyline size recommended is 5/8" unless otherwise specified by mast manufacturer. 6"–12" catenary (approximately 1,000 lb tension) recommended on initial tensioning.

CAUTION: WHEN THE TWO "A" LINES ON THE DRAWWORKS SIDE OF THE MAST ARE USED AS LOAD GUYS, THE MINIMUM LINE SIZE SHALL BE 3/4" IPS 6 × 31 CLASS OR BETTER, AND THE "Z" DIMENSION SHALL NOT BE LESS THAN 60'.

Figure 4-5. Recommended guying pattern—general conditions [3].

1. All guy line anchors should have a minimum breaking or pull-out strength at least equal to two times the maximum total calculated anchor load in the direction of the resultant load, and in the absence of manufacturer's recommendations, values in Table 4-3 are recommended.
2. Representative pull tests for the area, size, and type of anchor involved and made by recognized testing methods should be made and recorded. Records should be maintained by the installer for temporary anchors and by the lease owner for permanent anchors. Permanent anchors should be visually inspected prior to use. If damage or deterioration is apparent, the anchor should be tested.
3. Metal components of anchors should be galvanized or otherwise protected against corrosion. Sucker rods should not be used in anchor construction. Anchor location should be marked with a stake if projections aboveground are subject to bending or other abuse.
4. Anchor location should avoid old pit or other disturbed areas.

Mast Foundation for Portable Masts with Guy Lines

Foundations must consider ground conditions, location preparation, and supplemental footing as required to provide a stable base for mast erection and to support the mast during the most extreme loading encountered. A recommended location preparation to provide ground conditions for safe operations is shown in Figure 4-6.

Supplemental Footing

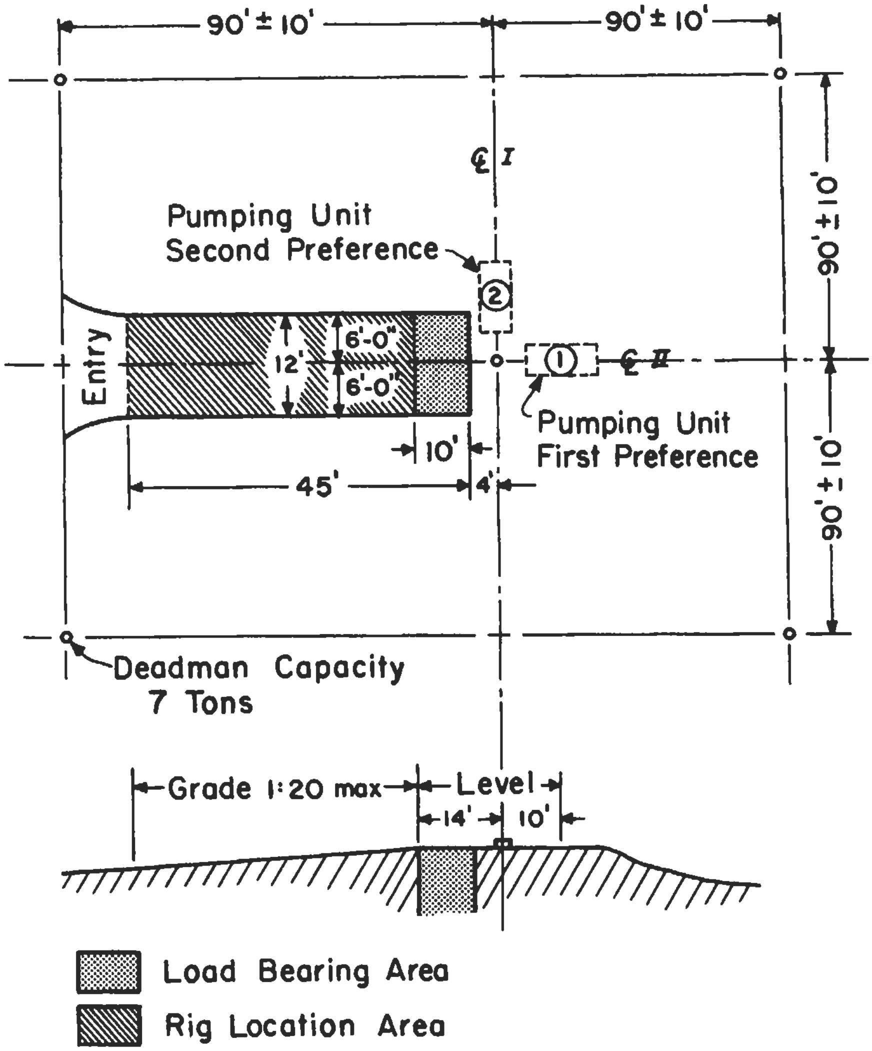
Supplemental footing must be provided to distribute the concentrated loads from the mast and mast mount to the ground. The manufacturer's load distribution diagram indicates the magnitude and location of these concentrated loads. If the manufacturer's load distribution diagram is not available, supplemental footing should be provided to carry the maximum hook load encountered, plus the gross weight of mast and mast mount weight during mast erection. The area

Table 4-3
Recommended Guyline Anchor Spacing and Loads
See Par. C. 16a and Figure C.1 [3]

1	2	3	4	5	6	7
	Doubles Mast		Singles Mast		Pole Mast	
Minimum Spacing X or Y Dimension See Fig. A.1, feet	Anchor Test Load, tons	Anchor Test Angle from Horiz. to Well Center Line	Anchor Test Load, tons	Anchor Test Angle from Horiz. to Well Center Line	Anchor Test Load, tons	Anchor Test Angle from Horiz. to Well Center Line
20	N.A.	N.A.	3.7	70°	7.0	67°
25	15.6	71°	—	—	—	—
30	13.7	67°	3.1	60°	—	—
40	11.0	60°	2.8	53°	4.0	49°
50	9.3	54°	2.7	45°	—	—
60	8.4	49°	2.7	45°	3.5	45°
70	7.8	45°	2.7	45°	—	—
80	7.4	45°	2.7	45°	3.0	45°
90	7.0	45°	2.7	45°	—	—

NOTE: *Preferred, X greater than Y. Limits, Y must not exceed 1.25X and Z must be equal to or less than 1.5Y, but not less than Y. (Fig. C.1)*

CAUTION: THE ADDITION OF WINDSCREENS OR THE RACKING OF PIPE ABOVE GROUND LEVEL CAN SIGNIFICANTLY INCREASE THE ABOVE ANCHOR REQUIREMENTS.



Load Bearing Area: Compacted sand or gravel requiring picking for removal or better base. Safe bearing capacity desired—Min., 8000 psf, level and drained. Rig Location Area: May grade away from well along centerline II at max. drop of 1:20. Should be level across grades parallel to centerline I. Safe bearing capacity desired—min., 6000 psf. Allow maneuvering entry for drive in or back in. Drainage of entire area required. See Table 4-3.

Figure 4-6. Portable mast location preparation [3].

and type of supplemental footing must ensure that the safe bearing capacity of soils on location is not exceeded.

Precautions and Procedures for Low-Temperature Operations

A survey of 13 drilling contractors operating 193 drilling rigs in northern Canada and Alaska indicated that there is a wide range of experience and operating practices under extremely low-temperature conditions. A sizable number of portable masts failed in the lowering or raising process in winter. Thus the exposure to low-temperature failures focuses on mast lowering and raising operations. Based on reports, however, this operation has been accomplished successfully in temperatures as low as -50°F. While the risk may be considerably greater because of the change in physical characteristics of steel at low temperatures, operators may carry on "normal" operations even at extremely low temperatures. This may be accomplished by closely controlled inspection procedures and careful handling and operation to reduce damage and impact loading during raising and lowering operations. At present, there seems to be no widely accepted or soundly supported basis for establishing a critical temperature for limiting the use of these oilfield structures. Experience in the operation of trucks and other heavy equipment exposed to impact forces indicates that -40°F may be the threshold of the temperature range, at which the risk of structural failure may increase rapidly. Precautionary measures should be more rigidly practiced at this point. The following recommended practices are included for reference:

1. To the extent possible, raising and lowering the mast at the "warmest" time of the day; use any sunlight or predictable atmospheric conditions. Consider the wind velocity factors.
2. Use any practical, available means, such as high-pressure steam timber bonfires, to warm sections of the mast.
3. Take up and loosen mast raising lines several times to assure the free movement of all parts.
4. Warm up engines and check the proper functioning of all machinery to assure that there will be no malfunctions that would result in sudden braking or jarring of the mast. Mast travel, once begun, must be slow, smooth, and continuous.
5. Inspection and repair provided in the section titled "Recommended Practice for Maintenance and Use of Drilling and Well Servicing Structures" are extremely critical under low-temperature conditions. Masts should be maintained in excellent condition.
6. In making field welds, the temperature of structural members should preferably be above 0°F. In the weld areas, steel should be preheated before welding or cutting operations.

Derrick Efficiency Factor

Derrick efficiency factor (DEF) is often used to rate or classify derrick or mast structural capacity [1,7,8]. The derrick efficiency factor is defined as a ratio of actual load to an equivalent load that is four times the force in the derrick leg carrying the greatest load. Thus the ratio is

$$\text{DEF} = \frac{\text{Actual load}}{\text{Equivalent load}}(100\%) \tag{4-7}$$

The derrick efficiency factor can be found for static (dead load) conditions and dynamic conditions. In this section, only the static conditions will be considered.

Example

Find the derrick efficiency factor (under static conditions) for a derrick that is capable of lifting a 600,000 lb drill string with a block and tackle that has eight working lines between the crown block and the traveling block. The crown block weighs 9,000 lb and the traveling block weighs 4,500 lb. Assume that there are no other tools hanging in the derrick. The dead line is attached at the bottom of leg A as shown in Figure 4-7.

Table 4-4 gives the calculations of the force in each leg of the derrick due to the centered load (i.e., 613,500 lb), the hoist-line load (i.e., the fast-line load;

Table 4-4
Example of Derrick Efficiency Factor Calculation [9]

	Force in Individual Derrick Legs (lbs)			
	A	B	C	D
Hoist-line load			37,781.25	37,381.25
Centered load	153,375.00	153,375.00	153,375.00	153,375.00
Dead-line load	75,562.50			
	228,937.50	153,375.00	191,156.25	191,156.25

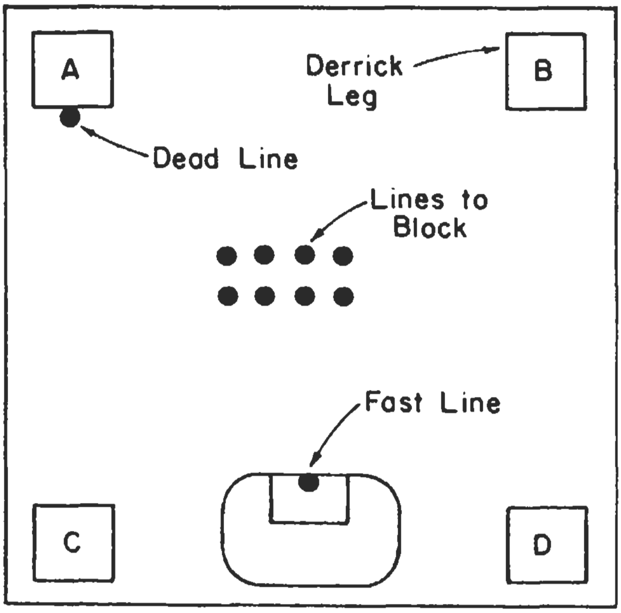


Figure 4-7. Projection of fast-line and deadline locations on rig floor [9].

75, 562.50 lb, divided by 2 since this load is shared by legs C and D) and the dead-line load at leg A (i.e., 75,562.50 lb).

The actual load on the derrick is the sum of the bottom row in Table 4-4.

The equivalent load is four times the force in leg A, which is the largest load of all four legs.

Thus,

$$\text{DEF} = \frac{764,625.00}{4(228,937.50)}(100\%)$$

$$= 83.50\%$$

HOISTING SYSTEM

A hoisting system, as shown in Figure 4-8, is composed of the drawworks, traveling block, crown block, extra line storage spool, various clamps, hooks, and wire rope.

Normally, a hoisting system has an even number of working lines between the traveling block and the crown block. The fast line is spooled onto the drawworks' hoisting drum. The dead line is anchored to the rig floor across from the drawworks. The weight indicator is a load cell incorporated in the dead line anchor.

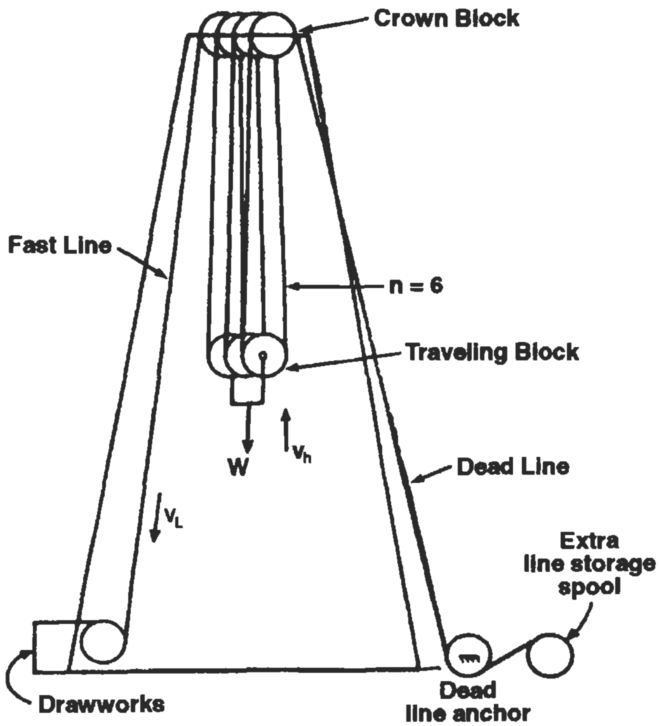


Figure 4-8. Schematic of simplified hoisting system on rotary drilling rig [9].

The mechanical advantage of the hoisting system is determined by the block and tackle and the number of working lines between the crown block and the traveling block [7].

Thus, for the static condition (i.e., no friction losses in the sheaves at the blocks), F_f (lb), the force in the fast line to hold the hook load, is

$$F_f = \frac{W_h}{j} \tag{4-8}$$

where W_h is the weight of the traveling block plus the weight of the drill string suspended in the hole, corrected for buoyancy effects in pounds; and j is the number of working lines between the crown block and traveling block. Under these static conditions, F_d (lb), the force in the dead line, is

$$F_d = \frac{W_h}{j} \tag{4-9}$$

The mechanical advantage (ma) under these static conditions is

$$\text{ma(static)} = \frac{W_h}{F_f} = j \tag{4-10}$$

When the hook load is lifted, friction losses in crown block and traveling block sheaves occur. It is normally assumed that these losses are approximately 2% deduction per working line. Under dynamic conditions, there will be an efficiency factor for the block and tackle system to reflect these losses. The efficiency will be denoted as the hook-to-drawwork efficiency (e_h). The force in the fast line under dynamic conditions (i.e., hook is moving) will be

$$F_f = \frac{W_h}{e_h j} \tag{4-11}$$

Equation 4-9 remains unchanged by the initiation of hook motion (i.e., the force in the dead line is the same under static or dynamic conditions). The mechanical advantage (ma) under dynamic conditions is

$$\text{ma (dynamic)} = e_h j \tag{4-12}$$

The total load on the derrick under dynamic conditions, F_t (lb), will be

$$F_t = W_h + \frac{W_h}{e_h j} + \frac{W_h}{j} + W_c + W_t \tag{4-13}$$

where W_c is the weight of the crown block, and W_t is the weight of tools suspended in the derrick, both in pounds.

Example

For dynamic conditions, find the total load on a derrick that is capable of lifting a 600,000-lb drill string with an 8-working line block and tackle. The crown block weighs 9,000 lb and the traveling block weighs 4,500 lb. Assume that there are no other tools hanging in the derrick and that the deadline is attached to the rig floor across from the drawworks in its normal position (see Figure 4-7). Assume the standard deduction of 2% per working line to calculate e_h.

$$e_h = 1.00 - 0.02(8) = 0.84$$

From Equation 4-13

$$F_t = 604{,}500 + \frac{600{,}000}{0.84(8)} + \frac{600{,}000}{8} + 9{,}000$$

$$= 604{,}500 + 89{,}286 + 75{,}000 + 9{,}000$$

$$= 786{,}786 \text{ lb}$$

Drawworks

The drawworks is the key operating component of the hoisting system. On most modern rotary drilling rigs, the prime movers either operate the hoisting drum within the drawworks or operate the rotary table through the transmission within the drawworks. Thus the drawworks is a complicated mechanical system with many functions [1,7].

Functions

The drawworks does not carry out only hoisting functions on the rotary drilling rig. In general, the functions of the drawworks are as follows:

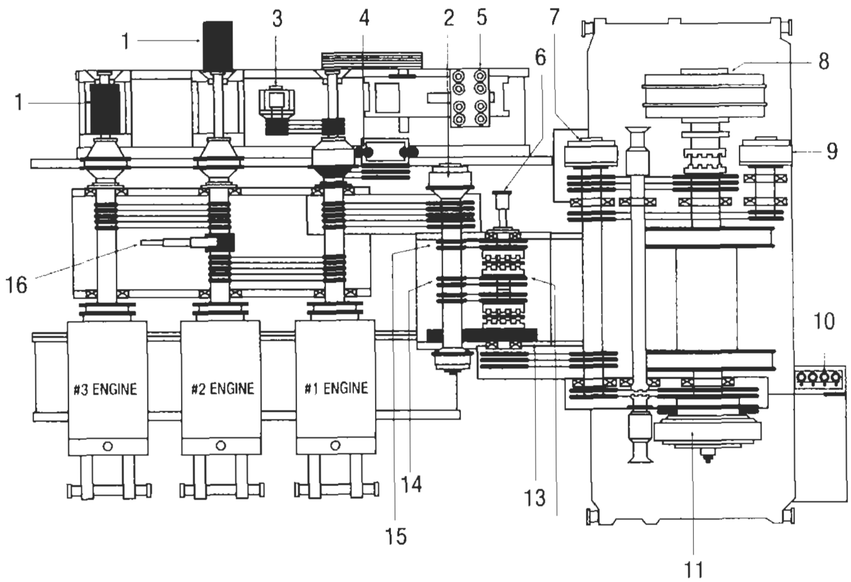
1. Transmit power from the prime movers (through the transmission) to its hoisting drum to lift drill string, casing string, or tubing string, or to pull in excess of these string loads to free stuck pipe.
2. Provide the braking systems on the hoist drum for lowering drill string, casing string, or tubing string into the borehole.
3. Transmit power from the prime movers (through the transmission) to the rotary drive sprocket to drive the rotary table.
4. Transmit power to the catheads for breaking out and making up drill string, casing string, and tubing string.

Figure 4-9 is a schematic of drawworks together with the prime mover power source.

Design

The drawworks basically contains the hoist drum, the transmissions, the brake systems, the clutch systems, rotary drive sprocket, and cathead. Figure 4-10 shows a schematic of the drawworks.

The power is provided to the drawworks by the prime movers at the master clutch (see Figure 4-9) and is transmitted to the master clutch shaft via sprockets and roller chain drives. The speed and the torque from the prime movers are controlled through the compound. The compound is a series of sprockets, roller chain drives, and clutches that allow the driller to control the power to the



1. Drive to pump
2. Master clutch
3. Generator
4. Air compressor
5. Washdown pump
6. Sand reel drive
7. Drum high air clutch
8. Auxiliary brake
9. Rotary drive air clutch countershaft
10. Driller's console
11. Drum low air clutch
12. High gear
13. Reverse gear
14. Intermediate gear
15. Low gear
16. Power flow selector*

*Note: This item is shown as a manually operated clutch. This, of course, on an actual rig would be air actuated.

Figure 4-9. Power train on a drawworks with accessories.

drawworks. The driller operates the compound and the drawworks (and other rig functions) from a driller's console (see Figure 4-11).

With the compound, the driller can obtain as many as 12 gears working through the drawworks transmission.

In Figure 4-11, the driller's console is at the left of the drawworks. Also, the hoisting drum and sand reel can be seen. The driller's brake control is between the driller's console and the drawworks to control the brake systems of the hoisting drum.

Hoisting Drum. The hoisting drum (usually grooved) is probably the most important component on the drawworks. It is through the drum that power is transmitted to lift the drill string with the drilling line (wire rope) wound on the drum. From the standpoint of power requirements for hoisting, the ideal

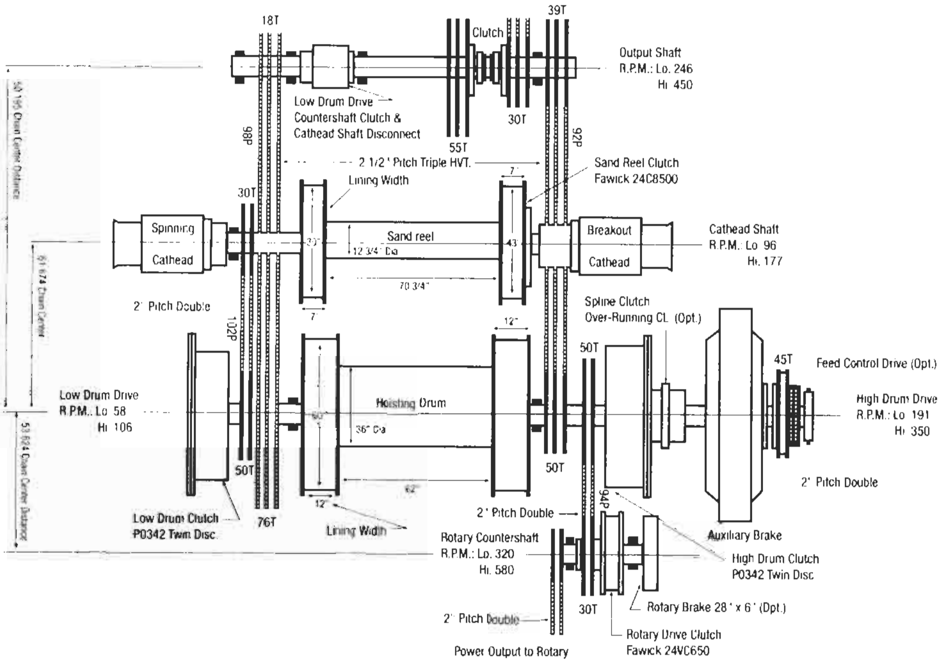
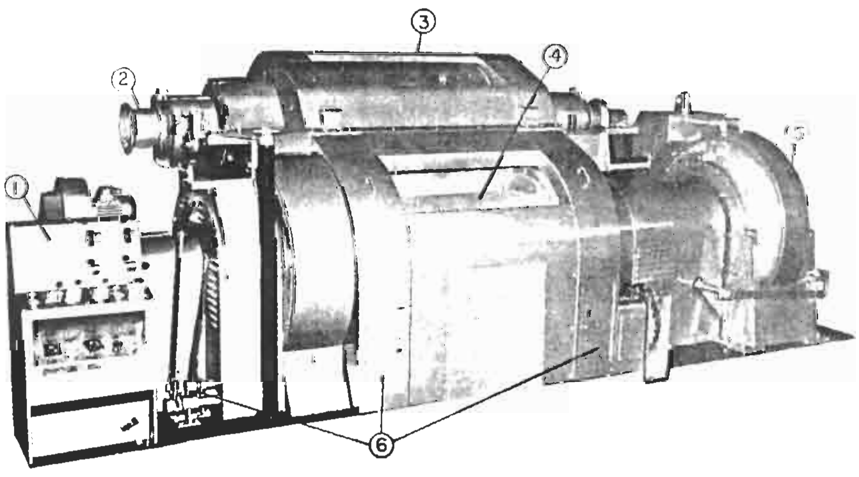


Figure 4-10. The drive group of a large DC electric rig. Note that this rig may be equipped with either two or three traction motors.



1. Driller's console. 2. Spinning cathead. 3. Sand reel. 4. Main drum (grooved). 5. Hydromatic brake. 6. Manual brakes (with inspection plates indicated).

Figure 4-11. The hoist on the rig floor.

drum would have a diameter as small as possible and a width as great as possible. From the standpoint of drilling line wear and damage, the hoisting drum would have the largest drum diameter. Therefore, the design of the hoisting drum must be compromised to obtain an optimum design. Thus, the hoist drum is usually designed to be as small as practical, but the drum is designed to be large enough to permit fast line speeds in consideration of operation and economy.

Often it is necessary to calculate the line-carrying capacity of the hoist drum. The capacity or length of drilling line in the first layer on the hoist drum L_1 (ft) is

$$L_1 = \frac{\pi}{12}(D + d)\frac{\ell}{d} \tag{4-14}$$

where D is the drum diameter, d is the line diameter, and ℓ is the hoisting drum length, all in inches.

The length of the second layer, L_2 (ft) is

$$L_2 = \frac{\pi}{12}(D + 3d)\frac{\ell}{d} \tag{4-15}$$

The length of the n^{th} layer L_n (ft) is

$$L_n = \frac{\pi}{12}[D + (2n - 1)d]\frac{\ell}{d} \tag{4-16}$$

where n is the total number of layers on the hoisting drum.

The total length of drilling line on the hoisting drum, L_t (ft), will be the sum of all the layers:

$$L_t = \frac{\pi}{12}(D + h)\frac{\ell h}{d^2} \tag{4-17}$$

where h is the hoist drum flange height, in inches.

Example

A hoist drum has an inside length of 48 in. and an outside diameter of 30 in. The outside diameter of the flange is 40 in. The drilling line diameter is 1 in. Find the total line capacity of the drum. The flange length is

$$h = \frac{40 - 30}{2} = 5 \text{ in.}$$

The total length (capacity) is

$$L_t = \frac{\pi}{12}(30 + 5)\frac{(48)(5)}{(1)^2}$$

$$= 2199 \text{ ft}$$

Transmission and Clutch. The transmission in the drawworks generally has six to eight speeds. Large rigs can have more gears in the drawworks transmission. More gearing capacity is available when the compound is used. This transmission uses a combination of sprockets and roller chain drives and gears to accomplish the change of speeds and torque from the prime movers (via the compound). The clutches used in the transmitting of prime mover power to the drawworks are jaw-type positive clutches and friction-type clutches. In modern drawworks, nearly all clutches are pneumatically operated from the driller's console. The driller's console also controls the shifting of gears within the drawworks.

Torque converters used in most drawworks are designed to absorb shocks from the prime movers or the driven equipment and to multiply the input torque. Torque converters are used in conjunction with internal combustion prime movers when these engines are used directly to drive the drawworks. More modern drawworks are driven by electric drives since such prime movers usually simplify the drawworks.

Brakes. The brake systems of the drawworks are used to slow and stop the movement of the large weights that are being lowered into the borehole. The brake system will be in continuous use when a round trip is made. The principal brake of the drawworks is the friction-type mechanical brake system. But when this brake system is in continuous use, it would generate a great deal of heat. Therefore, an auxiliary brake system is used to slow the lowering speeds before the friction-type mechanical brake system is employed to stop the lowering motion. Hydraulic brake system and electromagnetic brake system are the basic types of auxiliary brake systems in use. The hydraulic brake system uses fluid friction (much like a torque converter) to absorb power as equipment is lowered. The electromagnetic brake system uses two opposed magnetic fields supplied by external electrical current to control the speed of the hoisting drum. The auxiliary brake system can only control the speed of lowering and cannot be used to stop the lowering as does the mechanical friction-type brake system.

Catheads. The catheads are small rotating spools located on the sides of the drawworks. The cathead is used as a power source to carry out routine operations on the rig floor and in the vicinity of the rig. These operations include making up and breaking out drill pipe and casing, pulling single joints of pipe and casing from the pipe rack to the rig floor. The sand reel is part of this mechanism. This small hoisting drum carries a light wire rope line (sand line) through the crown to carry out pulling operations on the rig floor or in the vicinity of the rig.

Power Rating

In general, the drawworks is rated by its input horsepower. But it used to be rated by depth capability along with a specific size of drill pipe to which the depth rating pertains. The drawworks horsepower input required HP_{in} for hoisting operations is

$$HP_{in} = \frac{Wv_h}{33,000e_h e_m} \tag{4-18}$$

where W is the hook load in lb, v_h is the hoisting velocity of the traveling block in ft/min, e_h is the hook-to-drawworks efficiency, and e_m is the mechanical efficiency within the drawworks and coupling between the prime movers and the drawworks (usually taken as about 0.85).

Example

It is required that the drawworks input power be able to lift 600,000 lb at a rate of 50 ft/min. There are eight working lines between the traveling block and the crown block. Three input power systems are available: 1,100, 1,400, and 1,800 hp. Which of the three will be the most appropriate? The value of e_h is

$$e_h = 1.00 - 0.02(8) = 0.84$$

The input horsepower is

$$HP_{in} = \frac{600,000(50)}{33,000(0.84)(0.85)}$$

$$= 1273.2$$

The input power system requires 1400 hp.

Drilling and Production Hoisting Equipment [9,10]

Drilling and production hoisting equipment includes:

1. *Crown block sheaves and bearings:* The stationary pulley system at the top of the derrick or mast.
2. *Traveling blocks:* A heavy duty pulley system that hangs in the derrick and travels up and down with the hoisted tools. It is connected to the crown block with a wire rope that ultimately runs to the hoisting drum.
3. *Block-to-hook adapters:* A metal piece that attaches to the bottom of the traveling block and serves as the mount for the hook.
4. *Connectors and link adapters.*
5. *Drilling hooks:* The hook that attaches to the traveling block to connect the bail of the swivel.
6. *Tubing and sucker rod hooks:* Hooks connected to the traveling block for tubing and sucker-rod hoisting operations.
7. *Elevator links:* The elevator is a hinged clamp attached to the hook and is used to hoist drill pipe, tubing, and casing. The actual clamp is in a pair of links that in turn attaches to a bail supported on the hook.
8. *Casing, tubing and drill pipe elevators.*
9. *Sucker rod elevator.*
10. *Rotary swivel bail adaptors:* A bail adaptor that allows the bail of the swivel to be grasped and hoisted with elevators.
11. *Rotary swivels.* The swivel connecting the nonrotating hook and the rotating kelley while providing a nonrotating connection through which mud enters the kelley.
12. *Spiders:* The component of the elevator that latches onto the hoisted item.
13. *Deadline tiedowns:* The deadline is the nonmoving end of the wire rope from the hoisting down through the crown and traveling blocks. This end is anchored at ground level with a tiedown.
14. *Kelley spinners, when used as tension members:* An adapter between the swivel and the kelley that spins the kelley for rapid attachment and disattachment to joints of drill pipe.
15. *Rotary tables, as structural members:* The rotary table rotates to turn the drill string. It is also used to support the drill string during some phases of operation.

16. *Tension members of subsea handling equipment.*
17. *Rotary slips:* Wedging devices used to clamp the tool string into the rotary table. The wedging action is provided by friction.

Material Requirements

Castings. Steel castings used in the manufacture of the main load carrying components of the drilling and production hoisting equipment shall conform to ASTM A781: "Common Requirements for Steel and Alloy Castings for General Industrial Use," and either an individual material specification listed therein or a proprietary material specification that as a minimum conforms to ASTM A781.

Forgings. Steel forgings used in the manufacture of the main load carrying components of the equipment shall conform to ASTM A668: "Steel Forgings, Carbon and Alloy, for General Industrial Use" and ASTM A778: "Steel Forgings, General Requirements." A material specification listed in ASTM A788 or a proprietary specification conforming to the minimum requirements of ASTM A788 may be used.

Plates, Shapes, and Bar Stock. Structural material used in the manufacture of main load carrying components of the equipment shall conform to applicable ASTM or API specifications covering steel shapes, plates, bars, or pipe, or a proprietary specification conforming to the minimum requirements of applicable ASTM or appropriate standard. Structural steel shapes having a specified minimum yield strength less than 33,000 psi, or steel pipe having a specified minimum yield strength less than 35,000 psi shall not be used.

Design Rating and Testing

All hoisting equipment shall be rated in accordance with the requirements specified herein. Such ratings shall consist of a maximum load rating for all items, and a main-bearing rating for crown blocks, traveling blocks, and swivels. *The traveling block and crown block ratings are independent of wire rope size and strength.* Such ratings shall be calculated as specified herein and in accordance with good engineering practices. The ratings determined herein are intended to apply to new equipment only.

Maximum Load Rating. The maximum load ratings shall be given in tons (2,000-lb units). The size class designation shall represent the dimensional interchangeability and the maximum rated load of equipment specified herein. The recommended size classes are as follows (ton):

5	40	350
10	65	500
15	100	650
25	150	750
	250	1,000

For purpose of interchangeability contact radii shall comply with Table 4-5.

Maximum Load Rating Bases. The maximum load rating will be based on the design safety factor and the yield strength of the material. Crown block beams are an exception and shall be rated and tested in accordance with API Spec 4E: "Specification for Drilling and Well Servicing Structures."

Table 4-5
Recommended Hoisting Tool Contact Surface Radii
(All dimensions in inches) [9]

1		2		3		4		5		6		7		8		9	
Rating		Traveling Block & Hook Bail See Fig. 4-16								Hook & Swivel Bail See Fig. 4-17							
Short tons	Metric tons	A_1 Max in.	mm	A_2 Min in.	mm	B_1 Min in.	mm	B_2 Max in.	mm	E_1 Min in.	mm	E_2 Max in.	mm	F_1 Max in.	mm	F_2 Min in.	mm
25-40	22,7-36,3	2¾	69,85	2¾	69,85	3¼	82,55	3	76,20	2	50,80	1½	38,10	3	76,20	3	76,20
41-65	37,2-59	2¾	69,85	2¾	69,85	3¼	82,55	3	76,20	2	50,80	1¾	44,45	3½	88,90	3½	88.90
66-100	59,9-91	2¾	69,85	2¾	69,85	3¼	82,55	3	76,20	2¼	57,15	2	50,80	4	101,60	4	101,60
101-150	91,7-136	2¾	69.85	2¾	69,85	3¼	82,55	3	76,20	2½	63,50	2¼	57,15	4½	114,30	4½	114,30
151-250	137,1-227	4	101,60	4	101,60	3¼	82,55	3	76,20	2¾	69,85	2½	63,50	4½	114,30	4½	114,30
251-350	227,9-318	4	101,60	4	101,60	3¼	82,55	3	76,20	3	76,20	2¾	69,85	4½	114,30	4½	114,30
351-500	318,7-454	4	101,60	4	101,60	3½	88,90	3¼	82,55	3½	88,90	3¼	82,55	4½	114,30	4½	114,30
501-650	454,9-591	4	101,60	4	101,60	3½	88,90	3¼	82,56	3½	88,90	3¼	82,55	4½	114,30	4½	114.30
651-750	591,1-681	6	152,40	6	152,40	3¼	88,90	3¼	82,55	4¼	107,95	4	101,60	4½	114,30	4½	114,30
751-1000	681,9-908	6	152,40	6	152,40	6¼	158,75	6	152,40	5¼	133,35	5	127,00	5	127,00	5	127,00

10		11		12		13		14		15		16		17		18	
Elevator Link & Hook Link Ear See Fig. 4-18								Elevator Link & Elevator Link Ear See Fig. 4-18								Rating	
C_1 Max in.	mm	C_2 Min in.	mm	D_1 Min in.	mm	D_2 Max in.	mm	G_1 Max in.	mm	G_2 Min in.	mm	H_1 Min in.	mm	H_2 Max in.	mm	Short tons	Metric tons
1½	38,10	1¼	38,10	1¼	31,75	⅞	22,23			1	25,40			2	50,80	25-40	22,7-36,3
2½	63,50	2½	63,50	1¼	31,75	⅞	22,23			1	25,40			2	50,80	41-65	37,2-59
2½	63,50	2½	63,50	1½	38,10	1⅛	28,58			1	25,40			2	50,80	66-100	59,9-91
2½	63,50	2½	63,50	1½	38,10	1⅛	28,58	15/16	23,82	1½	38,10	2	50,80	2	50,80	101-150	91,7-136
4	101,60	4	101,60	1¾	44,45	1⅜	34,93	1 7/32	30,94	1⅞	47,63	2¾	69,85	2¾	69.85	151-250	137,1-227
4	101,60	4	101,60	1¾	44,45	1⅜	34,93	1 15/32	37,31	2	50,80	2¾	69,85	2¾	69.85	251-350	227,9-318
4	101,60	4¾	120,65	2¼	57,15	1⅞	47,63	1⅞	47,63	2	50,80	3¼	82,55	3¼	82,55	351-500	318,7-454
4	101,60	4¾	120,65	2¼	57,15	1⅞	47,63	2¼	57,15	2⅜	60,32	5	127,00	5	127,00	501-650	454,9-591
4	101,60	5	127,00	2½	63,50	2½	63,50	2¼	57,15	2⅜	60,32	5	127,00	5	127,00	651-750	591,1-681
4½	114,30	5	127,00	3	76,20	2¾	69,85	2¾	69,85	2⅞	73,03	6¼	158,75	6¼	158,75	751-1000	681,9-908

Crown Blocks. For crown block design, see API Specification 4F. Crown block sheaves and bearings shall be designed in accordance with API Specification 8A.

Spacer Plates. Spacer plates of traveling blocks, not specifically designed to lend support to the sheave pin, shall not be considered in calculating the rated capacity of the block.

Sheave Pins. In calculations transferring the individual sheave loads to the pins of traveling blocks, these loads shall be considered as uniformly distributed over a length of pin equal to the length of the inner bearing race, or over an equivalent length if an inner race is not provided.

Design Factor. The design safety factors shall be calculated as follows (see Figure 4-12) for the relationship between the design safety factor and rating:

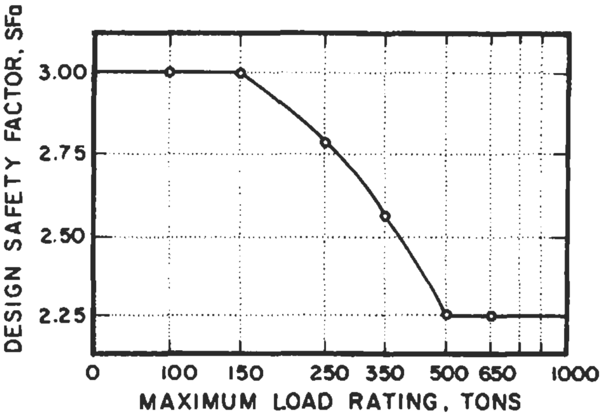


Figure 4-12. Design safety factor and rating relationships [9].

Calculated Rating (ton)	Yield Strength Design Safety Factor, SF_D
150 or less	3.00
Over 150 to 500	$3.00 - \dfrac{0.75(R^* - 150)}{350}$
Over 500	2.25

* R = rating in tons (2,000-lb units).

Mechanical Properties. The mechanical properties used for design shall be the minimum values allowed by the applicable material specification or shall be the minimum values determined by the manufacturer in accordance with the test procedures specified in ASTM A370: "Methods and Definitions for Mechanical Testing of Steel Products," or by mill certification for mill products. The yield point shall be used in lieu of yield strength for those materials exhibiting a yield point. Yield strength shall be determined at 0.2% offset.

Shear Strength. For the purpose of calculations involving shear, the ratio of yield strength in shear-to-yield strength in tension shall be 0.58.

Extreme Low Temperature. Maximum load ratings shall be established at room temperature and shall be valid down to 0°F (-18°C). *The equipment at rated loads when temperature is less than 0°F is not recommended unless provided for by the supplemental requirements. When the equipment is operating at lower temperatures, the lower impact absorbing characteristics of many steels must be considered.*

Test Unit. To assure the integrity of design calculations, a test shall be made on one full size unit that in all respects represents the typical product. For a family of units of the same design concept but of varying sizes, or ratings, one test will be sufficient to verify the accuracy of the calculation method used, if the item tested is approximately midway of the size and rating range of the family, and the test results are applicable equally to all units in that family. Significant changes in design concept or the load rating will require supportive load testing.

Parts Testing. Individual parts of a unit may be tested separately if the holding fixtures simulate the load conditions applicable to the part in the assembled unit.

Test Fixtures. Test fixtures shall support the unit (or part) in essentially the same manner as in actual service, and with essentially the same areas of contact on the load-bearing surfaces.

Test Procedure.

1. The test unit shall be loaded to the maximum rated load. After this load has been released, the unit shall be checked for useful functions. The useful function of all equipment parts shall not be impaired by this loading.
2. Strain gages may be applied to the test unit at all points where high stresses are anticipated, provided that the configuration of the units permits such techniques. The use of finite element analysis, models, brittle lacquer, etc., is recommended to confirm the proper location of strain gages. Three-element strain gages are recommended in critical areas to permit determination of the shear stresses and to eliminate the need for exact orientation of the gages.
3. The maximum test load to be applied to the test unit shall be $0.80 \times R \times SF_D$, but not less than 2R. Where R equals the calculated load rating in tons, SF_D is design safety factor.
4. The unit shall be loaded to the maximum test load carefully, reading strain gage values and observing for yielding. The test unit may be loaded as many times as necessary to obtain adequate test data.
5. Upon completion of the load test, the unit shall be disassembled and the dimensions of each part shall be checked carefully for evidence of yielding.

Determination of Load Rating. The maximum load rating may be determined from design and stress distribution calculations or from data acquired during a load test. Stress distribution calculations may be used to load rate the equipment only if the analysis has been shown to be within acceptable engineering allowances as verified by a load test on one member of the family of units of the same design. The stresses at that rating shall not exceed the allowed values. Localized yielding shall be permitted at areas of contact. In a unit that has been load tested, the critical permanent deformation determined by strain gages or other suitable means shall not exceed 0.002 in./in. If the stresses exceed the allowed values, the affected part or parts must be revised to obtain the desired rating. Stress distribution calculations may be used to load rate the equipment only if the analysis has been shown to be within acceptable engineering allowances as verified by a load test of one member of the family of units of the same design.

Alternate Test Procedure and Rating. Destructive testing may be used provided an accurate yield and tensile strength for the material used in the equipment has been determined. This may be accomplished by using tensile test specimens of the actual material and determining the yield strength to ultimate strength ratio. This ratio is then used to obtain the rating R (ton) of the equipment by the following equation:

$$R = \frac{\left(\frac{YS}{TS}\right) L_B}{SF_D} \tag{4-19}$$

where SF_D = yield strength design safety factor
YS = yield strength in psi
TS = ultimate tensile strength in psi
L_B = breaking load in tons

Load Testing Apparatus. The load apparatus used to simulate the working load on the test unit shall be calibrated in accordance with ASTM E-4: "Standard Methods of Verification of Testing Machines," so as to assure that the prescribed test load is obtained.

Block Bearing Rating. The bearing rating of crown and traveling blocks shall be determined by

$$W_b = \frac{NW_r}{714} \tag{4-20}$$

where W_b = calculated block bearing rating in tons
N = number of sheaves in the block
W_r = individual sheave bearing rating at 100 rpm for 3,000-hr minimum life for 90% of bearings in pounds

Swivel Bearing Rating. The bearing rating of swivels shall be determined by

$$W_s = \frac{W_r}{1600} \tag{4-21}$$

where W_s = calculated main thrust-bearing rating at 100 rpm in tons
W_r = main bearing thrust rating at 100 rpm for 3000-hr minimum life for 90% of bearings in pounds

Traveling Block Hood Eye Opening Rating. The traveling block top handling member shall, for 500-ton size class and larger, have a static load rating based on safety factors given in the preceding paragraph titled "Design Factor."

Design Changes. When any change made in material, dimension, or construction might decrease the calculated load or bearing ratings, the unit changed shall be rerated, and retested if necessary. Parts of the modified unit that remain unchanged from the original design need not be retested, provided such omission does not alter the test results of the other components.

Records. The manufacturer shall keep records of all calculations and tests. When requested by prospective purchaser or by a user of the equipment, the manufacturer shall examine the details of computations, drawings, tests, or other supporting data necessary to demonstrate compliance with the specification. It shall be understood that such information is for the sole use of the user or prospective purchaser for checking the API rating, and the manufacturer shall not be required to release the information from his custody.

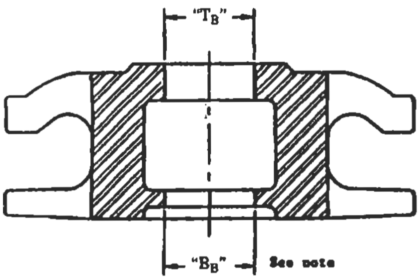
Elevators

Drill pipe elevators for taper shoulder and square shoulder weld-on tool joints shall have bore dimensions as specified in Table 4-6.

Table 4-6
Drill Pipe Elevator Bores
(All dimensions in inches) [9]

1	2	3		4		5		6		7
		Weld-On Tool Joints								
		Taper Shoulder				Square Shoulder				
Tool Joint Designation Reference	Drill Pipe Size and Style (All Weights and Grades)	Neck Diam. D_{TE} Max.[1]		Elev. Bore		Neck Diam. D_{SE} Max.[2]		Elev. Bore		Elev. Marking
		in.	mm	in.	mm	in.	mm	in.	mm	
NC 26 (2⅜ IF)	2⅜ EU	2 9/16	65.09	2 21/32	67.47	*		*		2⅜ EU
NC 31 (2⅞ IF)	2⅞ EU	3 3/16	80.96	3 9/32	83.34	3 3/16	80.96	3⅜	87.73	2⅞ EU
NC 38 (3½ IF)	3½ EU	3⅞	98.43	3 31/32	100.81	3⅞	98.43	4 1/16	103.19	3½ EU
NC 40 (4 FH)	3½ EU	3⅞	98.43	3 31/32	100.81	3⅞	98.43	4 1/16	103.19	
NC 40 (4 FH)	4 IU	4 3/16	106.36	4 5/32	101.86	4⅛	104.78	4 5/16	109.54	4 IU
NC 46 (4 IF)	4 EU	4½	114.30	4 25/32	121.44	4½	114.30	4 13/16	122.24	
	4½ IU	4 11/16	119.06	4 25/32	121.44	4⅝	117.48	4 13/16	122.24	4 EU
	4½ IEU	4 11/16	119.06	4 25/32	121.44	4⅝	117.48	4 13/16	122.24	4½ IU
4½ FH**	4½ IU	4 11/16	119.06	4 25/32	121.44	4⅝	117.48	4 13/16	122.24	4½ IEU
	4½ IEU	4 11/16	119.06	4 25/32	121.44	4⅝	117.48	4 13/16	122.24	
NC 50 (4½ IF)	4½ EU	5	127.00	5¼	133.35	5	127.00	5 5/16	134.94	4½ EU
	5 IEU	5¼	130.18	5¼	133.35	5⅛	130.18	5 5/16	134.94	5 IEU
5½ FH**	5 IEU	5⅛	130.18	5¼	133.35	5⅛	130.18	5 5/16	134.94	
5½ FH**	5 IEU	5 11/16	144.46	5 13/16	147.64	5 11/16	144.46	5⅞	149.23	5½ IEU
6⅝ FH	6⅝ IEU	6 57/64	175.02	7 1/32	178.66					6⅝

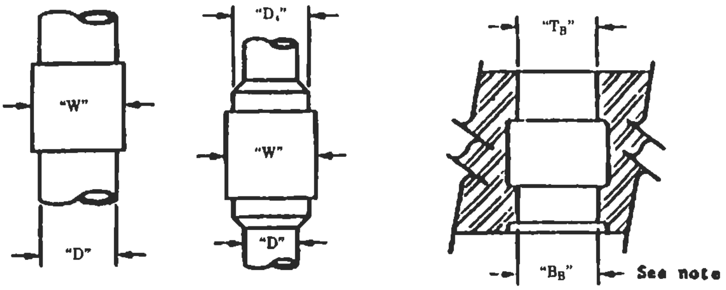
NOTE: *Elevators with the same bores are the same elevators.*
*Not manufactured.
**Obsolescent connection.
[1]Dimension D_{TE} from API Spec 7, Table 4.2.
[2]Dimension D_{SE} from API Spec 7, Appendix H.



Casing		Elevator Bores			
"D" Casing Dia.		"T_B" Top Bore ±1/64 ±.40 mm		"B_B" Bottom Bore +1/32 −1/64	+.79 −.40
in.	*mm*	in.	*mm*	in.	*mm*
4½	*114.30*	4.594	*116.69*	4.504	*116.69*
5	*127.00*	5.125	*130.18*	5.125	*130.18*
5½	*139.70*	5.625	*142.88*	5.625	*142.88*
6⅝	*168.28*	6.750	*171.45*	6.750	*171.45*
7	*177.80*	7.125	*180.98*	7.125	*180.98*
7⅝	*193.68*	7.781	*197.64*	7.781	*197.64*
7¾	*196.85*	7.906	*200.81*	7.908	*200.81*
8⅝	*219.08*	8.781	*223.04*	8.781	*223.04*
9⅝	*244.48*	9.781	*248.44*	9.781	*248.44*
10¾	*273.05*	10.938	*277.83*	10.938	*277.83*
11¾	*298.45*	11.938	*303.23*	11.938	*303.23*
13⅜	*339.73*	13.563	*344.50*	13.582	*344.50*
18	*406.40*	16.219	*411.96*	16.219	*411.96*
18⅝	*473.08*	18.875	*479.43*	18.875	*479.43*
20	*508.00*	20.281	*515.14*	20.281	*515.14*

NOTE: *Bottom bore "B_B" is optional, some elevator designs do not have a bottom bore.*

Table 4-6
(continued)



Tubing "D" Size O.D. in.	mm	Non-Upset Tubing "W" Collar Dia. in.	mm	"T$_B$" Top Bore +1,64 in.	±.40 mm	"B$_B$" Bottom Bore +1/32 -1/64 in.	+.79 -.40 mm	External Upset Tubing "W" Collar Dia. in.	mm	"D$_4$" Upset Dia. in.	mm	"T$_B$" Top Bore ±1/64 in.	±.40 mm	"B$_B$" Bottom Bore +1/32 -1/64 in.	+.79 -.40 mm
1.050	26,67	1.313	33,35	1.125	28,58	1.125	28,58	1.680	42,16	1.315	33,40	1.422	36,12	1.422	36,12
1.315	33,40	1.660	42,16	1.390	35,31	1.390	35,31	1.900	48,26	1.489	37,31	1.578	40,08	1.578	40,08
1.660	42,16	2.054	52,17	1.734	44,04	1.734	44,04	2.200	55.88	1.812	46,02	1.922	48,82	1.922	48.82
1.900	48,26	2.200	65,88	1.984	60,39	1.984	50,39	2.500	63,50	2.093	55,70	2.203	56,03	2.203	56,03
$2\frac{3}{8}$	60,32	2.875	73,03	2.453	62,31	2.453	62,31	3.063	77,80	2.593	65,89	2.703	68,58	2.703	68,58
$2\frac{7}{8}$	73,03	3.500	88,90	2.953	75,01	2.953	75,01	3.666	93,17	3.093	78,56	3.203	81.36	3.203	81,36
$3\frac{1}{2}$	88,90	4.250	107,95	3.678	90,88	3.578	90,88	4.500	114,30	3.750	96,25	3.859	98,02	3.859	98,02
4	101.60	4.750	120,65	4.078	103,58	4.078	109,58	5.000	127,00	4.250	107,95	4.359	110,74	4.359	110.74
$4\frac{1}{2}$	114,30	5.200	132,08	4.593	116,69	4.593	116,69	5.583	141,30	4.750	120,65	4.859	123,44	4.859	123.44

CAUTION: DO NOT USE EXTERNAL UPSET TUBING ELEVATORS ON NON-UPSET TUBING.

NOTE: *Bore "B$_B$" is optional, some elevator designs do not have a bottom bore.*

The permissible tolerance on the outside diameter immediately behind the tubing upset may cause problems with slip-type elevators.

Rotary Swivels

Rotary Swivel Pressure Testing. The assembled pilot model of rotary swivels shall be statically pressure tested. All cast members in the rotary swivel hydraulic circuit shall be pressure tested in production. This test pressure shall be shown on the cast member.

The test pressure shall be twice the working pressure up to 5,000 psi (incl.). For working pressures above 5,000 psi, the test pressure shall be one and one-half times the working pressure.

Swivel Gooseneck Connection. The angle between the gooseneck centerline and vertical shall be 15°. The swivel gooseneck connections shall be 2, $2\frac{1}{2}$, 3, $3\frac{1}{2}$, 4, or 5-in. nominal line pipe size as specified on the purchase order (see Figure 4-13). Threads on the gooseneck connection shall be internal line pipe threads conforming to API Standard 5B: "Threading, Gaging, and Thread Inspection of Casing, Tubing, and Line Pipe Threads." Rotary swivel gooseneck connections shall be marked with the size and type of thread, such as 3 API LP THD.

Rotary Hose Safety Chain Attachment. Swivels with gooseneck connections in 2 in. or larger shall have a suitable lug containing a $1\frac{1}{8}$-in. hole to accommodate the clevis of a chain having a breaking strength of 16,000 lb. The location of the lug is the choice of the manufacturer.

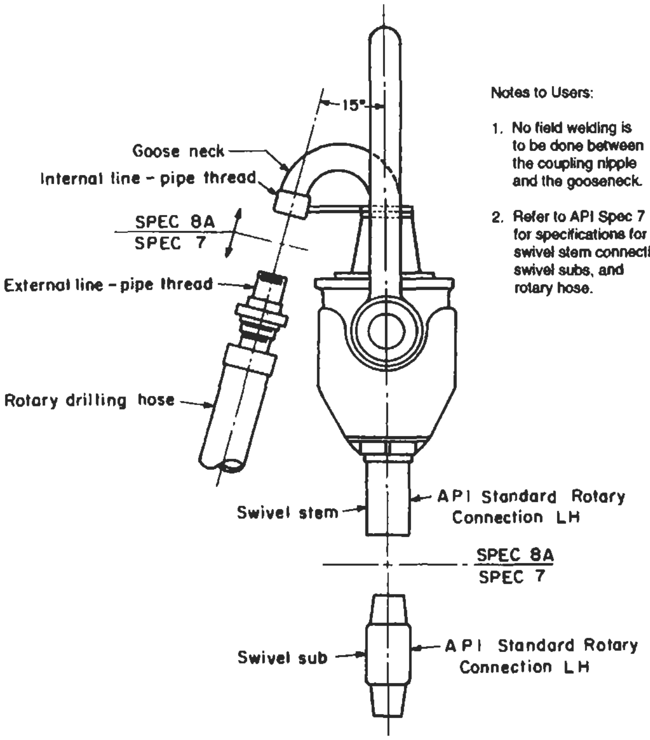


Figure 4-13. Rotary swivel connections [9].

Sheaves for Hoisting Blocks

The sheave diameter shall be the overall diameter D as shown in Figure 4-14. Sheave diameters shall, wherever practicable, be determined in accordance with recommendations given in the section titled "Wire Rope."

Grooves for drilling and casing line sheaves shall be made for the rope size specified by the purchaser. The bottom of the groove shall have a radius R, Table 4-7, subtending an arc of 150°. The sides of the groove shall be tangent to the ends of the bottom arc. Total groove depth shall be a minimum of 1.33d and a maximum of 1.75d (d is the nominal rope diameter shown in Figure 4-14).

In the same manner, grooves for sand-line sheaves shall be made for the rope size specified by the purchaser. The bottom of the groove shall have a radius R, Table 4-6, subtending an arc of 150°. The sides of the groove shall be tangent to the ends of the bottom arc. Total groove depth shall be a minimum of 1.75d and a maximum of 3d, and d is nominal rope diameter (see Figure 4-14, B).

Sheaves should be replaced or reworked when the groove radius decreases below the values shown in Table 4-8. Use sheave gages as shown in Figure 4-15. Figure 4-15 A shows a sheave with a minimum groove radius, and Figure 4-15 B shows a sheave with a tight groove.

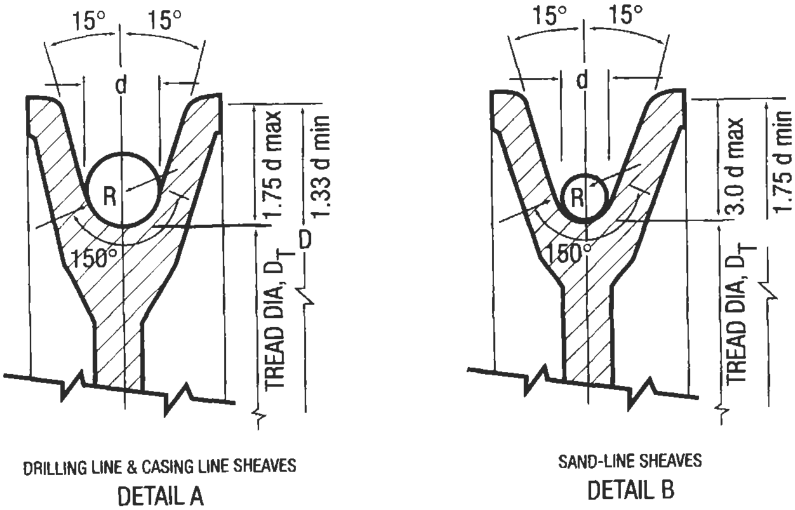


Figure 4-14. Sheave grooves [9].

Table 4-7
Groove Radii for New and Reconditioned Sheaves and Drums
(All dimensions in inches) [9]

1	2	1	2	1	2
Wire Rope Nominal Size	**Radii**	**Wire Rope Nominal Size**	**Radii**	**Wire Rope Nominal Size**	**Radii**
1/4	.137	1 5/8	.876	3 3/8	1.807
5/16	.167	1 3/4	.939	3 1/2	1.869
3/8	.201	1 7/8	1.003	3 3/4	1.997
7/16	.234	2	1.070	4	2.139
1/2	.271	2 1/8	1.137	4 1/4	2.264
9/16	.303	2 1/4	1.210	4 1/2	2.396
5/8	.334	2 3/8	1.273	4 3/4	2.534
3/4	.401	2 1/2	1.338	5	2.663
7/8	.468	2 5/8	1.404	5 1/4	2.804
1	.543	2 3/4	1.481	5 1/2	2.929
1 1/8	.605	2 7/8	1.544	5 3/4	3.074
1 1/4	.669	3	1.607	6	3.198
1 3/8	.736	3 1/8	1.664		
1 1/2	.803	3 1/4	1.731		

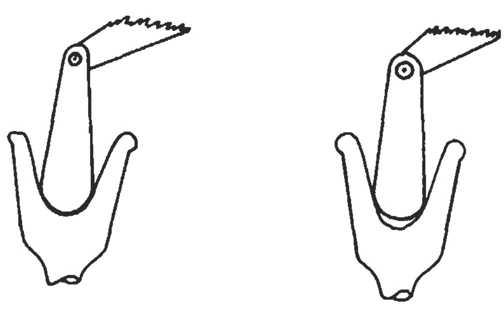
Standard Machine Tolerance

Contact Surface Radii

Figures 4-16, 4-17, 4-18, and Table 4-5 show recommended radii of hoisting tool contact surfaces. These recommendations cover hoisting tools used in drilling, and tubing hooks, but all other workover tools. Contact radii are intended to cover *only points of contact* between two elements and are not intended to define other physical dimensions of the connecting parts.

Table 4-8
Minimum Groove Radii for Worn Sheaves and Drums
(All dimensions in inches) [9]

1	2	1	2	1	2
Wire Rope Nominal Size	Radii	Wire Rope Nominal Size	Radii	Wire Rope Nominal Size	Radii
1/4	.129	1 5/8	.833	3 3/8	1.730
5/16	.160	1 3/4	.897	3 1/2	1.794
3/8	.190	1 7/8	.959	3 3/4	1.918
7/16	.220	2	1.019	4	2.050
1/2	.256	2 1/8	1.079	4 1/4	2.178
9/16	.288	2 1/4	1.153	4 1/2	2.298
5/8	.320	2 3/8	1.217	4 3/4	2.434
3/4	.380	2 1/2	1.279	5	2.557
7/8	.440	2 5/8	1.339	5 1/4	2.691
1	.513	2 3/4	1.409	5 1/2	2.817
1 1/8	.577	2 7/8	1.473	5 3/4	2.947
1 1/4	.639	3	1.538	6	3.075
1 3/8	.699	3 1/8	1.598		
1 1/2	.759	3 1/4	1.658		



DETAIL A DETAIL B

Figure 4-15. Use of sheave gages [9].

Inspection, Nondestructive Examination, and Compliance

Inspection. While work on the contract of the purchaser is being performed, the purchaser's inspector shall have reasonable access to the appropriate parts of the manufacturer's works concerning the manufacture of the equipment ordered hereunder. Inspection shall be made at the works prior to shipment, unless otherwise specified, and shall be conducted so as not to interfere unnecessarily with the works' operation or production schedules.

Nondestructive Examination. The manufacturer shall have a reasonable written nondestructive examination program to assure that the equipment manufactured is suitable for its intended use. If the purchaser's inspector desires to witness these operations, the manufacturer shall give reasonable notice of the time at which the examinations are to be performed.

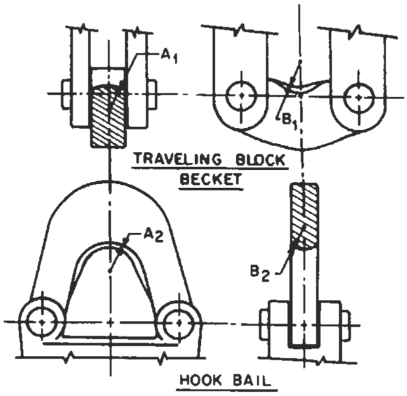


Figure 4-16. Traveling block and hook bail contact surface radii [9].

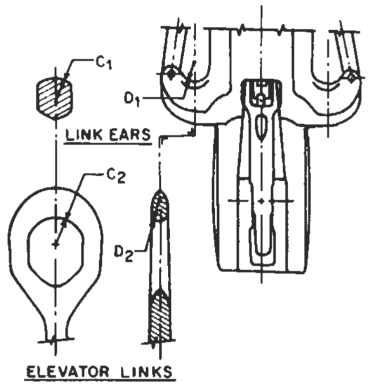


Figure 4-17. Elevator link and link ear contact surface radii [9].

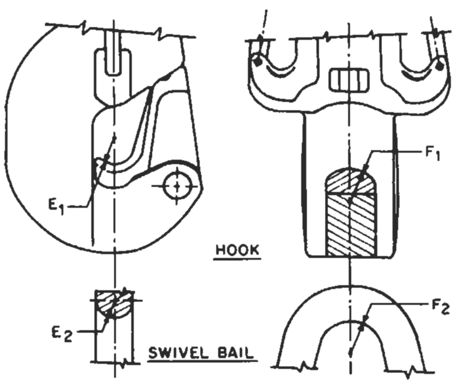


Figure 4-18. Hook and swivel bail contact surface radii [9].

Compliance. The manufacturer is responsible for complying with all of the provisions of the specification.

Supplementary Requirements

Magnetic Particle Examination. All accessible surfaces of the main load carrying components of the equipment shall be examined by a magnetic particle examination method or technique conforming to the requirements of ASTM E709: "Recommended Practice for Magnetic Particle Examination." Acceptance limits shall be as agreed upon by the manufacturer and the purchaser.

Liquid Penetrant Examination. All accessible surfaces of the main load carrying components of the equipment shall be examined by a liquid penetrant examination or technique conforming to the requirements of ASTM E165: "Recommended Practice for the Liquid Penetrant Examination Method." Acceptance limits shall be as agreed upon by the manufacturer and the purchaser.

Ultrasonic Examination. Main load carrying components of the equipment shall be ultrasonically examined in accordance with applicable ASTM standards. The extent of examination, method of examination, and basis for acceptance shall be agreed upon by the manufacturer and purchaser.

Radiographic Examination. Main load carrying components of the equipment shall be examined by means of gamma rays or x-rays. The procedure used shall be in accordance with applicable ASTM standards. Types and degrees of discontinuities considered shall be compared to the reference radiographs of ASTM as applicable. The extent of examination and the basis for acceptance shall be agreed upon by the manufacturer and purchaser.

Traceability. The manufacturer shall have reports of chemical analysis, heat treatment, and mechanical property tests for the main load carrying components of the equipment.

Welding. Where welding is involved in the critical load path of main load carrying components, recognized standards shall be used to qualify welders and procedures.

Extreme Low Temperature. Equipment intended for operation at temperatures below 0°F may require special design and/or materials.

Hoisting Tool Inspection and Maintenance Procedures

Inspection

Frequency of Inspection. Field inspection of drilling, production, and workover hoisting equipment in an operating condition should be made on a regular basis. A thorough on-the-job shutdown inspection should be made on a periodic basis, typically at 90 to 120-day intervals, or as special circumstances may require.

Critical loads may be experienced; for example, severe loads, impact loads such as jarring, pulling on stuck pipe, and/or operating at low temperatures. If in the judgment of the supervisor a critical load has occurred, or may occur, an on-the-job shutdown inspection equivalent to the periodic field inspection should be conducted before and after the occurrence of such loading. If critical loads are unexpectedly encountered, the inspection should be conducted immediately after such an occurrence.

When necessary, disassembly inspection of hoisting equipment should be made in a suitably equipped facility.

Methods of Inspection. Hoisting equipment should be inspected on a regular basis for cracks, loose fits or connections, elongation of parts, and other signs of wear, corrosion, or overloading. Any equipment showing cracks, excessive wear, etc., should be removed from service.

The periodic or critical load inspection in the field should be conducted by the crew with the inspector. For the periodic or critical load inspection, all foreign matter should be removed from surfaces inspected. Total field disassembly is generally not practical, and is not recommended, except as may be indicated in the detailed procedure for each tool.

Equipment, if necessary, should be disassembled in a suitably equipped facility and inspected for excessive wear, cracks, flaws, or deformation. Corrections should be made in accordance with the recommendations of the manufacturer. Before inspection, all foreign material, such as dirt, paint, grease, oil, scale, etc., should be removed from the inspected areas by a suitable method. The equipment should be disassembled as much as necessary to permit inspection of all load bearing parts, and the inspection should be made by trained, competent personnel.

Maintenance and Repairs

A regular preventive maintenance program should be established for all hoisting tools. Written maintenance procedures should be given to the crew or maintenance personnel. Maintenance procedures should be specified for each tool, as well as the specific lubricants to be used, and should be based on the tool manufacturer's recommendation. This recommended practice includes generalized procedures that are considered a minimum program. Care should be taken that instruction plates, rating plates, and warning labels are not missing, damaged, or illegible.

If repairs are not performed by the manufacturer, such repairs should be made in accordance with methods or procedures approved by the manufacturer. Minor cracks or defects, which may be removed without influence on safety or operation of the equipment, can be removed by grinding or filing. Following repair, the part should again be inspected by an appropriate method to ensure that the defect has been completely removed.

Antifriction bearings play an important part in the safe performance of the tool. The most likely requirements for bearing placement are very loose or bent cages (retainers), corrosion, abrasion, inadequate (or improper) lubrication, and spalling from fatigue. Excessive clearance may indicate improper adjustment or assembly and should be corrected. Repair of antifriction bearings should not be attempted by field or shop personnel. Consultation with the equipment manufacturer is recommended in case of unexplained or repeated bearing failure.

If the tool or part is defective beyond repair, it should be destroyed immediately.

Welding should not be done on hoisting tools without consulting the manufacturer. Without full knowledge of the design criteria, the materials used and the proper procedures (stress relieving, normalizing, tempering, etc.), it is possible to reduce the capacity of a tool sufficiently to make its continued use dangerous.

Inspection and maintenance (lubrication) of wire rope used in hoisting should be carried out on a regular basis. Wire rope inspection and maintenance recommendations are included in API RP 9B, "Application, Care and Use of Wire Rope for Oil Field Service" (see "Wire Rope").

Inspection and Maintenance Illustrations

Figures 4-19 through 4-36 are self-explanatory illustrations of generalized inspection and maintenance recommendations for each of the hoisting tools.

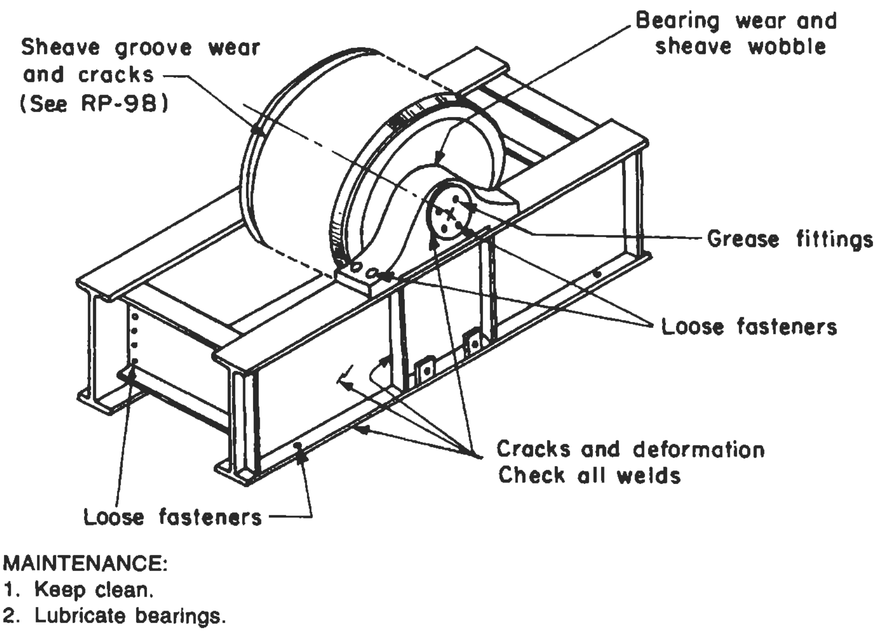
Wire Rope

Wire rope includes (1) bright (uncoated), galvanized, and drawn-galvanized wire rope of various grades and construction, (2) mooring wire rope, (3) torpedo lines, (4) well-measuring wire, (5) well-measuring strand, (6) galvanized wire guy strand, and (7) galvanized structural rope and strand [11,12].

Material

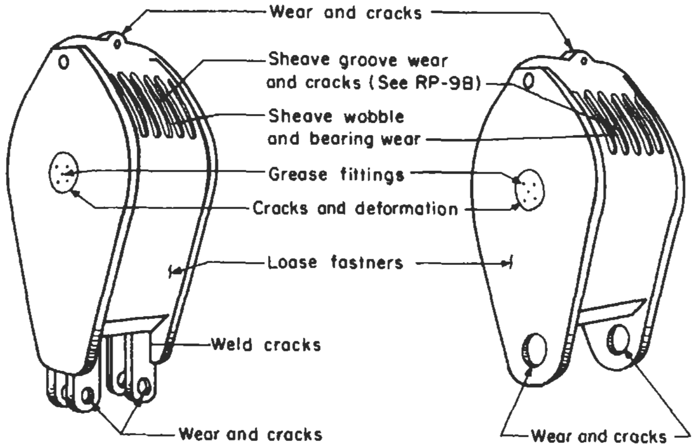
Wire used in the manufacture of wire rope is made from (1) acid or basic open-hearth steel, (2) basic oxygen steel, or (3) electric furnace steel. Wire tested before and after fabrication shall meet different tensile and torsional requirements as specified in Tables 4-9 and 4-10.

(text continued on page 563)



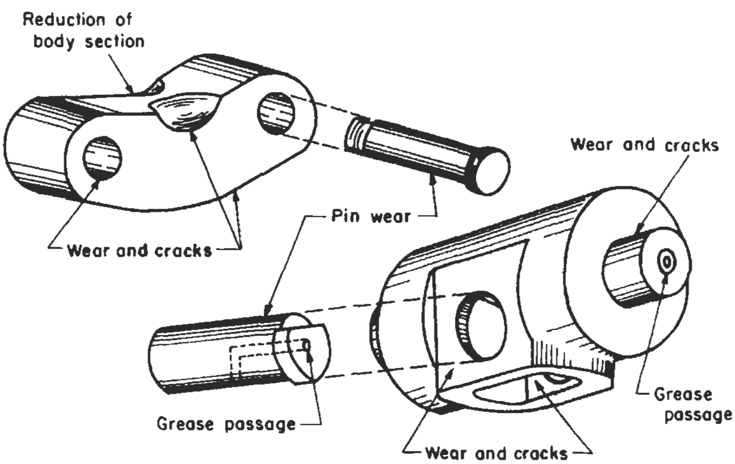
MAINTENANCE:
1. Keep clean.
2. Lubricate bearings.
3. Remove any rust and weather protect as required.
4. Check and secure all fasteners.

Figure 4-19. Crown block [10].



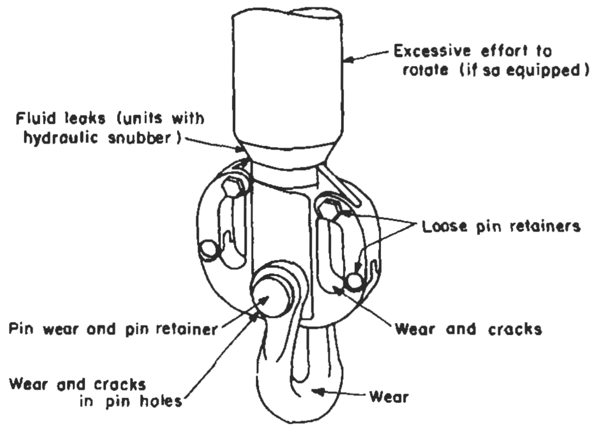
MAINTENANCE:
1. Keep clean.
2. Lubricate bearings.
3. Remove any rust and weather protect as required.
4. Check and secure all fasteners.

Figure 4-20. Traveling block [10].



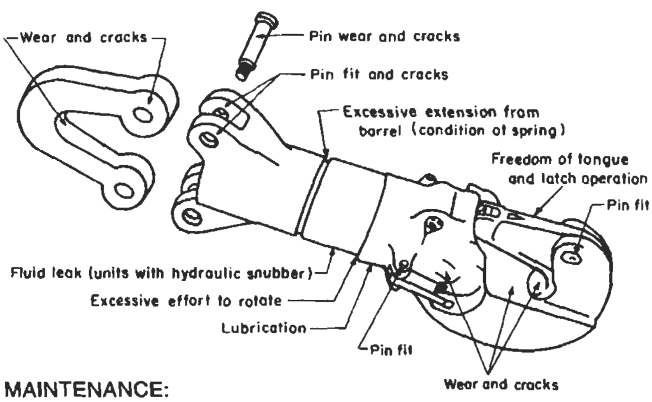
MAINTENANCE:
1. Keep clean.
2. Grease coat wear surface of clevis.
3. Remove any rust and weather protect as required.
4. Check and secure all pins.

Figure 4-21. Block-to-hook adapter [10].



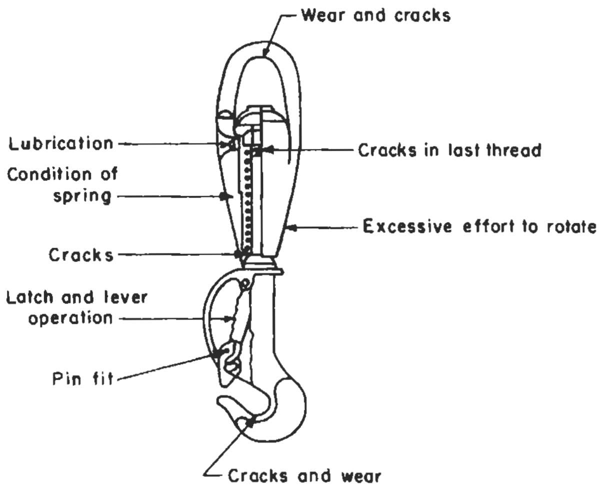
MAINTENANCE:
1. Keep clean.
2. Grease coat wear surfaces.
3. On units with hydraulic snubber check oil level and change oil at intervals recommended by manufacturer.
4. Oil pins not accessible to grease lubrication.
5. Remove any rust and weather protect as required.
6. Check and secure pins and fasteners.

Figure 4-22. Link adapter [10].



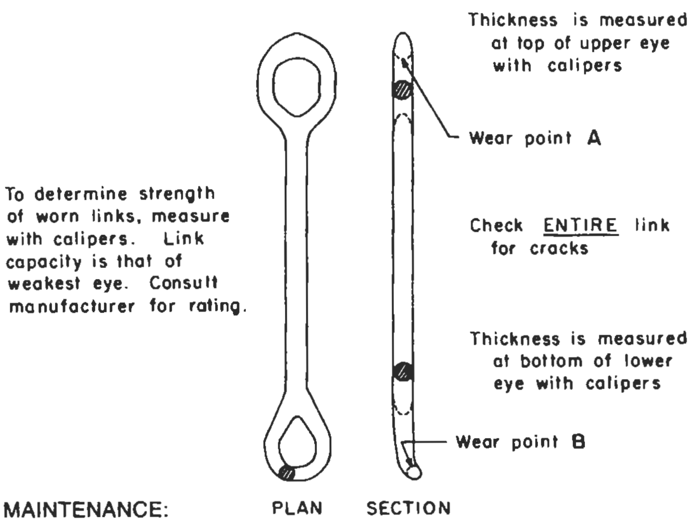
MAINTENANCE:
1. Keep clean.
2. Grease coat latching mechanism, link arms, and saddle.
3. Lube all grease fittings.
4. On units with hydraulic snubber check oil level and change oil at intervals recommended by manufacturer.
5. Oil pins not accessible to grease lubrication.
6. Remove any rust and weather protect as required.
7. Check and secure pins and fasteners.

Figure 4-23. Drilling hook [10].



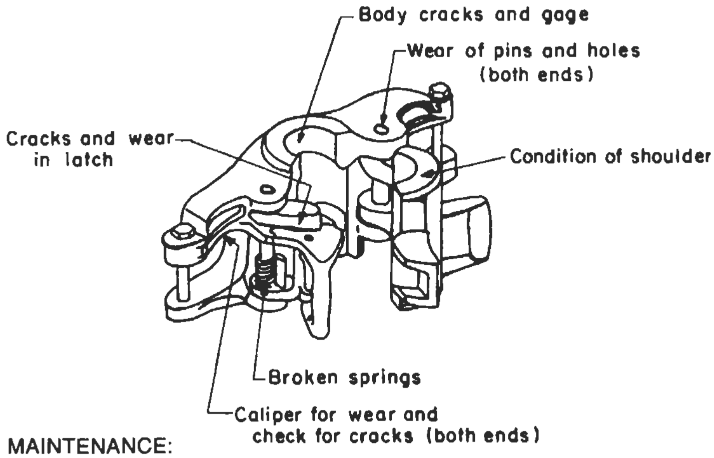
MAINTENANCE:
1. Keep clean.
2. Grease coat latching mechanism, hook, and bail throat.
3. Grease main bearing.
4. Oil pins not accessible to grease lubrication.
5. Remove any rust and weather protect as required.
6. Check and secure pins and fasteners.

Figure 4-24. Tubing and sucker rod hook [10].



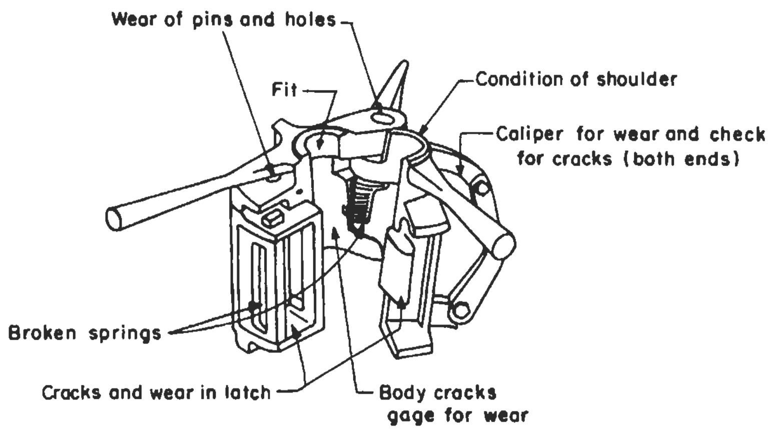
MAINTENANCE:
1. Keep clean.
2. Grease coat upper and lower eye wear surfaces.
3. Remove any rust and weather protect as required.

Figure 4-25. Elevator link [10].



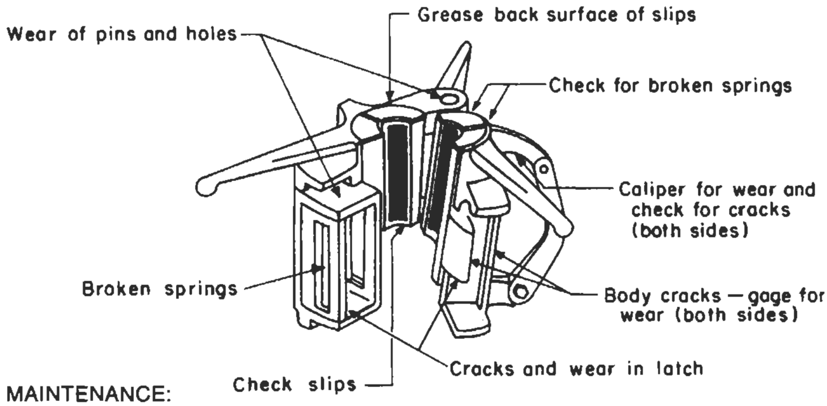
MAINTENANCE:
1. Keep clean.
2. Grease coat link arm wear surfaces, latch lug and bore seat on bottleneck elevators.
3. Lubricate hinge pin.
4. Remove any rust and weather protect as required.
5. Check and secure pins and fasteners.

Figure 4-26a. Casing, tubing, and drill pipe elevators, side door elevators [10].



MAINTENANCE:
1. Keep clean.
2. Grease coat link arm wear surfaces, latch lug and bore seat on bottleneck elevators.
3. Lubricate hinge pin.
4. Remove any rust and weather protect as required.
5. Check and secure pins and fasteners.

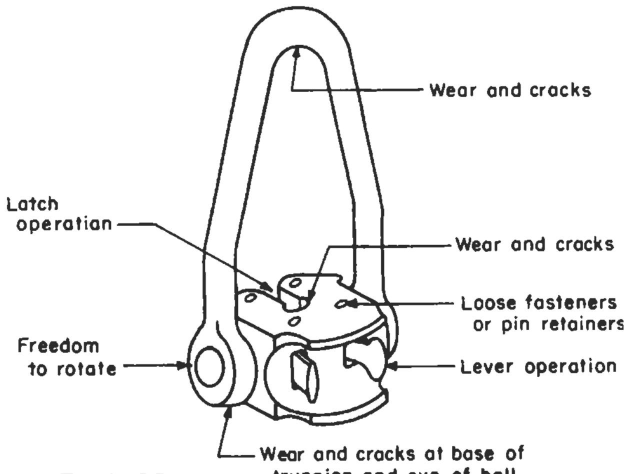
Figure 4-26b. Casing, tubing, and drill pipe elevators, and center latch elevators [10].



MAINTENANCE:

1. Keep clean.
2. Grease coat link arm wear surfaces and latch lug.
3. Lubricate hinge pin.
4. Remove any rust and weather protect as required.
5. Clean inserts. Replace when worn.
6. Tighten all loose fasteners.

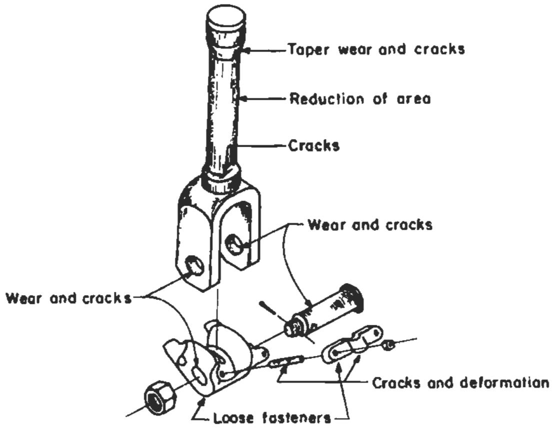
Figure 4-26c. Casing, tubing, and drill pipe elevators, slip type elevators [10].



MAINTENANCE:

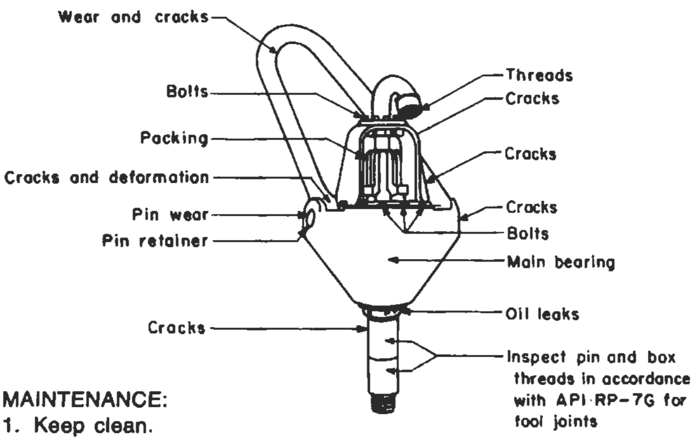
1. Keep clean.
2. Grease coat rod seating area, bail throat and latch mechanism.
3. Oil pins not accessible to grease lubrication.
4. Remove any rust and weather protect as required.
5. Check and secure pins and fasteners.

Figure 4-27. Sucker rod elevators [10].



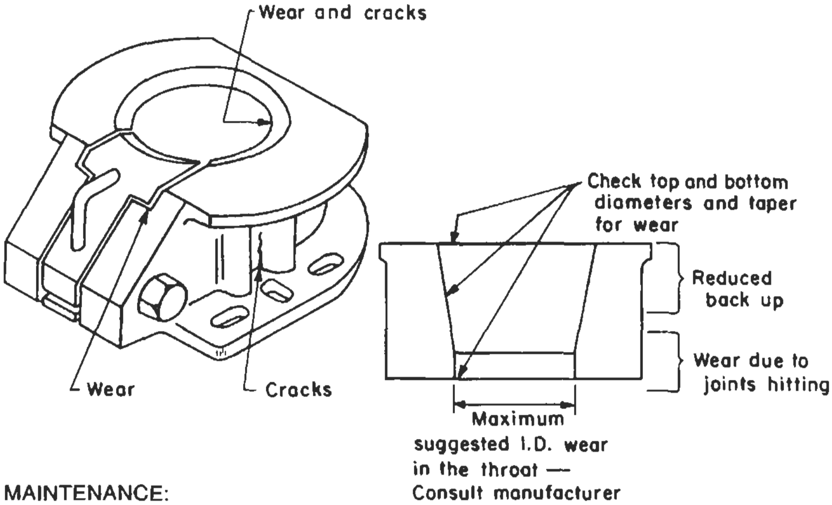
MAINTENANCE:
1. Keep clean.
2. Lubricate pivot and pin.
3. Remove any rust and weather protect as required.
4. Check and secure pins and fasteners.

Figure 4-28. Swivel bail adapter [10].



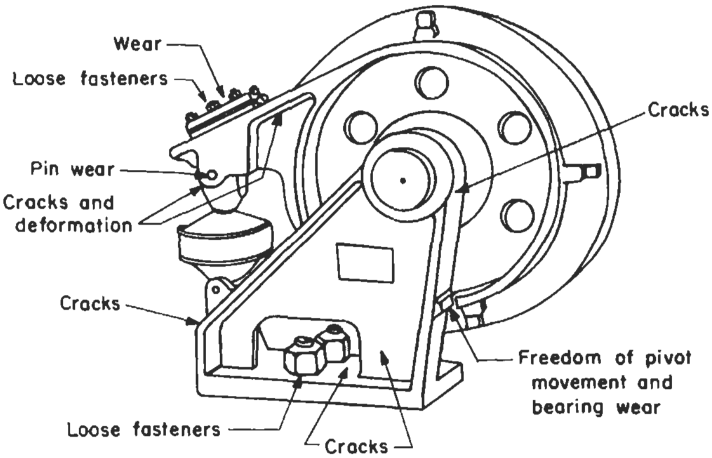
MAINTENANCE:
1. Keep clean.
2. Grease coat bail throat wear surface.
3. Lubricate bail pins, oil seals, upper bearing, and packing.
4. Check oil level as recommended by manufacturer.
5. Change oil at intervals recommended by manufacturer.
6. Remove any rust and weather protect as required.
7. Check and secure fasteners.
8. Protect threads at the gooseneck inlet and on the coupling nipple when not assembled during handling. Thread protection should be used.

Figure 4-29. Rotary swivel [10].



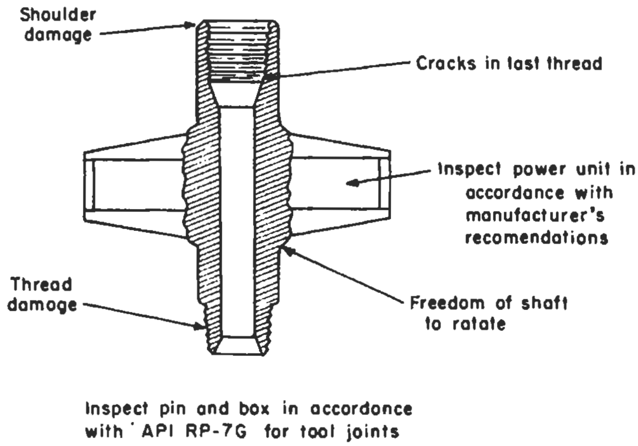
MAINTENANCE:
1. Keep clean.
2. Lubricate taper before each trip.
3. Remove any rust and weather protect as required.

Figure 4-30. Spider [10].



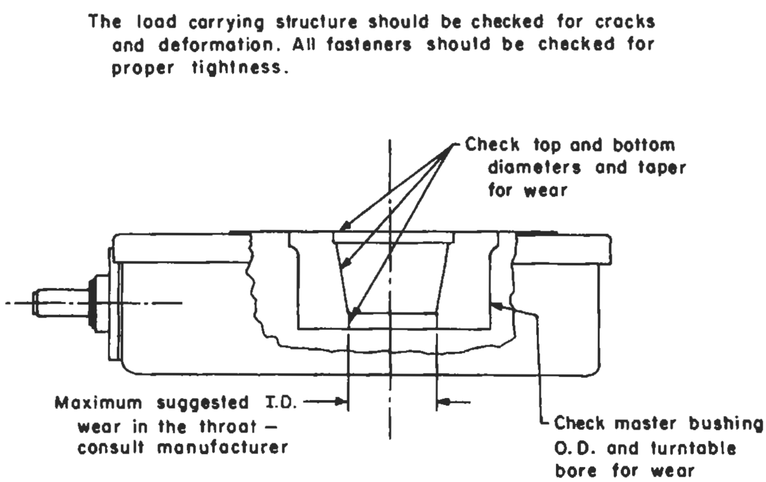
MAINTENANCE:
1. Keep clean.
2. Grease coat surface of wire line spool.
3. On units equipped with load cell for weight indicator, lubricate pivot bearing.
4. Remove any rust and weather protect as required.

Figure 4-31. Deadline anchor [10].



MAINTENANCE:
1. Keep clean.
2. Use thread compound on pin and box, and apply proper makeup torque in accordance with API RP-7G recommendations.
3. Maintain power unit in accordance with manufacturer's recommendations.
4. Remove any rust and weather protect as required.
5. Check and secure fasteners.

Figure 4-32. Kelly spinner [10].



MAINTENANCE:
1. Keep clean.
2. Remove any rust and weather protect as required.

Figure 4-33. Rotary table [10].

INSPECTION:
Heave compensator designs vary considerably among manufacturers, therefore, manufacturers' recommendations should be closely followed. In general, load carrying members should be checked for wear, cracks, flaws, and deformation. For compensators with integral traveling block and hook adaptor, the procedures defined in Figures. 4-16 and 4-17 apply to those parts of the assembly.

MAINTENANCE:
1. Keep clean.
2. Follow manufacturer's recommendations for specific unit.
3. For compensators with integral traveling block and hook adaptor, the procedures in Figures. 4-16 and 4-17 apply to those parts of the assembly.
4. Remove any rust and weather protect as required.

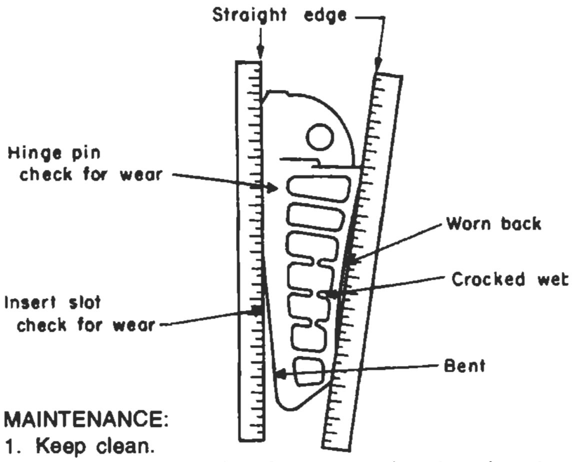
Figure 4-34. Heave compensator [10].

Ref. 4, p. 23

INSPECTION:
Tension members for sub sea handling equipment should be inspected according to the manufacturers' recommendations. In general, tension members should be checked for wear, cracks, reduction of area and elongation.

MAINTENANCE:
1. Tension members in sub sea handling equipment should be maintained in accordance with manufacturer's recommendations.

Figure 4-35. Tension members of subsea handling equipment [10].



MAINTENANCE:
1. Keep clean.
2. Check the insert slots for wear and replace inserts as required.
3. Lubricate hinge pin.
4. Remove any rust and protect as required.
5. Use straight edge to detect uneven wear or damage.
6. *Caution:* Do not use wrong size slips—Match pipe and slip size.

Figure 4-36. Rotary slips [10].

Table 4-9
Mechanical Properties of Individual Rope Wires (After Fabrication) [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)	(22)
		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength				
Wire Size Nominal Diameter		Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.
in.	mm	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.
0.010	0.25	17	76	17	76	241	20	89	21	93	222	21	93	23	102	202	23	102	25	111	176
0.011	0.28	20	89	22	98	219	23	102	25	111	202	26	116	28	125	183	28	125	30	133	160
0.012	0.30	24	107	26	116	201	28	125	30	133	185	31	138	33	147	168	33	147	35	156	146
0.013	0.33	28	125	30	133	185	33	147	35	156	171	36	160	38	169	155	39	173	41	182	135
0.014	0.36	33	147	35	156	172	38	169	40	178	159	42	187	44	196	144	45	200	47	209	126
0.015	0.38	38	169	40	178	161	44	196	46	205	148	48	214	50	222	134	52	231	54	240	117
0.016	0.41	43	191	45	200	150	50	222	52	231	139	55	245	57	254	126	59	262	62	276	109
0.017	0.43	49	218	51	227	142	56	249	58	258	130	61	271	65	289	118	66	294	70	311	103
0.018	0.46	55	245	57	254	134	62	276	66	294	124	69	307	73	325	112	74	329	78	347	97
0.019	0.48	60	267	64	285	126	70	311	74	329	117	77	342	81	360	106	83	369	87	387	93
0.020	0.51	67	298	71	316	120	77	342	81	360	110	85	378	89	396	100	92	409	96	427	87
0.021	0.53	74	329	78	347	114	85	378	89	396	105	94	418	98	436	95	100	445	106	471	83
0.022	0.56	81	360	85	378	109	94	418	98	436	101	102	454	108	480	90	110	489	116	516	79
0.023	0.58	89	396	93	414	105	102	454	108	480	96	112	498	118	525	86	121	538	127	565	75
0.024	0.61	97	431	101	449	100	111	494	117	520	92	122	543	128	569	82	132	587	138	614	71
0.025	0.64	104	463	110	489	96	120	534	126	560	88	133	592	139	618	78	142	632	150	667	68
0.026	0.66	113	503	119	529	92	130	578	136	605	85	143	636	151	672	75	154	685	162	721	65
0.027	0.69	122	543	128	569	88	140	623	148	658	82	154	685	162	721	72	166	738	174	774	63
0.028	0.71	131	583	137	609	86	151	672	159	707	79	166	738	174	774	69	178	792	188	836	60
0.029	0.74	140	623	148	658	83	162	721	170	756	76	177	787	187	832	66	191	850	201	894	58
0.030	0.76	150	667	158	703	80	173	770	181	805	73	190	845	200	890	64	205	912	215	956	55
0.031	0.79	160	712	168	747	77	184	818	194	863	71	203	903	213	947	62	218	970	230	1,023	54
0.032	0.81	171	761	179	796	74	196	872	206	916	68	215	956	227	1,010	59	232	1,032	244	1,085	51
0.033	0.84	181	805	191	850	72	209	930	219	974	67	229	1,019	241	1,072	57	247	1,099	259	1,1550	50
0.034	0.86	192	854	202	898	70	221	983	233	1,036	65	244	1,085	256	1,139	56	261	1,161	275	1,223	49
0.035	0.89	204	907	214	952	68	234	1,041	246	1,094	63	257	1,143	271	1,205	54	277	1,232	291	1,294	47
0.036	0.91	215	956	227	1,010	67	248	1,103	260	1,156	61	273	1,214	287	1,277	52	293	1,303	309	1,374	45
0.037	0.94	227	1,010	239	1,063	65	261	1,161	275	1,223	59	288	1,281	302	1,343	50	309	1,374	325	1,446	43
0.038	0.97	240	1,068	252	1,121	63	276	1,228	290	1,290	58	303	1,348	319	1,419	49	326	1,450	342	1,521	43
0.039	0.99	253	1,125	265	1,179	61	291	1,294	305	1,357	56	319	1,419	335	1,490	47	343	1,526	361	1,606	41
0.040	1.02	265	1,179	279	1,241	59	305	1,357	321	1,428	54	335	1,490	353	1,570	46	361	1,606	379	1,686	40
0.041	1.04	279	1,241	293	1,303	58	321	1,428	337	1,499	53	352	1,566	370	1,646	45	378	1,681	398	1,770	39
0.042	1.07	293	1,303	308	1,370	56	336	1,495	354	1,575	52	370	1,646	388	1,726	43	397	1,766	417	1,855	37
0.043	1.09	306	1,361	322	1,432	55	352	1,566	370	1,646	50	387	1,721	407	1,810	42	416	1,850	438	1,948	36
0.044	1.12	320	1,423	336	1,495	54	369	1,641	387	1,721	49	405	1,801	425	1,890	41	436	1,939	458	2,037	36
0.045	1.14	334	1,486	352	1,566	52	385	1,712	405	1,801	48	423	1,882	445	1,979	40	455	2,024	479	2,131	35
0.046	1.17	349	1,552	367	1,632	51	402	1,788	422	1,877	48	442	1,966	464	2,064	39	475	2,113	499	2,220	34
0.047	1.19	365	1,624	383	1,704	50	419	1,864	441	1,962	47	461	2,051	485	2,157	38	495	2,202	521	2,317	33
0.048	1.22	380	1,690	400	1,779	49	437	1,944	459	2,042	46	481	2,139	505	2,246	37	517	2,300	543	2,415	32
0.049	1.24	396	1,761	416	1,850	48	455	2,024	479	2,131	45	500	2,224	526	2,340	36	538	2,393	566	2,518	32
0.050	1.27	411	1,828	433	1,926	48	474	2,108	498	2,215	44	521	2,317	547	2,433	35	560	2,491	588	2,615	31
0.051	1.30	428	1,904	450	2,002	47	492	2,188	518	2,304	43	541	2,406	569	2,531	34	582	2,589	612	2,722	30
0.052	1.32	445	1,979	467	2,077	46	512	2,277	538	2,393	42	563	2,504	591	2,629	34	605	2,691	636	2,829	29
0.053	1.35	462	2,055	486	2,162	45	531	2,362	559	2,486	41	584	2,598	614	2,731	33	628	2,793	660	2,936	29
0.054	1.37	479	2,131	503	2,237	44	551	2,451	579	2,575	40	605	2,691	637	2,833	32	651	2,896	685	3,047	28
0.055	1.40	496	2,206	522	2,322	43	571	2,540	601	2,673	39	628	2,793	660	2,936	31	676	3,007	710	3,158	27
0.056	1.42	515	2,291	541	2,406	42	592	2,633	622	2,767	39	650	2,891	684	3,042	31	700	3,114	736	3,274	27
0.057	1.45	532	2,366	560	2,491	41	612	2,722	644	2,865	38	674	2,998	708	3,149	30	724	3,220	762	3,389	26
0.058	1.47	551	2,451	579	2,575	41	634	2,820	666	2,962	37	697	3,100	733	3,260	29	750	3,336	788	3,505	26
0.059	1.50	569	2,531	599	2,664	40	655	2,913	689	3,065	36	721	3,207	757	3,367	29	775	3,447	815	3,625	25
0.060	1.52	589	2,620	619	2,753	39	678	3,016	712	3,167	36	745	3,314	783	3,483	28	800	3,558	842	3,745	24
0.061	1.55	608	2,704	640	2,847	38	700	3,114	736	3,274	35	769	3,421	809	3,598	27	827	3,678	869	3,865	23
0.062	1.57	628	2,793	660	2,936	38	722	3,211	760	3,380	35	795	3,536	835	3,714	27	854	3,799	898	3,994	23
0.063	1.60	648	2,882	682	3,034	37	745	3,314	783	3,483	34	820	3,647	862	3,834	26	881	3,919	927	4,123	23
0.064	1.63	668	2,971	702	3,122	36	768	3,416	808	3,594	33	845	3,759	889	3,954	26	909	4,043	955	4,248	22
0.065	1.65	689	3,065	725	3,225	36	793	3,527	833	3,705	33	872	3,879	916	4,074	25	937	4,168	985	4,381	22
0.066	1.68	710	3,158	746	3,318	35	816	3,630	858	3,816	32	898	3,994	944	4,199	25	965	4,292	1,015	4,515	22
0.067	1.70	731	3,251	769	3,421	35	840	3,736	884	3,932	32	924	4,110	972	4,323	24	994	4,421	1,044	4,644	21
0.068	1.73	753	3,349	791	3,518	34	865	3,848	909	4,043	31	952	4,234	1,000	4,448	24	1,023	4,550	1,075	4,782	21
0.069	1.75	774	3,443	814	3,621	34	890	3,959	936	4,163	31	979	4,355	1,029	4,577	23	1,053	4,684	1,107	4,924	20
0.070	1.78	797	3,545	837	3,723	33	916	4,074	962	4,279	30	1,007	4,479	1,059	4,710	23	1,083	4,817	1,139	5,066	20
0.071	1.80	819	3,643	861	3,830	33	942	4,190	990	4,404	30	1,035	4,604	1,089	4,844	22	1,113	4,951	1,171	5,209	20
0.072	1.83	841	3,741	885	3,936	32	967	4,301	1,017	4,524	29	1,064	4,733	1,118	4,973	22	1,144	5,089	1,202	5,346	19
0.073	1.85	864	3,843	908	4,039	32	994	4,421	1,044	4,644	29	1,093	4,862	1,149	5,111	22	1,175	5,226	1,235	5,493	19
0.074	1.88	887	3,945	933	4,150	31	1,021	4,541	1,073	4,773	29	1,122	4,991	1,180	5,249	21	1,207	5,369	1,269	5,645	18
0.075	1.91	911	4,052	957	4,257	31	1,047	4,657	1,101	4,897	29	1,152	5,124	1,212	5,391	21	1,239	5,511	1,303	5,796	18
0.076	1.93	935	4,159	983	4,372	30	1,075	4,782	1,131	5,031	29	1,183	5,262	1,243	5,529	20	1,271	5,653	1,337	5,947	18
0.077	1.96	958	4,261	1,008	4,484	30	1,103	4,906	1,159	5,155	28	1,213	5,395	1,275	5,671	20	1,304	5,800	1,370	6,094	17
0.078	1.98	983	4,372	1,033	4,595	29	1,131	5,031	1,189	5,289	28	1,244	5,533	1,308	5,818	20	1,337	5,947	1,405	6,249	17
0.079	2.01	1,008	4,484	1,060	4,715	29	1,159	5,155	1,219	5,422	27	1,275	5,671	1,341	5,965	19	1,371	6,098	1,441	6,410	17

Table 4-9
(continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)	(22)
		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength				
Wire Size Nominal Diameter		Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.
in.	mm	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.
0.080	2.03	1,033	4,595	1,085	4,826	29	1,188	5,284	1,248	5,551	27	1,307	5,814	1,374	6,112	19	1,405	6,249	1,477	6,570	16
0.081	2.06	1,058	4,706	1,112	4,946	29	1,217	5,413	1,279	5,689	27	1,339	5,956	1,407	6,258	19	1,439	6,401	1,313	6,730	16
0.082	2.08	1,083	4,817	1,139	5,066	29	1,246	5,542	1,310	5,827	26	1,371	6,098	1,441	6,410	18	1,473	6,552	1,549	6,890	16
0.083	2.11	1,110	4,937	1,166	5,186	28	1,276	5,676	1,342	5,969	26	1,404	6,245	1,476	6,565	18	1,509	6,712	1,587	7,059	16
0.084	2.13	1,136	5,053	1,194	5,311	28	1,306	5,809	1,372	6,103	26	1,436	6,387	1,510	6,716	18	1,544	6,868	1,624	7,224	15
0.085	2.16	1,162	5,169	1,222	5,435	28	1,337	5,947	1,405	6,249	25	1,470	6,539	1,546	6,877	18	1,580	7,028	1,662	7,393	15
0.086	2.18	1,189	5,289	1,249	5,556	27	1,367	6,080	1,437	6,392	25	1,503	6,685	1,581	7,032	18	1,617	7,192	1,699	7,557	15
0.087	2.21	1,216	5,409	1,278	5,685	27	1,398	6,218	1,470	6,539	25	1,538	6,841	1,616	7,188	18	1,654	7,357	1,738	7,731	15
0.088	2.24	1,243	5,529	1,307	5,814	27	1,429	6,356	1,503	6,685	24	1,573	6,997	1,653	7,353	17	1,691	7,522	1,777	7,904	15
0.089	2.26	1,270	5,649	1,336	5,943	26	1,462	6,503	1,536	6,832	24	1,607	7,148	1,689	7,513	17	1,728	7,686	1,816	8,078	15
0.090	2.29	1,299	5,778	1,365	6,072	26	1,493	6,641	1,569	6,979	24	1,642	7,304	1,726	7,677	17	1,766	7,855	1,856	8,255	15
0.091	2.31	1,326	5,898	1,394	6,201	26	1,525	6,783	1,603	7,130	23	1,678	7,464	1,764	7,846	17	1,804	8,024	1,896	8,433	15
0.092	2.34	1,355	6,027	1,425	6,338	25	1,558	6,930	1,638	7,286	23	1,714	7,624	1,802	8,015	16	1,843	8,198	1,937	8,616	14
0.093	2.36	1,384	6,156	1,454	6,467	25	1,591	7,077	1,673	7,442	23	1,750	7,784	1,840	8,184	16	1,882	8,371	1,978	8,798	14
0.094	2.39	1,413	6,285	1,485	6,605	25	1,624	7,224	1,708	7,597	23	1,786	7,944	1,878	8,353	16	1,921	8,545	2,019	8,981	14
0.095	2.41	1,442	6,414	1,516	6,743	24	1,658	7,375	1,743	7,753	22	1,823	8,109	1,917	8,527	16	1,961	8,723	2,061	9,167	14
0.096	2.44	1,471	6,543	1,547	6,881	24	1,692	7,526	1,778	7,909	22	1,861	8,278	1,957	8,705	15	2,001	8,900	2,103	9,354	13
0.097	2.46	1,501	6,676	1,577	7,014	24	1,726	7,677	1,814	8,069	22	1,898	8,442	1,996	8,878	15	2,041	9,078	2,145	9,541	13
0.098	2.49	1,531	6,810	1,609	7,157	24	1,761	7,833	1,851	8,233	22	1,936	8,611	2,036	9,056	15	2,082	9,261	2,188	9,732	13
0.099	2.51	1,561	6,943	1,641	7,299	23	1,795	7,984	1,887	8,393	21	1,975	8,785	2,077	9,238	15	2,123	9,443	2,231	9,923	13
0.100	2.54	1,592	7,081	1,674	7,446	23	1,830	8,140	1,924	8,558	21	2,013	8,954	2,117	9,416	15	2,165	9,630	2,276	10,124	13
0.101	2.57	1,622	7,215	1,706	7,588	23	1,866	8,300	1,962	8,727	21	2,052	9,127	2,158	9,599	15	2,206	9,812	2,320	10,319	13
0.102	2.59	1,654	7,357	1,738	7,731	23	1,902	8,460	2,000	8,896	21	2,092	9,305	2,200	9,786	15	2,249	10,004	2,365	10,520	12
0.103	2.62	1,685	7,495	1,771	7,877	22	1,938	8,620	2,038	9,065	20	2,131	9,479	2,241	9,968	15	2,291	10,190	2,409	10,715	12
0.104	2.64	1,717	7,637	1,805	8,029	22	1,974	8,780	2,076	9,234	20	2,172	9,661	2,284	10,159	15	2,335	10,386	2,455	10,920	12
0.105	2.67	1,749	7,780	1,839	8,180	22	2,011	8,945	2,115	9,408	20	2,212	9,839	2,326	10,346	14	2,378	10,577	2,500	11,120	12
0.106	2.69	1,781	7,922	1,873	8,331	22	2,048	9,110	2,154	9,581	20	2,253	10,021	2,369	10,537	14	2,422	10,773	2,546	11,325	12
0.107	2.72	1,814	8,069	1,907	8,482	21	2,086	9,279	2,192	9,750	20	2,294	10,204	2,412	10,729	14	2,466	10,969	2,592	11,529	12
0.108	2.74	1,847	8,215	1,941	8,634	21	2,124	9,448	2,232	9,928	19	2,336	10,391	2,456	10,924	14	2,511	11,169	2,639	11,738	12
0.109	2.77	1,880	8,362	1,976	8,789	21	2,162	9,617	2,272	10,106	19	2,377	10,573	2,499	11,116	14	2,555	11,365	2,687	11,952	12
0.110	2.79	1,913	8,509	2,011	8,945	21	2,200	9,786	2,312	10,284	19	2,420	10,764	2,544	11,316	13	2,601	11,569	2,735	12,165	12
0.111	2.82	1,946	8,656	2,046	9,101	21	2,239	9,959	2,353	10,466	19	2,462	10,951	2,588	11,511	13	2,647	11,774	2,783	12,379	12
0.112	2.84	1,980	8,807	2,082	9,261	20	2,278	10,133	2,394	10,649	19	2,505	11,142	2,633	11,712	13	2,693	11,978	2,831	12,592	12
0.113	2.87	2,014	8,958	2,118	9,421	20	2,317	10,306	2,435	10,831	18	2,548	11,334	2,678	11,912	13	2,739	12,183	2,879	12,806	11
0.114	2.90	2,048	9,110	2,154	9,581	20	2,356	10,479	2,476	11,013	18	2,592	11,529	2,724	12,116	13	2,786	12,392	2,928	13,024	11
0.115	2.92	2,084	9,270	2,190	9,741	20	2,396	10,657	2,518	11,200	18	2,635	11,720	2,771	12,325	13	2,833	12,601	2,979	13,251	11
0.116	2.95	2,118	9,421	2,226	9,901	20	2,436	10,835	2,560	11,387	18	2,679	11,916	2,817	12,530	12	2,880	12,810	3,028	13,469	11
0.117	2.97	2,154	9,581	2,264	10,070	19	2,477	11,018	2,604	11,583	18	2,724	12,116	2,864	12,739	12	2,928	13,024	3,078	13,691	11
0.118	3.00	2,189	9,737	2,301	10,235	19	2,517	11,196	2,647	11,774	17	2,769	12,317	2,911	12,948	12	2,977	13,242	3,129	13,918	11
0.119	3.02	2,224	9,892	2,338	10,399	19	2,558	11,378	2,690	11,965	17	2,814	12,517	2,958	13,157	12	3,024	13,451	3,180	14,145	11
0.120	3.05	2,260	10,052	2,376	10,568	19	2,599	11,560	2,733	12,156	17	2,860	12,721	3,006	13,371	12	3,074	13,673	3,232	14,376	10
0.121	3.07	2,296	10,213	2,414	10,737	19	2,641	11,747	2,777	12,352	17	2,906	12,926	3,055	13,589	12	3,123	13,891	3,283	14,603	10
0.122	3.10	2,333	10,377	2,453	10,911	18	2,683	11,934	2,821	12,548	17	2,951	13,126	3,103	13,802	11	3,173	14,114	3,335	14,834	10
0.123	3.12	2,370	10,542	2,492	11,084	18	2,725	12,121	2,865	12,744	17	2,998	13,335	3,152	14,020	11	3,222	14,331	3,388	15,070	10
0.124	3.15	2,406	10,702	2,530	11,253	18	2,768	12,312	2,910	12,944	17	3,045	13,544	3,201	14,238	11	3,273	14,558	3,441	15,306	9
0.125	3.18	2,444	10,871	2,570	11,431	18	2,811	12,503	2,955	13,144	16	3,092	13,753	3,250	14,456	11	3,324	14,785	3,494	15,541	9
0.126	3.20	2,481	11,035	2,609	11,605	18	2,854	12,695	3,000	13,344	16	3,140	13,967	3,301	14,683	11	3,374	15,008	3,548	15,782	9
0.127	3.23	2,519	11,205	2,649	11,783	18	2,897	12,886	3,045	13,544	16	3,187	14,176	3,351	14,905	11	3,426	15,239	3,602	16,022	9
0.128	3.25	2,557	11,374	2,689	11,961	17	2,941	13,082	3,091	13,749	16	3,235	14,389	3,401	15,128	10	3,478	15,470	3,656	16,262	9
0.129	3.28	2,595	11,543	2,729	12,139	17	2,984	13,273	3,138	13,958	16	3,284	14,607	3,452	15,354	10	3,530	15,701	3,711	16,507	9
0.130	3.30	2,634	11,716	2,770	12,321	17	3,029	13,473	3,185	14,167	16	3,333	14,825	3,503	15,581	10	3,582	15,933	3,766	16,751	9
0.131	3.33	2,672	11,885	2,810	12,499	17	3,074	13,673	3,232	14,376	16	3,381	15,039	3,555	15,813	10	3,635	16,168	3,821	16,996	8
0.132	3.35	2,711	12,059	2,851	12,681	17	3,119	13,873	3,279	14,585	15	3,431	15,261	3,607	16,044	10	3,687	16,400	3,877	17,245	8
0.133	3.38	2,751	12,236	2,893	12,868	17	3,164	14,073	3,326	14,794	15	3,481	15,483	3,659	16,275	10	3,741	16,640	3,933	17,494	8
0.134	3.40	2,790	12,410	2,934	13,050	17	3,210	14,278	3,374	15,008	15	3,530	15,701	3,712	16,511	9	3,795	16,880	3,989	17,743	8
0.135	3.43	2,830	12,588	2,976	13,237	16	3,256	14,483	3,422	15,221	15	3,580	15,924	3,764	16,742	9	3,849	17,120	4,047	18,001	8
0.136	3.45	2,870	12,766	3,018	13,424	16	3,301	14,683	3,471	15,439	15	3,631	16,151	3,817	16,978	9	3,904	17,365	4,104	18,255	8
0.137	3.48	2,911	12,948	3,061	13,615	16	3,347	14,887	3,519	15,653	15	3,683	16,382	3,871	17,218	9	3,959	17,610	4,162	18,513	8
0.138	3.51	2,951	13,126	3,103	13,802	16	3,394	15,097	3,568	15,870	15	3,733	16,604	3,925	17,458	9	4,014	17,854	4,220	18,771	8
0.139	3.53	2,992	13,308	3,146	13,993	16	3,441	15,306	3,617	16,088	14	3,785	16,836	3,979	17,699	9	4,069	18,099	4,277	19,024	8
0.140	3.56	3,033	13,491	3,189	14,185	16	3,489	15,519	3,667	16,311	14	3,837	17,067	4,033	17,939	8	4,125	18,348	4,337	19,291	7
0.141	3.58	3,074	13,673	3,232	14,376	16	3,535	15,724	3,717	16,533	14	3,889	17,298	4,089	18,188	8	4,181	18,597	4,395	19,549	7
0.142	3.61	3,116	13,860	3,276	14,572	16	3,583	15,937	3,767	16,756	14	3,942	17,534	4,144	18,433	8	4,237	18,846	4,455	19,816	7
0.143	3.63	3,158	14,047	3,320	14,767	15	3,632	16,155	3,818	16,982	14	3,995	17,770	4,199	18,677	8	4,294	19,100	4,314	20,078	7
0.144	3.66	3,200	14,234	3,364	14,963	15	3,680	16,369	3,868	17,205	14	4,048	18,006	4,256	18,931	8	4,351	19,353	4,575	20,350	7
0.145	3.68	3,242	14,420	3,408	15,159	15	3,728	16,582	3,920	17,436	14	4,102	18,246	4,312	19,180	8	4,409	19,611	4,635	20,616	7
0.146	3.71	3,285	14,612	3,453	15,359	15	3,777	16,800	3,971	17,663	14	4,155	18,481	4,369	19,433	8	4,466	19,865	4,696	20,888	7
0.147	3.73	3,328	14,803	3,498	15,559	15	3,827	17,022	4,023	17,894	14	4,209	18,722	4,425	19,682	8	4,525	20,127	4,757	21,159	7
0.148	3.76	3,371	14,994	3,543	15,759	15	3,876	17,240	4,074	18,121	13	4,264	18,966	4,482	19,936	8	4,583	20,385	4,819	21,435	7
0.149	3.78	3,413	15,181	3,589	15,964	15	3,925	17,458	4,127	18,357	13	4,318	19,206	4,540	20,194	7	4,642	20,648	4,880	21,706	6

Table 4-9 (continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)	(22)
Wire Size Nominal Diameter		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength				
		Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.
in.	mm	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.
0.150	3.81	3,457	15,377	3,635	16,168	15	3,976	17,685	4,180	18,593	13	4,374	19,456	4,598	20,452	7	4,701	20,910	4,943	21,986	6
0.151	3.84	3,501	15,572	3,681	16,373	14	4,026	17,908	4,232	18,824	13	4,428	19,696	4,656	20,710	7	4,761	21,177	5,005	22,262	6
0.152	3.86	3,545	15,768	3,727	16,578	14	4,076	18,130	4,286	19,064	13	4,484	19,945	4,714	20,968	7	4,820	21,439	5,068	22,542	6
0.153	3.89	3,589	15,964	3,773	16,782	14	4,127	18,357	4,339	19,300	13	4,541	20,198	4,773	21,230	7	4,881	21,711	5,131	22,823	6
0.154	3.91	3,634	16,164	3,820	16,991	14	4,179	18,588	4,393	19,540	13	4,596	20,443	4,832	21,493	7	4,941	21,978	5,195	23,107	6
0.155	3.94	3,679	16,364	3,867	17,200	14	4,230	18,815	4,446	19,776	13	4,653	20,697	4,891	21,755	7	5,002	22,249	5,258	23,388	6
0.156	3.96	3,724	16,564	3,914	17,409	14	4,281	19,042	4,501	20,020	13	4,710	20,950	4,952	22,026	7	5,063	22,520	5,323	23,677	6
0.157	3.99	3,768	16,760	3,962	17,623	14	4,334	19,278	4,556	20,265	13	4,767	21,204	5,011	22,289	7	5,125	22,796	5,387	23,961	6
0.158	4.01	3,814	16,965	4,010	17,836	14	4,386	19,509	4,610	20,505	12	4,824	21,457	5,072	22,560	7	5,186	23,067	5,452	24,250	6
0.159	4.04	3,859	17,165	4,057	18,046	14	4,438	19,740	4,666	20,754	12	4,882	21,715	5,132	22,827	7	5,248	23,343	5,518	24,544	6
0.160	4.06	3,905	17,369	4,105	18,259	13	4.491	19,976	4,721	20,999	12	4,940	21,973	5,194	23,103	6	5,311	23,623	5,583	24,833	5
0.161	4.09	3,952	17,578	4,154	18,477	13	4,544	20,212	4,778	21,253	12	4,999	22,236	5,255	23,374	6	5,373	23,899	5,649	25,127	5
0.162	4.11	3,998	17,783	4,203	18,695	13	4,597	20,447	4,833	21,497	12	5,057	22,494	5,317	23,650	6	5,437	24,184	5,715	25,420	5
0.163	4.14	4,044	17,988	4,252	18,913	13	4,651	20,688	4,889	21,746	12	5,116	22,756	5,378	23,921	6	5,500	24,464	5,782	25,718	5
0.164	4.17	4,091	18,197	4,301	19,131	13	4,704	20,923	4,946	22,000	12	5,175	23,018	5,441	24,202	6	5,563	24,744	5,849	26,016	5
0.165	4.19	4,138	18,406	4,350	19,349	13	4,759	21,168	5,003	22,253	12	5,235	23,285	5,503	24,477	6	5,628	25,033	5,916	26,314	5
0.166	4.22	4,186	18,619	4,400	19,571	13	4,814	21,412	5,060	22,507	12	5,294	23,548	5,566	24,758	6	5,692	25,318	5,984	26,617	5
0.167	4.24	4,232	18,824	4,450	19,794	13	4,868	21,643	5,118	22,765	12	5,355	23,819	5,629	25,038	6	5,756	25,603	6,052	26,919	5
0.168	4.27	4,280	19,037	4,500	20,016	13	4,923	21,898	5,175	23,108	12	5,415	24,086	5,693	25,322	6	5,821	25,892	6,119	27,217	5
0.169	4.29	4,329	19,255	4,551	20,243	13	4,977	22,138	5,233	23,276	11	5,476	24,357	5,756	25,603	6	5,886	26,181	6,188	27,524	5
0.170	4.32	4,377	19,469	4,601	20,465	13	5,033	22,387	5,291	23,534	11	5,537	24,629	5,821	25,892	6	5,951	26,470	6,257	27,831	5
0.171	4.34	4,426	19,687	4,652	20,692	12	5,089	22,636	5,349	23,792	11	5,597	24,895	5,885	26,176	6	6,018	26,768	6,326	28,138	5
0.172	4.37	4,474	19,900	4,704	20,923	12	5,145	22,885	5,409	24,059	11	5,659	25,171	5,949	26,461	6	6,084	27,062	6,396	28,449	5
0.173	4.39	4,523	20,118	4,755	21,150	12	5,201	23,134	5,467	24,317	11	5,721	25,447	6,015	26,755	6	6,150	27,355	6,466	28,761	5
0.174	4.42	4,572	20,336	4,806	21,377	12	5,257	23,383	5,527	25,584	11	5,784	25,727	6,080	27,044	6	6,217	27,653	6,535	29,068	5
0.175	4.45	4,622	20,559	4,859	21,613	12	5,314	23,637	5,586	24,847	11	5,846	26,003	6,146	27,337	6	6,284	27,951	6,606	29,383	5
0.176	4.47	4,670	20,772	4,910	21,840	12	5,371	23,890	5,647	25,118	11	5,909	26,283	6,212	27,631	6	6,351	28.249	6,677	29,699	5
0.177	4.50	4,720	20,995	4,962	22,071	12	5,429	24,148	5,707	25,385	11	5,971	26,559	6,277	27,920	6	6,419	28,552	6,749	30,020	5
0.178	4.52	4,771	21,221	5,015	22,307	12	5,486	24.402	5,768	25,656	11	6,034	26,839	6,344	28,218	6	6,487	28,854	6,819	30,331	5
0.179	4.55	4,820	21,439	5,068	22,542	12	5,544	24,660	5,828	25,923	11	6,098	27,124	6,410	28,512	6	6,555	29,157	6,891	30,651	5
0.180	4.57	4,871	21,666	5,121	22,778	12	5,601	24,913	5,889	26,194	11	6,162	27,409	6,478	28,814	6	6,624	29,464	6,964	30,976	5
0.181	4.60	4,922	21,893	5,174	23,014	12	5,660	25,176	5,950	26,466	11	6,226	27,693	6,546	29,117	6	6,692	29,766	7,036	31,296	5
0.182	4.62	4,973	22,120	5,228	23,254	12	5,718	25,434	6,012	26,741	10	6,291	27,982	6,613	29,415	6	6,762	30,077	7,108	31,616	5
0.183	4.65	5,023	22,342	5,281	23,490	11	5,777	25,696	6,073	27,013	10	6,355	28,267	6,681	29,717	6	6,832	30,389	7,182	31,946	5
0.184	4.67	5,075	22,574	5,335	23,730	11	5,836	25,959	6,136	27,293	10	6,419	28,552	6,749	30,020	5	6,901	30,696	7,255	32,270	5
0.185	4.70	5,127	22,805	5,389	23,970	11	5,896	26,225	6,198	27,569	10	6,485	28,845	6,187	30,322	5	6,971	31,007	7,329	32,599	5
0.186	4.72	5,178	23,032	5,444	24,125	11	5,955	26,488	6,261	27,849	10	6,550	29,134	6,886	30,629	5	7,041	31,318	7,403	32,929	5
0.187	4.75	5,230	23,263	5,498	24,455	11	6,015	26,755	6,323	28,125	10	6,616	29,428	6,956	30,940	5	7,113	31,639	7,477	33,258	5
0.188	4.78	5,283	23,499	5,553	24,700	11	6,074	27,017	6,386	28,405	10	6,683	29,726	7,025	31,247	5	7,184	31,954	7,552	33,591	5
0.189	4.80	5,335	23,730	5,609	24,949	11	6,135	27,288	6,449	28,685	10	6,748	30,015	7,094	31,554	5	7,255	32,270	7,627	33,925	5
0.190	4.83	5,387	23,961	5,663	25,189	11	6,195	27,555	6,513	28,970	10	6,815	30,313	7,165	31,870	5	7,326	32,586	7,702	34,258	5
0.191	4.85	5,441	24,202	5,720	25,443	11	6,257	27,831	6,577	29,254	10	6,882	30,611	7,234	32,177	5	7,398	32,906	7,778	34,597	5
0.192	4.88	5,493	24,433	5,775	25,687	11	6,317	28,098	6,641	29,539	10	6,949	30,909	7,305	32,493	5	7,470	33,227	7,854	34,935	4
0.193	4.90	5,547	24,673	5,831	25,936	11	6,378	28,369	6,706	29,828	10	7,016	31,207	7,376	32,808	5	7,543	33,551	7,929	35,268	4
0.194	4.93	5,600	24,909	5,888	26,190	11	6,440	28,645	6,770	30,113	10	7,084	31,510	7,448	33,129	5	7,615	33,872	8,005	35,606	4
0.195	4.95	5,654	25,149	5,944	26,439	11	6,501	28,916	6,835	30,402	10	7,152	31,812	7,518	33,440	5	7,688	34,196	8,082	35,949	4
0.196	4.98	5,708	25,389	6,000	26,688	11	6,564	29,197	6,900	30,691	10	7,220	32,115	7,590	33,760	5	7,762	34,525	8,160	36,296	4
0.197	5.00	5,761	25,625	6,057	26,942	10	6,626	29,472	6,966	30,985	10	7,288	32,417	7,662	34,081	5	7,835	34,850	8,237	36,638	4
0.198	5.03	5,816	25,870	6,114	27,195	10	6,689	29,753	7,032	31,278	10	7,357	32,724	7,735	34,405	5	7,909	35,179	8,315	36,985	4
0.199	5.05	5,870	26,110	6,172	27,453	10	6,751	30,028	7,097	31,567	10	7,427	33,035	7,807	34,726	5	7,983	35,508	8,393	37,332	4
0.200	5.08	5,925	26,354	6,229	27,707	10	6,814	30,309	7,164	31,865	10	7,496	33,342	7,880	35,050	5	8,057	35,838	8,471	37,679	4
0.201	5.11	5,980	26,599	6,286	27,960	10	6,877	30,589	7,229	32,155	10	7,565	33,649	7,953	35,375	5	8,132	36,171	8,550	38,030	4
0.202	5.13	6,035	26,844	6,345	28,223	10	6,940	30,869	7,296	32,453	10	7,634	33,956	8,026	35,700	5	8,208	36,509	8,628	38,377	4
0.203	5.16	6,091	27,093	6,403	28,481	10	7,004	31,154	7,364	32,755	10	7,704	34,267	8,100	36.029	5	8,283	36,843	8,707	38,729	4
0.204	5.18	6,146	27,337	6,462	28,743	10	7,068	31,438	7,430	33,049	10	7,775	34,583	8,173	36,354	5	8,358	37,176	8,786	39,080	4
0.205	5.21	6,202	27,586	6,520	29,001	10	7,132	31,723	7,498	33,351	10	7,846	34,899	8,248	36,687	5	8,434	37,514	8,866	39,436	4
0.206	5.23	6,258	27,836	6,578	29,259	10	7,196	32,008	7,566	33,654	10	7,916	35,210	8,322	37,016	5	8,510	37,852	8,946	39,792	4
0.207	5.26	6,314	28,085	6,638	29,526	10	7,261	32,297	7,633	33,952	10	7,987	35,526	8,397	37,350	5	8,586	38,191	9,026	40,148	4
0.208	5.28	6,371	28,338	6,697	29,788	10	7,326	32,586	7,702	34,258	10	8,058	35,842	8,472	37,683	5	8,663	38,533	9,107	40,508	4
0.209	5.31	6,427	28,587	6,757	30,055	10	7,391	32,875	7,771	34,565	10	8,131	36,167	8,547	38,017	5	8,740	38,876	9,188	40,868	4
0.210	5.33	6,484	28,841	6,816	30,318	10	7,457	33,169	7,839	34,868	10	8,202	36,482	8,622	38,351	5	8,817	39,218	9,269	41,229	4
0.211	5.36	6,540	29,090	6,876	30,584	10	7,522	33,458	7,908	35,175	10	8,274	36,803	8,698	38,689	5	8,895	39,565	9,351	41,593	4
0.212	5.38	6,598	29,348	6,936	30,851	10	7,587	33,747	7,977	35,482	10	8,346	37,123	8,774	39,027	5	8,972	39,907	9,432	41,954	4
0.213	5.41	6,655	29,601	6,997	31,123	10	7,654	34,045	8,046	35,789	10	8,419	37,448	8,851	39,369	5	9,050	40,254	9,514	42,318	4
0.214	5.44	6,713	29,859	7,057	31,390	10	7,720	34,339	8,116	36,100	10	8,492	37,772	8,928	39,712	5	9,129	40,606	9,597	42,687	4
0.215	5.46	6,770	30,113	7,118	31,661	10	7,786	34,632	8,186	36,411	9	8,564	38,093	9,004	40,050	5	9,207	40,953	9,679	43,052	4
0.216	5.49	6,829	30,375	7,179	31,932	10	7,853	34,930	8,255	36,718	9	8,639	38,426	9,082	40,397	5	9,286	41,304	9,762	43,421	4
0.217	5.51	6,886	30,629	7,240	32,204	10	7,920	35,228	8,326	37,034	9	8,712	38,751	9,158	40,735	5	9,365	41,656	9,845	43,791	4
0.218	5.54	6,945	30,891	7,301	32,475	10	7,987	35,526	8,397	37,350	9	8,786	39,080	9,236	41,082	5	9,445	42,011	9,929	44,164	4
0.219	5.56	7,003	31,149	7,363	32,751	10	8,054	35,824	8.468	37,666	9	8,860	39,409	9,314	41,429	5	9.524	42,363	10,012	44,533	4

Table 4-9
(continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)	(22)
Wire Size Nominal Diameter		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength					Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength				
		Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.	Individual Minimum		Average Minimum		Min.
in.	mm	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.	lb	N	lb	N	Tor.
0.220	5.59	7,063	31,416	7,425	33,026	10	8,122	36,127	8,538	37,977	9	8,934	39,738	9,392	41,776	4	9,604	42,719	10,096	44,907	4
0.221	5.61	7,121	31,674	7,487	33,302	10	8,190	36,429	8,610	38,297	9	9,009	40,072	9,471	42,127	4	9,685	43,079	10,181	45,285	4
0.222	5.64	7,181	31,941	7,549	33,578	10	8,257	36,727	8,681	38,613	9	9,083	40,401	9,549	42,474	4	9,765	43,435	10,265	45,659	4
0.223	5.66	7,240	32,204	7,612	33,858	10	8,326	37,034	8,752	38,929	9	9,158	40,735	9,628	42,825	4	9,846	43,795	10,350	46,037	4
0.224	5.69	7,300	32,470	7,674	34,134	10	8,395	37,341	8,825	39,254	9	9,234	41,073	9,708	43,181	4	9,926	44,151	10,436	46,419	4
0.225	5.72	7,359	32,733	7,737	34,414	10	8,463	37,643	8,897	39,574	9	9,309	41,406	9,787	43,533	4	10,007	44,511	10,521	46,797	4
0.226	5.74	7,419	33,000	7,799	34,690	10	8,532	37,950	8,970	39,899	9	9,385	41,744	9,867	43,888	4	10,089	44,876	10,607	47,180	4
0.227	5.77	7,479	33,267	7,863	34,975	10	8,601	38,257	9,043	40,223	9	9,461	42,083	9,947	44,244	4	10,171	45,241	10,693	47,562	4
0.228	5.79	7,540	33,538	7,926	35,255	10	8,671	38,569	9.115	40,544	9	9,537	42,421	10,027	44,600	4	10,253	45,605	10,779	47,945	4
0.229	5.82	7,600	33,805	7,990	35,540	10	8,740	38,876	9.188	40,868	9	9,614	42,763	10,108	44,960	4	10,335	45,970	10,865	48,328	4
0.230	5.84	7,661	34,076	8,053	35,820	10	8,810	39,187	9,262	41,197	9	9,691	43,106	10,187	45,312	4	10,418	46,339	10,952	48,714	4
0.231	5.87	7,722	34,347	8,118	36,109	10	8,879	39,494	9,335	41,522	9	9,768	43,448	10,268	45,672	4	10,501	46,708	11,039	49,101	4
0.232	5.89	7,782	34,614	8,182	36,394	10	8,950	39,810	9,408	41,847	9	9,845	43,791	10,349	46,032	4	10,584	47,078	11,126	49,488	4
0.233	5.92	7,844	34,890	8,246	36,678	9	9,021	40,125	9,483	42,180	9	9,923	44,138	10,431	46,397	4	10,667	47,447	11,214	49,880	4
0.234	5.94	7,905	35,161	8,311	36,967	9	9,091	40,437	9,557	42,510	9	10,001	44,484	10,513	46,762	4	10,750	47,816	11,302	50,271	4
0.235	5.97	7,967	35,437	8,375	37,252	9	9,162	40,753	9,632	42,843	9	10,078	44,827	10,594	47,122	4	10,834	48,190	11,390	50,663	4
0.236	5.99	8,029	35,713	8,441	37,546	9	9.233	41,068	9,707	43,177	8	10,157	45,178	10,677	47,491	4	10,918	48,563	11,478	51,054	4
0.237	6.02	8,091	35,989	8,505	37,830	9	9,304	41,384	9,782	43,510	8	10,235	45,525	10,759	47,856	4	11,002	48,937	11,566	51,446	4
0.238	6.05	8,153	36,265	8,571	38,124	9	9,376	41,704	9,856	43,839	8	10,314	45,877	10,842	48,225	4	11,087	49,315	11,655	51,841	4
0.239	6.07	8,215	36,540	8,637	38,417	9	9,448	42,025	9,932	44,178	8	10,393	46,228	10,925	48,594	4	11,172	49,693	11,744	52,237	4
0.240	6.10	8,278	36,821	8,802	38,706	9	9,519	42,341	10,007	44,511	8	10,472	46,579	11,009	48,968	4	11,256	50,067	11,834	52,638	3
0.241	6.12	8,340	37,096	8,768	39,000	9	9,591	42,661	10,083	44,849	8	10,550	46,926	11,092	49,337	4	11,342	50,449	11,924	53,038	3
0.242	6.15	8,404	37,381	8,834	39,294	9	9,664	42,985	10,160	45,192	8	10,630	47,282	11,176	49,711	4	11,427	50,827	12,013	53,434	3
0.243	6.17	8,466	37,657	8,900	39,587	9	9,736	43,306	10,236	45,530	8	10,709	47,634	11,259	50,080	4	11,513	51,210	12,103	53,834	3
0.244	6.20	8,529	37,937	8,967	39,885	9	9,809	43,630	10,313	45,872	8	10,790	47,994	11,344	50,458	4	11,600	51,597	12,194	54,239	3
0.245	6.22	8,593	38,222	9,033	40,179	9	9,882	43,955	10,388	46,206	8	10,870	48,350	11,428	50,832	4	11,685	51,975	12,285	54,644	3
0.246	6.25	8,657	38,506	9,101	40,481	9	9,955	44,280	10,465	46,548	8	10,950	48,706	11,512	51,205	4	11,772	52,362	12,376	55,048	3
0.247	6.27	8,720	38,787	9,168	40,779	9	10,029	44,609	10,543	46,895	8	11,031	49,066	11,597	51,583	4	11,859	52,749	12,467	55,453	3
0.248	6.30	8,785	39,076	9,235	41,077	9	10,102	44,934	10,620	47,238	8	11,112	49,426	11,682	51,962	4	11,946	53,136	12,558	55,858	3
0.249	6.32	8,848	39,356	9,302	41,375	9	10,176	45,263	10,698	47,585	8	11,193	49,786	11,767	52,340	4	12,032	53,518	12,650	56,267	3
0.250	6.35	8,912	39,641	9,370	41,678	9	10,249	45,588	10,775	47,927	8	11,275	50,151	11,853	52,722	4	12,120	53,910	12,742	56,676	3

Table 4-10
Mechanical Properties of Individual Rope Wires (Before Fabrication) [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Wire Size Nominal Diameter		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength		
in.	mm	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.
0.010	0.25	17	76	254	20	89	234	22	98	218	24	107	190
0.011	0.28	21	93	231	24	107	213	27	120	198	29	129	173
0.012	0.30	25	111	212	29	129	195	32	142	182	34	151	158
0.013	0.33	29	129	195	34	151	180	37	165	168	40	178	146
0.014	0.36	34	151	181	39	173	167	43	191	156	46	205	136
0.015	0.38	39	173	169	45	200	156	49	218	145	53	236	126
0.016	0.41	44	196	158	51	227	146	56	249	136	60	267	118
0.017	0.43	50	222	149	57	254	137	63	280	128	68	302	111
0.018	0.46	56	249	141	64	285	130	71	316	121	76	338	105
0.019	0.48	62	276	133	72	320	123	79	351	114	85	378	100
0.020	0.51	69	307	126	79	351	116	87	387	108	94	418	94
0.021	0.53	76	338	120	87	387	111	96	427	103	103	458	90
0.022	0.56	83	369	115	96	427	106	105	467	98	113	503	86
0.023	0.58	91	405	110	105	467	101	115	512	94	124	552	82
0.024	0.61	99	440	105	114	507	97	125	556	90	135	600	78
0.025	0.64	107	476	101	123	547	93	136	605	86	146	649	75
0.026	0.66	116	516	97	133	592	89	147	654	83	158	703	72
0.027	0.69	125	556	93	144	641	86	158	703	80	170	756	70
0.028	0.71	134	596	90	155	689	83	170	756	77	183	814	67
0.029	0.74	144	641	87	166	738	80	182	810	74	196	872	65
0.030	0.76	154	685	84	177	787	77	195	867	72	210	934	62
0.031	0.79	164	729	81	189	841	75	208	925	69	224	996	60
0.032	0.81	175	778	78	201	894	72	221	983	67	238	1,059	58
0.033	0.84	186	827	76	214	952	70	235	1,045	65	253	1,125	57
0.034	0.86	197	876	74	227	1,010	68	250	1,112	63	268	1,192	55
0.035	0.89	209	930	72	240	1,068	66	264	1,174	61	284	1,263	53
0.036	0.91	221	983	70	254	1,130	64	280	1,245	60	301	1,339	52
0.037	0.94	233	1,036	68	268	1,192	62	295	1,312	58	317	1,410	50
0.038	0.97	246	1,094	66	283	1,259	61	311	1,383	56	334	1,486	49
0.039	0.99	259	1,152	64	298	1,326	59	327	1,454	55	352	1,566	48
0.040	1.02	272	1,210	62	313	1,392	57	344	1,530	53	370	1,646	46
0.041	1.04	286	1,272	61	329	1,463	56	361	1,606	52	388	1,726	45
0.042	1.07	300	1,334	59	345	1,535	55	379	1,686	51	407	1,810	44
0.043	1.09	314	1,397	58	361	1,606	53	397	1,766	50	427	1,899	43
0.044	1.12	328	1,459	57	378	1,681	52	415	1,846	48	447	1,988	42
0.045	1.14	343	1,526	55	395	1,757	51	434	1,930	47	467	2,077	41
0.046	1.17	358	1,592	54	412	1,833	50	453	2,015	46	487	2,166	40
0.047	1.19	374	1,664	53	430	1,913	49	473	2,104	45	508	2,260	39
0.048	1.22	390	1,735	52	448	1,993	48	493	2,193	44	530	2,357	38
0.049	1.24	406	1,806	51	467	2,077	47	513	2,282	43	552	2,455	38
0.050	1.27	422	1,877	50	486	2,162	46	534	2,375	42	574	2,553	37
0.051	1.30	439	1,953	49	505	2,246	45	555	2,469	42	597	2,655	36
0.052	1.32	456	2,028	48	525	2,335	44	577	2,566	41	620	2,758	35
0.053	1.35	474	2,108	47	545	2,424	43	599	2,664	40	644	2,865	35
0.054	1.37	491	2,184	46	565	2,513	42	621	2,762	39	668	2,971	34

Table 4-10
(continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Wire Size Nominal Diameter		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength		
in.	mm	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.
0.055	1.40	509	2,264	45	586	2,607	41	644	2,865	38	693	3,082	33
0.056	1.42	528	2,349	44	607	2,700	41	667	2,967	38	718	3,194	33
0.057	1.45	546	2,429	43	628	2,793	40	691	3,074	37	743	3,305	32
0.058	1.47	565	2,513	43	650	2,891	39	715	3,180	36	769	3,421	32
0.059	1.50	584	2,598	42	672	2,989	38	739	3,287	36	795	3,536	31
0.060	1.52	604	2,687	41	695	3,091	38	764	3,398	35	821	3,652	30
0.061	1.55	624	2,776	40	718	3,194	37	789	3,509	35	848	3,772	30
0.062	1.57	644	2,865	40	741	3,296	37	815	3,625	34	876	3,896	29
0.063	1.60	665	2,958	39	764	3,398	36	841	3,741	33	904	4,021	29
0.064	1.63	685	3,047	38	788	3,505	35	867	3,856	33	932	4,146	28
0.065	1.65	707	3,145	38	813	3,616	35	894	3,977	32	961	4,275	28
0.066	1.68	728	3,238	37	837	3,723	34	921	4,097	32	990	4,404	28
0.067	1.70	750	3,336	37	862	3,834	34	948	4,217	31	1,019	4,533	27
0.068	1.73	772	3,434	36	887	3,945	33	976	4,341	31	1,049	4,666	27
0.069	1.75	794	3,532	36	913	4,061	33	1,004	4,466	30	1,080	4,804	26
0.070	1.78	817	3,634	35	939	4,177	32	1,033	4,595	30	1,111	4,942	26
0.071	1.80	840	3,736	35	966	4,297	32	1,062	4,724	29	1,142	5,080	26
0.072	1.83	863	3,839	34	992	4,412	31	1,091	4,853	29	1,173	5,218	25
0.073	1.85	886	3,941	34	1,019	4,533	31	1,121	4,986	29	1,205	5,360	25
0.074	1.88	910	4,048	33	1,047	4,657	30	1,151	5,120	28	1,238	5,507	24
0.075	1.91	934	4,154	33	1,074	4,777	30	1,182	5,258	28	1,271	5,653	24
0.076	1.93	959	4,266	32	1,103	4,906	30	1,213	5,395	27	1,304	5,800	24
0.077	1.96	983	4,372	32	1,131	5,031	29	1,244	5,533	27	1,337	5,947	23
0.078	1.98	1,008	4,484	31	1,160	5,160	29	1,276	5,676	27	1,371	6,098	23
0.079	2.01	1,034	4,599	31	1,189	5,289	28	1,308	5,818	26	1,406	6,254	23
0.080	2.03	1,059	4,710	30	1,218	5,418	28	1,340	5,960	26	1,441	6,410	22
0.081	2.06	1,085	4,826	30	1,248	5,551	28	1,373	6,107	26	1,476	6,565	22
0.082	2.08	1,111	4,942	30	1,278	5,685	27	1,406	6,254	25	1,511	6,721	22
0.083	2.11	1,138	5,062	29	1,309	5,822	27	1,440	6,405	25	1,548	6,886	22
0.084	2.13	1,165	5,182	29	1,339	5,956	27	1,473	6,552	25	1,584	7,046	21
0.085	2.16	1,192	5,302	29	1,371	6,098	26	1,508	6,708	24	1,621	7,210	21
0.086	2.18	1,219	5,422	28	1,402	6,236	26	1,542	6,859	24	1,658	7,375	21
0.087	2.21	1,247	5,547	28	1,434	6,378	26	1,577	7,014	24	1,696	7,544	21
0.088	2.24	1,275	5,671	28	1,466	6,521	25	1,613	7,175	23	1,734	7,713	20
0.089	2.26	1,303	5,796	27	1,499	6,668	25	1,648	7,330	23	1,772	7,882	20
0.090	2.29	1,332	5,925	27	1,531	6,810	25	1,684	7,490	23	1,811	8,055	20
0.091	2.31	1,360	6,049	27	1,564	6,957	24	1,721	7,655	23	1,850	8,229	20
0.092	2.34	1,390	6,183	26	1,598	7,108	24	1,758	7,820	22	1,890	8,407	19
0.093	2.36	1,419	6,312	26	1,632	7,259	24	1,795	7,984	22	1,930	8,585	19
0.094	2.39	1,449	6,445	26	1,666	7,410	24	1,832	8,149	22	1,970	8,763	19
0.095	2.41	1,479	6,579	25	1,700	7,562	23	1,870	8,318	22	2,011	8,945	19
0.096	2.44	1,509	6,712	25	1,735	7,717	23	1,909	8,491	21	2,052	9,127	18
0.097	2.46	1,539	6,845	25	1,770	7,873	23	1,947	8,660	21	2,093	9,310	18
0.098	2.49	1,570	6,983	25	1,806	8,033	23	1,986	8,834	21	2,135	9,496	18
0.099	2.51	1,601	7,121	24	1,841	8,189	22	2,026	9,012	21	2,177	9,683	18

Table 4-10
(continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Wire Size Nominal Diameter		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength		
in.	mm	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.
0.100	2.54	1,633	7,264	24	1,877	8,349	22	2,065	9,185	20	2,220	9,875	18
0.101	2.57	1,664	7,401	24	1,914	8,513	22	2,105	9,363	20	2,263	10,066	18
0.102	2.59	1,696	7,744	24	1,951	8,678	22	2,146	9,545	20	2,307	10,262	17
0.103	2.62	1,728	7,686	23	1,988	8,843	21	2,186	9,723	20	2,350	10,453	17
0.104	2.64	1,761	7,833	23	2,025	9,007	21	2,228	9,910	20	2,395	10,653	17
0.105	2.67	1,794	7,980	23	2,063	9,176	21	2,269	10,093	19	2,439	10,849	17
0.106	2.69	1,827	8,126	23	2,101	9,345	21	2,311	10,279	19	2,484	11,049	17
0.107	2.72	1,860	8,273	22	2,139	9,514	21	2,353	10,466	19	2,529	11,249	16
0.108	2.74	1,894	8,425	22	2,178	9,688	20	2,396	10,657	19	2,575	11,454	16
0.109	2.77	1,928	8,576	22	2,217	9,861	20	2,438	10,844	19	2,621	11,658	16
0.110	2.79	1,962	8,727	22	2,256	10,035	20	2,482	11,040	18	2,668	11,867	16
0.111	2.82	1,996	8,878	22	2,296	10,213	20	2,525	11,231	18	2,715	12,076	16
0.112	2.84	2,031	9,034	21	2,336	10,391	20	2,569	11,427	18	2,762	12,285	16
0.113	2.87	2,066	9,190	21	2.376	10,568	19	2,613	11,623	18	2,809	12,494	15
0.114	2.90	2,101	9,345	21	2,416	10,746	19	2,658	11,823	18	2,857	12,708	15
0.115	2.92	2,137	9,505	21	2,457	10,929	19	2,703	12,023	18	2,906	12,926	15
0.116	2.95	2,172	9,661	21	2,498	11,111	19	2,748	12,223	17	2,954	13,139	15
0.117	2.97	2,209	9,826	20	2,540	11,298	19	2,794	12,428	17	3,003	13,357	15
0.118	3.00	2,245	9,986	20	2,582	11,485	18	2,840	12,632	17	3,053	13,580	15
0.119	3.02	2,281	10,146	20	2,624	11,672	18	2,886	12,837	17	3,102	13,798	15
0.120	3.05	2,318	10,310	20	2,666	11,858	18	2,933	13,046	17	3,153	14,025	14
0.121	3.07	2.355	10.475	20	2,709	12,050	18	2,980	13,255	17	3,203	14,247	14
0.122	3.10	2,393	10,644	19	2,752	12,241	18	3,027	13,464	17	3,254	14,474	14
0.123	3.12	2,431	10,813	19	2,795	12,432	18	3,075	13,678	16	3,305	14,701	14
0.124	3.15	2,468	10,978	19	2,839	12,628	18	3,123	13,891	16	3,357	14,932	14
0.125	3.18	2,507	11,151	19	2,883	12,824	17	3,171	14,105	16	3,409	15,163	14
0.126	3.20	2,545	11,320	19	2,927	13,019	17	3,220	14,323	16	3,461	15,395	14
0.127	3.23	2,584	11,494	19	2,971	13,215	17	3,269	14,541	16	3,514	15,630	14
0.128	3.25	2.623	11,667	18	3,016	13,415	17	3,318	14,758	16	3,567	15,866	13
0.129	3.28	2,662	11,841	18	3,061	13,615	17	3,368	14,981	16	3,620	16,102	13
0.130	3.30	2,702	12,018	18	3,107	13,820	17	3,418	15,203	15	3,674	16,342	13
0.131	3.33	2,741	12,192	18	3,153	14,025	17	3,468	15,426	15	3,728	16,582	13
0.132	3.35	2,781	12,370	18	3,199	14,229	16	3,519	15,653	15	3,782	16,822	13
0.133	3.38	2,822	12,552	18	3,245	14,434	16	3,570	15,879	15	3,837	17,067	13
0.134	3.40	2,862	12,730	18	3,292	14,643	16	3,621	16,106	15	3,892	17,312	13
0.135	3.43	2,903	12,913	17	3,339	14,852	16	3,672	16,333	15	3,948	17,561	13
0.136	3.45	2,944	13,095	17	3,386	15,061	16	3,724	16,564	15	4,004	17,810	13
0.137	3.48	2,986	13,282	17	3,433	15,270	16	3,777	16,800	15	4,060	18,059	13
0.138	3.51	3,027	13,464	17	3,481	15,483	16	3,829	17,031	14	4,117	18,312	12
0.139	3.53	3,069	13,651	17	3,529	15,697	15	3,882	17,267	14	4,173	18,562	12
0.140	3.56	3,111	13,838	17	3,578	15,915	15	3,935	17,503	14	4,231	18,819	12
0.141	3.58	3,153	14,025	17	3,626	16,128	15	3,989	17,743	14	4,288	19,073	12
0.142	3.61	3,196	14,216	17	3,675	16,346	15	4,043	17,983	14	4,346	19,331	12
0.143	3.63	3,239	14,407	16	3,725	16,569	15	4,097	18,223	14	4,404	19,589	12
0.144	3.66	3,282	14,598	16	3,774	16,787	15	4,152	18,468	14	4,463	19,851	12

Table 4-10
(continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Wire Size Nominal Diameter		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength		
in.	mm	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.
0.145	3.68	3,325	14,790	16	3,824	17,009	15	4,207	18,713	14	4,522	20,114	12
0.146	3.71	3,369	14,985	16	3,874	17,232	15	4,262	18,957	14	4,581	20,376	12
0.147	3.73	3,413	15,181	16	3,925	17,458	15	4,317	19,202	13	4,641	20,643	12
0.148	3.76	3,457	15,377	16	3,975	17,681	14	4,373	19,451	13	4,701	20,910	11
0.149	3.78	3,501	15,572	16	4,026	17,908	14	4,429	19,700	13	4,761	21,177	11
0.150	3.81	3,546	15,773	16	4,078	18,139	14	4,486	19,954	13	4,822	21,448	11
0.151	3.84	3,591	15,973	15	4,129	18,366	14	4,542	20,203	13	4,883	21,720	11
0.152	3.86	3,636	16,173	15	4,181	18,597	14	4,599	20,456	13	4,944	21,991	11
0.153	3.89	3,681	16,373	15	4,233	18,828	14	4,657	20,714	13	5,006	22,267	11
0.154	3.91	3.727	16,578	15	4,286	19,064	14	4,714	20,968	13	5,068	22,542	11
0.155	3.94	3,773	16,782	15	4,338	19,295	14	4,772	21,226	13	5,130	22,818	11
0.156	3.96	3,819	16,987	15	4,391	19,531	14	4,831	21,488	13	5,193	23,098	11
0.157	3.99	3,865	17,192	15	4,445	19,771	14	4,899	21,746	13	5,256	23,379	11
0.158	4.01	3,912	17,401	15	4,498	20,007	13	4,948	22,099	12	5,319	23,659	11
0.159	4.04	3,958	17,605	15	4,552	20,247	13	5,007	22,271	12	5,383	23,944	11
0.160	4.06	4,005	17,814	14	4,606	20,487	13	5,067	22,538	12	5,447	24,228	10
0.161	4.09	4,053	18,028	14	4,661	20,732	13	5,127	22,805	12	5,511	24,513	10
0.162	4.11	4,100	18,237	14	4,715	20,972	13	5,187	23,072	12	5,576	24,802	10
0.163	4.14	4,148	18,450	14	4,770	21,217	13	5,247	23,339	12	5,641	25,091	10
0.164	4.17	4,196	18,664	14	4,825	21,462	13	5,308	23,610	12	5,706	25,380	10
0.165	4.19	4,244	18,877	14	4,881	21,711	13	5,369	23,881	12	5,772	25,674	10
0.166	4.22	4,293	19,095	14	4,937	21,960	13	5,430	24,153	12	5,838	25,967	10
0.167	4.24	4,341	19,309	14	4,993	22,209	13	5,492	24,428	12	5,904	26,261	10
0.168	4.27	4,390	19,527	14	5,049	22,458	13	5,554	24,704	12	5,970	26,555	10
0.169	4.29	4,440	19,749	14	5,105	22,707	12	5,616	24,980	12	6,037	26,853	10
0.170	4.32	4,489	19,967	14	5,162	22,961	12	5,679	25,260	11	6,104	27,151	10
0.171	4.34	4,539	20,189	13	5,219	23,214	12	5,741	25,536	11	6,172	27,453	10
0.172	4.37	4,589	20,413	13	5,277	23,472	12	5,804	25,816	11	6,240	27,756	10
0.173	4.39	4,639	20,634	13	5,334	23,726	12	5,868	26,101	11	6,308	28,058	10
0.174	4.42	4,689	20,857	13	5,392	23,984	12	5,932	26,396	11	6,376	28,360	10
0.175	4.45	4,740	21,064	13	5,450	24,242	12	5,996	26,670	11	6,514	28,974	9
0.176	4.47	4,790	21,306	13	5,509	24,504	12	6,060	26,955	11	6,514	28,974	9
0.177	4.50	4,841	21,533	13	5,568	24,766	12	6,124	27,240	11	6,584	29,286	9
0.178	4.52	4,893	21,764	13	5,627	25,029	12	6,189	27,529	11	6,653	29,593	9
0.179	4.55	4,944	21,991	13	5,686	25,291	12	6,254	27,818	11	6,723	29,904	9
0.180	4.57	4,996	22,222	13	5,745	25,554	12	6,320	28,111	11	6,794	30,220	9
0.181	4.60	5,048	22,454	13	5,805	25,821	12	6,386	28,405	11	6,864	30,531	9
0.182	4.62	5,100	22,685	13	5,865	26,088	11	6,452	28,698	11	6,935	30,847	9
0.183	4.65	5,152	22,916	12	5,925	26,354	11	6,518	28,992	11	7,007	31,167	9
0.184	4.67	5,205	23,152	12	5,986	26,626	11	6,584	29,286	10	7,078	31,483	9
0.185	4.70	5,258	23,388	12	6,047	26,897	11	6,651	29,584	10	7,150	31,803	9
0.186	4.72	5,311	23,623	12	6.108	27,168	11	6,718	29,882	10	7,222	32,123	9
0.187	4.75	5,364	23,859	12	6,169	27,440	11	6,786	30,184	10	7,295	32,448	9
0.188	4.78	5,418	24,099	12	6,230	27,711	11	6,854	30,487	10	7,368	32,773	9
0.189	4.80	5,472	24,339	12	6,292	27,987	11	6,921	30,785	10	7,441	33,098	9

Table 4-10
(continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Wire Size Nominal Diameter		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength		
in	mm	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.
0.190	4.83	5,525	24,575	12	6,354	28,263	11	6,990	31,092	10	7,514	33,422	9
0.191	4.85	5,580	24,820	12	6,417	28,543	11	7,058	31,394	10	7,588	33,751	9
0.192	4.88	5,634	2,5060	12	6,469	28,819	11	7,127	31,701	10	7,662	34,081	8
0.193	4.90	5,689	25,305	12	6,542	29,099	11	7,196	32,008	10	7,736	34,410	8
0.194	4.93	5,744	25,549	12	6,605	29,379	11	7,266	32,319	10	7,810	34,739	8
0.195	4.95	5,799	25,794	12	6,668	29,659	11	7,335	32,626	10	7,885	35,072	8
0.196	4.98	5,854	26,039	12	6,732	29,944	11	7,405	32,937	10	7,961	35,411	8
0.197	5.00	5,909	26,283	11	6,796	30,229	10	7,475	33,249	10	8,036	35,744	8
0.198	5.03	5,965	26,532	11	6,860	30,513	10	7,536	33,365	10	8,112	36,082	8
0.199	5.05	6,021	26,781	11	6,924	30,798	10	7,617	33,880	10	8,188	36,420	8
0.200	5.08	6,077	27,030	11	6,989	31,087	10	7,688	34,196	9	8,264	36,758	8
0.201	5.11	6,133	27,280	11	7,053	31,372	10	7,759	34,512	9	8,341	37,101	8
0.202	5.13	6,190	27,533	11	7,118	31,661	10	7,830	34,828	9	8,418	37,443	8
0.203	5.16	6,247	27,787	11	7,184	31,954	10	7,902	35,148	9	8,495	37,786	8
0.204	5.18	6,304	28,040	11	7,249	32,244	10	7,974	35,468	9	8,572	38,128	8
0.205	5.21	6,361	28,294	11	7,315	32,537	10	8,047	35,793	9	8,650	38,475	8
0.206	5.23	6,418	28,547	11	7,381	32,831	10	8,119	36,113	9	8,728	38,822	8
0.207	5.26	6,476	28,805	11	7,447	33,124	10	8,192	36,438	9	8,806	39,169	8
0.208	5.28	6,534	29,063	11	7,514	33,422	10	8,265	36,763	9	8,885	39,520	8
0.209	5.31	6,592	29,321	11	7,581	33,720	10	8,339	37,092	9	8,964	39,872	8
0.210	5.33	6,650	29,579	11	7,648	34,018	10	8,412	37,417	9	9,043	40,223	8
0.211	5.36	6,708	29,837	11	7,715	34,316	10	8,486	37,746	9	9,123	40,579	8
0.212	5.38	6,767	30,100	11	7,782	34,614	10	8,560	38,075	9	9,202	40,930	8
0.213	5.41	6,826	30,362	11	7,850	34,917	10	8,635	38,408	9	9,282	41,286	8
0.214	5.44	6,885	30,624	10	7,918	35,219	10	8,710	38,742	9	9,363	41,647	7
0.215	5.46	6,944	30,887	10	7,986	35,522	9	8,784	39,071	9	9,443	42,002	7
0.216	5.49	7,004	31,154	10	8,054	35,824	9	8,860	39,409	9	9,524	42,363	7
0.217	5.51	7,063	31,416	10	8,123	36,131	9	8,935	39,743	9	9,605	42,723	7
0.218	5.54	7,123	31,683	10	8,192	36,438	9	9,011	40,081	9	9,687	43,088	7
0.219	5.56	7,183	31,950	10	8,261	36,745	9	9,087	40,419	9	9,768	43,448	7
0.220	5.59	7,244	32,221	10	8,330	37,052	9	9,163	40,757	8	9,850	43,813	7
0.221	5.61	7,304	32,488	10	8,400	37,363	9	9,240	41,100	8	9,933	44,182	7
0.222	5.64	7,365	32,760	10	8,469	37,670	9	9,316	41,438	8	10,015	44,547	7
0.223	5.66	7,426	33,031	10	8,539	37,981	9	9,393	41,780	8	10,098	44,916	7
0.224	5.69	7,487	33,302	10	8,610	38,297	9	9,471	42,127	8	10,181	45,285	7
0.225	5.72	7,548	33,574	10	8,680	38,609	9	9,548	42,470	8	10,264	45,654	7
0.226	5.74	7,609	33,845	10	8,751	38,924	9	9,626	42,816	8	10,348	46,028	7
0.227	5.77	7,671	34,121	10	8,822	39,240	9	9,704	43,163	8	10,432	46,402	7
0.228	5.79	7,733	34,396	10	8,893	39,556	9	9,782	43,510	8	10,516	46,775	7
0.229	5.82	7,795	34,672	10	8,964	39,872	9	9,861	43,862	8	10,600	47,149	7

Table 4-10
(continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Wire Size Nominal Diameter		Level 2 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 3 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 4 Bright (Uncoated) or Drawn-Galvanized Breaking Strength			Level 5 Bright (Uncoated) or Drawn-Galvanized Breaking Strength		
in.	mm	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.	lb	N	Tor.
0.230	5.84	7,857	34,948	10	9,036	40,192	9	9,939	44,209	8	10,685	47,527	7
0.231	5.87	7,920	35,228	10	9,107	40,508	9	10,018	44,560	8	10,770	47,905	7
0.232	5.89	7,982	35,504	10	9,179	40,828	9	10,097	44,911	8	10,855	48,283	7
0.233	5.92	8,045	35,784	9	9,252	41,153	9	10,177	45,267	8	10,940	48,661	7
0.234	5.94	8,108	36,064	9	9,324	41,473	9	10,257	45,623	8	11,026	49,044	7
0.235	5.97	8,171	36,345	9	9,397	41,798	9	10,336	45,975	8	11,112	49,426	7
0.236	5.99	8,235	36,629	9	9,470	42,123	8	10,417	46,335	8	11,198	49,809	7
0.237	6.02	8,298	36,910	9	9,543	42,447	8	10,497	46,691	8	11,284	50,191	7
0.238	6.05	8,362	37,194	9	9,616	42,772	8	10,578	47,051	8	11,371	50,578	7
0.239	6.07	8,426	37,479	9	9,690	43,101	8	10,659	47,411	8	11,458	50,965	7
0.240	6.10	8,490	37,764	9	9,763	43,426	8	10,740	47,772	8	11,545	51,352	6
0.241	6.12	8,554	38,048	9	9,837	43,755	8	10,821	48,132	8	11,633	51,744	6
0.242	6.15	8,619	38,337	9	9,912	44,089	8	10,903	48,497	8	11,720	52,131	6
0.243	6.17	8,683	38,622	9	9,986	44,418	8	10,984	48,857	8	11,807	52,522	6
0.244	6.20	8,748	38,911	9	10,061	44,751	8	11,067	49,226	8	11,897	52,918	6
0.245	6.22	8,813	39,200	9	10,135	45,060	8	11,149	49,591	7	11,985	53,309	6
0.246	6.25	8,879	39,494	9	10,210	45,414	8	11,231	49,955	7	12,074	53,705	6
0.247	6.27	8,944	39,783	9	10,286	45,752	8	11,314	50,325	7	12,163	54,101	6
0.248	6.30	9,010	40,076	9	10,361	46,086	8	11,397	50,694	7	12,252	54,497	6
0.249	6.32	9,075	40,366	9	10,437	46,424	8	11,480	51,063	7	12,341	54,893	6
0.250	6.35	9,141	40,659	9	10,512	46,757	8	11,564	51,437	7	12,431	55,293	6

(text continued from page 544)

Galvanized Wire Rope. Galvanized wire rope shall be made of wire having a tightly adherent, uniform and continuous coating of zinc applied after final cold drawing, by the electrodeposition process or by the hot-galvanizing process. The minimum weight of zinc coating shall be as specified in Table 4-11.

Drawn-Galvanized Wire Rope. Drawn-galvanized wire rope shall be made of wire having a tightly adherent, uniform, and continuous coating of zinc applied at an intermediate stage of the wire drawing operation, by the electrodeposition process or by the hot-galvanizing process. The minimum weight of zinc coating shall be as specified in Table 4-12.

Properties and Tests for Wire and Wire Rope

Selection of Test Specimens. For the test of individual wires and of rope, a 10-ft (3.05-m) section shall be cut from a finished piece of unused and undamaged

Table 4-11
Weight of Zinc Coating for Galvanized Rope Wire [12]

(1)	(2)	(3)	(4)
Diameter of Wire		Minimum Weight of Zinc Coating	
in.	mm	oz./ft²	kg/m²
0.028 to 0.047	0.71 to 1.19	0.20	0.06
0.048 to 0.054	1.22 to 1.37	0.40	0.12
0.055 to 0.063	1.40 to 1.60	0.50	0.15
0.064 to 0.079	1.63 to 2.01	0.60	0.18
0.080 to 0.092	2.03 to 2.34	0.70	0.21
0.093 and larger	2.36 and larger	0.80	0.24

Table 4-12
Weight of Zinc Coating for Drawn-Galvanized Rope Wire [12]

(1)	(2)	(3)	(4)
Diameter of Wire		Minimum Weight of Zinc Coating	
in.	mm	oz./ft²	kg/m²
0.018 to 0.028	0.46 to 0.71	0.10	0.03
0.029 to 0.060	0.74 to 1.52	0.20	0.06
0.061 to 0.090	1.55 to 2.29	0.30	0.09
0.091 to 0.140	2.31 to 3.56	0.40	0.12

wire rope; such sample must be new or in an unused condition. The total wire number to be tested shall be equal to the number of wires in any one strand, and the wires shall be selected from all strands of the rope. The specimens shall be selected from all locations or positions so that they would constitute a complete composite strand exactly similar to a regular strand in the rope. The specimen for all "like-positioned" (wires symmetrically placed in a strand) wires to be selected so as to use as nearly as possible an equal number from each strand.

Any unsymmetrically placed wires, or marker wires, are to be disregarded entirely. Center wires are subject to the same stipulations that apply to symmetrical wires.

Selection and testing of wire prior to rope fabrication will be adequate to ensure the after-fabrication wire rope breaking strength and wire requirements can be met. Prior to fabrication, wire tests should meet the requirements of Table 4-10.

Conduct of Tests. The test results of each test on any one specimen should be associated and may be studied separately from other specimens.

If, when making any individual wire test on any wire, the first specimen fails, not more than two additional specimens from the same wire shall be tested. The average of any two tests showing failure or acceptance shall be used as the value to represent the wire. The test for the rope may be terminated at any time sufficient failures have occurred to be the cause for rejection.

The purchaser may at his or her expense test all of the wires if the results of the selected tests indicate that further checking is warranted.

Tensile Requirements of Individual Wire. Specimens shall not be less than 18 in. (457 mm) long, and the distance between the grips of the testing machine shall not be less than 12 in. (305 mm). The speed of the movable head of the testing machine, under no load, shall not exceed 1 in./min (0.4 mm/s). Any specimen breaking within $\frac{1}{4}$ in. (6.35 mm) from the jaws shall be disregarded and a retest made.

Note: The diameter of wire can more easily and accurately be determined by placing the wire specimen in the test machine and applying a load not over 25% of the breaking strength of the wire.

The breaking strength of either bright (uncoated) or drawn-galvanized wires of the various grades shall meet the values shown in Table 4-9 or Table 4-10 for the size wire being tested. Wire tested after rope fabrication allows one wire in 6×7 classification, or three wires in 6×19 and 8×19 classifications and 18×7 and 19×7 constructions, or six wires in 6×37 classification or nine wires in 6×61 classification, or twelve wires in 6×91 classification wire rope to fall below, but not more than 10% below, the tabular value for individual minimum. If, when making the specified test, any wires fall below, but not more than 10% below, the individual minimum, additional wires from the same rope shall be tested until there is cause for rejection or until all of the wires in the rope have been tested. Tests of individual wires in galvanized wire rope and of individual wires in strand cores and in independent wire rope cores are not required.

Torsional Requirements of Individual Wire. The distance between the jaws of the testing machine shall be 8 in. ± $\frac{1}{16}$ in. (203 mm ± 1 mm). For small diameter wires, where the number of turns to cause failure is large, and in order to save testing time, the distance between the jaws of the testing machine may be less than 8 in. (203 mm). One end of the wire is to be rotated with respect to the other end at a uniform speed not to exceed sixty 360° (6.28 rad) twists per minute, until breakage occurs. The machine must be equipped with an automatic counter to record the number of twists causing breakage. One jaw shall be fixed axially and the other jaw movable axially and arranged for applying tension weights to wire under test. Tests in which breakage occurs within $\frac{1}{8}$ in. (3.18 mm) of the jaw shall be discarded.

In the torsion test, the wires tested must meet the values for the respective grades and sizes as covered by Table 4-12 or Table 4-13. In wire tested after rope fabrication, it will be permissible for two wires in 6×7 classification or five wires in 6×19 and 8×19 classifications and 18×7 and 19×7 constructions or ten wires in 6×37 classification or fifteen wires in 6×61 classification, or twenty wires in 6×91 classification rope to fall below, but not more than 30% below, the specified minimum number of twists for the individual wire being tested.

During the torsion test, tension weights as shown in Table 4-13 shall be applied to the wire tested.

The minimum torsions for individual bright (uncoated) or drawn-galvanized wire of the grades and sizes shown in Columns 7, 12, and 17 of Tables 4-9 and 4-10 shall be the number of 360° (6.28 rad) twists in an 8-in. (203 mm) length that the wire must withstand before breakage occurs. Torsion tests of individual wires in galvanized wire rope and of individual wires in strand cores and independent wire rope cores are not required.

When the distance between the jaws of the testing machine is less than 8 in. (203 mm), the minimum torsions shall be reduced in direct proportion to the change in jaw spacing, or determined by

Table 4-13
Applied Tension for Torsional Tests [12]

(1)	(2)	(3)	(4)
Wire Size Nominal Diameter (in)	(mm)	Minimum Applied Tension* (lb)	(N)
0.011 to 0.016	0.28 to 0.42	1	4
0.017 to 0.020	0.43 to 0.52	2	9
0.021 to 0.030	0.53 to 0.77	4	18
0.031 to 0.040	0.78 to 1.02	6	27
0.041 to 0.050	1.03 to 1.28	8	36
0.051 to 0.060	1.29 to 1.53	9	40
0.061 to 0.070	1.54 to 1.79	11	49
0.071 to 0.080	1.80 to 2.04	13	58
0.081 to 0.090	2.05 to 2.30	16	71
0.091 to 0.100	2.31 to 2.55	19	85
0.101 to 0.110	2.56 to 2.80	21	93
0.111 to 0.120	2.81 to 3.06	23	102
0.121 to 0.130	3.07 to 3.31	25	111

*Weights shall not exceed twice the minimums listed.

$$T_s = \frac{(T_L)(L_s)}{(L_L)} \tag{4-22}$$

where T_s = minimum torsions for short wire
T_L = minimum torsions for 8-in. (203-mm) length as given in Table 4-9 for size and grade of wire
L_s = distance between testing-machine jaws for short wire in in. (mm)
L_L = 8 in. (203 mm)

Breaking Strength Requirements for Wire Rope. The nominal strength of the various grades of finished wire rope with fiber core shall be as specified in Tables 4-14, 4-15, and 4-16. The nominal strength of the various grades of wire rope having a strand core or an independent wire-rope core shall be as specified in Tables 4-17 through 4-22. The nominal strength of the various types of flattened strand wire rope shall be specified in Table 4-23. The nominal strength of the various grades of drawn-galvanized wire rope shall be specified in Tables 4-14 through 4-23.

When testing finished wire-rope tensile test specimens to their breaking strength, suitable sockets shall be attached by the correct method. The length of test specimen shall not be less than 3 ft (0.91 m) between sockets for wire ropes up to 1-in. (25.4 mm) diameter and not less than 5 ft (1.52 m) between sockets for wire ropes $1\frac{1}{8}$-in. (28.6 mm) to 3-in. (77 mm) diameter. On wire ropes larger than 3 in. (77 mm), the clear length of the test specimen shall be at least 20 times the rope diameter. The test shall be valid if failure occurs 2 in. (50.8 mm) from the sockets or holding mechanism.

Due to the variables in sample preparation and testing procedures, it is difficult to determine the true strength. Thus, the actual breaking strength during test shall be at least 97.5% of the nominal strength as shown in the applicable table. If the first specimen fails at a value below the 97.5% nominal strength value, a second test shall be made, and if the second test meets the strength requirements, the wire rope shall be accepted.

Table 4-14
Classification Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Fiber Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
				Nominal Strength					
Nominal Diameter		Approx. Mass		Plow Steel			Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
3/8	9.5	0.21	0.31	10,200	45.4	4.63	11,720	52.1	5.32
7/16	11.5	0.29	0.43	13,800	61.4	6.26	15,860	70.5	7.20
1/2	13	0.38	0.57	17,920	79.7	8.13	20,600	91.6	9.35
9/16	14.5	0.48	0.71	22,600	101	10.3	26,000	116	11.8
5/8	16	0.59	0.88	27,800	124	12.6	31,800	141	14.4
3/4	19	0.84	1.25	39,600	176	18.0	45,400	202	20.6
7/8	22	1.15	1.71	53.400	238	24.2	61,400	273	27.9
1	26	1.50	2.23	69,000	307	31.3	79,400	353	36.0

Table 4-15
6¥19 and 6¥37 Classification Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Fiber Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
				Nominal Strength								
Nominal Diameter		Approx. Mass		Plow Steel			Improved Plow Steel			Extra Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
1/2	13	0.42	0.63	18,700	83.2	8.48	21,400	95.2	9.71	23,600	105	10.7
9/16	14.5	0.53	0.79	23,600	106	10.7	27,000	120	12.2	29,800	132	13.5
5/8	16	0.66	0.98	29,000	129	13.2	33,400	149	15.1	36,600	163	16.6
3/4	19	0.95	1.41	41,400	184	18.8	47,600	212	21.6	52,400	233	23.8
7/8	22	1.29	1.92	56,000	249	25.4	64,400	286	29.2	70,800	315	32.1
1	26	1.68	2.50	72,800	324	33.0	83,600	372	37.9	92,000	409	41.7
1 1/8	29	2.13	3.17	91,400	407	41.5	105,200	468	47.7	115,600	514	52.4
1 1/4	32	2.63	3.91	112,400	500	51.0	129,200	575	58.5	142,200	632	64.5
1 3/8	35	3.18	4.73				155,400	691	70.5	171,000	760	77.6
1 1/2	38	3.78	5.63				184,000	818	83.5	202,000	898	91.6
1 5/8	42	4.44	6.61				214,000	952	97.1	236,000	1050	107
1 3/4	45	5.15	7.66				248,000	1100	112	274,000	1220	124
1 7/8	48	5.91	8.80				282,000	1250	128	312,000	1390	142
2	52	6.72	10.0				320,000	1420	146	352,000	1560	160

Manufacture and Tolerances

Strand Construction. The 6×7 classification ropes shall contain six strands that are made up of 3 through 14 wires, of which no more than 9 are outside wires fabricated in one operation.* See Table 4-14 and Figure 4-37.

(text continued on page 571)

**One operation strand*—When the king wire of the strand becomes so large (manufacturer's discretion) that it is considered undesirable, it is allowed to be replaced with a seven-wire strand manufactured in a separate stranding operation. This does not constitute a two-operation strand.

Table 4-16
18×7 Construction Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Fiber Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
				Nominal Strength*					
Nominal Diameter		Approx. Mass		Improved Plow Steel			Extra Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
1/2	13	0.43	0.64	19,700	87.6	8.94	21,600	96.1	9.80
9/16	14.5	0.55	0.82	24,800	110	11.2	27,200	121	12.3
5/8	16	0.68	1.01	30,600	136	13.9	33,600	149	15.2
3/4	19	0.97	1.44	43,600	194	19.8	48,000	214	21.8
7/8	22	1.32	1.96	59,000	262	26.8	65,000	289	29.5
1	26	1.73	2.57	76,600	341	34.7	84,400	375	38.3
1 1/8	29	2.19	3.26	96,400	429	43.7	106,200	472	48.2
1 1/4	32	2.70	4.02	118,400	527	53.7	130,200	579	59.1
1 3/8	35	3.27	4.87	142,600	634	64.7	156,800	697	71.1
1 1/2	38	3.89	5.79	168,800	751	76.6	185,600	826	84.2

*These strengths apply only when a test is conducted with both ends fixed. When in use, the strength of these ropes may be significantly reduced if one end is free to rotate.

Table 4-17
6×19 Classification Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Independent Wire-Rope Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
				Nominal Strength								
Nominal Diameter		Approx. Mass		Improved Plow Steel			Extra Improved Plow Steel			Extra Extra Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
1/2	13	0.46	0.68	23,000	102	10.4	26,600	118	12.1	29,200	130	13.2
9/16	14.5	0.59	0.88	29,000	129	13.2	33,600	149	15.2	37,000	165	16.8
5/8	16	0.72	1.07	35,800	159	16.2	41,200	183	18.7	45,400	202	20.6
3/4	19	1.04	1.55	51,200	228	23.2	58,800	262	26.7	64,800	288	29.4
7/8	22	1.42	2.11	69,200	308	31.4	79,600	354	36.1	87,600	389	39.7
1	26	1.85	2.75	89,800	399	40.7	103,400	460	46.9	113,800	506	51.6
1 1/8	29	2.34	3.48	113,000	503	51.3	130,000	678	59.0	143,000	636	64.9
1 1/4	32	2.89	4.30	138,800	617	63.0	159,800	711	72.5	175,800	782	79.8
1 3/8	35	3.50	5.21	167,000	743	75.7	192,000	854	87.1	212,000	943	96.2
1 1/2	38	4.16	6.19	197,800	880	89.7	228,000	1010	103	250,000	1112	113
1 5/8	42	4.88	7.26	230,000	1020	104	264,000	1170	120	292,000	1300	132
1 3/4	45	5.67	8.44	266,000	1180	121	306,000	1360	139	338,000	1500	153
1 7/8	48	6.50	9.67	304,000	1350	138	348,000	1550	158	384,000	1710	174
2	52	7.39	11.0	344,000	1630	156	396,000	1760	180	434,000	1930	197

Table 4-18
6×37 Classification Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Independent Wire-Rope Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
				Nominal Strength								
Nominal Diameter		Approx. Mass		Improved Plow Steel			Extra Improved Plow Steel			Extra Extra Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
1/2	13	0.46	0.68	23,000	102	10.4	26,600	118	12.1	29,200	130	13.2
9/16	14.5	0.59	0.88	29,000	129	13.2	33,600	149	15.2	37,000	165	16.8
5/8	16	0.72	1.07	35,800	159	16.2	41,200	183	18.7	45,400	202	20.6
3/4	19	1.04	1.55	51,200	228	23.2	58,800	262	26.7	64,800	288	29.4
7/8	22	1.42	2.11	69,200	308	31.4	79,600	354	36.1	87,600	389	39.7
1	26	1.85	2.75	89,800	399	40.7	103,400	460	46.9	113,800	506	51.6
1 1/8	29	2.34	3.48	113,000	503	51.3	130,000	578	59.0	143,000	636	64.9
1 1/4	32	2.89	4.30	138,800	617	63.0	159,800	711	72.5	175,800	782	79.8
1 3/8	35	3.50	5.21	167,000	743	75.7	192,000	854	87.1	212,000	943	96.2
1 1/2	38	4.16	6.19	197,800	880	89.7	228,000	1010	103	250,000	1112	113
1 5/8	42	4.88	7.26	230,000	1020	104	264,000	1170	120	292,000	1300	132
1 3/4	45	5.67	8.44	266,000	1180	121	306,000	1360	139	338,000	1500	153
1 7/8	48	6.50	9.67	304,000	1350	138	348,000	1550	158	384,000	1710	174
2	52	7.39	11.0	344,000	1530	156	396,000	1760	180	434,000	1930	197
2 1/8	54	8.35	12.4	384,000	1710	174	442,000	1970	200	488,000	2170	221
2 1/4	58	9.36	13.9	430,000	1910	195	494,000	2200	224	544,000	2420	247
2 3/8	60	10.4	15.5	478,000	2130	217	548,000	2440	249	604,000	2690	274
2 1/2	64	11.6	17.3	524,000	2330	238	604,000	2690	274	664,000	2950	301
2 5/8	67	12.8	19.0	576,000	2560	261	658,000	2930	299	728,000	3240	330
2 3/4	71	14.0	20.8	628,000	2790	285	736,000	3270	333	794,000	3530	360
2 7/8	74	15.3	22.8	682,000	3030	309	796,000	3540	361	864,000	3840	392
3	77	16.6	24.7	740,000	3290	336	856,000	3810	389	936,000	4160	425
3 1/8	80	18.0	26.8	798,000	3550	362	920,000	4090	417	1,010,000	4490	458
3 1/4	83	19.5	29.0	858,000	3820	389	984,000	4380	447	1,086,000	4830	493
3 3/8	87	21.0	31.3	918,000	4080	416	1,074,000	4780	487	1,164,000	5180	528
3 1/2	90	22.7	33.8	982,000	4370	445	1,144,000	5090	519	1,242,000	5520	563
3 3/4	96	26.0	38.7	1,114,000	4960	505	1,290,000	5740	585	1,410,000	6270	640
4	103	29.6	44.0	1,254,000	5580	569	1,466,000	6520	665	1,586,000	7050	720

Table 4-19
6×61 Classification Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Independent Wire-Rope Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
				Nominal Strength					
Nominal Diameter		Approx. Mass		Improved Plow Steel			Extra Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
3 1/2	90	22.7	33.8	966,000	4300	438	1,110,000	4940	503
3 3/4	96	26.0	38.7	1,098,000	4880	498	1,264,000	5620	573
4	103	29.6	44.0	1,240,000	5520	562	1,426,000	6340	647
4 1/4	109	33.3	49.6	1,388,000	6170	630	1,598,000	7110	725
4 1/2	115	37.4	55.7	1,544,000	6870	700	1,776,000	7900	806
4 3/4	122	41.7	62.1	1,706,000	7590	774	1,962,000	8730	890
5	128	46.2	68.8	1,874,000	8340	850	2,156,000	9590	978

Table 4-20
6×91 Classification Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Independent Wire-Rope Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
				Nominal Strength					
Nominal Diameter		Approx. Mass		Improved Plow Steel			Extra Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
4	103	29.6	44.1	1,178,000	5240	534	1,354,000	6020	614
$4^1/_4$	109	33.3	49.6	1,320,000	5870	599	1,518,000	6750	689
$4^1/_2$	115	37.4	55.7	1,468,000	6530	666	1,688,000	7510	766
$4^3/_4$	122	41.7	62.1	1,620,000	7210	735	1,864,000	8290	846
5	128	46.2	68.7	1,782,000	7930	808	2,048,000	9110	929
$5^1/_4$	135	49.8	74.1	1,948,000	8670	884	2,240,000	9960	1016
$5^1/_2$	141	54.5	81.1	2,120,000	9430	962	2,438,000	10800	1106
$5^3/_4$	148	59.6	88.7	2,296,000	10200	1049	2,640,000	11700	1198
6	154	65.0	96.7	2,480,000	11000	1125	2,852,000	12700	1294

Table 4-21
8×19 Classification Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Independent Wire-Rope Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
				Nominal Strength					
Nominal Diameter		Approx. Mass		Improved Plow Steel			Extra Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
$^1/_2$	13	0.47	0.70	20,200	89.9	9.16	23,400	104	10.5
$^9/_{16}$	14.5	0.60	0.89	25,600	114	11.6	29,400	131	13.3
$^5/_8$	16	0.73	1.09	31,400	140	14.2	36,200	161	16.4
$^3/_4$	19	1.06	1.58	45,000	200	20.4	51,800	230	23.5
$^7/_8$	22	1.44	2.14	61,000	271	27.7	70,000	311	31.8
1	26	1.88	2.80	79,200	352	35.9	91,000	405	41.3
$1^1/_8$	29	2.39	3.56	99,600	443	45.2	114,600	507	51.7

Table 4-22
19×7 Construction Wire Rope, Bright (Uncoated) or Drawn-Galvanized Wire, Wire Strand Core [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
				Nominal Strength*					
Nominal Diameter		Approx. Mass		Improved Plow Steel			Extra Improved Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
$^1/_2$	13	0.45	0.67	19,700	87.6	8.94	21,600	96.1	9.80
$^9/_{16}$	14.5	0.58	0.86	24,800	110	11.2	27,200	121	12.3
$^5/_8$	16	0.71	1.06	30,600	136	13.9	33,600	149	15.2
$^3/_4$	19	1.02	1.52	43,600	194	19.8	48,000	214	21.8
$^7/_8$	22	1.39	2.07	59,000	262	26.8	65,000	289	29.5
1	26	1.82	2.71	76,600	341	34.7	84,400	375	38.3
$1^1/_8$	29	2.30	3.42	96,400	429	43.7	106,200	472	48.2
$1^1/_4$	32	2.84	4.23	118,400	527	53.7	130,200	579	59.1
$1^3/_8$	35	3.43	5.10	142,600	634	64.7	156,800	697	71.1
$1^1/_2$	38	40.8	6.07	168,800	751	76.6	185,600	826	84.2

*These strengths apply only when a test is conducted with both ends fixed. When in use, the strength of these ropes may be significantly reduced if one end is free to rotate.

Table 4-23
6×25 "B," 6×27 "H," 6×30 "G," 6×31 "V"
Flattened Strand Construction Wire Rope
Bright (Uncoated) or Drawn-Galvanized Wire [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
				Typical Nominal Strength*					
Nominal Diameter		Approx. Mass		Improved Plow Steel			Extra Imp. Plow Steel		
in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
1/2	13	0.47	0.70	25,400	113	11.5	28,000	125	12.7
9/16	14.5	0.60	0.89	32,000	142	14.5	35,200	157	16.0
5/8	16	0.74	1.10	39,400	175	17.9	43,400	193	19.7
3/4	19	1.06	1.58	56,400	251	25.6	62,000	276	28.1
7/8	22	1.46	2.17	76,000	330	34.5	83,800	373	38.0
1	26	1.89	2.81	98,800	439	44.8	108,800	484	49.3
1 1/8	29	2.39	3.56	124,400	553	56.4	137,000	609	62.1
1 1/4	32	2.95	4.39	152,600	679	69.2	168,000	747	76.2
1 3/8	35	3.57	5.31	183,600	817	83.3	202,000	898	91.6
1 1/2	38	4.25	6.32	216,000	961	98.0	238,000	1,060	108
1 5/8	42	4.99	7.43	254,000	1,130	115	280,000	1,250	127
1 3/4	45	5.74	8.62	292,000	1,300	132	322,000	1,430	146
1 7/8	48	6.65	9.90	334,000	1,490	151	368,000	1,640	167
2	52	7.56	11.2	378,000	1,680	171	414,000	1,840	188

(text continued from page 567)

The 6×19 classification ropes shall contain 6 strands that are made up of 15 through 26 wires, of which no more than 12 are outside wires fabricated in one operation. See Tables 4-15 and 4-17 and Figures 4-38 to 4-43.

The 6×37 classification ropes shall contain six strands that are made up of 27 through 49 wires, of which no more than 18 are outside wires fabricated in one operation. See Tables 4-15 and 4-18 and Figures 4-44 to 4-51.

The 6×61 classification ropes shall contain six strands that are made up of 50 through 74 wires, of which no more than 24 are outside wires fabricated in one operation. See Table 4-19 and Figures 4-52 and 4-53.

The 6×91 classification wire rope shall have six strands that are made up of 75 through 109 wires, of which no more than 30 are outside wires fabricated in one operation. See Table 4-20 and Figures 4-54 and 4-55.

The 8×19 classification wire rope shall have eight strands that are made up of 15 through 26 wires, of which no more than 12 are outside wires fabricated in one operation. See Table 4-21 and Figures 4-56 and 4-57.

The 18×7 and 19×7 wire rope shall contain 18 or 19 strands, respectively. Each strand is made up of seven wires. It is manufactured counterhelically laying an outer 12-strand layer over an inner 6×7 or 7×7 wire rope. This produces a rotation-resistant characteristic. See Tables 4-16 and 4-22 and Figures 4-58 and 4-59.

The 6×25 "B," 6×27 "H," 6×30 "G," and 6×31 "V" flattened strand wire rope shall have six strands with 24 wires fabricated in two operations around a semitriangular shaped core. See Table 4-23 and Figures 4-60 to 4-63.

In the manufacture of uniform-diameter wire rope, wires shall be continuous. If joints are necessary in individual wires, they shall be made, prior to fabrication

(text continued on page 575)

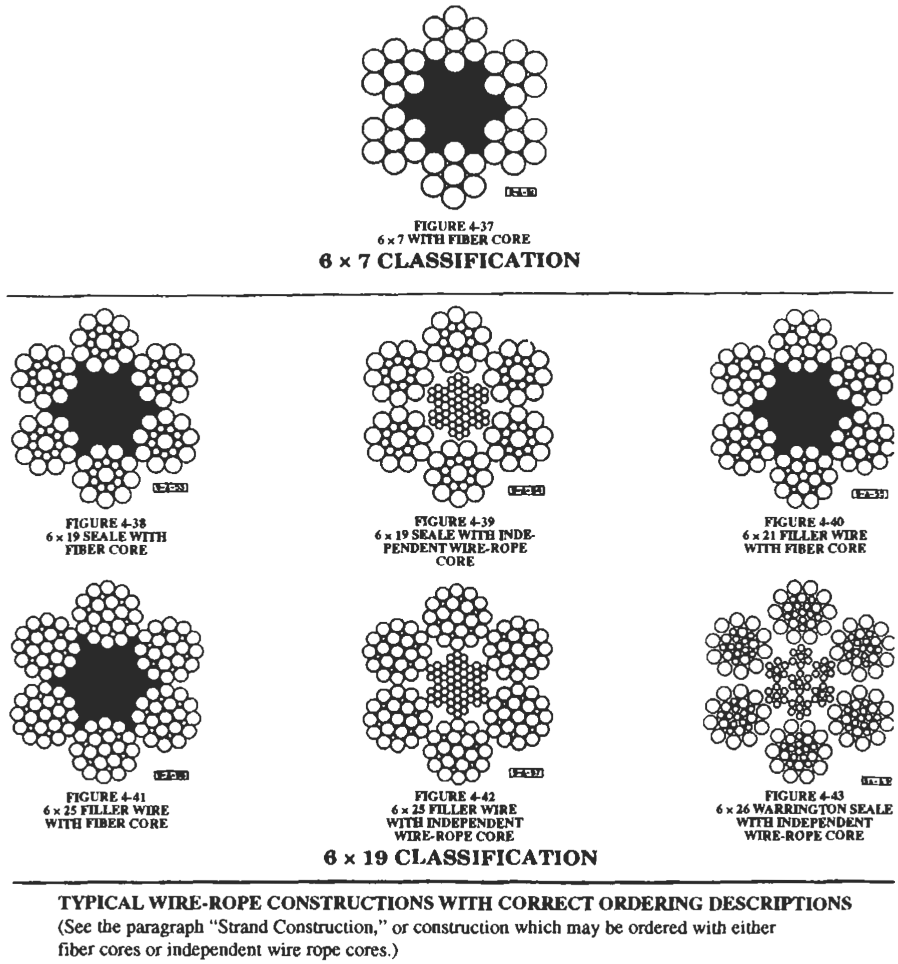
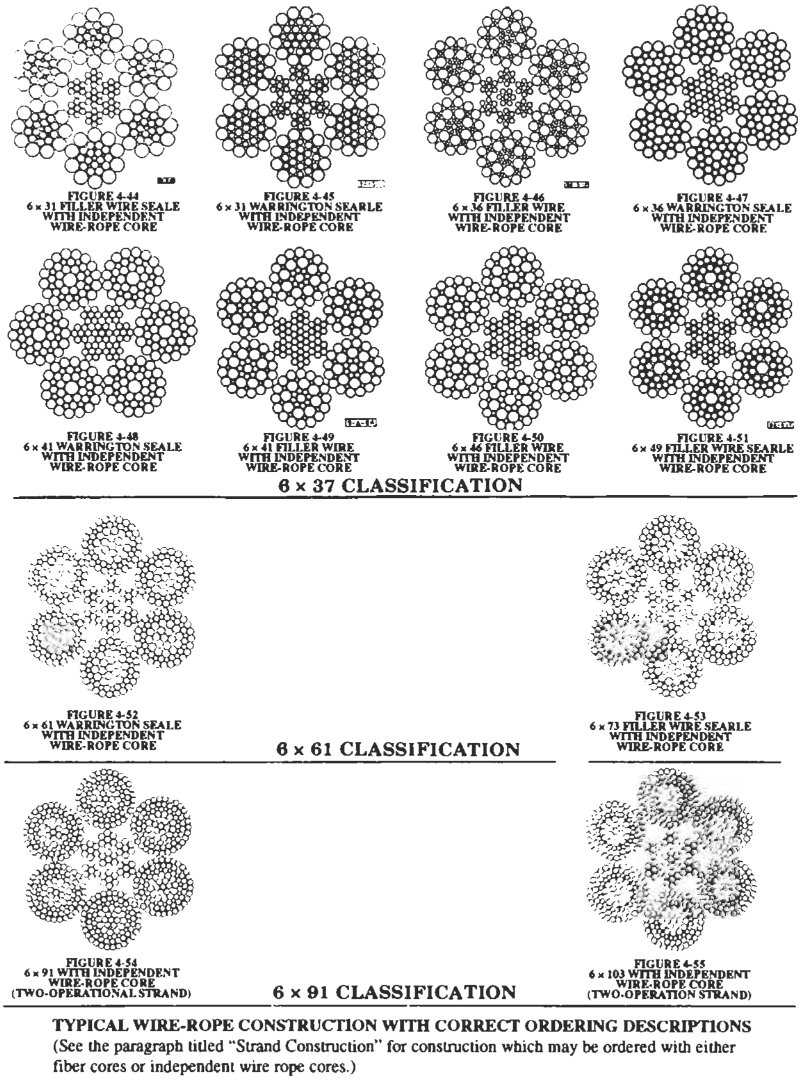
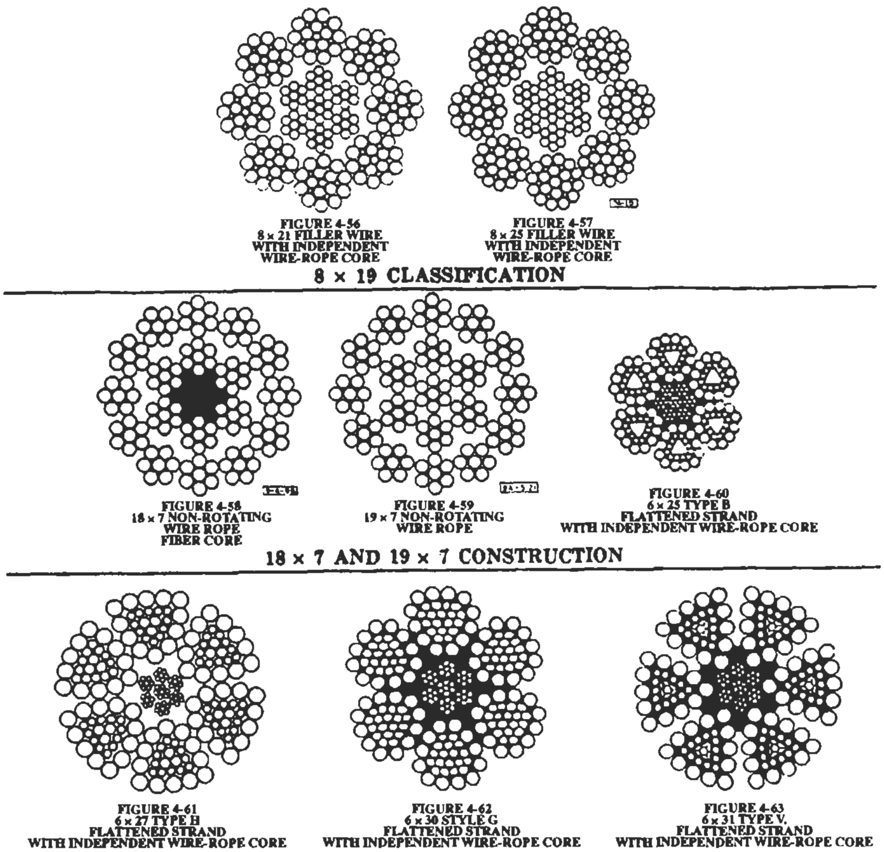


Figure 4-37. 6×7 with fiber core [12].
Figure 4-38. 6×19 seale with fiber core [12].
Figure 4-39. 6×19 seale with independent wire-rope core [12].
Figure 4-40. 6×21 filler wire with fiber core [12].
Figure 4-41. 6×25 filler wire with fiber core [12].
Figure 4-42. 6×25 filler wire with independent wire-rope core [12].
Figure 4-43. 6×26 Warrington seale with independent wire-rope core [12].



TYPICAL WIRE-ROPE CONSTRUCTION WITH CORRECT ORDERING DESCRIPTIONS
(See the paragraph titled "Strand Construction" for construction which may be ordered with either fiber cores or independent wire rope cores.)

Figure 4-44. 6×31 filler wire seale with independent wire-rope core [12].
Figure 4-45. 6×31 Warrington seale with independent wire-rope core [12].
Figure 4-46. 6×36 filler wire with independent wire-rope core [12].
Figure 4-47. 6×36 Warrington seale with independent wire-rope core [12].
Figure 4-48. 6×41 Warrington seale with independent wire-rope core [12].
Figure 4-49. 6×41 filler wire with independent wire-rope core [12].
Figure 4-50. 6×46 filler wire with independent wire-rope core [12].
Figure 4-51. 6×49 filler wire seale with independent wire-rope core [12].
Figure 4-52. 6×61 Warrington seale with independent wire-rope core [12].
Figure 4-53. 6×73 filler wire seale with independent wire-rope core [12].
Figure 4-54. 6×91 with independent wire-rope core (two-operation strand) [12].
Figure 4-55. 6×103 with independent wire-rope core (two-operation strand) [12].



TYPICAL WIRE-ROPE CONSTRUCTIONS WITH CORRECT ORDERING DESCRIPTIONS
(See the paragraph titled "Strand Construction" for construction which may be ordered with either fiber cores or independent wire rope cores.)

Figure 4-56. 8×21 filler wire with independent wire-rope core [12].
Figure 4-57. 8×25 filler wire with independent wire-rope core [12].
Figure 4-58. 18×7 nonrotating wire rope with fiber core [12].
Figure 4-59. 19×7 nonrotating wire rope [12].
Figure 4-60. 6×25 type B flattened strand with independent wire-rope core [12].
Figure 4-61. 6×27 type H flattened strand with independent wire-rope core [12].
Figure 4-62. 6×30 style G flattened strand with independent wire-rope core [12].
Figure 4-63. 6×31 type V flattened strand with independent wire-rope core [12].

(text continued from page 571)

of the strand, by brazing or electric welding. Joints shall be spaced in accordance with the equation

$$J = 24D \tag{4-23}$$

where J = minimum distance between joints in main wires in any one strand in in. mm
D = nominal diameter of wire rope in in. mm

Wire rope is most often furnished preformed, but can be furnished nonpreformed, upon special request by the purchaser. A preformed rope has the strands shaped to the helical form they assume in the finished rope before the strands have been fabricated into the rope. The strands of such preformed rope shall not spring from their normal position when the seizing bands are removed. Cable tool is one of the few applications for which nonpreformed is still used.

The Lay of Finished Rope. Wire rope shall be furnished right lay or left lay and regular lay or Lang lay as specified by the purchaser (see Figure 4-64). If not otherwise specified on the purchase order, right-lay, regular-lay rope shall be furnished. For 6×7 wire ropes, the lay of the finished rope shall not exceed eight times the nominal diameter. For 6×19, 6×37, 6×61, 6×91, and 8×19 wire rope, the lay of the finished rope shall not exceed $7\frac{1}{4}$ times the nominal diameter. For flattened strand rope designations 6×25 "B," 6×27 "H," 6×30 "G," and 6×31 "V," the lay of the finished rope shall not exceed eight times the nominal diameter.

Diameter of Ropes and Tolerance Limits. The diameter of a wire rope shall be the diameter of a circumscribing circle and shall be measured at least 5 ft (1.52 m) from properly seized end with a suitable caliper (see Figure 4-65). The diameter tolerance* of wire rope shall be

Nominal inch diameter: −0% to +5%
Nominal mm diameter: −1% to +4%

Diameter of Wire and Tolerance Limits. In separating the wire rope for gaging of wire, care must be taken to separate the various sizes of wire composing the different layers of bright (uncoated), drawn-galvanized, or galvanized wires in the strand. In like-positioned wires total variations of wire diameters shall not exceed the values of Table 4-24.

Fiber Cores. For all wire ropes, all fiber cores shall be hard-twisted, best-quality, manila, sisal, polypropylene, or equivalent. For wire ropes of uniform diameter, the cores shall be of uniform diameter and hardness, effectively supporting the strands. Manila and sisal cores shall be thoroughly impregnated with a suitable lubricating compound free from acid. Jute cores shall not be used.

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*A question may develop as to whether or not the wire rope complies with the oversize tolerance. In such cases, a tension of not less than 10% nor more than 20% of nominal required breaking strength is applied to the rope, and the rope is measured while under this tension.

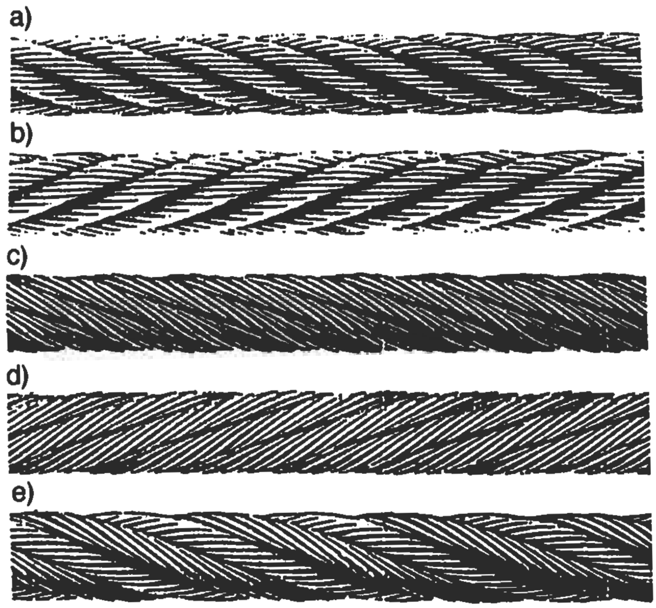

Figure 4-64. Right and left lay, and regular and Lang lay [12].

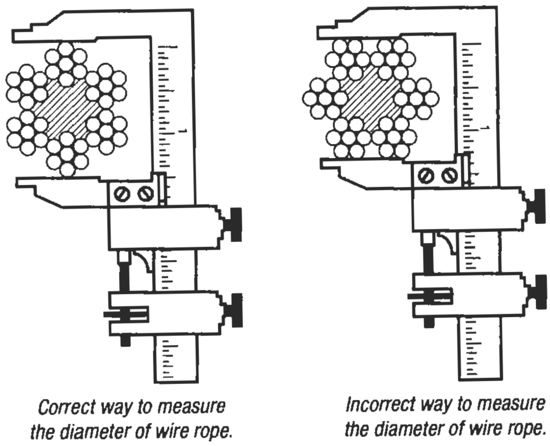

Figure 4-65. Measurement of diameter [12].

Table 4-24
Wire Diameter Tolerance [12]

(1)	(2)	(3)	(4)	(5)	(6)
		Total Variation			
Wire Diameters		Uncoated (bright) and Drawn Galvanized Wires		Galvanized Wires	
inches	mm	inches	mm	inches	mm
0.018 - 0.027	0.46 - 0.69	0.0015	0.038	—	—
0.028 - 0.059	0.70 - 1.50	0.0020	0.051	0.0035	0.089
0.060 - 0.092	1.51 - 2.34	0.0025	0.064	0.0045	0.114
0.093 - 0.141	2.35 - 3.58	0.0030	0.076	0.0055	0.140
0.142 and larger	3.59 and larger	0.0035	0.075	0.0075	0.190

Lengths. Length of wire rope shall be specified by the purchaser. If minimum length is critical to the application, it shall be specified and conform to the following tolerances.

1300 ft (400 m): -0 to +5%
> 1300 ft (400 m): Original tolerance
+ 66 ft (20 m) per each additional 3280 ft (1000 m) or part thereof.

If minimum is not critical to the application, it shall conform to the following tolerances.

1300 ft (400 m): ±2.5%
> 1300 ft (400 m): Original tolerance
± 33 ft (10 m) per each additional 3280 ft (1000 m) or part thereof.

Lubrication. All wire rope, unless otherwise specified, shall be lubricated and impregnated in the manufacturing process with a suitable compound for the application in amounts best adapted to individual territories. This lubricant should thoroughly protect the ropes internally and externally to minimize rust or corrosion until the rope is put in service.

Mooring Wire Rope

Mooring wire rope is used as anchor lines in spread mooring systems, and shall comply with the all the provisions of Wire Rope.

Wire rope for this use should be one operation, right lay, regular lay, independent wire rope core, preformed, galvanized or bright. The nominal strength of galvanized and bright mooring wire rope shall be as specified in Table 4-25. For bright mooring wire ropes, the wire grade shall comply with the requirements for Extra Improved Plow Steel, Table 4.2.8 or ISO Std 2232* value of 1770 N/mm^2.

*International Organization for Standardization, Standard 2232-1973, "Drawn Wire for General Purpose Non-Alloy Steel Wire Ropes—Specifications," available from American National Standards Institute, 1430 Broadway, New York, New York 10018.

Table 4-25
6×19, 6×37, and 6×61 Construction Mooring Wire Rope, Independent Wire-Rope Core [12]

1	2	3	4	5	6	7	8	9	10	11
Construction Classification	Nominal Diameter		Approximate Mass		Nominal Strength					
					Galvanized			Bright		
	in.	mm	lb/ft	kg/m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
6x19	1	26	1.85	2.75	93,060	414	42.2	96,800	426	43.5
6x19	1⅛	29	2.34	3.48	117,000	520	53.1	119,000	530	54.1
6x19, 6x37	1¼	32	2.89	4.30	143,800	640	65.2	145,000	646	65.9
6x19, 6x37	1⅜	35	3.50	5.21	172,800	769	78.4	174,000	773	78.8
6x19, 6x37	1½	38	4.16	6.19	205,200	913	93.1	205,000	911	92.9
6x19, 6x37	1⅝	42	4.88	7.26	237,600	1,060	108	250,000	1,110	113
6x19, 6x37	1¾	45	5.67	8.44	275,400	1,230	125	287,000	1,280	130
6x19, 6x37	1⅞	48	6.50	9.67	313,200	1,390	142	327,000	1,450	148
6x19, 6x37	2	52	7.39	11.0	356,400	1,590	162	369,000	1,640	167
6x19, 6x37	2⅛	54	8.35	12.4	397,800	1,770	180	413,000	1,840	188
6x19, 6x37	2¼	58	9.36	13.9	444,600	1,980	202	461,000	2,060	209
6x19, 6x37	2⅜	60	10.4	15.5	493,200	2,190	224	528,000	2,350	239
6x19, 6x37	2½	64	11.6	17.0	543,600	2,420	247	604,000	2,690	274
6x37	2⅝	67	12.8	18.6	595,800	2,650	270	658,000	2,930	299
6x37	2¾	71	14.0	20.9	649,800	2,890	295	736,000	3,270	333
6x37	2⅞	74	15.3	22.7	705,600	3,140	320	796,000	3,540	361
6x37	3	77	16.6	24.6	765,000	3,400	347	856,000	3,810	389
6x37	3⅛	80	18.0	26.6	824,400	3,670	374	920,000	4,090	417
6x37, 6x61	3¼	83	19.5	28.6	885,600	3,940	402	984,000	4,380	447
6x37, 6x61	3⅜	87	21.0	31.4	952,200	4,240	432	1,074,000	4,780	487
6x37, 6x61	3½	90	22.7	33.6	1,015,000	4,520	460	1,144,000	5,090	519
6x37, 6x61	3¾	96	26.0	38.2	1,138,000	5,060	516	1,290,000	5,740	585
6x37, 6x61	4	103	29.6	44.0	1,283,000	5,710	582	1,466,000	6,520	665
	4¼	109	33.3	49.3	1,438,000	6,400	652	1,606,000	7,140	728
	4½	115	37.4	54.9	1,598,000	7,110	725	1,774,000	7,890	805
	4¾	122	41.7	61.8	1,766,000	7,860	801	1,976,000	8,790	896

NOTE: For tests see Paragraph titled "Acceptance."

Torpedo Lines

Torpedo lines shall be bright (uncoated) or drawn-galvanized, and shall be right, regular lay. The lay of the finished rope shall not exceed eight times the nominal diameter.

Torpedo lines shall be made of five strands of five wires each, or five strands of seven wires each. The strands of the 5×5 construction shall have one center wire and four outer wires of one diameter, fabricated in one operation. The five strands shall be laid around one fiber or cotton core (see Figure 4-66). The strands of the 5×7 construction shall have one center wire and six outer wires of one diameter, fabricated in one operation. The strands shall be laid around one fiber or cotton core (see Figure 4-67).

The four outer wires in each strand of the 5×5 construction [both bright (uncoated) and drawn-galvanized] and all the wires in each strand of the 5×7 construction [both bright (uncoated) and drawn-galvanized] shall have the breaking strengths as in Tables 4-15 and 4-16 for the specified grade and applicable wire size. The center wire of the 5×5 construction shall be hard drawn or annealed and shall not be required to meet the minimum breaking strength specified for the outer wires (the center wire represents about 5% of the total metallic area of the rope and is substantially a filler wire).

The nominal strength of torpedo lines shall be as specified in Tables 4-26 and 4-27. When testing finished ropes to their breaking strength, suitable sockets

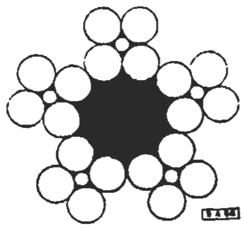

Figure 4-66. 5×5 construction torpedo line [12].

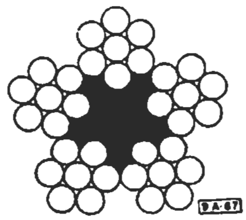

Figure 4-67. 5×7 construction torpedo line [12].

Table 4-26
5×5 Construction Torpedo Lines [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Nominal Diameter of Rope		Approx. Mass		Nominal Strength					
				Plow Steel			Improved Plow Steel		
in.	mm	lb/100 ft	kg/100 m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
1/8	3.18	2.21	3.29	1,120	4.98	0.51	1,290	5.74	0.59
9/64	3.57	2.80	4.16	1,410	6.27	0.64	1,620	7.21	0.74
5/32	3.97	3.46	5.15	1,740	7.74	0.79	2,000	8.90	0.91
3/16	4.76	4.98	7.41	2,490	11.08	1.13	2,860	12.72	1.30
1/4	6.35	8.86	13.91	4,380	19.48	1.99	5,030	22.37	2.28
5/16	7.94	13.80	20.54	6,780	30.16	3.08	7,790	34.65	3.53

Table 4-27
5×7 Construction Torpedo Lines [12]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Nominal Diameter of Rope		Approx. Mass		Nominal Strength					
				Plow Steel			Improved Plow Steel		
in.	mm	lb/100 ft	kg/100 m	lb	kN	Metric Tonnes	lb	kN	Metric Tonnes
1/8	3.18	2.39	3.56	1,210	5.38	0.55	1,400	6.23	0.64
9/64	3.57	3.02	4.49	1,530	6.81	0.69	1,760	7.83	0.80
5/32	3.97	3.73	5.55	1,890	8.41	0.86	2,170	9.65	0.98
3/16	4.76	5.38	8.01	2,700	12.01	1.23	3,110	13.83	1.41
1/4	6.35	9.55	14.21	4,760	21.17	2.16	5,470	24.33	2.48
5/16	7.94	14.90	22.17	7,380	32.83	3.35	8,490	37.77	3.85

or other acceptable means of holding small cords shall be used. The length of tension test specimens shall be not less than 1 ft (0.305 m) between attachments. If the first specimen fails at a value below the specified nominal strength, two additional specimens from the same rope shall be tested, one of which must comply with the nominal strength requirement.

The diameter of the ropes shall be not less than the specified diameter. Torpedo-line lengths shall vary in 500-ft (152.4 m) multiples.

Well-Measuring Wire

Well-measuring wire shall be in accordance with Table 4-28, and shall consist of one continuous piece of wire without brazing or welding of the finished wire. The wire shall be made from the best quality of specified grade of material with good workmanship and shall be free from defects that might affect its appearance or serviceability. Coating on well-measuring wire shall be optional with the purchaser.

A specimen of 3-ft (0.91 m) wire shall be cut from each coil of well-measuring wire. One section of this specimen shall be tested for elongation simultaneously with the test for tensile strength. The ultimate elongation shall be measured on a 10-in. (254 mm) specimen at instant of rupture, which must occur within the 10-in. (254 mm) gage length. To determine elongation, a 100,000-psi (690-mPa) stress shall be imposed upon the wire at which the extensometer is applied. Directly to the extensometer reading shall be added 0.4% to allow for the initial elongation occurring before application of extensometer.

The remaining section of the 3-ft (0.91-m) test specimen shall be gaged for size and tested for torsional requirements.

If, in any individual test, the first specimen fails, not more than two additional specimens from the same wire shall be tested. The average of any two tests showing failure or acceptance shall be used as the value to represent the wire.

Well-Measuring Strand

Well-measuring strand shall be bright (uncoated) or drawn-galvanized.

Well-measuring strand shall be left lay. The lay of the finished strand shall not exceed 10 times the nominal diameter.

Well-measuring strands may be of various combinations of wires but are commonly furnished in 1×16 (1-6-9) and 1×19 (1-6-12) constructions.

Well-measuring strands shall conform to the properties listed in Table 4-29.

To test finished strands to their breaking strength, suitable sockets or other acceptable means of holding small cords shall be used.

Wire Guy Strand and Structural Rope and Strand

Galvanized wire guy strand shall conform to ASTM A-475: "Zinc-Coated Steel Wire Strand."* Aluminized wire guy strand shall conform to ASTM A-474: "Aluminum Coated Steel Wire Strand."* Galvanized structural strand shall conform to ASTM A-586: "Zinc-Coated Steel Structural Strand."* Galvanized structural rope shall conform to ASTM A-603: "Zinc-Coated Steel Structural Wire Rope."*

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*American Society for Testing and Materials, 1916 Race Street, Philadelphia, Pennsylvania 19103.

Table 4-28
Requirements for Well-Measuring Wire, Bright or Drawn-Galvanized Carbon Steel* [12]

1	2	3	4	5	6	7
Nominal Diameter in.	**0.066**	**0.072**	**0.082**	**0.092**	**0.105**	**0.108**
mm	**1.68**	**1.83**	**2.08**	**2.34**	**2.67**	**2.74**
Tolerance on diameter in. ±0.001		±0.001	±0.001	±0.001	±0.001	±0.001
mm	±0.03	±0.03	±0.03	±0.03	±0.03	±0.03
Breaking strength						
Minimum lb	811	961	1239	1547	1966	2109
kN	3.61	4.27	5.51	6.88	8.74	9.38
Maximum lb	984	1166	1504	1877	2421	2560
kN	4.38	5.19	6.69	8.35	10.77	11.38
Elongation to 10 in. (254 mm), per cent Minimum	1½	1½	1½	1½	1½	1½
Torsions, minimum number of twists in 8 in. (203 mm)	32	29	26	23	20	19

*For well-measuring wire of other materials or coatings, refer to supplier for physical properties.

Table 4-29
Requirements for Well-Servicing Strand Bright or Drawn-Galvanized Carbon Steel [12]

1		2	3
Nominal Diameter	Inches	3/16"	1/4"
	MM	4.8	6.4
Tolerances on Diameter	Inches	–0" +.013"	–0" +.015"
	MM	–.048 +.288	–.064 +.320
Nominal Breaking Strength	Lbs.	4700	8200
	KN	20.9	36.5
Approximate Mass	Lbs./100′	7.3	12.7
	kg/100′	3.3	5.8

Packing and Marking

Finished wire rope, unless otherwise specified, shall be shipped on substantial round-head reels. Reels on which sand lines, drilling lines, or casing lines are shipped shall have round arbor holes of 5 in. (127 mm) to $5\frac{3}{4}$-in. (146-mm) diameter. When reel is full of rope, there shall be a clearance of not less than 2 in. (51 mm) between the full reel and the outside diameter of the flange.

The manufacturer shall protect the wire rope on reels from damage by moisture, dust, or dirt with a water-resistant covering of builtup material, such as tar paper and burlap, or similar material.

The following data shall be plainly marked on the face of the wire-rope reel:

1. Name of manufacturer.
2. Reel number.
3. API monogram only by authorized manufacturers.

4. Grade (plow steel, improved plow steel, or extra improved plow steel).
5. Diameter of rope, in. (mm).
6. Length of rope, ft (m).
7. Type of construction (Warrington, Seale, or Filler Wire).
8. Type of core (fiber, wire, plastic, or fiber and plastic).

Inspection and Rejection

The manufacturer will, on request of the purchaser, conduct tests as called for in specifications on reasonable notice from the purchaser. During the tests, the manufacturer will afford opportunity to the purchaser's representative to present.

The manufacturer, when delivering wire rope with the API monogram and grade designation, should warrant that such material complies with the specification. The wire rope rejected under specifications should not be wound on reels bearing the API monogram, or sold as API wire rope. When the wire rope wound on reels bearing the API monogram is rejected, the monogram shall be removed.

It is recommended that whenever possible, the purchaser, upon receipt, shall test all new wire rope purchased in accordance with specifications. If a rope fails to render satisfactory service, it is impractical to retest such used rope. It is therefore required that the purchaser shall preserve at least one test specimen of all new rope purchased, length of specimen to be at least 10 ft (3.05 m), properly identified by reel number, etc. Care must be taken that no damage will result by storage of specimen.

If the purchaser is not satisfied with the wire rope service, he or she shall send the properly preserved sample or a sample of the rope from an unused section to any testing laboratory mutually agreed upon by the purchaser and the manufacturer, with instructions to make a complete API test, and notify the manufacturer to have a representative present. If the report indicates compliance with specifications, the purchaser shall assume cost of testing; otherwise, the manufacturer shall assume the expense and make satisfactory adjustments not exceeding full purchase price of the rope. If the report indicates noncompliance with specifications, the testing laboratory shall forward a copy of the test report to the manufacturer.

Wire-Rope Sizes and Constructions

Typical sizes and constructions of wire rope for oilfield service are shown in Table 4-30. Because of the variety of equipment designs, the selection of other constructions than those shown is justifiable.

In oilfield service, wire rope is often referred to as wire line or cable. For clarity, these various expressions are incorporated in this recommended practice.

Field Care and Use of Wire Rope

Handling on Reel. When handling wire rope on a reel with a binding or lifting chain, wooden blocks should always be used between the rope and the chain to prevent damage to the wire or distortion of the strands in the rope. Bars for moving the reel should be used against the reel flange, and not against the rope. The reel should not be rolled over or dropped on any hard, sharp object to protect the rope, and should not be dropped from a truck or platform to avoid damage to the rope and the reel.

Table 4-30
Typical Sizes and Constructions of Wire Rope for Oilfield Service [12]

1	2	3	4
Service and Well Depth	Wire Rope Diameter in.	(mm)	Wire Rope Description (Regular Lay)
Rod and Tubing Pull Lines			
Shallow	1/2 to 3/4 incl.	*(13 to 19)*	6x25 FW or 6x26 WS or 6x31 WS or 18x7[1] or 19x7[1]. PF, LL[1], IPS or EIPS, IWRC
Intermediate	3/4, 7/8	*(19, 22)*	
Deep	7/8 to 1 1/8 incl.	*(22 to 29)*	
Rod Hanger Lines	1/4	*(6.5)*	6x19, PF, RL, IPS, FC
Sand Lines			
Shallow	1/4 to 1/2 incl.	*(6.5 to 13)*	6x7 Bright or Galv.[2], PF, RL, PS or IPS, FC
Intermediate	1/2, 9/16	*(13, 14.5)*	
Deep	9/16, 5/8	*(14.5, 16)*	
Drilling Lines—Cable Tool (Drilling and Cleanout)			
Shallow	5/8, 3/4	*(16, 19)*	6x21 FW, PF or NPF, RL or LL, PS or IPS, FC
Intermediate	3/4, 7/8	*(19, 22)*	
Deep	7/8, 1	*(22, 26)*	
Casing Lines—Cable Tool			
Shallow	3/4, 7/8	*(19, 22)*	6x25 FW or 6x26 WS, PF, RL, IPS or EIPS, FC or IWRC
Intermediate	7/8, 1	*(22, 26)*	
Deep	1, 1/8	*(26, 29)*	
Drilling Lines—Coring and Slim-Hole Rotary Rigs			
Shallow	7/8, 1	*(22, 26)*	6x26 WS, PF, RL, IPS or EIPS, IWRC
Intermediate	1, 1 1/8	*(26, 29)*	6x19 S or 6x26 WS, PF, RL, IPS or EIPS, IWRC
Drillings Lines—Rotary Rigs			
Shallow	1, 1 1/8	*(26, 29)*	6x19 S or 6x21 S or 6x25 FW or FS, PF, RL, IPS or EIPS, IWRC
Intermediate	1 1/8, 1 1/4	*(29, 32)*	
Deep	1 1/4 to 1 3/4 incl.	*(32, 45)*	
Winch Lines—Heavy Duty	5/8 to 7/8 incl.	*(16 to 22)*	6x26 WS or 6x31 WS, PF, RL, IPS or EIPS, IWRC
	7/8 to 1 1/8 incl.	*(22 to 29)*	6x36 WS, PF, RL, IPS or EIPS, IWRC
Horsehead Pumping-Unit Lines			
Shallow	1/2 to 1 1/8 incl.[3]	*(13 to 29)*	6x19 Class or 6x37 Class or 19x7, PF, IPS, FC or IWRC
Intermediate	5/8 to 1 1/8 incl.[4]	*(16 to 29)*	6x19 Class or 6x37 Class, PF, IPS, FC or IWRC
Offshore Anchorage Lines	7/8 to 2 3/4 incl.	*(22 to 70)*	6x19 Class, Bright or Galv., PF, RL, IPS or EIPS, IWRC
	1 3/8 to 4 3/4 incl.	*(35 to 122)*	6x37 Class, Bright or Galv., PF, RL, IPS or EIPS, IWRC
	3 3/4 to 4 3/4 incl.	*(96 to 122)*	6x61 Class, Bright or Galv., PF, RL, IPS or EIPS, IWRC
Mast Raising Lines[5]	1 3/8 and smaller	*(thru 35)*	6x19 Class, PF, RL, IPS or EIPS, IWRC
	1 1/2 and larger	*(38 and up)*	6x37 Class, PF, RL, IPS or EIPS, IWRC
Guideline Tensioner Line	3/4	*(19)*	6x25 FW, PF, RL, IPS or EIPS, IWRC
			Wire Rope Description (Lang Lay)
Riser Tensioner Line	1 1/2, 2	*(38,51)*	6x37 Class or PF, RL, IPS or EIPS, IWRC

Abbreviations:

WS — Warrington-Seale
S — Seale
FS — Flattened strand
FW — Filler-Wire
PS — Plow steel
IPS — Improved plow steel
EIPS — Extra improved plow steel
PF — Preformed
NPF — Non-preformed
RL — Right lay
LL — Left lay
FC — Fiber core
IWRC — Independent wire rope core

[1]Single line pulling of rods and tubing requires left lay construction or 18 x 7 or 19 x 7 construction. Either left lay or right lay may be used for multiple line pulling.
[2]Bright wire sand lines are regularly furnished; galvanized finish is sometimes required.
[3]Applies to pumping units having one piece of wire rope looped over an ear on the horsehead and both ends fastened to a polished-rod equalizer yoke.
[4]Applies to pumping units having two vertical lines (parallel) with sockets at both ends of each line.
[5]See API Spec 4E: *Specification for Drilling and Well Servicing Structures.*

Rolling the reel in or allowing it to stand in any harmful medium such as mud, dirt, or cinders should be avoided. Planking or cribbing will be of assistance in handling the reel as well as in protecting the rope against damage.

Handling during Installation. Blocks should be strung to give a minimum of wear against the sides of sheave grooves. It is also good practice in changing lines to suspend the traveling block from the crown on a single line. This tends to limit the amount of rubbing on guards or spacers, as well as chances for kinks. This is also very effective in pull-through and cutoff procedure.

The reel should be set up on a substantial horizontal axis, so that it is free to rotate as the rope is pulled off, and the rope will not rub against derrick members or other obstructions while being pulled over the crown. A snatch block with a suitable size sheave should be used to hold the rope away from such obstructions.

A suitable apparatus for jacking the reel off the floor and holding it so that it can turn on its axis is desirable. Tension should be maintained on the wire rope as it leaves the reel by restricting the reel movement. A timber or plank provides satisfactory brake action. When winding the wire rope onto the drum, sufficient tension should be kept on the rope to assure tight winding.

To replace a worn rope with a new one, a swivel-type stringing grip for attaching the new rope to the old rope is recommended. This will prevent transferring the twist from one piece of rope to the other. Ensure that the grip is properly applied. The new rope should not be welded to the old rope to pull it through the system.

Care should be taken to avoid kinking a wire rope since a kink can be cause for removal of the wire rope or damaged section. Wire ropes should not be struck with any object, such as a steel hammer, derrick hatchet, or crowbar, that may cause unnecessary nicks or bruises. Even a soft metal hammer can damage a rope. Therefore, when it is necessary to crowd wraps together, any such operation should be performed with the greatest care; and a block of wood should be interposed between the hammer and rope.

Solvent may be detrimental to a wire rope. If a rope becomes covered with dirt or grit, it should be cleaned with a brush.

After properly securing the wire rope in the drum socket, the number of excess or dead wraps or turns specified by the equipment manufacturer should be maintained. Whenever possible, a new wire rope should be run under controlled loads and speeds for a short period after installation. This will help to adjust the rope to working conditions. If a new coring or swabbing line is excessively wavy when first installed, two to four sinker bars may be added on the first few trips to straighten the line.

Care of Wire Rope in Service. The recommendations for handling a reel should be observed at all times during the life of the rope. The design factor should be determined by the following:

$$\text{Design factor} = \frac{B}{W} \tag{4-24}$$

where B = nominal strength of the wire rope in pounds
W = fast line tension

When a wire rope is operated close to the minimum design factor, the rope and related equipment should be in good operating condition. At all times, the operating personnel should minimize shock, impact, and acceleration or deceleration of loads. Successful field operations indicate that the following design factors should be regarded as minimum:

	Minimum Design Factor
Cable-tool line	3
Sand line	3
Rotary drilling line	3
Hoisting service other than rotary drilling	3
Mast raising and lowering line	2.5
Rotary drilling line when setting casing	2
Pulling on stuck pipe and similar infrequent operations	2

Wire-rope life varies with the design factor; therefore, longer rope life can generally be expected when relatively high design factors are maintained.

To calculate the design factor for multipart string-ups, Figures 4-68 and 4-69 can be used to determine the value of W. W is the fast line tension and equals the fast line factor* times the hook load weight indicator reading. As an example, see below:

drilling line	$1\frac{3}{8}$ in. (35 mm) EIPS
number of lines	10
hook load	400,000 lb (181.4 tons)

Sheaves are roller bearing type.

From Figure 4-68, Case A, the fast line factor is 0.123. The fast line tension is then 400,00 lb (181.4 t) × 0.123 = 49,200 lb (22.3 t) = W. Following the formula above, the design factor is then the nominal strength of $1\frac{3}{8}$ in. (35 mm) EIPS drilling line divided by the fast line tension, or 192,000 lb (87.1 tons) ÷ 49,200 lb (22.3 t) = 3.9.

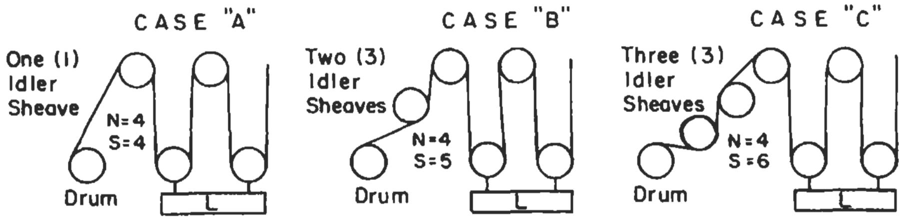
When working near the minimum design factor, consideration should be given to the efficiencies of wire rope bent around sheaves, fittings, or drums. Figure 4-70 shows how rope can be affected by bending.

Rope should be kept tightly and evenly wound on the drums. Sudden, severe stresses are injurious to a wire rope, and should be reduced to a minimum. Experience has indicated that wear increases with speed; economy results from moderately increasing the load and diminishing the speed. Excessive speeds may injure wire rope. Care should be taken to see that the clamps used to fasten the rope for dead ending do not kink, flatten, or crush the rope.

Wire ropes are well lubricated when manufactured; however, this lubrication will not last throughout the entire service life of the rope. Periodically, therefore, the rope will need to be field lubricated. When necessary, lubricate the rope with a good grade of lubricant that will penetrate and adhere to the rope, and that is free from acid or alkali.

The clamps used to fasten lines for dead ending shall not kink, flatten, or crush the rope. The rotary line dead-end tie down is equal in importance to any other part of the system. The dead-line anchorage system shall be equipped with a drum and clamping device strong enough to withstand the loading, and designed to prevent wire line damage that would affect service over the sheaves in the system.

*The fast line factor is calculated considering the tensions needed to overcome sheave bearing friction.



L = Load; S = No. of Sheaves; N = No. of Rope Parts Supporting Load

FAST LINE TENSION = FAST LINE FACTOR X LOAD

1	2	3	4	5	6	7	8	9	10	11	12	13
	Plain Bearing Sheaves						Roller Bearing Sheaves					
	K = 1.09*						K = 1.04*					
	Efficiency			Fast Line Factor			Efficiency			Fast Line Factor		
N	Case A	Case B	Case C	Case A	Case B	Case C	Case A	Case B	Case C	Case A	Case B	Case C
2	.880	.807	.740	.368	.620	.675	.943	.907	.872	.530	.551	.574
3	.844	.774	.710	.395	.431	.469	.925	.889	.855	.360	.375	.390
4	.810	.743	.682	.309	.336	.367	.908	.873	.839	.275	.286	.298
5	.778	.714	.655	.257	.280	.305	.890	.856	.823	.225	.234	.243
6	.748	.686	.629	.223	.243	.265	.874	.840	.808	.191	.198	.206
7	.719	.660	.605	.199	.216	.236	.857	.824	.793	.167	.173	.180
8	.692	.635	.582	.181	.197	.215	.842	.809	.778	.148	.154	.161
9	.666	.611	.561	.167	.182	.198	.826	.794	.764	.135	.140	.145
10	.642	.589	.540	.156	.170	.185	.811	.780	.750	.123	.128	.133
11	.619	.568	.521	.147	.160	.175	.796	.766	.736	.114	.119	.124
12	.597	.547	.502	.140	.152	.166	.782	.752	.723	.106	.111	.115
13	.576	.528	.485	.133	.145	.159	.768	.739	.710	.100	.104	.108
14	.556	.510	.468	.128	.140	.153	.755	.725	.698	.095	.099	.102
15	.537	.493	.452	.124	.135	.147	.741	.713	.685	.090	.094	.097

$$\text{EFFICIENCY} = \frac{(K^N - 1)}{K^S N\,(K-1)} \qquad \text{Fast Line Factor} = \frac{1}{N \times \text{EFFICIENCY}}$$

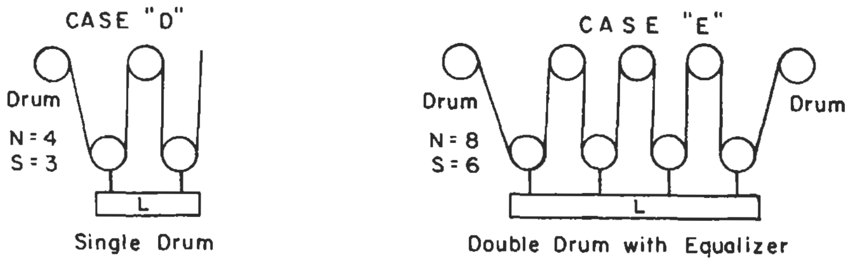
NOTE: The above cases apply also where the rope is dead ended at the lower or traveling block or derrick floor after passing over a dead sheave in the crown.

*In these tables the K factor for sheave friction is 1.09 for plain bearings and 1.04 for roller bearings. Other K factors can be used if recommended by the equipment manufacturer.

Figure 4-68. Efficiency of wire-rope reeving for multiple sheave blocks, Cases A, B, and C [11].

The following precautions should be observed to prevent premature wire breakage in drilling lines.

1. *Cable-tool drilling lines.* Movement of wire rope against metallic parts can accelerate wear. This can also create sufficient heat to form martensite, causing embrittlement of wire and early wire rope removal. Such also can be formed by friction against the casing or hard rock formation.



L = Load; S = No. of Sheaves; N = No. of Rope Parts Supporting Load
(Not counting equalizer)

FAST LINE TENSION = FAST LINE FACTOR X LOAD

1	2	3	4	5	6	7	8	9
	Plain Bearing Sheaves K = 1.09*				Roller Bearing Sheaves K = 1.04*			
	Efficiency		Fast Line Factor		Efficiency		Fast Line Factor	
N	Case D	Case E	Case D	Case E	Case D	Case E	Case D	Case E
2	.959	1.000	.522	.500	981	1.000	.510	.500
3	.920	. . . .	.362	. . . .	.962	. . . .	.346	. . . .
4	.883	.959	.283	.261	.944	.981	.265	.255
5	.848	. . . .	.236	. . . .	.926	. . . .	.216	. . . .
6	.815	.920	.204	.181	.909	.962	.183	.173
7	.784	. . . .	.182	. . . .	.892	. . . .	.160	. . . .
8	.754	.883	.166	.141	.875	.944	.143	.132
9	.726	. . . .	.153	. . . .	.859	. . . .	.130	. . . .
10	.700	.848	.143	.118	.844	.926	.119	.108
11	.674	. . . .	.135	. . . .	.828	. . . .	.110	. . . .
12	.650	.815	.128	.102	.813	.909	.101	.091
13	.628	. . . .	.122	. . . .	.799	. . . .	.096	. . . .
14	.606	.784	.118	.091	.785	.892	.091	.080
15	.586	. . . .	.114	. . . .	.771	. . . .	.086	. . . .

$$\text{CASE "D" EFFICIENCY} = \frac{(K^N - 1)}{K^S N\,(K-1)} \qquad \text{CASE "E" EFFICIENCY} = \frac{2(K^{\frac{N}{2}} - 1)}{K^{\frac{S}{2}}\,N\,(K-1)}$$

$$\text{FAST LINE FACTOR} = \frac{1}{N \times \text{EFFICIENCY}} \qquad \text{FAST LINE FACTOR} = \frac{1}{N \times \text{EFFICIENCY}}$$

NOTE: The above cases apply also where the rope is dead ended or the equalizer is located at the lower or traveling block or derrick floor after passing over a dead sheave in the crown.

*In these tables, the K factor for sheave friction is 1.09 for plain bearings and 1.04 for roller bearings. Other K factors can be used if recommended by the equipment manufacturer.

Figure 4-69. Efficiency of wire-rope reeving for multiple sheave blocks, Cases D and E [11].

2. *Rotary drilling lines.* Care should be taken to maintain proper winding of rotary drilling lines on the drawworks drum to avoid excessive friction that may result in the formation of martensite. Martensite may also be formed by excessive friction in worn grooves of sheaves, slippage in sheaves, or excessive friction resulting from rubbing against a derrick member. A line guide

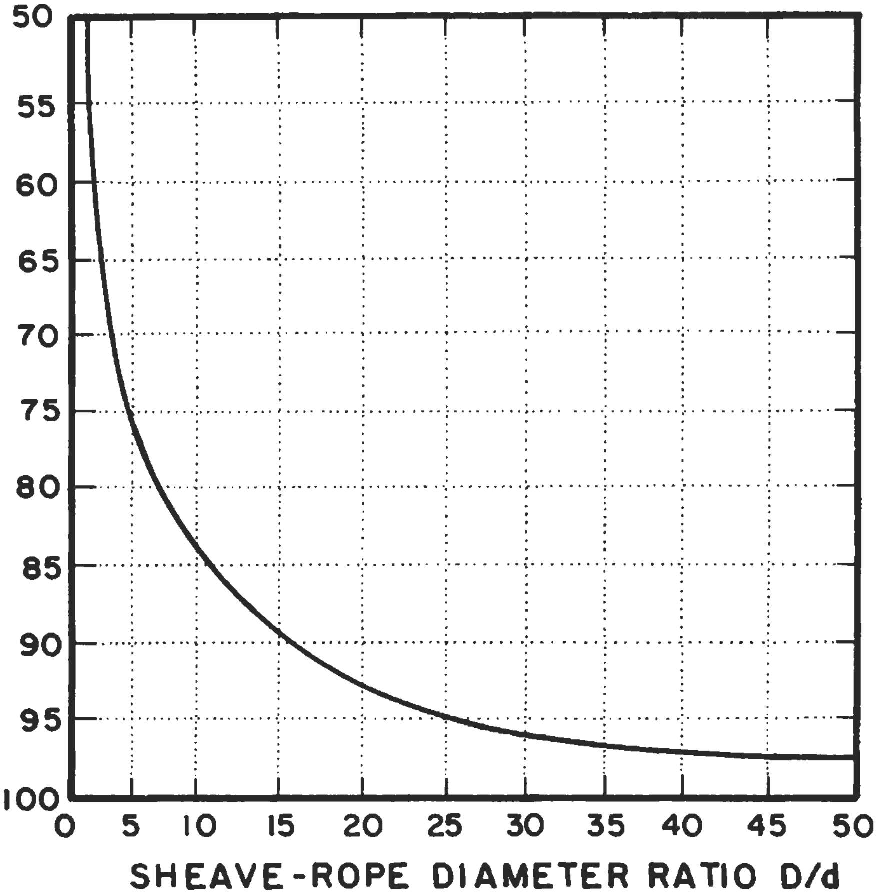


Figure 4-70. Efficiencies of wire ropes bent around stationary sheaves (static stresses only) [11].

should be employed between the drum and the fast line sheave to reduce vibration and to keep the drilling line from rubbing against the derrick.

Martensite is a hard, nonductile microconstituent formed when steel is heated above its critical temperature and cooled rapidly. In the case of steel of the composition conventionally used for rope wire, martensite can be formed if the wire surface is heated to a temperature near or somewhat in excess of 1400°F (760°C), and then cooled at a comparatively rapid rate. The presence of a martensite film at the surface of the outer wires of a rope that has been in service is evidence that sufficient frictional heat has been generated on the crown of the rope wires to momentarily raise the wire surface temperature to a point above the critical temperature range of the steel. The heated surface is then rapidly cooled by the adjacent cold metal within the wire and the rope structure, and an effective quenching results.

Figure 4-71A shows a rope that has developed fatigue fractures at the crown in the outer wires, and Figure 4-71B shows a photomicrograph (100× magnification) of a specimen cut from the crown of one of these outer wires. This photomicrograph clearly shows the depth of the martensite layer and the cracks produced by the inability of the martensite to withstand the normal flexing of the rope. The result is a disappointing service life for the rope. Most outer wire failures may be attributed to the presence of martensite.

Worn sheave and drum grooves cause excessive wear on the rope. All sheaves should be in proper alignment. The fast sheave should line up with the center of the hoisting drum. From the standpoint of wire-rope life, the condition and contour of sheave grooves are important and should be checked periodically. The sheave groove should have a radius not less than that in Table 4-31; otherwise, rope life can be reduced. Reconditioned sheave grooves should conform to the recommended radii for new and reconditioned sheaves as given in Table 4-32. Each operator should establish the most economical point at which sheaves should be regrooved by considering the loss in rope life that will result

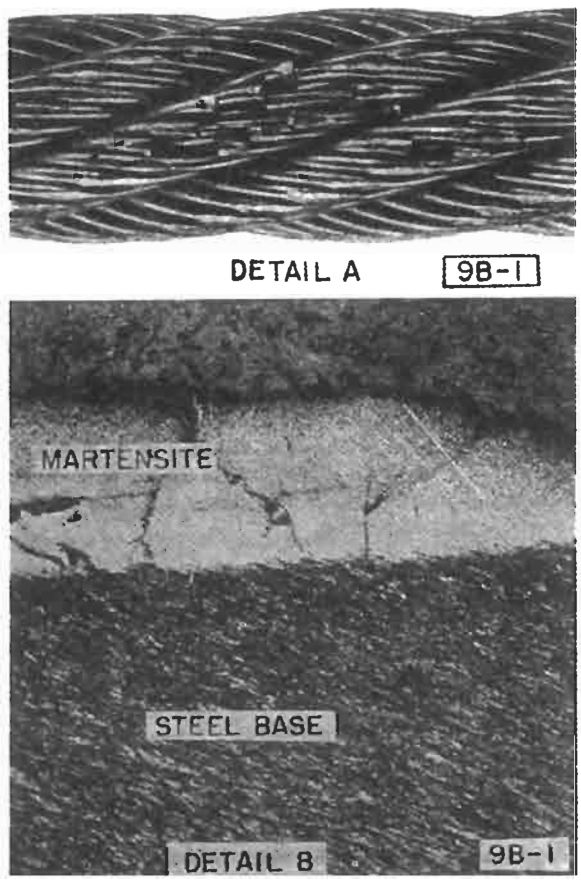


Figure 4-71. Fatigue fractures in outer wires caused by the formation of martensite [11].

Table 4-31
Minimum Groove Radii for Worn Sheaves [11]

1		2		1		2		1		2	
Wire Rope Nominal Size		Radii		Wire Rope Nominal Size		Radii		Wire Rope Nominal Size		Radii	
in.	*(mm)*	in.	*(mm)*	in.	*(mm)*	in.	*(mm)*	in.	*(mm)*	in.	*(mm)*
1/4	*(6.5)*	.129	*(3.28)*	1 5/8	*(42)*	.833	*(21.16)*	3 3/8	*(86)*	1.730	*(43.94)*
5/16	*(8)*	.160	*(4.06)*	1 3/4	*(45)*	.897	*(22.78)*	3 1/2	*(90)*	1.794	*(45.57)*
3/8	*(9.5)*	.190	*(4.83)*	1 7/8	*(48)*	.959	*(24.36)*	3 3/4	*(96)*	1.918	*(48.72)*
7/16	*(11)*	.220	*(5.59)*	2	*(52)*	1.025	*(26.04)*	4	*(103)*	2.050	*(52.07)*
1/2	*(13)*	.256	*(6.50)*	2 1/8	*(54)*	1.079	*(27.41)*	4 1/4	*(109)*	2.178	*(55.32)*
9/16	*(14.5)*	.288	*(7.32)*	2 1/4	*(58)*	1.153	*(29.29)*	4 1/2	*(115)*	2.298	*(58.37)*
5/8	*(16)*	.320	*(8.13)*	2 3/8	*(60)*	1.199	*(30.45)*	4 3/4	*(122)*	2.434	*(61.82)*
3/4	*(19)*	.380	*(9.65)*	2 1/2	*(64)*	1.279	*(32.49)*	5	*(128)*	2.557	*(64.95)*
7/8	*(22)*	.440	*(11.18)*	2 5/8	*(67)*	1.339	*(34.01)*	5 1/4	*(135)*	2.691	*(68.35)*
1	*(26)*	.513	*(13.03)*	2 3/4	*(71)*	1.409	*(35.79)*	5 1/2	*(141)*	2.817	*(71.55)*
1 1/8	*(29)*	.577	*(14.66)*	2 7/8	*(74)*	1.473	*(37.41)*	5 3/4	*(148)*	2.947	*(74.85)*
1 1/4	*(32)*	.639	*(16.23)*	3	*(77)*	1.538	*(39.07)*	6	*(154)*	3.075	*(78.11)*
1 3/8	*(35)*	.699	*(17.75)*	3 1/8	*(80)*	1.598	*(40.59)*				
1 1/2	*(38)*	.759	*(19.28)*	3 1/4	*(83)*	1.658	*(42.11)*				

Sheaves worn to these sizes can be detrimental to wire rope service and should be regrooved or removed from service.

Table 4-32
Minimum Groove Radii for New and Reconditioned Sheaves [11]

1		2		1		2		1		2	
Wire Rope Nominal Size		Radii		Wire Rope Nominal Size		Radii		Wire Rope Nominal Size		Radii	
in.	*(mm)*	in.	*(mm)*	in.	*(mm)*	in.	*(mm)*	in.	*(mm)*	in.	*(mm)*
1/4	*(6.5)*	.135	*(3.43)*	1 5/8	*(42)*	.876	*(22.25)*	3 3/8	*(86)*	1.807	*(45.90)*
5/16	*(8)*	.167	*(4.24)*	1 3/4	*(45)*	.939	*(23.85)*	3 1/2	*(90)*	1.869	*(47.47)*
3/8	*(9.5)*	.201	*(5.11)*	1 7/8	*(48)*	1.003	*(25.48)*	3 3/4	*(96)*	1.997	*(50.72)*
7/16	*(11)*	.234	*(5.94)*	2	*(52)*	1.085	*(27.56)*	4	*(103)*	2.139	*(54.33)*
1/2	*(13)*	.271	*(6.88)*	2 1/8	*(54)*	1.137	*(28.88)*	4 1/4	*(109)*	2.264	*(57.51)*
9/16	*(14.5)*	.303	*(7.70)*	2 1/4	*(58)*	1.210	*(30.73)*	4 1/2	*(115)*	2.396	*(60.86)*
5/8	*(16)*	.334	*(8.48)*	2 3/8	*(60)*	1.271	*(32.28)*	4 3/4	*(122)*	2.534	*(64.36)*
3/4	*(19)*	.401	*(10.19)*	2 1/2	*(64)*	1.338	*(33.99)*	5	*(128)*	2.663	*(67.64)*
7/8	*(22)*	.468	*(11.89)*	2 5/8	*(67)*	1.404	*(35.66)*	5 1/4	*(135)*	2.804	*(71.22)*
1	*(26)*	.543	*(13.79)*	2 3/4	*(71)*	1.481	*(37.62)*	5 1/2	*(141)*	2.929	*(74.40)*
1 1/8	*(29)*	.605	*(15.37)*	2 7/8	*(74)*	1.544	*(39.22)*	5 3/4	*(148)*	3.074	*(78.08)*
1 1/4	*(32)*	.669	*(16.99)*	3	*(77)*	1.607	*(40.82)*	6	*(154)*	3.198	*(81.23)*
1 3/8	*(35)*	.736	*(18.69)*	3 1/8	*(80)*	1.664	*(42.27)*				
1 1/2	*(38)*	.803	*(20.40)*	3 1/4	*(83)*	1.731	*(43.97)*				

Standard Machine Tolerance

from worn sheaves as compared to the cost involved in regrooving. When a new rope is to be installed on used sheaves, it is particularly important that the sheave grooves be checked as recommended. To ensure a minimum turning effort, all sheaves should be kept properly lubricated.

Seizing. Before cutting, a wire rope should be securely seized on each side of the cut by serving with soft wire ties. For socketing, at least two additional seizings should be placed at a distance from the end equal to the basket length of the socket. The total length of the seizing should be at least two rope diameters, and securely wrapped with a seizing iron. This is very important, as it prevents the rope untwisting and ensures equal tension in the strands when the load is applied.

The recommended procedure for seizing a wire rope is as follows:

a. The seizing wire should be wound on the rope by hand as shown in Figure 4-73 (1). The coils should be kept together and considerable tension maintained on the wire.
b. After the seizing wire has been wound on the rope, the ends of the wire should be twisted together by hand in a counterclockwise direction so that the twisted portion of the wires is near the middle of the seizing (see Figure 4-73 (2)).
c. Using "Carew" cutters, the twist should be tightened just enough to take up the slack (Figure 4-73 (3)). Tightening the seizing by twisting should not be attempted.
d. The seizing should be tightened by prying the twist away from the axis of the rope with the cutters as shown in Figure 4-73 (4).
e. The tightening of the seizing should be repeated as often as necessary to make the seizing tight.

Figure 4-72. Correct method of attaching clips to wire rope [11].

Figure 4-73. Putting a seizing on a wire rope [11].

f. To complete the seizing operation, the ends of the wire should be cut off as shown in Figure 4-73 (5), and the twisted portion of the wire tapped flat against the rope. The appearance of the finished seizing is illustrated in Figure 4-73 (6).

Socketing (Zinc Poured or Spelter).

Wire Rope Preparation. The wire rope should be securely seized or clamped at the end before cutting. Measure from the end of the rope a length equal to approximately 90% of the length of the socket basket. Seize or clamp at this point. Use as many seizings as necessary to prevent the rope from unlaying.

After the rope is cut, the end seizing should be removed. Partial straightening of the strands and/or wire may be necessary. The wires should then be separated and broomed out and the cores treated as follows:

1. *Fiber core*—Cut back length of socket basket.
2. *Steel core*—Separate and broom out.
3. *Other*—Follow manufacturer's recommendations.

Cleaning. The wires should be carefully cleaned for the distance inserted in the socket by one of the following methods:

Acid cleaning

1. *Improved plow steel and extra improved plow steel, bright and galvanized.* Use a suitable solvent to remove lubricant. The wires then should be dipped in commercial muriatic acid until thoroughly cleaned. The depth of immersion in acid must not be more than the broomed length. The acid should be neutralized by rinsing in a bicarbonate of soda solution. Fresh acid should be prepared when satisfactory cleaning of the wires requires more than one minute. (Prepare new solution—do not merely add new acid to old.) Be sure acid surface is free of oil or scum. The wires should be dried and then dipped in a hot solution of zinc-ammonium chloride flux. Use a concentration of 1 lb (454 g) of zinc-ammonium chloride in 1 gal (3.8 L) of water and maintain the solution at a temperature of 180°F (82°C) to 200°F (93°C).
2. *Stainless steel.* Use a suitable solvent to remove lubricant. The wires then should be dipped in a hot caustic solution, such as oakite, then in a hot water rinse, and finally dipped in one of the following solutions until thoroughly cleaned:
   a. commercial muriatic acid
   b. 1 part by weight of cupric chloride, 20 parts by weight of concentrated hydrochloric acid
   c. 1 part by weight of ferric chloride, 10 parts by weight of concentrated nitric or hydrochloric acid, 20 parts by weight of water.

   Use the above solutions at room temperature. The wires should then be dipped in clean hot water. A suitable flux may be used.

   Fresh solution should be prepared when satisfactory cleaning of the wires requires more than a reasonable time. (Prepare new solutions—do not merely add new solution to old solution.) Be sure solution surface is free of oil and scum.
3. *Phosphor bronze.* Use a suitable solvent to remove lubricant. The wires should then be dipped in commercial muriatic acid until thoroughly cleaned.
4. *Monel metal.* Use a suitable solvent to remove lubricant. The wires then should be dipped in the following solution until thoroughly cleaned: 1 part glacial acetic acid + 1 part concentrated nitric acid.

This solution is used at room temperature. The broom should be immersed from 30 to 90 s. The depth of immersion in the solution must not be more than broomed length. The wires should then be dipped in clean hot water.

Ultrasonic cleaning (all grades). An ultrasonic cleaner suitable for cleaning wire rope is permitted in lieu of the acid cleaning methods described previously.

Other cleaning methods. Other cleaning methods of proven reliability are permitted.

Attaching Socket. Preheat the socket to approximately 200°F (93°C). Slip socket over ends of wire. Distribute all wires evenly in the basket and flush with top of basket. Be sure socket is in line with axis of rope.

Use only zinc not lower in quality than high grade per ASTM Specification B-6. Heat zinc to a range allowing pouring at 950°F (510°C) to 975°F (524°C). Skim off any dross accumulated on the surface of the zinc bath. Pour molten zinc into the socket basket in one continuous pour if possible. Tap socket basket while pouring.

Final Preparation. Remove all seizings. Apply lubricant to rope adjacent to socket to replace lubricant removed by socketing procedure. Socket is then ready for service.

Splicing. Splicing wire rope requires considerable skill, and the instructions for splicing wire rope will be found in the catalogues of most of the wire-rope manufacturers, where the operation sequence is carefully described, and many clear illustrations are presented. These illustrations give, in fact, most of the information needed.

Socketing (Thermo-Set Resin). Before proceeding with thermo-set resin socketing, the manufacturer's instructions for using this product should be carefully read. Particular attention should be given to sockets designed specifically for resin socketing. Other thermo-set resins used may have specifications that differ from those shown in this section.

Seizing and Cutting the Rope. The rope manufacturer's directions for a particular size or construction of rope are to be followed with regard to the number, position, and length of seizings, and the seizing wire size to be used. The seizing, which will be located at the base of the installed fitting, must be positioned so that the ends of the wires to be embedded will be slightly below the level of the top of the fitting's basket. Cutting the rope can best be accomplished by using an abrasive wheel.

Opening and Brooming the Rope End. Prior to opening the rope end, place a short temporary seizing directly above the seizing that represents the base of the broom. The temporary seizing is used to prevent brooming the wires to full length of the basket, and also to prevent the loss of lay in the strands and rope outside the socket. Remove all seizings between the end of the rope and the temporary seizing. Unlay the strands comprising the rope. Starting with the IWRC, or strand core, open each strand and each strand of the rope, and broom or unlay the individual wires. (A fiber core may be cut in the rope at the base of the seizing. Some prefer to leave the core in. Consult the manufacturer's instructions.) When the brooming is completed, the wires should be distributed evenly within a cone so that they form an included angle of approximately 60°.

Some types of sockets require a different brooming procedure and the manufacturer's instructions should be followed.

Cleaning the Wires and Fittings. Different types of resin with different characteristics require varying degrees of cleanliness. The following cleaning procedure was used for one type of polyester resin with which over 800 tensile tests were made on ropes in sizes $\frac{1}{4}$ in. (6.5 mm) to $3\frac{1}{2}$-in. (90 mm) diameter without experiencing any failure in the resin socket attachment.

Thorough cleaning of the wires is required to obtain resin adhesion. Ultrasonic cleaning in recommended solvents (such as trichloroethylene or 1,1,1-trichloroethane or other nonflammable grease-cutting solvents) is the preferred method in accordance with OSHA standards. If ultrasonic cleaning is not available, trichloroethane may be used in brush or dip-cleaning; but fresh solvent should be used for each rope end fitting and should be discarded after use. After cleaning, the broom should be dried with clean compressed air or in another suitable fashion before proceeding to the next step. Using acid to etch the wires before resin socketing is *unnecessary and not recommended.* Also, the use of a flux on the wires before pouring the resin should be avoided since *this adversely affects bonding of the resin to the steel wires.* Since there is a variation in the properties of different resins, the manufacturer's instructions should be carefully followed.

Placement of the Fitting. The rope should be placed vertically with the broom up, and the broom should be closed and compacted to insert the broomed rope end into the fitting base. Slip on the fitting, removing any temporary banding or seizing as required. Make sure the broomed wires are uniformly spaced in the basket with the wire ends slightly below the top edge of the basket, and make sure the axis of the rope and the fitting are aligned. Seal the annular space between the base of the fitting and the exiting rope to prevent leakage of the resin from the basket. A nonhardening butyl rubber base sealant gives satisfactory performance. Make sure the sealant does not enter the socket base, so that the resin may fill the complete depth of the socket basket.

Pouring the Resin. Controlled heat-curing (no open flame) at a temperature range of 250 to 300°F (121 to 149°C) is recommended; and is required if ambient temperatures are less than 60°F (16°C) (which may vary with different resins). When controlled heat curing is not available and ambient temperatures are not less than 60°F (16°C), the attachment should not be disturbed and tension should not be applied to the socketed assembly for at least 24 hr.

Lubrication of Wire Rope after Socket Attachment. After the resin has cured, relubricate the wire rope at the base of the socket to replace the lubricant that was removed during the cleaning operation.

Resin Socketing Compositions. Manufacturer's directions should be followed in handling, mixing, and pouring the resin composition.

Performance of Cured Resin Sockets. Poured resin sockets may be moved when the resin has hardened. After ambient or elevated temperature cure recommended by the manufacturer, resin sockets should develop the nominal strength of the rope; and should also withstand, without cracking or breakage, shock loading sufficient to break the rope. Manufacturers of resin socketing material should be required to test to these criteria before resin materials are approved for this end use.

Attachment of Clips

The clip method of making wire-rope attachments is widely used. Drop-forged clips of either the U-bolt or the double-saddle type are recommended. When properly applied as described herein, the method develops about 80% of the rope strength in the case of six strand ropes.

When attaching clips, the rope length to be turned back when making a loop is dependent upon the rope size and the load to be handled. The recommended lengths, as measured from the thimble base, are given in Table 4-33. The thimble should first be wired to the rope at the desired point and the rope then bent around the thimble and temporarily secured by wiring the two rope members together.

Table 4-33
Attachment of Clips [11]

1	2	3	4
Diameter of Rope, in. *(mm)*	Number of Clips	Length of Rope Turned Back, in. *(mm)*	Torque, ft-lb *(N•m)*
1/8 *(3)*	2	3 1/4 *(83)*	4.5 *(6.1)*
3/16 *(5)*	2	3 3/4 *(95)*	7.5 *(10)*
1/4 *(6.5)*	2	4 3/4 *(121)*	15 *(20)*
5/16 *(8)*	2	5 1/4 *(133)*	30 *(41)*
3/8 *(9.5)*	2	6 1/2 *(165)*	45 *(61)*
7/16 *(11)*	2	7 *(178)*	65 *(88)*
1/2 *(13)*	3	11 1/2 *(292)*	65 *(88)*
9/16 *(14.5)*	3	12 *(305)*	95 *(129)*
5/8 *(16)*	3	12 *(305)*	95 *(129)*
3/4 *(19)*	4	18 *(457)*	130 *(176)*
7/8 *(22)*	4	19 *(483)*	225 *(305)*
1 *(26)*	5	26 *(660)*	225 *(305)*
1 1/8 *(29)*	6	34 *(864)*	225 *(305)*
1 1/4 *(32)*	7	44 *(1117)*	360 *(488)*
1 3/8 *(35)*	7	44 *(1120)*	360 *(488)*
1 1/2 *(38)*	8	54 *(1372)*	360 *(488)*
1 5/8 *(42)*	8	58 *(1473)*	430 *(583)*
1 3/4 *(45)*	8	61 *(1549)*	590 *(800)*
2 *(51)*	8	71 *(1800)*	750 *(1020)*
2 1/4 *(57)*	8	73 *(1850)*	750 *(1020)*
2 1/2 *(64)*	9	84 *(2130)*	750 *(1020)*
2 3/4 *(70)*	10	100 *(2540)*	750 *(1020)*
3 *(77)*	10	106 *(2690)*	1200 *(1630)*

NOTE 1: *If a pulley is used in place of a thimble for turning back the rope, add one additional clip.*

The first clip should be attached at a point about one base width from the last seizing on the dead end of the rope and tightened securely. The saddle of the clip should rest upon the long or main rope and the U-bolt upon the dead end. All clips should be attached in this manner (see Figure 4-74). The short end of the rope should rest squarely upon the main portion.

The second clip should be attached as near the loop as possible. The nuts for this clip should not be completely tightened when it is first installed. The recommended number of clips and the space between clips are given in Table 4-33. Additional clips should be attached with an equal spacing between clips. Before completely tightening the second and any of the additional clips, some stress should be placed upon the rope in order to take up the slack and equalize the tension on both sides of the rope.

When the clips are attached correctly, the saddle should be in contact with the long end of the wire rope and the U-bolt in contact with the short end of the loop in the rope as shown in Figure 4-72. The incorrect application of clips is illustrated in Figure 4-74.

The nuts on the second and additional clips should be tightened uniformly, by giving alternately a few turns to one side and then the other. It will be found that the application of a little oil to the threads will allow the nuts to be drawn tighter. After the rope has been in use a short time, the nuts on all clips should be retightened, as stress tends to stretch the rope, thereby reducing its diameter. The nuts should be tightened at all subsequent regular inspection periods. A half hitch, either with or without clips, is not desirable as it malforms and weakens wire rope.

Figure 4-75 illustrates, in a simplified form, the generally accepted methods of reeving (stringing up) in-line crown and traveling blocks, along with the location of the drawworks drum, monkey board, drill pipe fingers, and deadline anchor in relation to the various sides of the derrick. Ordinarily, the only two variables in reeving systems, as illustrated, are the number of sheaves in the crown and traveling blocks or the number required for handling the load, and the location of the deadline anchor. Table 4-34 gives the right-hand string-ups. The reeving sequence for the left-hand reeving with 12 lines on a seven-sheave crown-block and six-sheave traveling block illustrated in Figure 4-75 is given in Arrangement No. 1 of Table 4-34. The predominant practice is to use left-hand reeving and

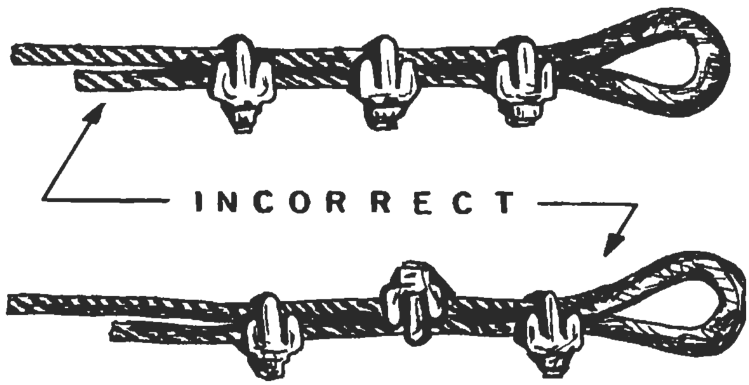


Figure 4-74. Incorrect methods of attaching clips to wire rope [11].

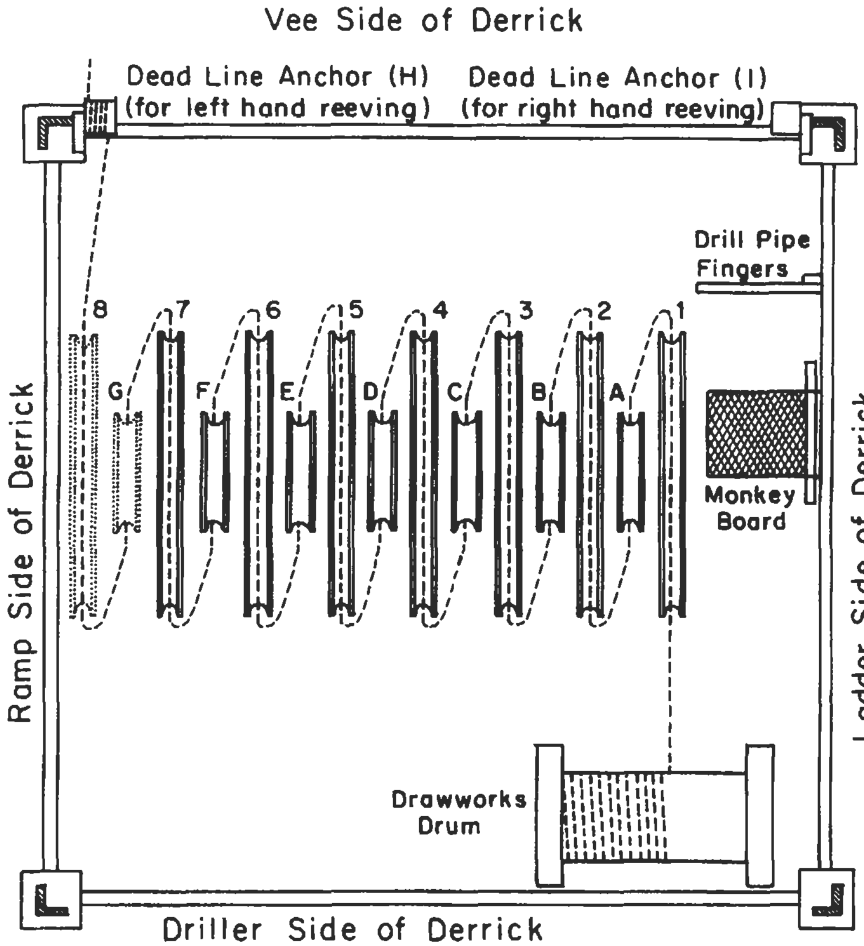


Figure 4-75. Typical reeving diagram for 14-line string-up with eight-sheave crown block and seven-sheave traveling block: left-hand reeving [11].

locate the deadline anchor to the left of the derrick vee. In selecting the best of the various possible methods for reeving casing or drilling lines, the following basic factors should be considered:

1. Minimum fleet angle from the drawworks drum to the first sheave of the crown block, and from the crown block sheaves to the traveling block sheaves.
2. Proper balancing of crown and traveling blocks.
3. Convenience in changing from smaller to larger number of lines, or from larger to smaller number of lines.
4. Location of deadline on monkey board side for convenience and safety of derrickman.

Table 4-34
Recommended Reeving Arrangements for 12-, 10-, 8-, and 6-Line String-ups Using 7-Sheave Crown Blocks with 6-Sheave Traveling Blocks and 6-Sheave Crown Blocks with 5-Sheave Travelina Blocks [11]

Arrange-ment No.	No. of Sheaves: Crown Block	No. of Sheaves: Trav. Block	Type of String-up	No. of Lines to	Reeving Sequence (Read From Left to Right Starting with Crown Block and Going Alternately From Crown to Traveling to Crown)															
1	8	7	Left Hand	14	Crown Block Trav. Block	1	A	2	B	3	C	4	D	5	E	6	F	7	G	8
2	8	7	Right Hand	14	Crown Block Trav. Block	8	G	7	F	6	E	5	D	4	C	3	B	2	A	1
3	7	6	Left Hand	12	Crown Block Trav. Block	1	A	2	B	3	C	4	D	5	E	6	F	7		
4	7	6	Right Hand	12	Crown Block Trav. Block	7	F	6	E	5	D	4	C	3	B	2	A	1		
5	7	6	Left Hand	10	Crown Block Trav. Block	1	A	2	B	3			D	5	E	6	F	7		
6	7	6	Right Hand	10	Crown Block Trav. Block	7	F	6	E	5			C	3	B	2	A	1		
7	6	5	Left Hand	10	Crown Block Trav. Block	1	A	2	B	3	C	4	D	5	E	6				
8	6	5	Right Hand	10	Crown Block Trav. Block	6	E	5	D	4	C	3	B	2	A	1				
9	6	5	Left Hand	8	Crown Block Trav. Block	1	A	2	B	3			D	5	E	6				
10	6	5	Right Hand	8	Crown Block Trav. Block	6	E	5	D	4			B	2	A	1				
11	6	5	Left Hand	8	Crown Block Trav. Block	1	A	2	B	3	C	4	D	5			G			
12	6	5	Right Hand	8	Crown Block Trav. Block	6	E	5	D	4	C	3	B	2			H			
13	6	5	Left Hand	6	Crown Block Trav. Block			2	B	3	C	4	D	5			G			
14	6	5	Right Hand	6	Crown Block Trav. Block			5	D	4	C	3	B	2			H			
15	6	5	Left Hand	6	Crown Block Trav. Block	1	A			3	C	4			E	6				
16	6	5	Right Hand	6	Crown Block Trav. Block	6	E			4	C	3			A	1				

5. Location of deadline anchor, and its influence upon the maximum rated static hook load of derrick.

Recommended Design Features

The proper design of sheaves, drums, and other equipment on which wire rope is used is very important to the service life of wire rope. It is strongly urged that the purchaser specify on his order that such material shall conform with recommendations set forth in this section.

The inside diameter of socket and swivel-socket baskets should be $\frac{5}{32}$ in. larger than the nominal diameter of the wire rope inserted. Alloy or carbon steel, heat treated, will best serve for sheave grooves. Antifriction bearings are recommended for all rotating sheaves.

Drums should be large enough to handle the rope with the smallest possible number of layers. Drums having a diameter of 20 times the nominal wire-rope diameter should be considered minimum for economical practice. Larger diameters than this are preferable. For well-measuring wire, the drum diameter should be as large as the design of the equipment will permit, but should not be less than 100 times the wire diameter. The recommended grooving for wire-rope drums is as follows:

a. On drums designed for multiple-layer winding, the distance between groove centerlines should be approximately equal to the nominal diameter of the wire rope plus one-half the specified oversize tolerance. For the best spooling condition, this dimension can vary according to the type of operation.

b. The curvature radius of the groove profile should be equal to the radii listed in Table 4-32.

c. The groove depth should be approximately 30% of the nominal diameter of the wire rope. The crests between grooves should be rounded off to provide the recommended groove depth.

Diameter of Sheaves. When bending conditions over sheaves predominate in controlling rope life, sheaves should be as large as possible after consideration has been given to economy of design, portability, etc. When conditions other than bending over sheaves predominate as in the case of hoisting service for rotary drilling, the size of the sheaves may be reduced without seriously affecting rope life. The following recommendations are offered as a *guide* to designers and users in selecting the proper sheave size.

$$D_T = d \times F \tag{4-25}$$

where DT = tread diameter of sheave in in. (mm) (see Figure 4-76),
d = nominal rope diameter in in. (mm), and
F = sheave-diameter factor, selected from Table 4-35.

It should be stressed that if sheave design is based on condition C, fatigue due to severe bending can occur rapidly. If other operation conditions are not present to cause the rope to be removed from service, this type of fatigue is apt to result in wires breaking where they are not readily visible to external examination. Any condition resulting in rope deterioration of a type that is difficult to judge by examination during service should certainly be avoided.

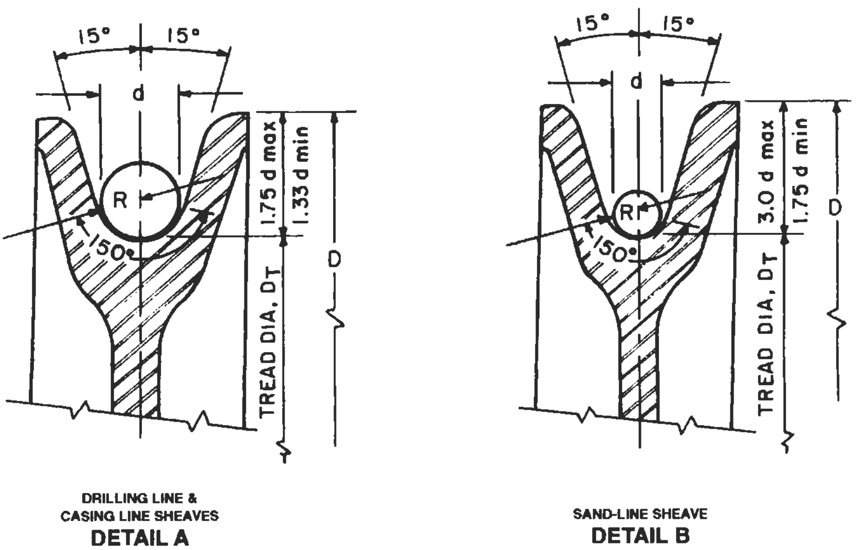


Figure 4-76. Sheave grooves [11].

Table 4-35
Sheave-Diameter Factors [11]

1	2	3	4
	Factor, F		
Rope Classification	Condition A	Condition B	Condition C
6x7	72	42	
6x17 Seale	56	33	
6x19 Seale	51	30	(See Fig. 3.1
6x21 Filler Wire	45	26	and
6x25 Filler Wire	41	24	Table 3.2)
6x31	38	22	
6x37	33	18	
8x19 Seale	36	21	
8x19 Warrington	31	18	
18x7 and 19x7	51	36	
Flattened Strand	51	45	*

*Follow manufacturer's recommendations.

Condition A—Where bending over sheaves is of major importance, sheaves at large as those determined by factors under condition A are recommended.

Condition B—Where bending over sheaves is important, but some sacrifice in rope life is acceptable to achieve portability, reduction in weight, economy of design, etc., sheaves at least as large as those determined by factors under condition B are recommended.

Condition C—Some equipment is used under operating conditions which do not reflect the advantage of the selection of sheaves by factors under conditions A or B. In such cases, sheave-diameter factors may be selected from Figure 4-76 and Table 4-34. As smaller factors are selected, the bending life of the wire rope is reduced and it becomes an increasingly important condition of rope service. Some conception of relative rope service with different rope constructions and/or different sheave sizes may be obtained by multiplying the ordinate found in Figure 4-76 by the proper construction factor indicated in Table 4-34.

The diameter of sheaves for well-measuring wire should be as large as the design of the equipment will permit but not less than 100 times the diameter of the wire.

Sheave Grooves. On all sheaves, the arc of the groove bottom should be smooth and concentric with the bore or shaft of the sheave. The centerline of the groove should be in a plane perpendicular to the axis of the bore or shaft of the sheave.

Grooves for drilling and casing line sheaves shall be made for the rope size specified by the purchaser. The groove bottom shall have a radius R (Table 4-32) subtending an arc of 150°. The sides of the groove shall be tangent to the ends of the bottom arc. Total groove depth shall be a minimum of 1.33d and a maximum of 1.75d (d is the nominal rope diameter shown in Figure 4-76).

Grooves for sand-line sheaves shall be made for the rope size specified by the purchaser. The groove bottom shall have a radius R (Table 4-32) subtending an arc of 150°. The sides of the groove shall be tangent to the ends of the bottom arc. Total groove depth shall be a minimum of 1.75d and a maximum of 3d (d is nominal rope diameter shown in Figure 4-77B).

Grooves on rollers of oil savers should be made to the same tolerances as the grooves on the sheaves.

Sheaves conforming to the specifications (Specification 8A) shall be marked with the manufacturer's name or mark, the sheave groove size and the sheave OD. These markings shall be cast or stamped on the outer rim of the sheave groove and stamped on the nameplate of crown and traveling blocks. For example, a 36-in. sheave with $1\frac{1}{8}$ in. groove shall be marked

AB CO 1 1/8 SPEC 8A

Sheaves should be replaced or reworked when the groove radius decreases below the values shown in Table 4-31. Use sheave gages as shown in Figure 4-77A shows a sheave with a minimum groove radius, and 4-77B shows a sheave with a tight groove.

Evaluation of Rotary Drilling Line

The total service performed by a rotary drilling line can be evaluated by considering the amount of work done by the line in various drilling operations (drilling, coring, fishing, setting casing, etc.), and by evaluating such factors as the stresses imposed by acceleration and deceleration loadings, vibration stresses, stresses imposed by friction forces of the line in contact with drum and sheave surfaces, and other even more indeterminate loads. However, for comparative purposes, an approximate evaluation can be obtained by computing only the work done by the line in raising and lowering the applied loads in making round trips, and in the operations of drilling, coring, setting casing, and short trips.

Round-Trip Operations. Most of the work done by a drilling line is that performed in making round trips (or half-trips) involving running the string of

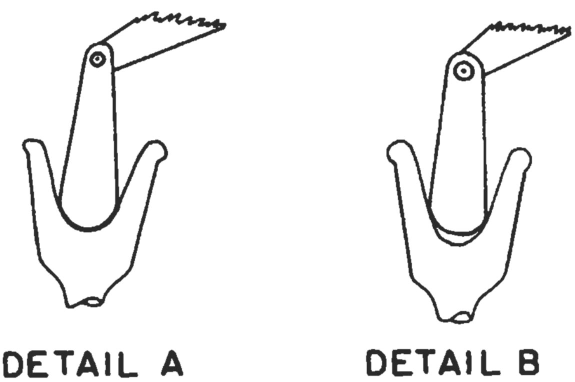


Figure 4-77. Use of sheave gage [11].

drill pipe into the hole and pulling the string out of the hole. The amount of work performed per round trip should be determined by

$$T_r = \frac{D(L_s + D)W_m}{10,560,000} + \frac{D\left(M + \frac{1}{2}C\right)}{2,640,000} \qquad (4\text{-}26)$$

where T_r = ton-miles (weight in tons times distance moved in miles)
D = depth of hole in feet
L_s = length of drill-pipe stand in feet
N = number of drill-pipe stands
W_m = effective weight per foot of drill-pipe from Figure 4-78 in pounds
M = total weight of traveling block-elevator assembly in pounds
C = effective weight of drill-collar assembly from Figure 4-78 minus the effective weight of the same length of drill-pipe from Figure 4-78 in pounds

Drilling Operations. The ton-miles of work performed in drilling operations is expressed in terms of work performed in making round trips, since there is a direct relationship as illustrated in the following cycle of drilling operations.

1. Drill ahead length of the kelly.
2. Pull up length of the kelly.
3. Ream ahead length of the kelly.
4. Pull up length of the kelly to add single or double.
5. Put kelly in rat hole.
6. Pick up single or double.
7. Lower drill stem in hole.
8. Pick up kelly.

Analysis of the cycle of operations shows that for any hole, the sum of operations 1 and 2 is equal to one round trip; the sum of operations 3 and 4 is equal to another round trip; the sum of operation 7 is equal to one-half a round trip; and the sum of operations 5, 6, and 8 may, and in this case does, equal another one-half round trip, thereby making the work of drilling the hole equivalent to three round trips to bottom, and the relationship can be expressed as

$$T_d = 3(T_2 - T_1) \qquad (4\text{-}27)$$

where Td = ton-mile drilling
T_1 = ton-miles for one round trip at depth D_1 (depth where drilling started after going in hole, in ft)
T_2 = ton-miles for one round trip at depth D_2 (depth where drilling stopped before coming out of hole in ft)

If operations 3 and 4 are omitted, then formula 4-27 becomes

$$T_d = 2(T_2 - T_1) \qquad (4\text{-}28)$$

Coring Operations. The ton-miles of work performed in coring operations, as for drilling operations, is expressed in terms of work performed in making round trips, since there is a direct relationship illustrated in the following cycle of coring operations.

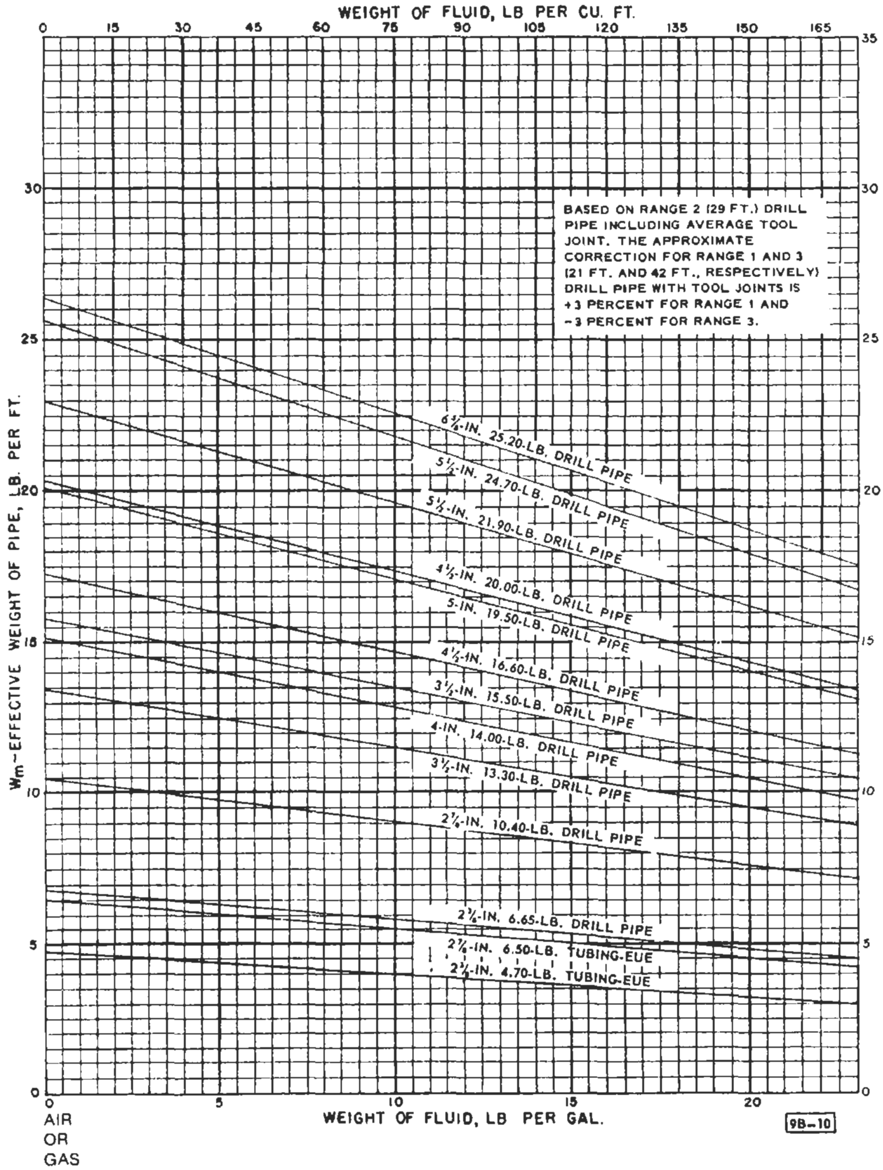


Figure 4-78. Effective weight of pipe-in drilling fluid [11].

1. Core ahead length of core barrel.
2. Pull up length of kelly.
3. Put kelly in rat hole.
4. Pick up single.
5. Lower drill stem in hole.
6. Pick up kelly.

Analysis of the cycle of operation shows that for any one hole the sum of operations 1 and 2 is equal to one round trip; the sum of operations 5 is equal to one-half a round trip; and the sum of operations 3, 4, and 6 may, and in this case does, equal another one-half round trip, thereby making the work of drilling the hole equivalent to two round trips to bottom, and the relationship can be expressed as

$$T_c = 2(T_4 - T_3) \qquad (4\text{-}29)$$

where T_c = ton-mile coring
T_3 = ton-miles for one round trip at depth D_3 (depth where coring started after going in hole, in feet)
T_4 = ton-miles for one round trip at depth D_4 (depth where coring stopped before coming out of hole, in feet)

Setting Casing Operations. The calculation of the ton-miles for the operation of setting casing should be determined as in round-trip operations as for drill pipe, but with the effective weight of the casing being used, and with the result being multiplied by one-half, since setting casing is a one-way (one-half round-trip) operation. Ton-miles for setting casing can be determined from

$$T_s = \left[\frac{D(L_{cs} + D)(W_{cm})}{10,560,000} + \frac{D\left(M + \frac{1}{2}C\right)}{2,640,000}\right]\left(\frac{1}{2}\right) \qquad (4\text{-}30)$$

Since no excess weight for drill collars need be considered, Equation 4-30 becomes

$$T_s = \left[\frac{D(L_{cs} + D)(W_{cm})}{10,560,000} + \frac{DM}{2,640,000}\right]\left(\frac{1}{2}\right) \qquad (4\text{-}31)$$

where T_s = ton-miles setting casing
L_{cs} = length of joint of casing in ft
W_{cm} = effective weight per foot of casing in lb/ft

The effective weight per foot of casing W_{cm} may be estimated from data given on Figure 4-78 for drill pipe (using the approximate lb/ft), or calculated as

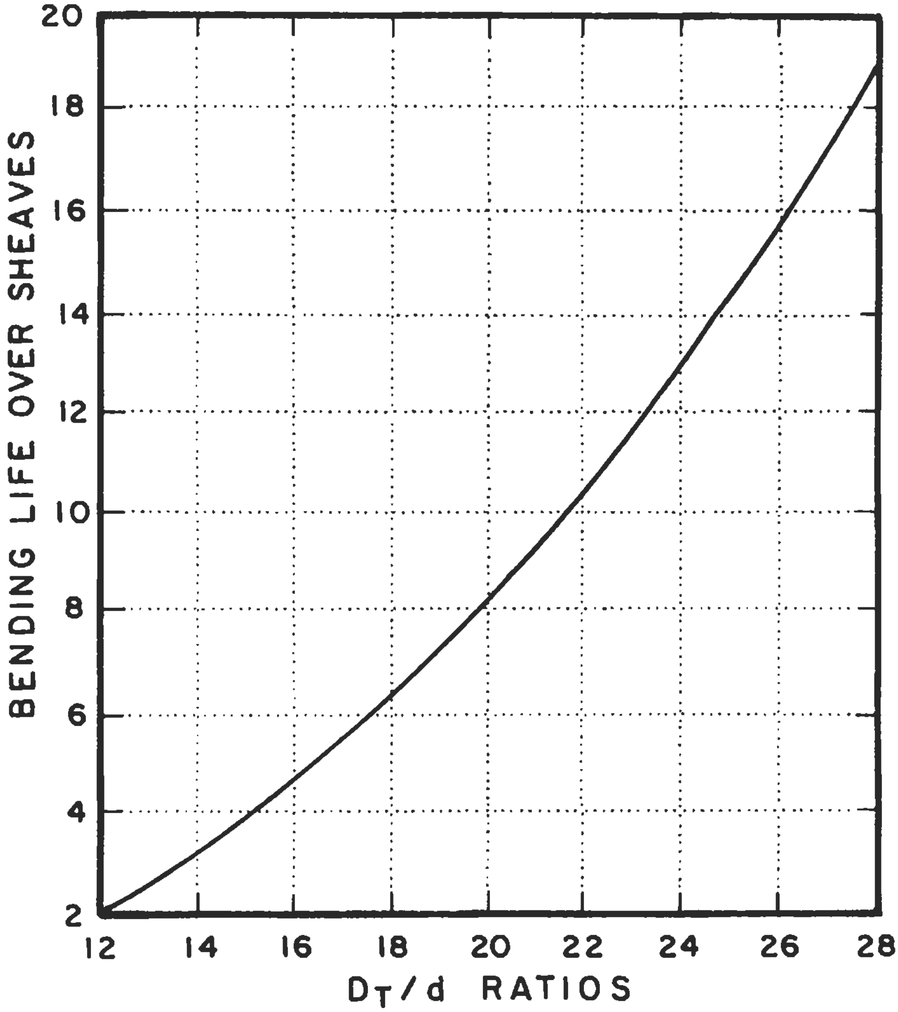
$$W_{cm} = W_{ca}(1 - 0.015B) \qquad (4\text{-}32)$$

where W_{ca} is weight per foot of casing in air in lb/ft
B is weight of drilling fluid from Figure 4-79 or Figure 4-80 in lb/gal

Short Trip Operations. The ton-miles of work performed in short trip operations, as for drilling and coring operations, is also expressed in terms of round trips. Analysis shows that the ton-miles of work done in making a short trip is equal to the difference in round trip ton-miles for the two depths in question. This can be expressed as

$$T_{ST} = T_6 - T_5 \qquad (4\text{-}33)$$

where T_{ST} = ton-miles for short trip
T_5 = ton-miles for one round trip at depth D_5 (shallower depth)
T_6 = ton-miles for one round trip at depth D_6 (deeper depth)



D_T = tread diameter of sheave, inches *(mm)* (see Fig. 4-73)
d = nominal rope diameter, inches *(mm)*.

Figure 4-79. Relative service for various D_T/d ratios for sheaves [11].*
See "Diameter of Sheaves," subparagraph titled "Variation for Different Service Applications."
*Based on laboratory tests involving systems consisting of sheaves only.

For the comparative evaluation of service from rotary drilling lines, the grand total of ton-miles of work performed will be the sum of the ton-miles for all round-trip operations (Equation 4-26), the ton-miles for all drilling operations (Equation 4-27), the ton-miles for all coring operations (Equation 4-29), the ton-miles for all casing setting operations (Equation 4-30), and the ton-miles for all short trip operations (Equation 4-33). By dividing the grand total ton-miles for

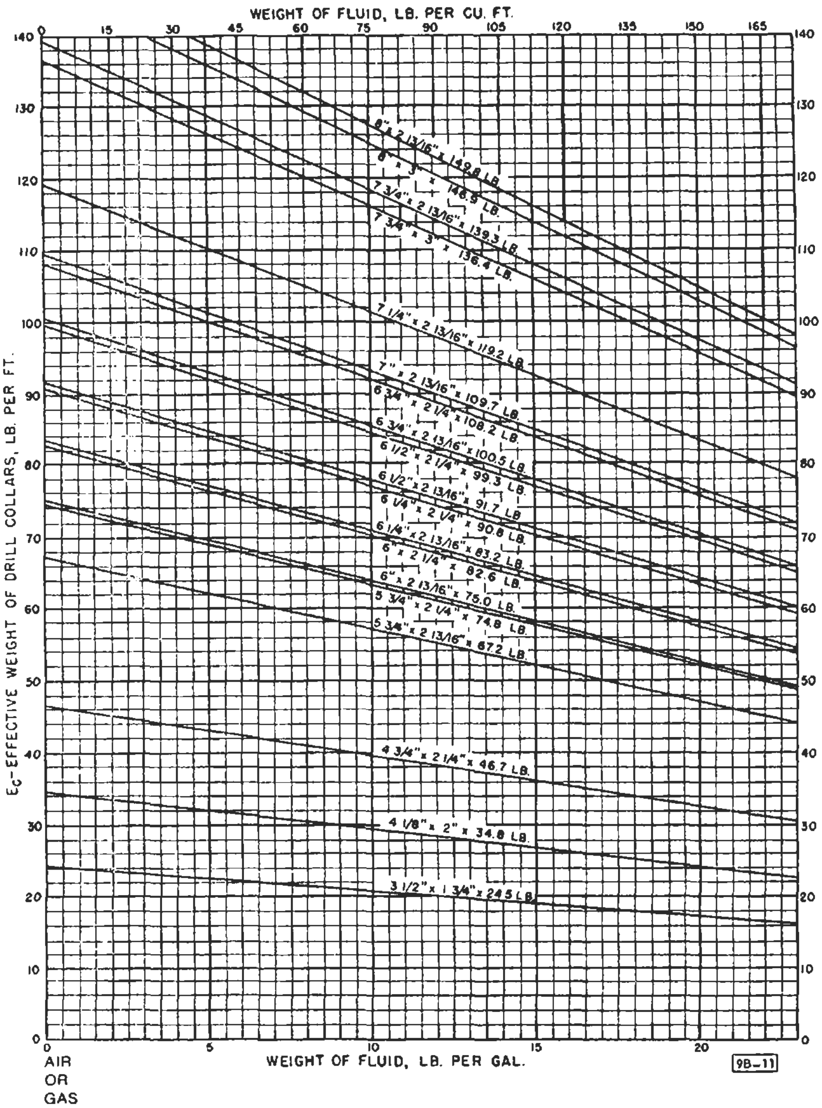
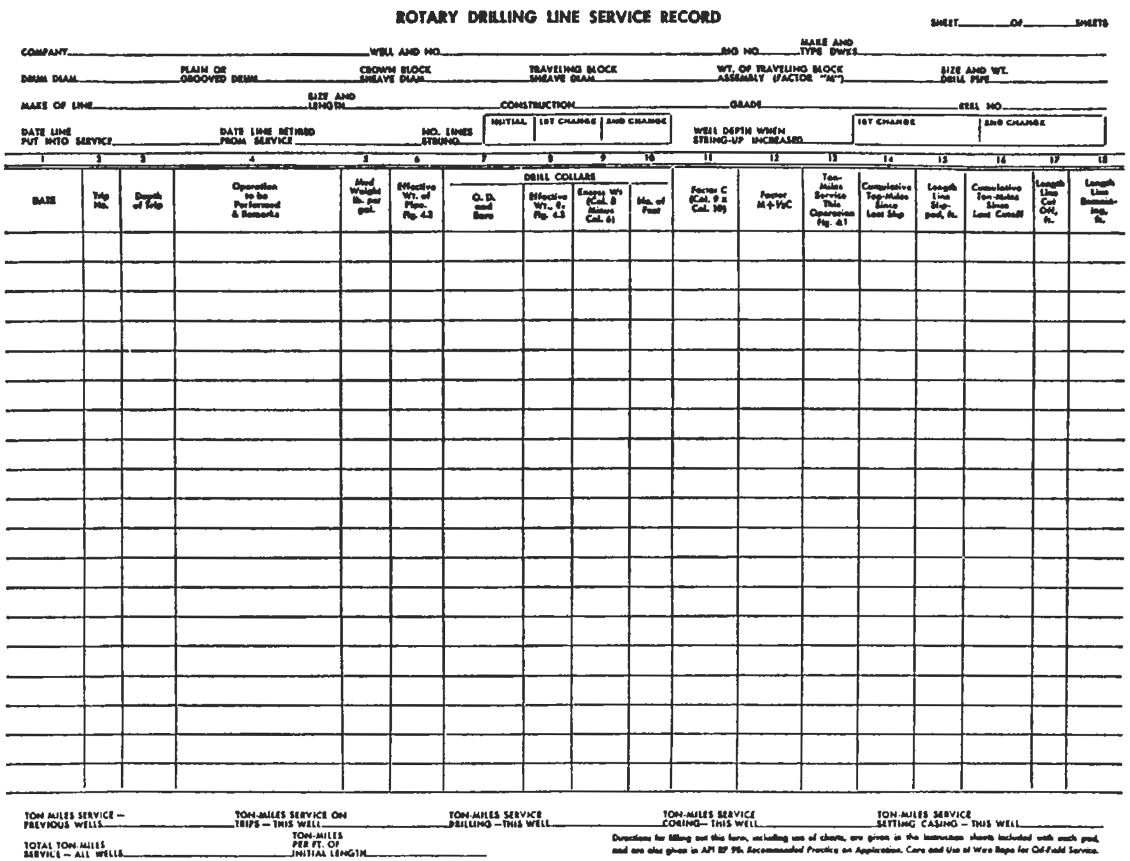


Figure 4-80. Effective weight of drill collars in drilling fluid [11].

all wells by the original length of line in feet, the evaluation of rotary drilling lines in ton-miles per foot on initial length may be determined.

Rotary Drilling Line Service-Record Form

Figure 4-81 is a rotary drilling line service-record form. It can be filled out on the bases of Figure 4-82 and previous discussion.



ROTARY DRILLING LINE SERVICE RECORD

SHEET ______ OF ______ SHEETS

COMPANY ______ WELL AND NO. ______ RIG NO. ______ MAKE AND TYPE DWKS. ______

DRUM DIAM. ______ PLAIN OR GROOVED DRUM ______ CROWN BLOCK SHEAVE DIAM. ______ TRAVELING BLOCK SHEAVE DIAM. ______ WT. OF TRAVELING BLOCK ASSEMBLY (FACTOR "M") ______ SIZE AND WT. DRILL PIPE ______

MAKE OF LINE ______ SIZE AND LENGTH ______ CONSTRUCTION ______ GRADE ______ REEL NO. ______

DATE LINE PUT INTO SERVICE ______ DATE LINE RETIRED FROM SERVICE ______ NO. LINES STRUNG ______ INITIAL | 1ST CHANGE | 2ND CHANGE — WELL DEPTH WHEN STRING-UP INCREASED ______ 1ST CHANGE | 2ND CHANGE

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
						Drill Collars											
Date	Trip No.	Depth of Trip	Operation to be Performed & Remarks	Mud Weight lb. per gal.	Effective Wt. of Pipe, Fig. 4-3	O.D. and Bore	Effective Wt., lb. Fig. 4-3	Excess Wt (Col. 8 Minus Col. 6)	No. of Feet	Factor C (Col. 9 x Col. 10)	Factor M+½C	Ton-Miles Service This Operation Fig. 4-1	Cumulative Ton-Miles Since Last Slip	Length Line Slipped, ft.	Cumulative Ton-Miles Since Last Cutoff	Length Line Cut Off, ft.	Length Line Remaining, ft.

TON-MILES SERVICE — PREVIOUS WELLS ______ TON-MILES SERVICE ON TRIPS — THIS WELL ______ TON-MILES SERVICE DRILLING — THIS WELL ______ TON-MILES SERVICE CORING — THIS WELL ______ TON-MILES SERVICE SETTING CASING — THIS WELL ______

TOTAL TON-MILES SERVICE — ALL WELLS ______ TON-MILES PER FT. OF INITIAL LENGTH ______

Directions for filling out this form, including use of charts, are given in the instruction sheets included with each pad, and are also given in API RP 9B: *Recommended Practice on Application, Care and Use of Wire Rope for Oil-Field Service.*

Figure 4-81. Facsimile of rotary drilling line service-record form [11].

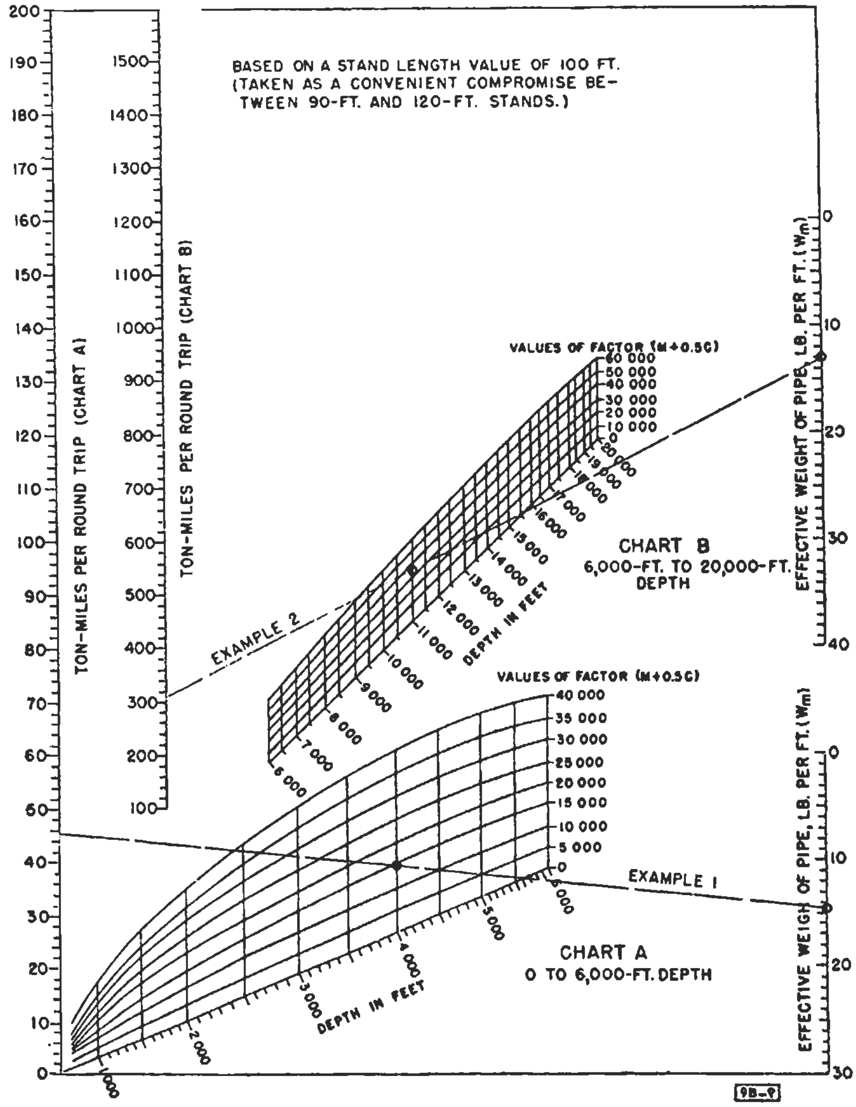


Figure 4-82. Rotary-drilling ton-mile charts [11].

Slipping and Cutoff Practice for Rotary Drilling Lines

Using a planned program of slipping and cutoff based upon increments of service can greatly increase the service life of drilling lines. Determining when to slip and cut depending only on visual inspection, will result in uneven wear, trouble with spooling (line "cutting in" on the drum), and long cutoffs, thus decreasing the service life. The general procedure in any program should be to supply an excess of drilling line over that required to string up, and to slip this excess through the system at such a rate that it is evenly worn and that the line removed by cutoff at the drum end has just reached the end of its useful life.

Initial Length of Line. The relationship between initial lengths of rotary lines and their normal service life expectancies is shown in Figure 4-83. Possible savings by the use of a longer line may be offset by an increased cost of handling for a longer line.

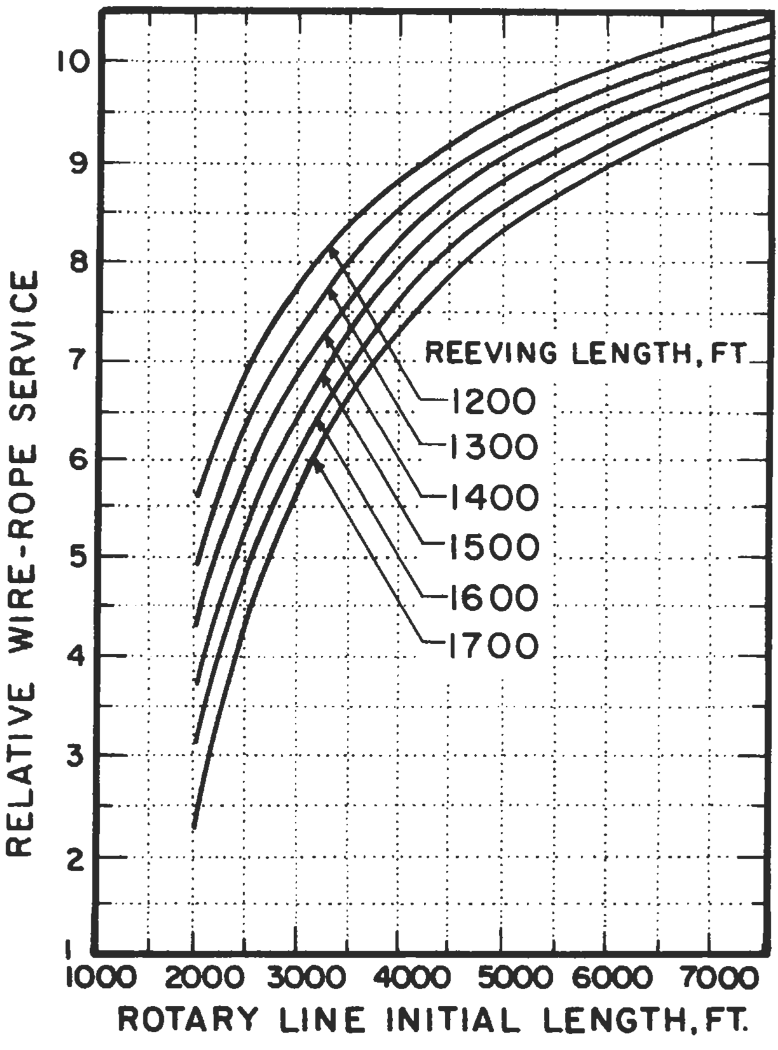
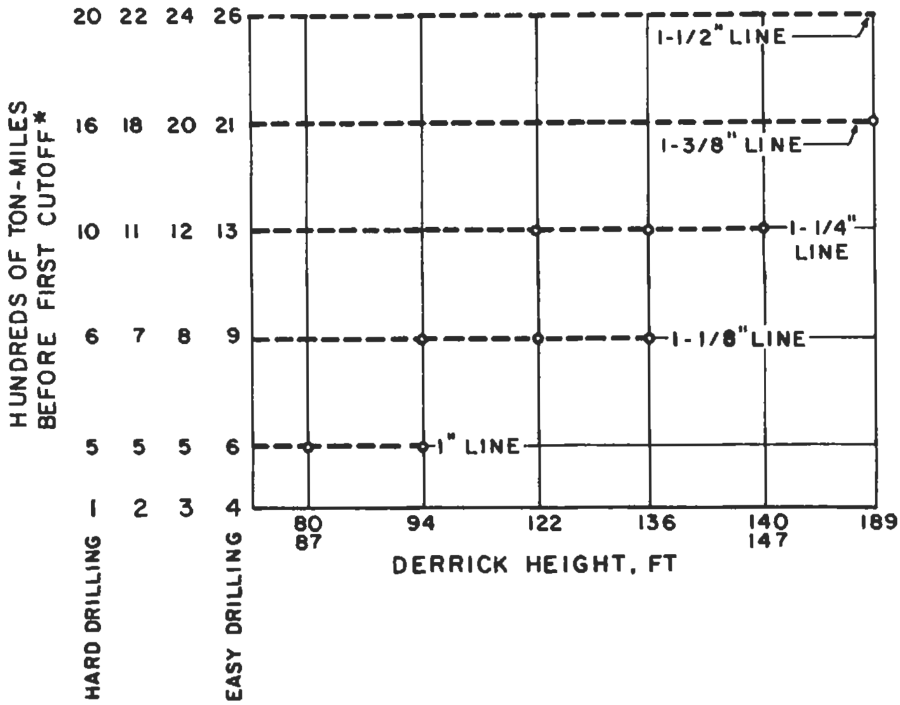


Figure 4-83. Relationship between rotary-line initial length and service life [11].*

*Empirical curves developed from general field experience.

Service Goal. A goal for line service in terms of ton-miles between cutoffs should be selected. This value can initially be determined from Figures 4-84 and 4-85 and later adjusted in accordance with experience. Figure 4-86 shows a graphical method of determining optimum cutoff frequency.

Variations in Line Service. Ton-miles of service will vary with the type and condition of equipment used, drilling conditions encountered, and the skill used



Explanation:
To determine (approximately) the desirable ton-miles before the first cutoff on a new line, draw a vertical line from the derrick height to the wireline size used. Project this line horizontally to the ton-mile figure given for the type of drilling encountered in the area. Subsequent cutoffs should be made at 100 ton-miles less than those indicated for 1⅛-in. and smaller lines, and at 200 ton-miles less than 1¼-in. and 1⅜-in. lines.

Figure 4-84. Ton-mile derrick height and line-size relationships [11].*

*The values for ton-miles before cutoff, as given in Figure 4-84 were calculated for improved plow steel with an independent wire-rope core and operating at a design factor of 5. When a design factor other than 5 is used, these values should be modified in accordance with Figure 4-85. The values given in Figure 4-84 are intended to serve as a guide for the selection of initial ton-mile values as explained in Par. "Service Goal." These values are conservative, and are applicable to all typical constructions of wire rope as recommended for the rotary drilling lines shown in Table 4-9.

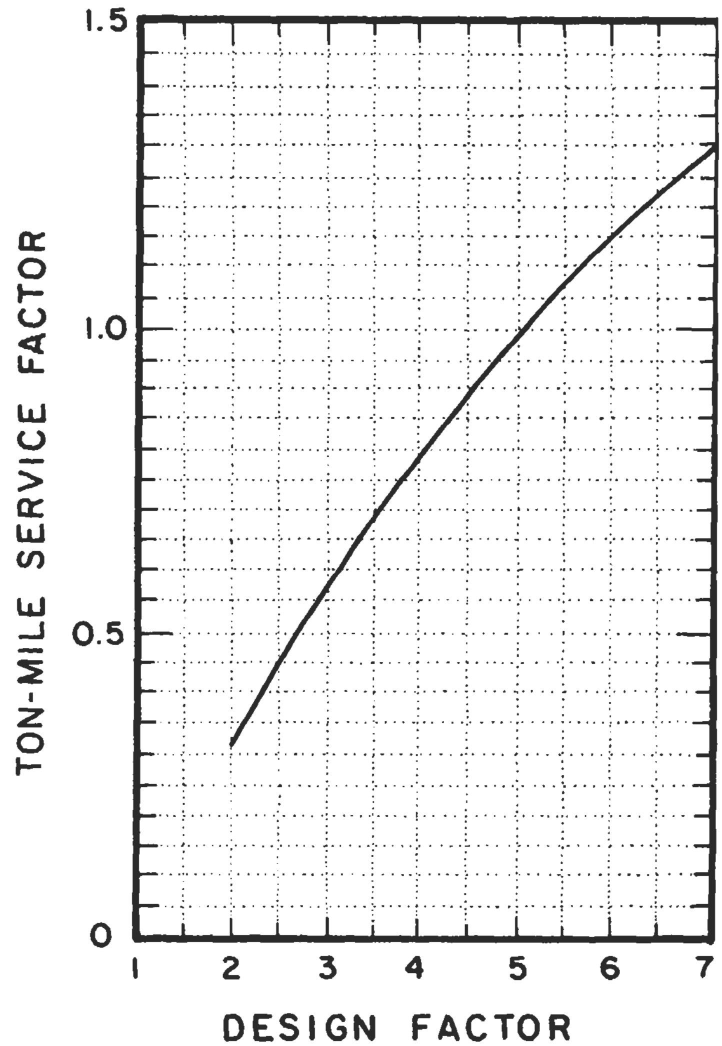


Figure 4-85. Relationship between design factors and ton-mile service factors [11].*

NOTE: Light loads can cause rope to wear out from fatigue prior to accumulation of anticipated ton-miles.

*Based on laboratory tests of bending over sheaves.

in the operation. A program should be "tailored" to the individual rig. The condition of the line as moved through the reeving system and the condition of the cutoff portions will indicate whether the proper goal was selected. In all cases, visual inspection of the wire rope by the operator should take precedence over any predetermined procedures. (See Figure 4-86 for a graphical comparison of rope services.)

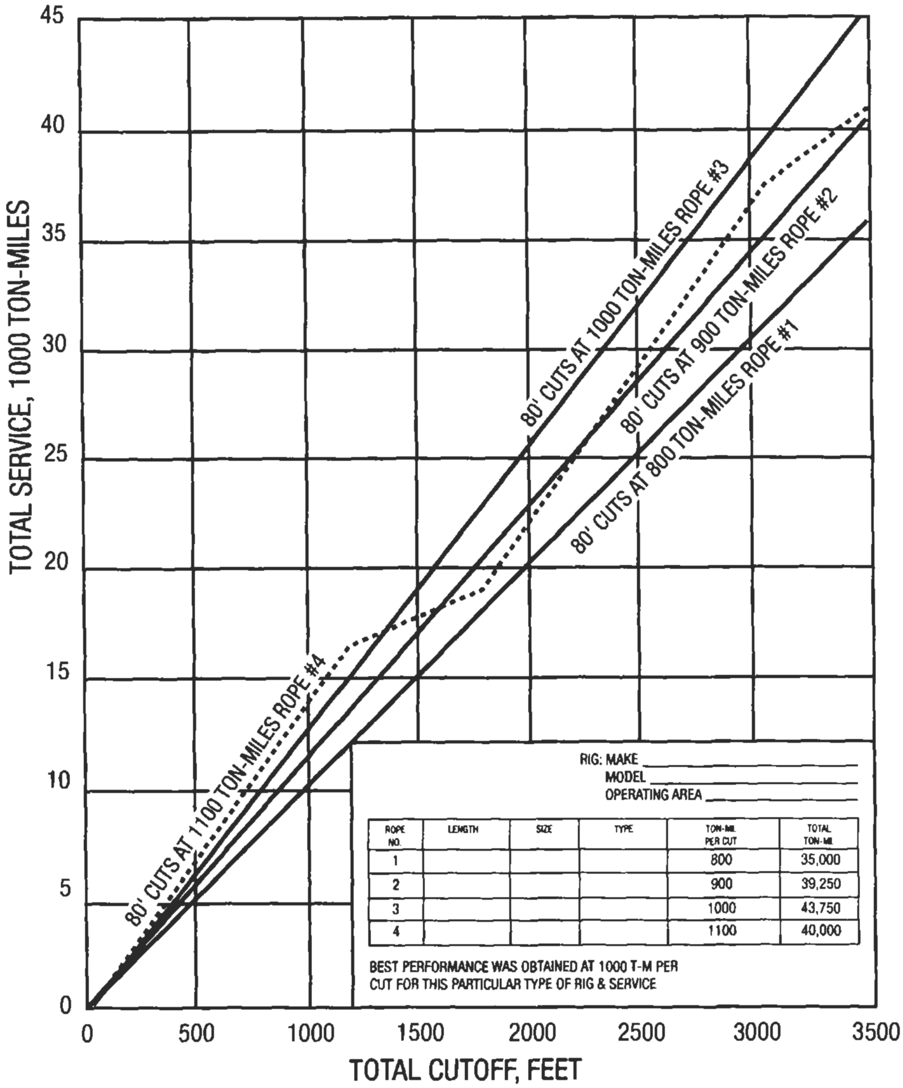


Figure 4-86. Graphic method of determining optimum frequency of cutoff to give maximum total ton-miles for a particular rig operating under certain drilling conditions [11].

Cutoff Length. The following factors should be considered in determining a cutoff length:

1. The excess length of line that can conveniently be carried on the drum.
2. Load-pickup points from reeving diagram.
3. Drum diameter and crossover points on the drum.

The crossover and pickup points should not repeat. This is done by avoiding cutoff lengths that are multiples of either drum circumference, or lengths between pickup points. Successful programs have been based on cutoff lengths ranging from 30 to 150 ft. Table 4-36 shows a recommended length of cutoff (number of drum laps) for each height derrick and drum diameter.

Slipping Program. The number of slips between cutoffs can vary considerably depending upon drilling conditions and the length and frequency of cutoffs. Slips should be increased if the digging is rough, if jarring jobs occur, etc. Slipping that causes too much line piles up on the drum, particularly an extra layer on the drum, before cutoff should be avoided. In slipping the line, the rope should be slipped an amount such that no part of the rope will be located for a second time in a position of severe wear. The positions of severe wear are the point of crossover on the drum and the sections in contact with the traveling-block and crown-block sheaves at the pickup position. The cumulative

Table 4-36
Recommended Cutoff Lengths in Terms of Drum Laps*
See Paragraph Titled "Cutoff Length" [11]

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Derrick or Mast Height, ft	Drum Diameter, in.													
	11	13	14	16	18	20	22	24	26	28	30	32	34	36
	Number of Drum Laps per Cutoff													
151 Up										15½	14½	13½	12½	11½
141 to 150								13½	12½	11½	11½	10½		
133 to 140						15½	14½	12½	11½	11½	10½	9½		
120 to 132				17½	15½	14½	12½	12½	11½	10½	9½	9½		
91 to 119		19½	17½	14½	12½	11½	10½	9½	9½	8½				
73 to 90		17½	14½	12½	11½									
Up through 72	12½	11½												

*In order to insure a change of the point of crossover on the drum, where wear and crushing are most severe, the laps to be cut off are given in multiples of one-half lap or ono quarter lap dependent upon the type of drum grooving.

Example:

Assumed conditions:

a. Derrick height: 138 ft
b. Wire-line size: $1\frac{1}{4}$ in.
c. Type Drilling: #3
d. Drum diameter: 28 in.
e. Design Factor: 3.

Solution:

1. From Fig. 4-84 determine that (for a line with a design factor of 5) the first cutoff would be made after 1200 ton-miles and additional cut-offs after each successive 1000 ton-miles.
2. Since a design factor of 3 applies, Fig. 4-85 indicates that these values should be multiplied by a factor of 0.58. Hence the first cutoff should be made after 696 ton-miles and additional cutoffs after each successive 580 ton-miles.
3. From Table 4-36 determine that 11½ drum laps (84 ft) should be removed at each cutoff.
4. Slip 21 ft every 174 ton-miles for four times and cut off after the fourth slip. Thereafter, slip 21 ft every 145 ton-miles and cut off on the fourth slip.

number of feet slipped between cutoffs should be equal to the recommended feet for ton-mile cutoff. For example, if cutting off 80 ft every 800 ton-miles, 20 ft should be slipped every 200 ton-miles, and the line cut off on the fourth slip.

Field Troubles and Their Causes

All wire rope will eventually deteriorate in operation or have to be removed simply by virtue of the loads and reversals of load applied in normal service. However, many conditions of service or inadvertent abuse will materially shorten the normal life of a wire rope of proper construction although it is properly applied. The following field troubles and their causes give some of the field conditions and practices that result in the premature replacement of wire rope. It should be borne in mind that in all cases the contributory cause of removal may be one or more of these practices or conditions.

Wire-Rope Trouble	Possible Cause
Rope broken (all strands).	Overload resulting from severe impact, kinking, damage, localized wear, weakening of one or more strands, or rust-bound condition and loss of elasticity. Loss of metallic area due to broken wires caused by severe bending.
One or more whole strands parted.	Overloading, kinking, divider interference, localized wear, or rust-bound condition. Fatigue, excessive speed, slipping, or running too loosely. Concentration of vibration at dead sheave or dead-end anchor.
Excessive corrosion.	Lack of lubrication. Exposure to salt spray, corrosive gases, alkaline water, acid water, mud, or dirt. Period of inactivity without adequate protection.
Rope damage by careless handling in hauling to the well or location.	Rolling reel over obstructions or dropping from car, truck, or platform. The use of chains for lashing, or the use of lever against rope instead of flange. Nailing through rope to flange.
Damage by improper socketing.	Improper seizing that allows slack from one or more strands to work back into rope; improper method of socketing or poor workmanship in socketing, frequently shown by rope being untwisted at socket, loose or drawn.
Kinks, dog legs, and other distorted places.	Kinking the rope and pulling out the loops such as in improper coiling or unreeling. Improper winding on the drum. Improper tie-down. Open-drum reels having longitudinal spokes too widely spaced. Divider interference. The addition of improperly spaced cleats increase the drum diameter. Stressing while rope is over small sheave or obstacle.

Wire-Rope Trouble	Possible Cause
Damage by hooking back slack too tightly to girt.	Operation of walking beam causing a bending action on wires at clamp and resulting in fatigue and cracking of wires, frequently before rope goes down into hole.
Damage or failure on a fishing job.	Rope improperly used on a fishing job, resulting in damage or failure as a result of the nature of the work.
Lengthening of lay and reduction of diameter.	Frequently produced by some type of overloading, such as an overload resulting in a collapse of the fiber core in swabbing lines. This may also occur in cable-tool lines as a result of concentrated pulsating or surging forces that may contribute to fiber-core collapse.
Premature breakage of wires.	Caused by frictional heat developed by pressure and slippage, regardless of drilling depth.
Excessive wear in spots.	Kinks or bends in rope due to improper handling during installation or service. Divider interference; also, wear against casing or hard shells or abrasive formations in a crooked hole. Too infrequent cutoffs on working end.
Spliced rope.	A splice is never as good as a continuous piece of rope, and slack is liable to work back and cause irregular wear.
Abrasion and broken wires in a straight line. Drawn or loosened strands. Rapid fatigue breaks.	Injury due to slipping rope through clamps.
Reduction in tensile strength or damage to rope.	Excessive heat due to careless exposure to fire or torch.
Distortion of wire rope.	Damage due to improperly attached clamps or wire-rope clips.
High strands.	Slipping through clamps, improper seizing, improper socketing or splicing, kinks, dog legs, and core popping.
Wear by abrasion.	Lack of lubrication. Slipping clamp unduly. Sandy or gritty working conditions. Rubbing against stationary object or abrasive surface. Faulty alignment. Undersized grooves and sheaves.
Fatigue breaks in wires.	Excessive vibration due to poor drilling conditions, i.e., high speed, rope, slipping, concentration of vibration at dead sheave or dead-end anchor, undersized grooves and sheaves, and improper selection of rope construction. Prolonged bending action over spudder sheaves, such as that due to hard drilling.

Wire-Rope Trouble	Possible Cause
Spiraling or curling.	Allowing rope to drag or rub over pipe, sill, or any object during installation or operation. It is recommended that a block with sheave diameter 16 times the nominal wire-rope diameter, or larger, be used during installation of the line.
Excessive flattening or crushing.	Heavy overload, loose winding on drum, or cross winding. Too infrequent cutoffs on working end of cable-tool lines. Improper cutoff and moving program for cable-tool lines.
Bird-caging or core-popping.	Sudden unloading of line such as hitting fluid with excessive speed. Improper drilling motion or jar action. Use of sheaves of too small diameter or passing line around sharp bend.
Whipping off of rope.	Running too loose.
Cutting in on drum.	Loose winding on drum. Improper cutoff and moving program for rotary drilling lines. Improper or worn drum grooving or line turnback plate.

ROTARY EQUIPMENT

Rotary equipment refers to the pieces of surface equipment in drilling operations that actually rotate or impart rotating motion to the drill pipe. This equipment includes the upper members of the drill string such as the swivel, swivel sub kelly cock, kelly, lower kelly valve, and kelly sub (Figure 4-87), as well as the kelly bushing and rotatary table [13].

Swivel and Rotary Hose

Swivel

The swivel (Figure 4-88) suspends the kelly, allows for rotation of the kelly and drill string, and provides a connection for the rotary hose to allow mud circulation.

The rotary swivel is pressure tested periodically, and the test pressure is shown on the swivel nameplate. All cast members in the swivel hydraulic circuit are pressure tested in production, and the test pressure is shown on the cast member.

Rotary Hose

Although the rotary hose is not a rotating element, it is mentioned here due to its connection with the swivel. It is used as the flexible connector between the top of the standpipe and the swivel, and allows for vertical travel of the swivel and block (Figure 4-89). It is usually 45 ft or longer. Rotary hose specifications are provided in API Specification 7 [13]. Each hose assembly is

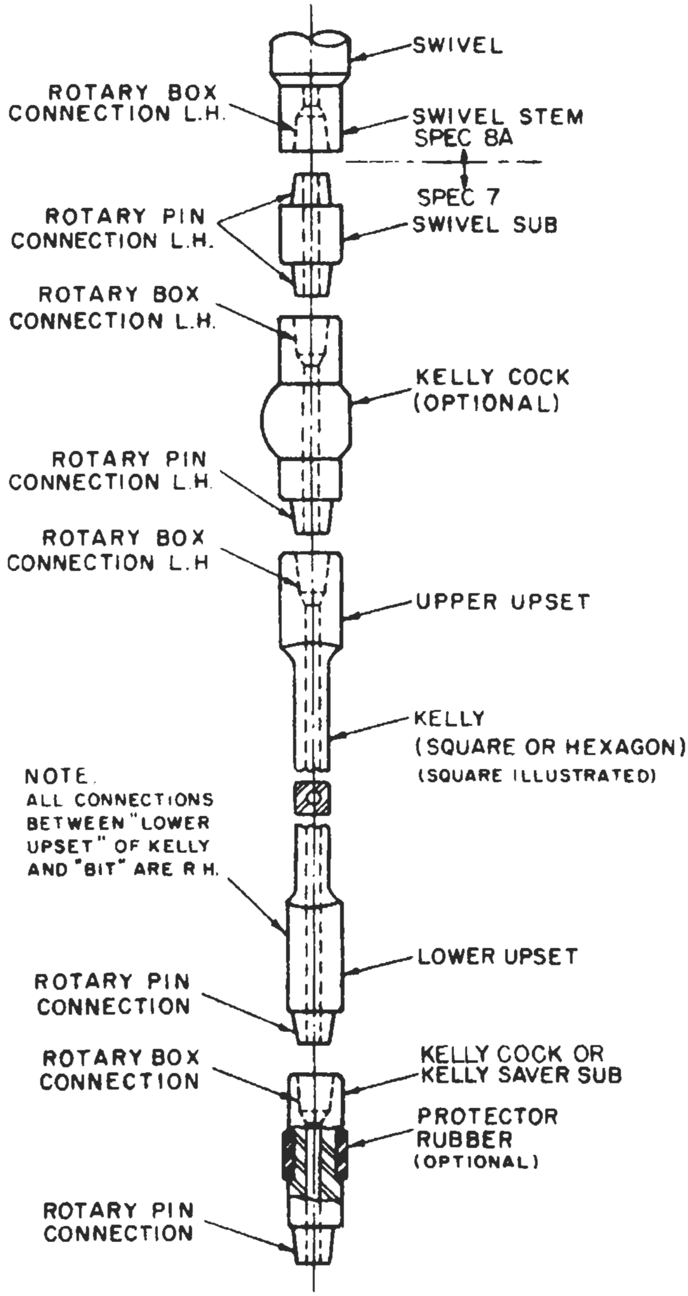


Figure 4-87. Rotary equipment—surface elements of drill string [13].

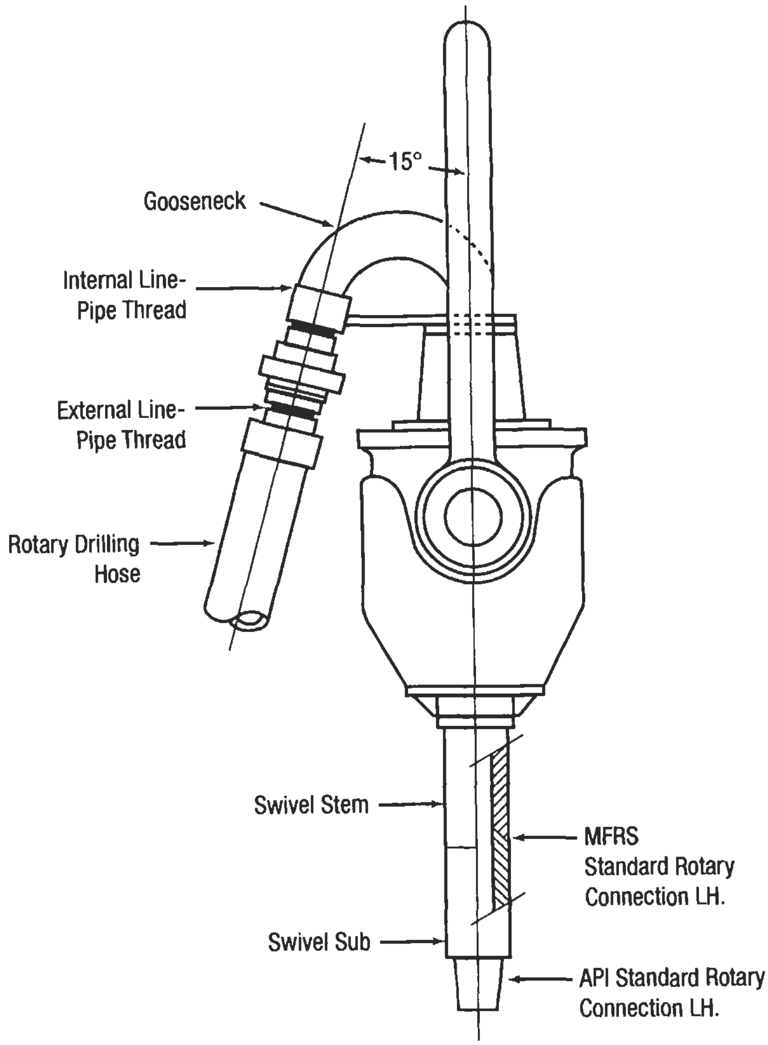


Figure 4-88. Swivel nomenclature [15].

individually tested at its applicable pressure and is held at this pressure for a minimum period of 1 min (test pressure). The maximum working pressure of the hose assembly includes the surge pressure and should be at least $2\frac{1}{2}$ times smaller than the minimum burst pressure of the hose.

Rotary hose external connections (i.e., the connection to the swivel gooseneck) are threaded with line-pipe thread as specified in API Specification 5B [14].

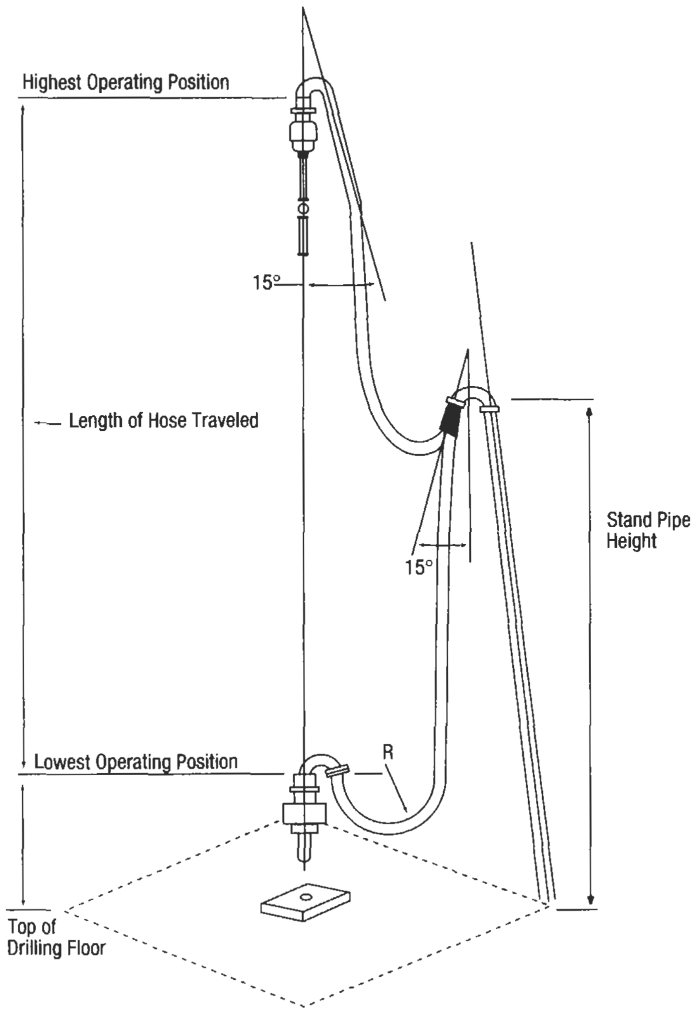


Figure 4-89. Rotary hose [15].

Drill-Stem Subs

Different types of drill-stem subs are shown in Figures 4-90 and 4-91. The classification of drill-stem subs is presented in Table 4-37.

Swivel Sub

The outside diameter of the swivel sub is at least equal to the diameter of the upper kelly box. The swivel sub should have a minimum of 8 in. of tong space length. The minimum bore diameter is equal to that of the kelly. The

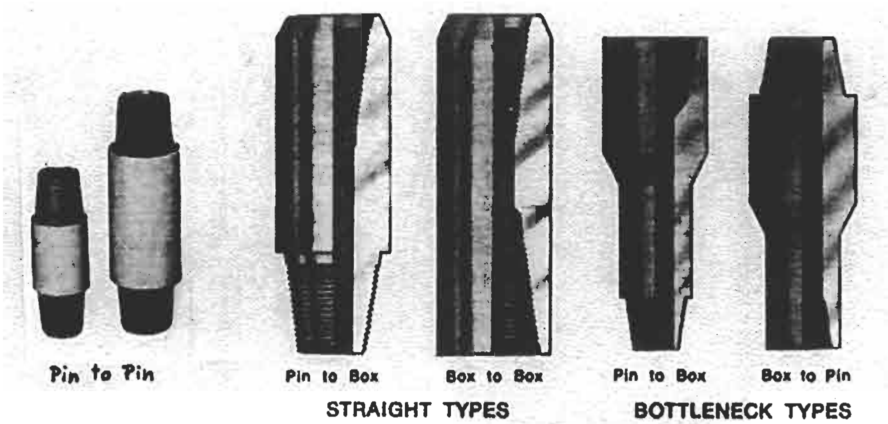


Figure 4-90. Drill-stem subs [13].

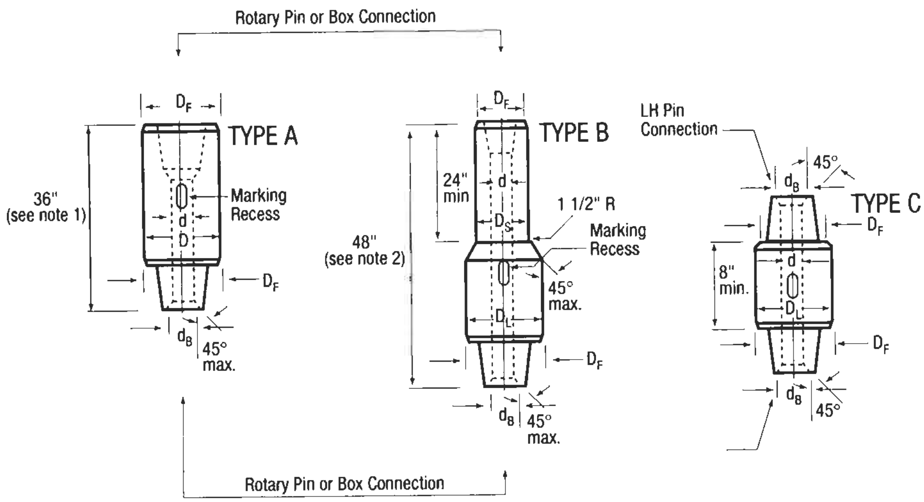


Figure 4-91. Types of drill-stem subs [13].

swivel sub is furnished with a pin-up and a pin-down rotary shouldered connection. Both connections are left handed.

Kelly Cock and Lower Kelly Valve

The kelly cock and lower kelly valve are manually operated valves in the circulating system.

Table 4-37
Drill-Stem Subs [13]

1	2	3	4
Type	**Class**	**Upper Connection to Assemble w/**	**Lower Connection to Assemble w/**
A or B	Kelly Sub	Kelly	Tool Joint
"	Tool Joint Sub	Tool Joint	Tool Joint
"	Crossover Sub	Tool Joint	Drill Collar
"	Drill Collar Sub	Drill Collar	Drill Collar
"	Bit Sub	Drill Collar	Bit
C	Swivel Sub	Swivel Sub	Kelly

Kelly Cock

The kelly cock (Figure 4-92) is located between the kelly joint and the swivel. The kelly cock will close the drill string if the swivel, drilling hose, or standpipe develops a leak or rupture and threatens to blowout. It also closes in the event that pressure within the hose exceeds the hose pressure rating. The specifications for kelly cocks are provided in API Specification 7 [1].

Lower Kelly Valves

Some kellys (Figure 4-93) are equipped with a mud check-valve that is placed immediately below the kelly. When mud pumps are shut off, this valve closes to save mud that would otherwise be spilled out onto the rig floor. To avoid loss of pressure across the tool, the kelly valve is either fully open or fully closed while in operation. It opens and closes automatically and automatically allows reverse flow.

Lower Kelly Cock

The lower kelly cock is often substituted for the lower kelly valve. It is operated manually, as shown in Figure 4-92. The specifications for lower kelly cocks are provided in API Specification 7 [1].

Kelly

The kelly (Figure 4-94) is a square-shaped or hexagonal-shaped pipe (drive section) that transmits power from the rotary table to the drill bit. Also, drilling mud is pumped downhole through the kelly. Kellys must conform to the dimensions specified for the respective sizes in API Specification 7 [1].

The drive section of the hexagonal kelly is stronger than the drive section of the square kelly when the appropriate kelly has been selected for a given casing size. For a given bending load, the stress level is less in the hexagonal kelly; thus the hexagonal kelly will operate for more cycles before failure.

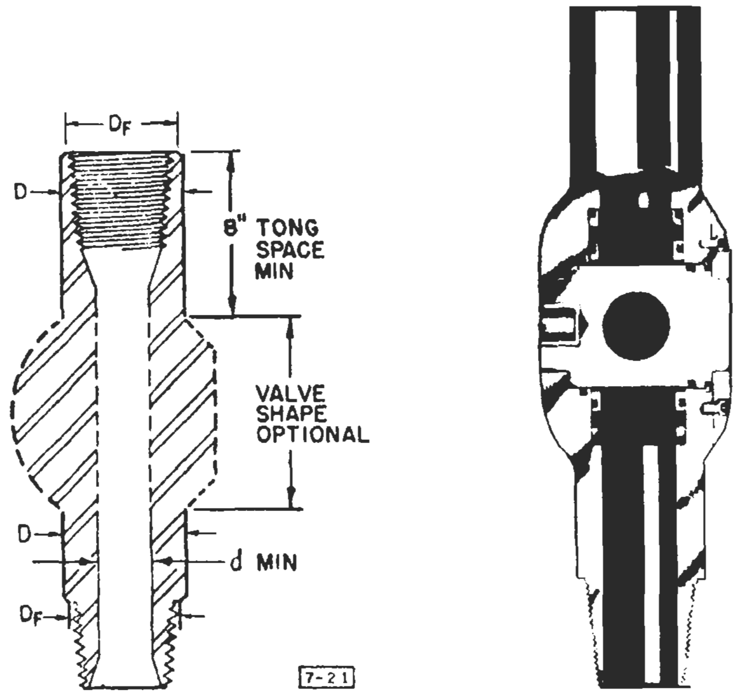


Figure 4-92. Upper kelly cock [13].

Square-Forged Kellys

In square-forged kellys, the decarburized zone has been removed from the corners of the fillet between the drive section and the upset to prevent fatigue cracks. Hexagonal kellys have machined surfaces and are generally free of decarburized zone in the drive section.

The life of the drive section as related to the fit with kelly drive bushings is generally greater when the square drive section is used. However, the use of adjustable drive bushings (adjustable bushings with wear) can drastically increase the life of the square drive section.

The important parts of the kelly that should be examined for wear are:

- the corners of the drive section (for surface wear)
- the junctions between the upsets and the drive section (for cracks)
- the straightness of the kelly.

Rotary Table and Bushings

Rotary Table

The rotary table (Figure 4-95) provides the rotary movement to the kelly. The master bushing of the rotary table encases the kelly bushing or pipe slips, as

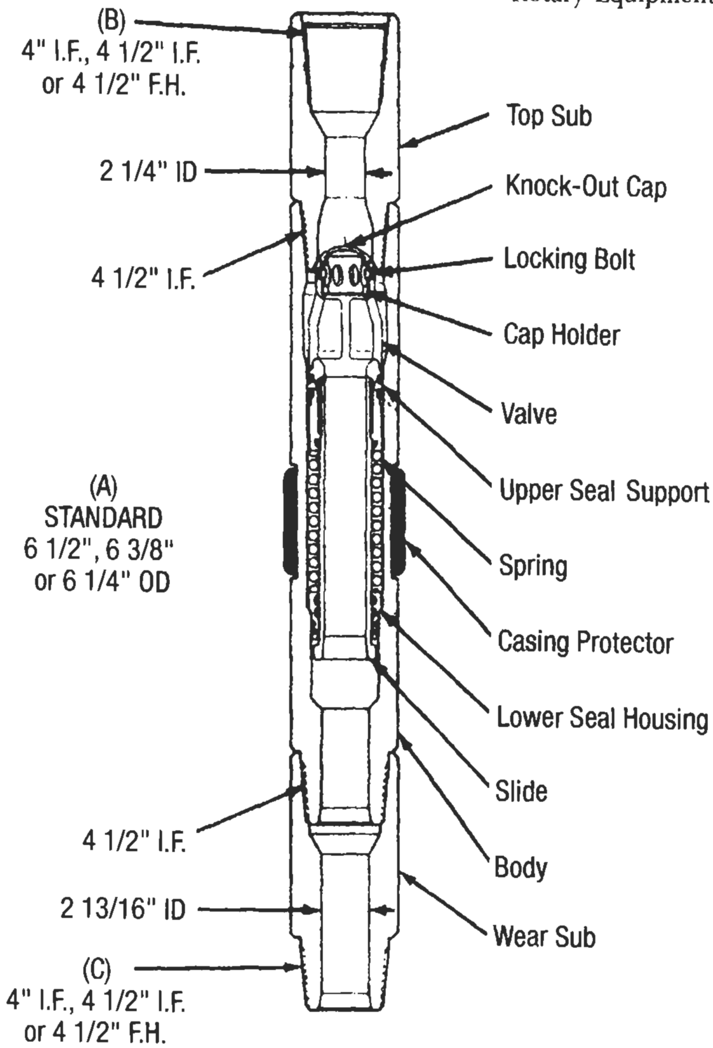


Figure 4-93. Lower kelly valve (mud saver) [16].

shown in Figure 4-96. As the rotary table turns, the master bushing, the kelly, the drill pipe, and the bit also turn. The rotary table is driven by the drawworks.

Master Bushings

There are two types of master bushings:

1. Square drive master bushings (Figure 4-97)
2. Pin drive master bushings (Figure 4-98)

(text continued on page 626)

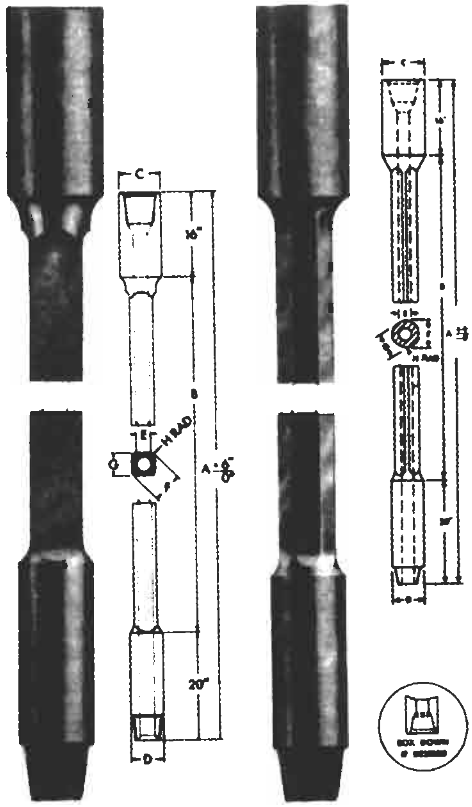


Figure 4-94. Square kelly and hexagonal kelly.

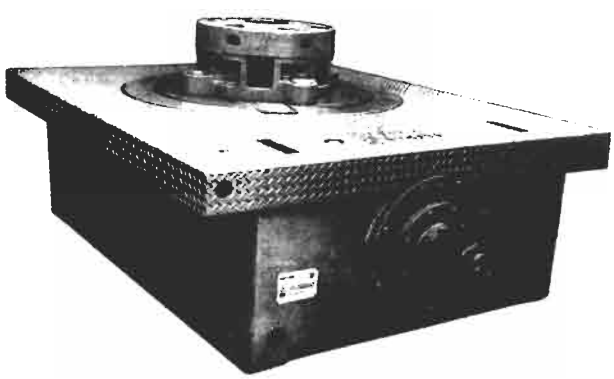


Figure 4-95. Rotary table with pin-drive master bushings [16].

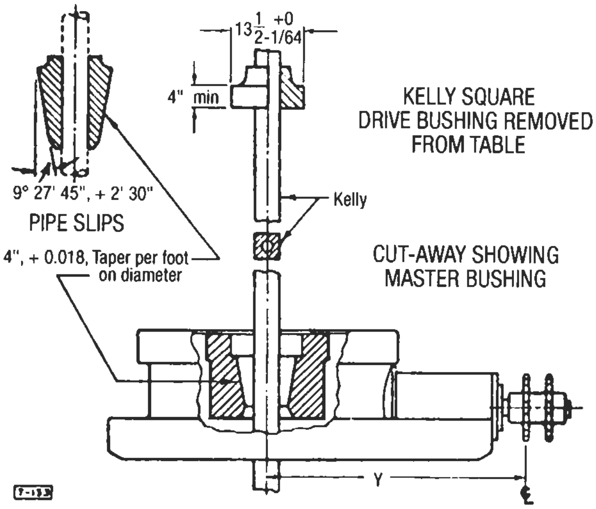


Figure 4-96. Rotary table with square drive bushings and slips [13].

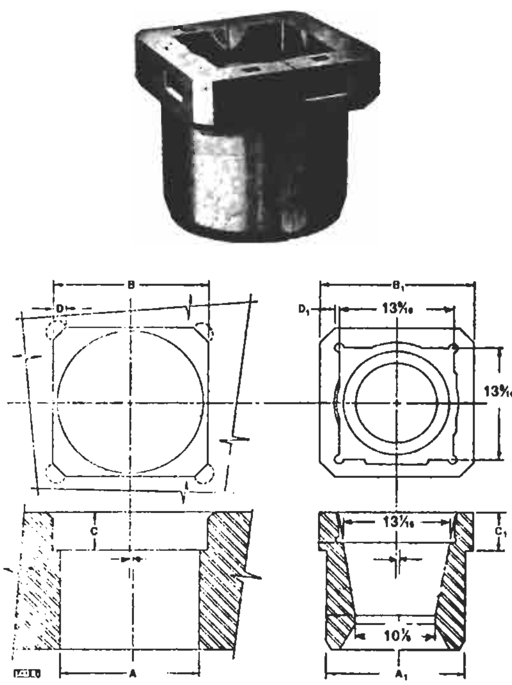


Figure 4-97. Rotary table opening and square drive master bushing [13].

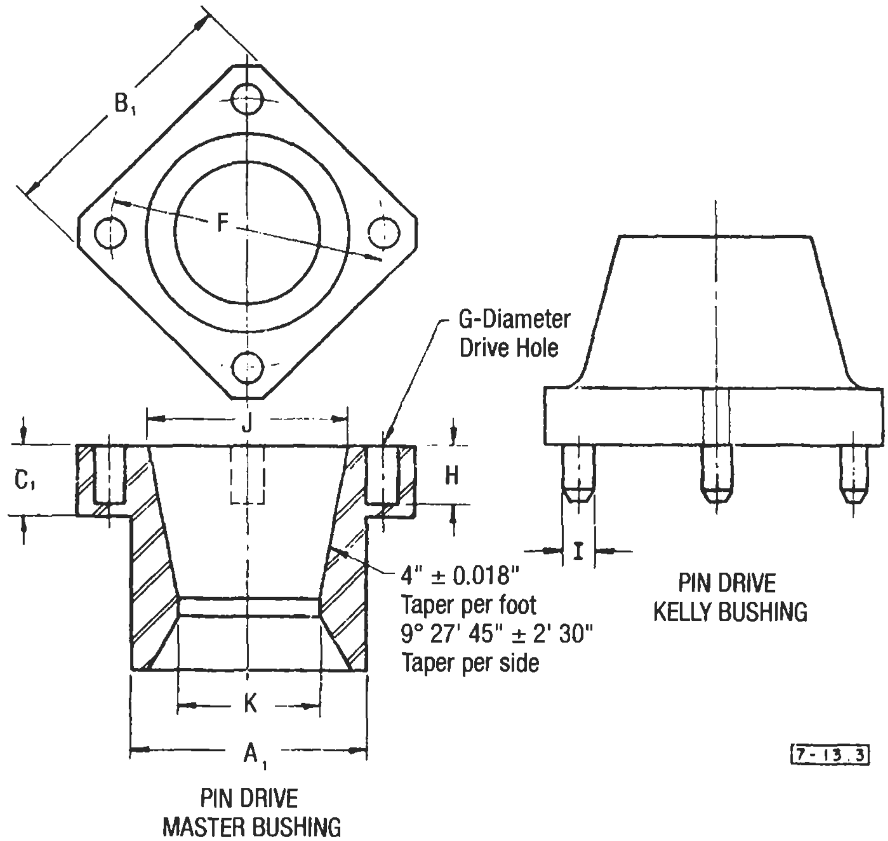


Figure 4-98. Pin-drive master bushing [13].

(text continued from page 623)

The API requirements for rotary table openings for square drive master bushings and the sizes of the square drive and pin-drive master bushings are specified in API Specification 7 [1].

Kelly Bushings

The kelly bushing attaches the kelly to the rotary table. It locks into the master bushing and transfers the torque produced by the table to the kelly. There are two types of kelly bushings [16]:

1. Square drive kelly bushings (aligned with square drive master bushings)
2. Pin drive kelly bushings (aligned with pin-drive master bushings)

MUD PUMPS

Mud pumps consume more than 60% of all the horsepower used in rotary drilling. Mud pumps are used to circulate drilling fluid through the mud circulation system while drilling. A pump with two fluid cylinders, as shown in Figure 4-99, is called a duplex pump. A three-fluid-cylinder pump, as shown in Figure 4-100, is called a triplex pump. Duplex pumps are usually double action, and triplex pumps are usually single action.

Mud pumps consist of a power input end and a fluid output end. The power input end, shown in Figure 4-101, transfers power from the driving engine (usually diesel or electric) to the pump crankshaft. The fluid end does the actual work of pumping the fluid. A cross-section of the fluid end is shown in Figure 4-102.

Pump Installation

Suction Manifold

The hydraulic horsepower produced by mud pumps depends mainly on the geometric and mechanical arrangement of the suction piping. If suction-charging centrifugal pumps (e.g., auxiliary pumps that help move the mud to the mud pump) are not used, the pump cylinders have to be filled by the hydrostatic head.

Incomplete filling of the cylinders can result in hammering, which produces destructive pressure peaks and shortens the pump life. Filling problems become more important with higher piston velocities. The suction pressure loss through the suction valve and seat is from 5 to 10 psi. Approximately 1.5 psi of pressure is required for each foot of suction lift. Since the maximum available atmospheric pressure is 14.7 psi (sea level), suction pits placed below the pump should be

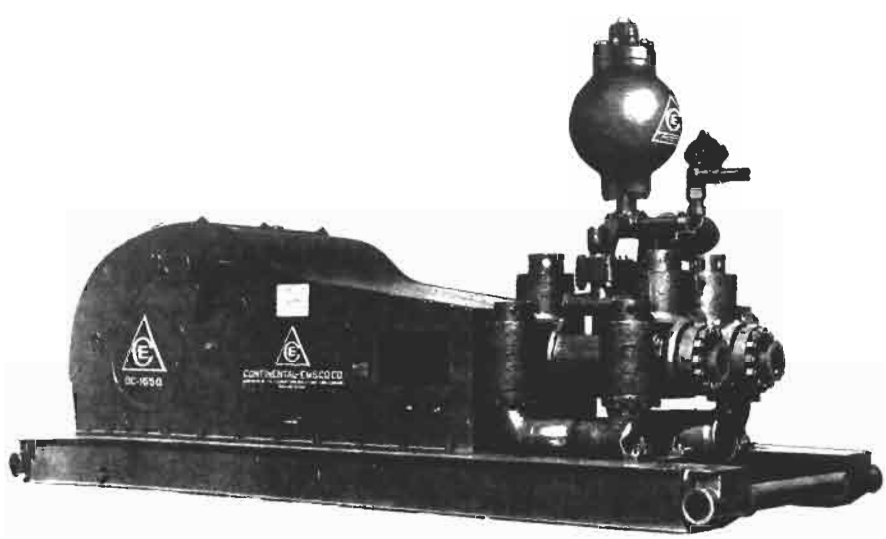


Figure 4-99. Duplex slush (mud) pump. *(Courtesy National Oilwell.)*

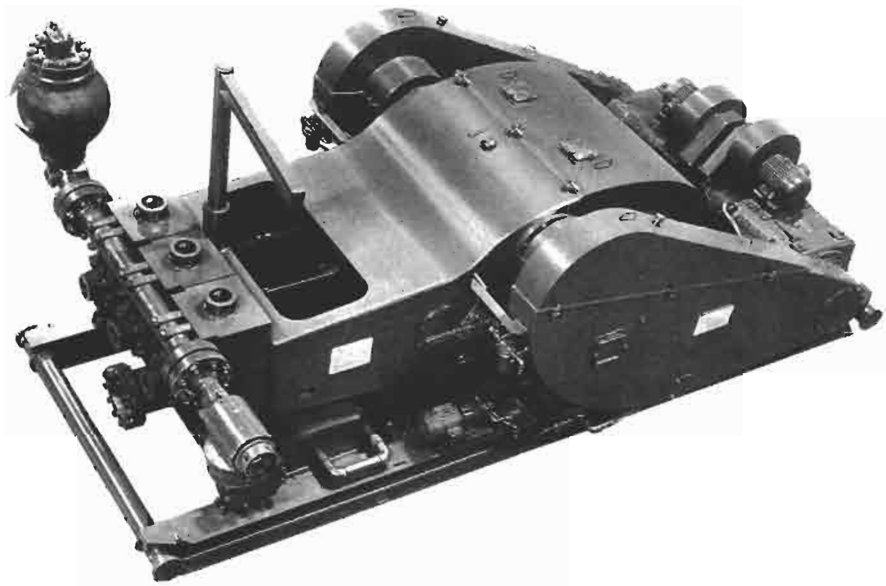


Figure 4-100. Triplex slush (mud) pump. *(Courtesy National Oilwell.)*

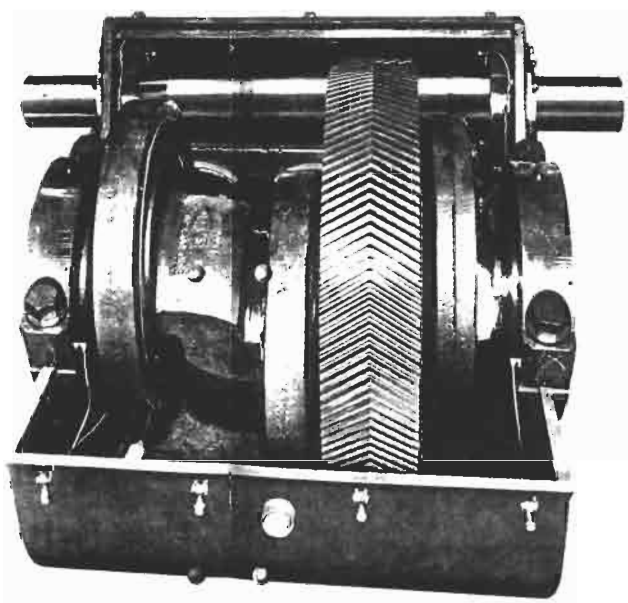


Figure 4-101. Power end of mud pump. *(Courtesy LTV Energy Products Company.)*

eliminated. Instead, suction tanks placed level with or higher than the pump should be used to ensure a positive suction head. Figure 4-103 shows an ideal suction arrangement with the least amount of friction and low inertia.

A poorly designed suction entrance to the pump can produce friction equivalent to 30 ft of pipe. Factors contributing to excessive suction pipe friction are an intake connection with sharp ends, a suction strainer, suction pipe with a small diameter, long runs of suction pipe, and numerous fittings along the

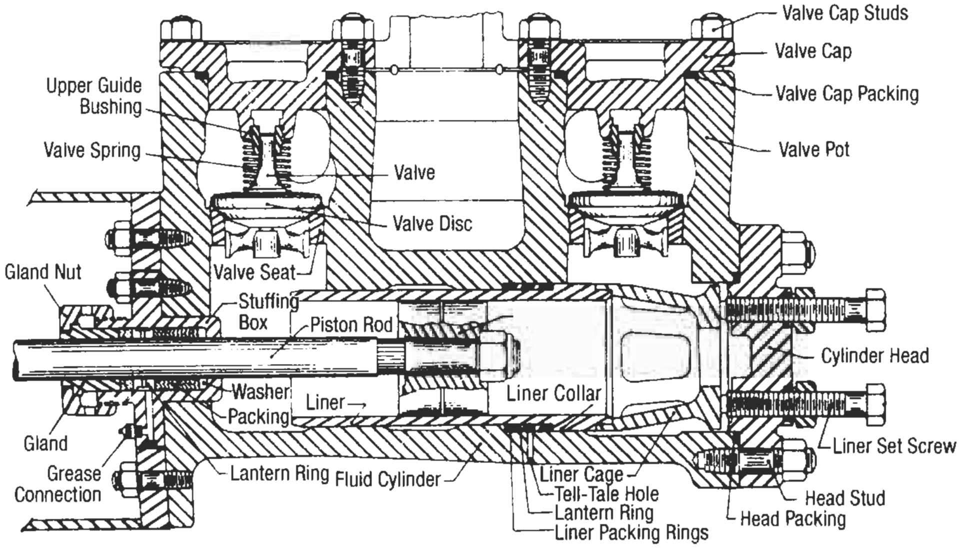


Figure 4-102. Cross-section of fluid end of mud pump. *(Courtesy International Association of Drilling Contractors.)*

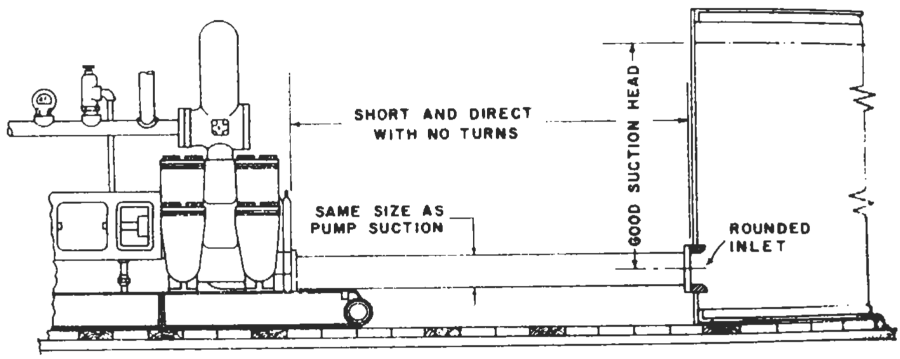


Figure 4-103. Installation of mud pump suction piping. *(Courtesy International Association of Drilling Contractors.)*

suction pipe. Minimizing the effect of inertia requires a reduction of the suction velocity and mud weight. It is generally practical to use a short suction pipe with a large diameter.

When a desirable suction condition cannot be attained, a charging pump becomes necessary. This is a common solution used on many modern rigs.

Cooling Mud

Mud temperatures of 150° can present critical suction problems. Under low pressure or vacuum existing in the cylinder on the suction stroke, the mud can boil, hence decreasing the suction effectiveness. Furthermore, hot mud accelerates the deterioration of rubber parts, particularly when oil is present. Large mud tanks with cooling surfaces usually solve the problem.

Gas and Air Separation

Entrained gas and air expands under the reduced pressure of the suction stroke, lowering the suction efficiency. Gas in water-base mud may also deteriorate the natural rubber parts used. Gases are usually separated with baffles or by changing mud composition.

Settling Pits

The normally good lubricating qualities of mud can be lost if cuttings, particularly fine sand, are not effectively separated from the mud. Adequate settling pits and shale shakers usually eliminate this trouble. Desanders are used occasionally.

Discharge Manifold

A poorly designed discharge manifold can cause shock waves and excessive pressure peaks. This manifold should be as short and direct as possible, avoiding any sharps turns. The conventional small atmospheric air chamber, often furnished with pumps, supplies only a moderate cushioning effect. For best results, this air chamber should be supplemented by a large atmospheric air chamber or by a precharged pulsation dampener.

Pump Operation

Priming

A few strokes of the piston in a dry liner may ruin the liner. When the pump does not fill by gravity or when the cylinders have been emptied by standing too long or by replacement of the piston and liner, it is essential to prime the pump through the suction valve cap openings.

Cleaning the Suction Manifold

Suction lines are often partly filled by settled sand and by debris from the pits, causing the pump to hammer at abnormally low speeds. Frequent inspection and cleaning of the suction manifold is required. The suction strainer can also be a liability if it is not cleaned frequently.

Cleaning the Discharge Strainer

The discharge strainer often becomes clogged with pieces of piston and valve rubber. This may increase the pump pressure that is not shown by the pressure gauge beyond the strainer. The strainer should be inspected and cleaned frequently to prevent a pressure buildup.

Lost Circulation Materials

Usually special solids, such as nut shells, limestone, expanded perlite, etc., are added to the drilling muds to fill or clog rock fractures in the open hole of a well. Most of these lost circulation materials can shorten the life of pump parts. They are especially hard on valves and seats when they accumulate on the seats or between the valve body and the valve disc.

Parts Storage

Pump parts for high-pressure service are made of precisely manufactured materials and should be treated accordingly. In storage at the rig, metal parts should be protected from rusting and physical damage, and rubber parts should be protected from distortion and from exposure to heat, light, and oil. In general, parts should remain in their original packages where they are usually protected with rust-inhibiting coatings and wrappings and are properly supported to avoid damage. Careless stacking of pistons may distort or cut the sealing lips and result in early failures. Hanging lip-type or O-ring packings on a hook or throwing them carelessly into a bin may ruin them. Metal parts temporarily removed from pumps should be thoroughly cleaned, greased, and stored like new parts.

Pump Performance Charts

The charts showing the performance of duplex pumps are shown in Table 4-38 [17]. The charts showing the performance of triplex pumps are shown in Table 4-39 [17]. A chart listing the pump output required for a given annular velocity is shown in Table 4-40 [18]. A chart listing the power input horsepower required for a given pump working pressure is shown in Table 4-41 [19].

Mud Pump Hydraulics

The required pump output can be approximated as follows [20–22]:
Minimum Q (gal/min):

$$Q_{min} = (30 \text{ to } 50)\ D_h \tag{4-36}$$

or

$$Q_{min} = 481\frac{D_h^2 - D_p^2}{\bar{\sigma} D_h} \tag{4-34}$$

where D_h = hole diameter in in.
D_p = pipe diameter in in.
$\bar{\sigma}$ = mud specific weight in lb/gal

(text continued on page 644)

Table 4-38
Mud Pump Performance—Duplex Pumps [17]
Pump Discharge Pressure (PSI) (Shaded Area); Pump Discharge Volume (Gal./Stroke) (Based on 100% Volumetric Efficieny)

MANUFACTURER: BETHLEHEM (DUPLEX)

MODEL	MAX. I.H.P.	MAX. S.P.M.	STROKE LENGTH	ROD SIZE	LINER SIZE (IN)													
					4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2	7-3/4	8
225 AND GEI-225-OB	275	80	14"	2"		1158		931		[illegible]		[illegible]		560		486		426
						4.35		5.35		6.44		7.62		8.90		10.28		11.77
325 AND GBF-325-OB	398	80	16"	2-1/4"		1478		1202		[illegible]		[illegible]		715		[illegible]		541
						4.85		5.97		7.24		8.65		10.0		11.6		13.25
450 AND GCF-450-OB	551	80	16"	2-1/2"		2112		[illegible]		[illegible]		1181		1008		[illegible]		758
						4.73		5.86		7.10		8.48		9.92		11.48		13.18
600 GCF-600-OB GCI-600-OB	712	70	18"	2-1/2"		2762		2230		[illegible]		1547		1318		1138		994
						5.32		6.58		8.00		9.53		11.18		13.01		14.82
B-1640	1999	90	16"	3-1/2"						4570		3900		3360		2920		2570
										[illegible]		[illegible]		9.33		[illegible]		12.59
F-850	848	70	16"	—		3066		2534		2129		1814		1564		1363		1198
G-35	468	100	14"	2-1/8"		1485		[illegible]		[illegible]		878		757		650		580
						4.330		5.330		6.424		7.614		8.900		10.280		11.756
G-45	606	100	16"	2-1/4"		1691		1392		1173		1001		863		751		660
						4.889		6.032		7.283		8.643		10.112		11.689		13.376
G-65	874	100	16"	2-1/2"		2480		2033		1708		1458		1255		1093		961
						4.760		5.902		7.154		8.514		9.982		11.560		13.246
G-85	1212	100	18"	2-3/4"		3066		2534		2128		1814		1564		1363		1198
						5.194		6.480		7.887		9.417		11.070		12.844		14.742
G-300	300	60	16"	2-1/4"		1460		1206		1013	[illegible]	864	800	745	694			
						4.883		6.033		7.283	7.967	8.633	9.383	10.017	10.883			
H-25	336	100	12"	2"	1383		1132		944		[illegible]		686		593			
					3.356		4.172		5.069		6.049		7.109		8.252			
H-150	231	95	12"	2"	1050		843		693		581		494		426			
					3.33		4.14		5.03		6.00		7.06		8.20			

MANUFACTURER: CONTINENTAL EMSCO B-LINE SLUSH PUMPS (DUPLEX)

MODEL	MAX. I.H.P.	MAX. S.P.M.	STROKE LENGTH	ROD SIZE	LINER SIZE (IN) 4-1/4	4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2	7-3/4	8	8-1/4
B-14	265	65	14"	2"			1520 3.8		1220 4.9		[illegible] 5.9		[illegible] 7.1		715 8.3		615 9.6		[illegible] 11.1		
B-1640	1670	75	16"	3-1/2"								4570 6.5		3900 7.9		3360 9.3		2960 10.9		2570 12.6	
C-12	165	65	12"	1-7/8"	[illegible] 2.7		[illegible] 3.4		880 4.2		[illegible] 5.1		[illegible] 6.1		520 7.1		445 8.3				
C-16	425	65	16"	2-1/2"					1790 5.3		[illegible] 6.5		[illegible] 7.8		1040 9.2		880 10.8		[illegible] 12.4		675 14.1
D-175	175	75	12"	1-7/8"		1130 3.0	1000 3.4	898 3.8	[illegible] 4.2		[illegible] 5.1		558 6.1		475 7.1						
D-300	300	70	14"	2"			1800 3.9	1430 4.4	1280 4.9		[illegible] 5.9		[illegible] 7.1		754 8.3		650 9.6	602 10.4			
D-500	500	65	16"	2-1/2"			2720 4.1	2350 4.8	2110 5.3		1710 6.5		1435 7.8		1225 9.3		1035 10.8	970 11.6			
D-700	808	75	16"	2-3/4"						2727 5.8	2463 6.4	2236 7.0	2044 7.7	1875 8.4	1726 9.1	1593 9.8	1478 10.6	1374 11.4	[illegible] 12.3	1187 13.1	
D-850	850	60	18"	3"							[illegible] 7.0	[illegible] 7.7	[illegible] 8.5	2240 9.2	2056 10.1	1895 10.9	1758 11.8	1629 12.7	[illegible] 13.6	1418 14.6	1328 15.6
G-35	421	90	14"	2-1/8"				1485 4.3		1227 5.3		1031 6.4		[illegible] 7.6		757 8.9		660 10.3		580 11.8	
G-45	545	90	16"	2-1/4"				1691 4.9		1397 6.0		1174 7.3		1001 8.6		863 10.1		751 11.7		660 13.4	
G-65	786	90	16"	2-1/2"				2460 4.8		2033 5.9		[illegible] 7.2		[illegible] 8.5		12.55 10.0		[illegible] 11.6		[illegible] 13.2	
G-85 J-85	1091	90	18"	2-3/4"				3068 5.2		2534 6.5		2129 7.9		1814 9.4		1564 11.1		1363 12.8		[illegible] 14.7	
H-25	302	90	12"	2"			1383 3.4		1132 4.2		[illegible] 5.1		790 6.0		885 7.1		593 8.3				

Table 4-38 (continued)

MANUFACTURER: CONTINENTAL EMSCO (DUPLEX)																				
MODEL	MAX. I.H.P.	MAX. S.P.M.	STROKE LENGTH	ROD SIZE	LINER SIZE (IN)															
					4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2	7-3/4	8	8-1/4
D-125	125	85	10"	1-3/4"	840 2.5	748 2.9	670 3.2	604 3.5	548 3.9	499 4.3	456 4.7	419 5.1	386 5.5	357 6.0	326 6.5	308 6.9				
D-175	175	75	12"	1-7/8"	1130 3.0	1000 3.4	898 3.8	807 4.2	731 4.6	686 5.1	608 5.6	558 6.1	514 6.6	475 7.1						
D-225	257	80	12"	1-7/8"	1551 3.0	1379 3.4	1234 3.8	1111 4.2	1007 4.6	916 5.1	838 5.6	769 6.1	708 6.6	654 7.1	607 7.7	565 8.3				
D-300	300	70	14"	2"		1600 3.9	1430 4.4	1280 4.9	1162 5.4	1060 5.9	965 6.5	886 7.1	815 7.7	754 8.3	698 8.9	650 9.6	602 10.3			
D-375	429	80	14"	2"		1991 3.9	1777 4.4	1600 4.9	1451 5.4	1318 5.9	1204 6.5	1104 7.1	1018 7.7	939 8.3	871 8.9	810 9.6	755 10.3			
DA-500	500	65	16"	2-1/2"		2720 4.2	2350 4.8	2110 5.3	1902 5.9	1710 6.5	1566 7.1	1435 7.8	1317 8.5	1225 9.2	1122 10.0	1035 10.8	970 11.6			
D-550 DB-550	635	75	16"	2-1/2"		2915 4.2	2590 4.8	2317 5.3	2090 5.9	1894 6.5	1727 7.1	1577 7.8	1449 8.5	1336 9.2	1235 10.0	1146 10.8	1067 11.6			
DA-700 DB-700	808	75	16"	2-3/4"					2727 5.8	2463 6.4	2236 7.0	2044 7.7	1875 8.4	1726 9.1	1593 9.8	1478 10.6	1374 11.4			
DA-850	850	60	18"	3"						2954 7.0	2680 7.7	2440 8.5	2240 9.2	2055 10.1	1895 10.9	1758 11.8	1629 12.7	1515 13.6	1418 14.6	1328 15.6
DB-850	992	70	18"	3"						2954 7.0	2680 7.7	2440 8.5	2240 9.2	2055 10.1	1895 10.9	1758 11.8	1629 12.7			
D-1000 DC-1000	1000	60	18"	3"						3480 7.0	3153 7.7	2871 8.5	2635 9.2	2418 10.1	2229 10.9	2068 11.8	1917 12.7	1782 13.6	1668 14.6	1562 15.6
DB-1000	1167	70	18"	3"						3480 7.0	3153 7.7	2871 8.5	2635 9.2	2418 10.1	2229 10.9	2068 11.8	1917 12.7			
D-1350 DC-1350	1575	70	18"	3-1/2"							4474 7.3	4058 8.1	3706 8.8	3392 9.7	3123 10.5	2880 11.4	2669 12.3			
D-1650 DC-1650	1925	70	18"	3-1/2"							5469 7.3	4960 8.1	4530 8.8	4146 9.7	3817 10.5	3520 11.4	3262 12.3			

MANUFACTURER: GARDNER-DENVER (DUPLEX)

MODEL	MAX. I.H.P	MAX. S.P.M.	STROKE LENGTH	ROD SIZE	LINER SIZE (IN) 5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2	7-3/4	8
FXK	255	70	14	2	1163 4.4	— —	[illegible] 5.4	— —	807 6.5	— —	686 7.7	636 8.3	583 9.0	[illegible] 9.6			
FXL	625	55	20	2-1/2	2569 6.0	— —	2115 7.4	— —	1777 8.9	— —	1514 10.6	1404 11.5	1306 12.5	1217 13.4	— —	1085 15.5	1000 16.6
FXO	122	70	10	1-3/4	772 3.2	— —	[illegible] 3.9	— —	536 4.7	— —	457 5.5	423 6.0	393 6.5	367 6.9			
FXQ	320	65	16	2	1319 5.0	— —	1232 6.2	— —	950 7.4	— —	810 8.8	751 9.5	696 10.2	651 11.0	— —	570 12 6	
FXZ	185	70	12	2	988 3.8	— —	817 4.6	— —	686 5.6	—	585 6.6	542 7.1	504 7.7	470 8.3			
GXH	1250	60	18						3942 7.7	— —	3281 9.3	3035 10.0	2793 10.9	2580 11.8	2408 12.7	2232 13.6	
GXN	500	70	14		2435 4.3	— —	1974 5.3	— —	1633 6.4	— —	1377 7.6	1271 8.2	1177 8.9	1084 9.5			
GXP	700	70	16		3060 4.8	— —	2470 5.9	— —	2040 7.1	— —	1712 8.5	1578 9.2	1460 10.0	1367 10.8	— —	1171 12 3	
GXQ	350	70	16		1470 5.0	— —	1185 6.0	— —	1000 7.3	— —	843 8.7	778 9.4	720 10.1	669 10.9			
GXR	1000	60	18						3113 7.8	2815 8.6	2578 9.4	2373 10.2	2194 11.1	2035 11.9	1903 12.8	1772 13.7	
KXF	700	70	16				2470 5.9	— —	2040 7.1	— —	1712 8.5	1578 9.2	1460 10.0	1367 10.8	— —	1171 12.3	
KXG	1000	60	18						3113 7.8	2815 8.6	2578 9.4	2373 10.2	2194 11.1	2035 11.9	1903 12.8	1772 13.7	
KXJ	1500	60	18						4845 7.5	— —	4025 9.0	3640 10.0	3350 10.9	3085 11.8			

Table 4-38
(continued)

MANUFACTURER: IDECO (DUPLEX)					LINER SIZE (IN)																
MODEL	MAX. I.H.P.	MAX. S.P.M.	STROKE LENGTH	ROD SIZE	3-3/4	4	4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2	7-3/4	8
MM-200 MM-200F	200	80	10	1-7/8	2000 1.7	1825 2.0	1460 2.5	1302 2.8	1163 3.1	1045 3.5	920 3.9	[illegible] 4.2	[illegible] 4.6	728 5.0	667 5.5	617 5.9					
MM-300 MM-300GB	300	80	12	2	2500 2.0	2380 2.3	1830 3.0	1632 3.4	1458 3.8	1309 4.2	[illegible] 4.6	1052 5.2	[illegible] 5.6	904 6.1	832 6.6	768 7.1	712 7.7	662 8.3			
MM-450	450	80	12	2-1/4			2830 2.9	2510 3.3	2225 3.7	2008 4.1	[illegible] 4.5	[illegible] 5.0	[illegible] 5.5	1374 6.0	1265 6.5	1165 7.0	1082 7.6	1000 8.2			
MM-550 MM-550F	550	65	15	2-1/2				3120 4.0	2775 4.5	2480 5.0	[illegible] 5.5	[illegible] 6.1	[illegible] 6.7	1690 7.3	1550 8.0	1425 8.7	1320 9.4	1220 10.1			
MM-600 MM-600C	600	65	16	2-1/2					2830 4.8	2540 5.3	[illegible] 5.9	[illegible] 6.5	[illegible] 7.2	1725 7.8	1582 8.5	1458 9.3	1348 10.0	1250 10.8	1165 11.5	1086 12.4	
MM-700 MM-700F	700	65	16	2-3/4						3038 5.2	2730 5.8	[illegible] 6.4	[illegible] 7.0	2048 7.7	1878 8.4	1725 9.1	1595 9.9	1487 10.6	1375 11.4	1285 12.3	
MM-900	900	65	16	3							[illegible] 5.2	3250 6.0	[illegible] 6.8	— —	2459 8.2	— —	2085 9.7	1933 10.4	1796 11.2	1670 12.1	1562 12.9
MM-1000 MM-1000GB	1000	65	16	3									[illegible] 6.8	2990 7.5	2735 8.2	2510 8.9	2325 9.7	2155 10.5	2000 11.3		
MM-1250 MM-1250GB	1250	65	18	3-1/8									[illegible] 7.6	3350 8.4	3065 9.2	2620 10.0	2600 10.8	2405 11.7	2230 12.6	2079 13.5	1940 14.5
MM-1450F MM-1450GB	1450	65	18	3-1/8									4270 7.6	3880 8.4	3560 9.2	3270 10.0	3010 10.8	2790 11.7	2590 12.6		
MM-1625 MM-1625F	1625	65	18	3-3/8									4920 7.4	4450 8.2	4060 8.9	3790 9.7	3430 10.6	3170 11.5	2940 12.4		
MM-1750F	1750	65	18	3-3/8									5000 7.4	4800 8.2	4380 8.9	4020 9.7	3700 10.6	3410 11.5	3175 12.4		

MANUFACTURER: NATIONAL SUPPLY (DUPLEX)

MODEL	MAX I H P	MAX S P M	STROKE LENGTH	ROD SIZE	LINER SIZE 3-1/2	3-3/4	4	4-1/4	4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2	7-3/4	8	8-1/4	8-1/2
C-100	150	75	10	1-3/4							915	825	745	680	625	570									
											3.2	3.5	3.9	4.3	4.7	5.1									
C-150 B	220	70	12	1-7/8							[illegible]	1085	965	895	820	[illegible]	690	[illegible]	[illegible]	[illegible]					
											3.8	4.2	4.7	5.1	5.6	6.1	6.6	7.2	7.7	8.3					
C-250	370	65	15	2-1/4							1810	1625	1465	1330	1215	1115	1025	945	875	810					
											4.6	5.1	5.7	6.2	6.8	7.5	8.1	8.8	9.5	10.2					
C-350	600	60	18	2-3/8							2665	2400	2170	1965	1790	1640	1510	1390	[illegible]	[illegible]	[illegible]	[illegible]			
											5.4	6.1	6.7	7.4	8.1	8.9	9.7	10.5	11.3	12.2	13.1	14.0			
D-50	80	75	10	1-1/2	1000	—	775	—	[illegible]	—	[illegible]														
					1.5	—	2.0	—	2.6	—	3.3														
E-500	590	70	14	2-5/8							[illegible]	2670	[illegible]	[illegible]	1970	1805	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]			
											4.1	4.6	5.1	5.6	6.2	6.8	7.4	8.0	8.7	9.4	10.1	10.8			
E-700	825	65	16	3-1/8										3000	—	[illegible]	—	[illegible]	—	[illegible]	—	[illegible]	—	—	1260
														6.1	—	7.4	—	8.9	—	10.4	—	12.0	—	—	14.7
G-700	700	70	14	2-5/8							[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
											4.1	4.6	5.1	5.6	6.2	6.8	7.4	8.0	8.7	9.4	10.1	10.8	11.5		
G-1000, G-1000B G-1000C	1000	65	16	3-1/8											[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
															6.8	7.4	8.1	8.9	9.6	10.4	11.2	12.0	12.9	13.7	14.7
H-850, H-850A	850	70	15	2-7/8									3320	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
													5.3	5.9	6.5	7.1	7.8	8.5	9.2	9.9	10.6	11.4	12.2		
H-1250	1250	65	16	3-1/8											4135	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]			
															6.8	7.4	8.1	8.9	9.6	10.4	11.2	12.0			
K-180	180	80	10	2					[illegible]	[illegible]	[illegible]	945	855	775	710	650	[illegible]	[illegible]	[illegible]	[illegible]					
									2.5	2.8	3.1	3.5	3.8	4.2	4.6	5.0	5.5	5.9	6.4	6.9					
K-280	280	75	12	2						[illegible]	[illegible]	[illegible]	1180	1075	980	900	830	[illegible]	[illegible]	[illegible]					
										3.4	3.8	4.2	4.6	5.1	5.6	6.1	6.6	7.1	7.7	8.3					
K-380	380	70	14	2-3/8						2100	1875	1675	1520	1370	[illegible]	1145	[illegible]	[illegible]	[illegible]	[illegible]					
										3.8	4.2	4.7	5.2	5.8	6.3	6.9	7.5	8.1	8.8	9.5					
K-500 K-500A	513	70	15	2-5/8						[illegible]	[illegible]	[illegible]	[illegible]	1765	[illegible]	1470	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]				
										3.9	4.4	4.9	5.5	6.0	6.6	7.3	7.9	8.6	9.3	10.0	10.8				
K-700 K-700A	700	65	16	2-7/8									[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
													5.7	6.3	6.9	7.6	8.3	9.0	9.8	10.5	11.3	12.2	13.0		
KSH-180	180	80	10	2			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	945	855	775	710										
							1.9	2.2	2.5	2.8	3.1	3.5	3.8	4.2	4.6										
KSH-280	280	75	12	2			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1300	1180	1075	980										
							2.3	2.6	3.0	3.4	3.8	4.2	4.6	5.1	5.6										
N-900	900	65	16	2-7/8											[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					
															6.9	7.6	8.3	9.0	9.8	10.5					
N-1000	1000	65	16	2-7/8									[illegible]	3560	[illegible]	[illegible]	2705	[illegible]	[illegible]	[illegible]					
													5.7	6.3	6.9	7.6	8.3	9.0	9.8	10.5					
N-1100	1100	65	16	3-1/8											3640	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					
															6.8	7.4	8.1	8.9	9.6	10.4					
N-1300	1300	65	16	3-1/8										4750	4300	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					
														6.1	6.8	7.4	8.1	8.8	9.6	10.4					
N-1600	1600	65	16	3-3/8											[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					
															6.6	7.3	8.0	8.7	9.4	10.2					

Table 4-38
(continued)

MANUFACTURER: OILWELL/WILSON SNYDER (DUPLEX)

MODEL	MAX. I.H.P.	MAX. S.P.M.	STROKE LENGTH	ROD SIZE	LINER SIZE (IN) 2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2	7-3/4	8
7P	282	55	18							1225	—	950	—	725	670	625	580			
										5.7	—	8.4	—	9.9	10.7	11.6	12.4			
12P	100	70	12					875	675	540										
								2.4	3.1	3.9										
12P-LD	100	70	12							540	[illegible]	[illegible]								
										3.9	4.7	5.7								
14P	238	60	14								1020	850	—	725	670	625	580			
											5.4	6.5	—	7.7	8.3	9.0	9.6			
14P-HD 214P	325	65	14								1325	1140	—	[illegible]	[illegible]	820	765			
											5.3	6.4	—	7.6	8.2	8.9	9.5			
16P	159	60	16							765	[illegible]	525								
										5.1	6.2	7.5								
18P	353	55	18							1700	—	1240	—	990	890	825	765			
										5.5	—	8.2	—	9.7	10.5	11.3	12.2			
20P	480	55	20							2120	—	1430	—	1200	1110	1020	950	—	825	775
										6.0	—	8.9	—	10.7	11.5	12.4	13.4	—	15.5	16.6
212P 712P	220	70	12	1-7/8						1200	—	800	—	690	640	600	550			
										3.8	—	5.6	—	6.6	7.2	7.7	8.3			
218P	450	60	18	2-1/4						1960	—	1330	—	1120	1039	960	890			
										5.5	—	8.2	—	9.7	10.5	11.3	12.2			
220P	600	60	20	2-1/2						2450	—	1625	—	1370	1260	1165	1080	—	940	880
										6.0	—	8.9	—	10.6	11.6	12.5	13.4	—	15.5	16.6
510P	116	85	10		1990	1500	1310	985	760	615										
					0.7	1.1	1.5	2.0	2.6	3.3										
610P	116	85	10								500	—	385							
											4.0	—	5.2							
612P	140	70	12							765	635	520								
										3.8	4.7	5.6								
612P-LD	140	70	12										480	—	—	—	350	—	305	
													6.1	—	—	—	8.3	—	9.5	
700P A700P	700	65	16	2-3/4							2725	2235	—	1875	1725	1593	1478	—	1260	
											5.8	7.0	—	8.4	9.1	9.9	10.6	—	12.2	
810P	116	85	10													250	—	250	—	250
																6.5	—	7.5	—	8.6
816P	700	65	16	2-3/4							2725	2235	—	1875	1725	1590	1478	—	1250	1197
											5.8	7.0	—	8.4	9.1	9.9	10.6	—	12.2	13.1
818P	1000	65	18	3-1/4								2980	—	2460	2270	2100	1940	—	1675	1560
												7.5	—	9.1	9.9	10.7	11.6	—	13.4	14.4
850P A850P	850	65	16	3-1/4							3500	2950	—	2370	2175	2006	1870	—	1600	
											5.4	6.7	—	8.0	8.8	9.5	10.2	—	11.9	
1400P	1400	65	18	3-1/2								4310	—	3560	—	3020				
												7.3	—	8.8	—	10.5				
1700P	1700	65	18	3-1/2								5000	—	4320	—	3640				
												7.3	—	8.8	—	10.5				
1000P A1000P	1000	65	18	3-1/4							3640	2990	—	2460	2275	2100	1940	—	1675	
											6.1	7.5	—	9.1	9.9	10.7	11.6	—	13.4	

Table 4-39
Mud Pump Performance—Triplex Pumps [17]
Pump Discharge Pressure (PSI) (Shaded Area); Pump Discharge Volume (Gal./Stroke) (Based on 100% Volumetric Efficiency)

MANUFACTURER: EXAMPLE (TRIPLEX)

MODEL	MAX. I.H.P.	MAX. S.P.M.	STROKE LENGTH	LINER SIZE (IN) 5	5-1/2	6	6-1/2	7
X	1250	120	12	5250 3.1	4342 3.7	3645 4.4	3135 5.2	2680 6.0

MANUFACTURER: ALFRED WIRTH (TRIPLEX)

MODEL	MAX I.H.P.	MAX. S.P.M.	STROKE LENGTH	LINER SIZE (IN) 4	4-1/4	4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7
TPK 6-1/4" x 9-1/4"/1000—1250*	1000	160	9-1/4	— —	— —	5049 1.9	4530 2.1	4089 2.4	3708 2.6	3379 2.9	3091 3.1	2839 3.4	2617 3.7	— —	— —	— —
TPK 7" x 10"/1300—1625*	1300	150	10	— —	— —	— —	— —	5243 2.6	4756 2.8	4334 3.1	3970 3.4	3643 3.7	3359 4.0	3103 4.3	2877 4.7	2678 5.0
TPK 7" x 12"/1600—2000*	1600	120	12	— —	— —	— —	— —	— —	— —	5559 3.7	5079 4.1	4664 4.4	4303 4.8	3978 5.2	3689 5.6	3428 6.0

*INTERMITTENT

MANUFACTURER: CONTINENTAL — EMSCO (TRIPLEX)

MODEL	MAX. I.H.P.	MAX. S.P.M.	STROKE LENGTH	LINER SIZE (IN) 2-1/2	2-5/6	2-3/4	3	3-1/4	3-1/2	3-3/4	4	4-1/4	4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2
F-350 F-350-HP	350	175	7	[illegible] .4	[illegible] .5	[illegible] .5	[illegible] .6	4100 .8	3525 .9	3080 1.0	2705 1.1	2390 1.3	2135 1.5	— —	1730 1.8	1570 2.0	1428 2.2	1309 2.4	1200 2.6						
F-500	500	170	7-1/2						4651 9	— —	[illegible] 1.2	[illegible] 1.4	[illegible] 1.6	[illegible] 1.7	2373 1.9	2154 2.1	[illegible] 2.3	1794 2.5	1648 2.8						
F-650	650	160	8									[illegible] 1.5	[illegible] 1.7	[illegible] 1.8	3070 2.0	2770 2.3	[illegible] 2.5	[illegible] 2.7	2128 2.9						
F-750	750	200	8						5000 1.0	[illegible] 1.1	[illegible] 1.3	[illegible] 1.5	3470 1.7	[illegible] 1.8	[illegible] 2.0	2550 2.3	2320 2.5	2125 2.7	1955 2.9						
F-800	800	150	9								[illegible] 1.5	— —	[illegible] 1.9	[illegible] 2.1	3590 2.3	3260 2.5	2965 2.8	2715 3.0	2490 3.3	2295 3.6					
F-1000	1000	140	10										[illegible] 2.1	[illegible] 2.3	[illegible] 2.6	[illegible] 2.8	3575 3.1	3270 3.4	3010 3.7	2770 4.0					
F-1300	1300	120	12														[illegible] 3.7	[illegible] 4.1	3781 4.4	3494 4.8	3260 5.2	[illegible] 5.6	[illegible] 6.0		
FA-1300	1300	120	12												[illegible] 3.1	— —	[illegible] 3.7	[illegible] 4.1	3791 4.4	3494 4.8	3234 5.2	[illegible] 5.6	[illegible] 6.0	[illegible] 6.4	2428 6.9
F-1600	1600	120	12														[illegible] 3.7	5078 4.1	4665 4.4	4299 4.8	4012 5.2	[illegible] 5.6	[illegible] 6.0		
FA-1600	1600	120	12												5500 3.1	— —	[illegible] 3.7	5078 4.1	4665 4.4	4299 4.8	3981 5.2	[illegible] 5.6	[illegible] 6.0	[illegible] 6.4	[illegible] 6.9

Table 4-39
(continued)

MANUFACTURER: GARDNER-DENVER (TRIPLEX)

MODEL	MAX. I.H.P.	MAX S.P.M	STROKE LENGTH	LINER SIZE (IN) 3	3-1/4	3-1/2	3-3/4	4	4-1/4	4-1/2	4-34	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7
PJ-8	275	175	8	3118	[illegible]	[illegible]	—	[illegible]	—	[illegible]	—	1122								
				0.7	0.9	1.0	—	1.3	—	1.7	—	2.0								
PO-7	700	130	7			4483	—	3434	—	2712	—	2198	—	1815	—	[illegible]				
						1.8	—	2.3	—	2.9	—	3.6	—	4.3	—	5.1				
PV-9	1000	125	9			[illegible]	—	[illegible]	—	[illegible]	—	2540	—	2050	—	[illegible]				
						2.2	—	2.9	—	3.7	—	4.6	—	5.6	—	6.6				
PY-7	500	160	7							3150	—	2500	—	2110	—	[illegible]	—	[illegible]	—	1330
										1.5	—	1.8	—	2.2	—	2.6	—	3.0	—	3.5
PZ-7	550	165	7							[illegible]	—	[illegible]	—	[illegible]	—	2000	—	[illegible]	—	1470
										1.5	—	1.8	—	2.2	—	2.6	—	3.0	—	3.5
PZ-8	750	165	8					[illegible]	—	[illegible]	—	3433	—	[illegible]	—	2385	[illegible]			
								1.3	—	1.7	—	2.0	—	2.5	—	2.9	3.2			
PZ-9	1000	150	9							5630	—	4485	—	3710	—	[illegible]	[illegible]	[illegible]	—	2285
										1.9	—	2.3	—	2.8	—	3.3	3.6	3.9	—	4.5
PZ-11	1600	130	11											[illegible]	—	[illegible]	—	[illegible]	—	3454
														3.4	—	4.0	—	4.7	—	5.5

MANUFACTURER: IDECO

MODEL	MAX I.H.P.	MAX. S.P.M	STROKE LENGTH	LINER SIZE (IN) 4	4-1/4	4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/2
IDECO 500	500	170	8	3588	3175	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1734	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
				1.3	1.5	1.7	1.8	2.0	2.3	2.5	2.7	2.9	3.2	3.4	3.7	4.0	
IDECO 800	800	150	10			[illegible]	—	[illegible]	—	[illegible]	—	2240	—	[illegible]	—	[illegible]	
						2.1	—	—	—	3.3	—	3.7	—	4.3	—	5.0	
IDECO 1000	1000	150	10			[illegible]	—	4322	—	[illegible]	—	3002	—	[illegible]	—	[illegible]	
						2.1	—	2.6	—	3.1	—	3.7	—	4.3	—	5.0	
IDECO 1300	1405	130	12					[illegible]	—	4514	—	[illegible]	—	[illegible]	—	[illegible]	[illegible]
								3.1	—	3.7	—	4.4	—	5.2	—	6.0	6.9
IDECO 1600	1732	130	12							5556	—	[illegible]	—	[illegible]	—	[illegible]	[illegible]
										3.7	—	4.4	—	5.2	—	6.0	6.9

MANUFACTURER: NATIONAL SUPPLY (TRIPLEX)

MODEL	MAX I.H.P.	MAX S.P.M	STROKE LENGTH	(LINER SIZE (IN) 3-1/2	3-3/4	4	4-1/4	4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4
7-P-50	500	165	7-3/4	[illegible]	—	[illegible]	—	[illegible]	—	2365	—	[illegible]	—	[illegible]	[illegible]				
				1.0	—	1.3	—	1.6	—	2.0	—	2.4	—	2.8	3.1				
8-P-80	800	160	8-1/2					[illegible]	—	3580	—	2940	—	[illegible]	[illegible]				
								1.8	—	2.2	—	2.6	—	3.1	3.4				
9-P-100	1000	150	9-1/4					[illegible]	—	[illegible]	—	[illegible]	—	[illegible]	[illegible]				
								1.9	—	2.4	—	2.9	—	3.4	3.7				
10-P-130	1300	140	10							[illegible]	—	[illegible]	—	[illegible]	[illegible]				
										2.5	—	3.1	—	3.7	4.0				
12-P-160	1600	120	12									[illegible]	—	[illegible]	—	[illegible]	[illegible]	[illegible]	[illegible]
												3.7	—	4.4	—	5.2	5.6	6.0	6.4

MANUFACTURER: OILWELL/WILSON SNYDER (TRIPLEX)																			
MODEL	MAX. I.H.P.	MAX. S.P.M.	STROKE LENGTH	LINER SIZE (IN)															
				4	4-1/4	4-1/2	4-3/4	5	5-1/4	5-1/2	5-3/4	6	6-1/4	6-1/2	6-3/4	7	7-1/4	7-1/2	7-3/4
350-PT	350	175	8	[illegible]	—	[illegible]	—	1500	—	1250	1144	1050	—	900	—	770			
				1.3	—	1.7	—	2.0	—	2.5	2.7	2.9	—	3.5	—	4.0			
850-PT	850	160	9	[illegible]	—	4400	—	3560	—	[illegible]	—	2470	—	2110					
				1.5	—	1.9	—	2.3	—	2.8	—	3.3	—	3.9					
1100-PT	1100	150	10			5000	—	4500	—	3700	—	3110	—	2650					
						2.1	—	2.5	—	3.1	—	3.7	—	4.3					
1400-PT A1400-PT	1400	150	10					5000	—	4723	4321	3968	—	3381	3135	2915	2718	2540	2378
								2.5	—	3.1	3.4	3.7	—	4.3	4.6	5.0	5.4	5.7	6.1
1700-PT A1700-PT	1700	150	12					5000	—	4723	4321	3968	—	3381	3135	2915	2718	2540	2378
								3.1	—	3.7	4.0	4.4	—	5.2	5.6	6.0	6.4	6.9	7.4
A560-PT	560	175	8	[illegible]	—	[illegible]	—	2420	—	[illegible]	[illegible]	1580	—	1430	—	1240			
				1.3	—	1.7	—	2.0	—	2.5	2.7	2.9	—	3.5	—	4.0			

Courtesy Hughes Christensen.

Table 4-40
Pump Output vs. Annular Velocity [18]

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
		Annular Velocity, Ft/min														
Hole Size, In.	Drill-pipe Size, In.	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150
4¾	2⅜	7	14	21	28	35	41	48	55	62	69	76	83	90	97	104
	2⅞	6	12	17	23	29	35	41	47	52	58	64	70	76	82	87
5⅝	2⅞	10	19	29	38	48	57	67	76	86	95	105	114	124	134	143
	3½	8	16	24	32	40	47	55	63	71	79	87	95	103	111	119
5¾	2⅞	11	21	32	43	54	64	75	86	96	107	118	129	139	150	161
	3½	9	18	27	36	45	55	64	73	82	91	100	109	118	127	136
6	3½	10	19	29	39	48	58	68	78	87	97	107	116	126	136	145
6⅛	3½	10	21	31	41	52	62	72	83	93	103	113	124	134	144	155
6¼	3½	11	22	33	44	55	66	77	88	98	109	120	131	142	153	164
6¾	3½	14	27	41	54	68	82	95	109	122	136	150	163	177	190	204
7⅝	3½	19	37	56	75	94	112	131	150	168	187	206	225	243	262	281
	4½	15	31	46	62	77	93	108	124	139	155	170	186	201	216	232
7⅞	3½	20	41	61	81	102	122	142	162	183	203	223	244	264	284	305
	4½	17	34	51	68	85	102	119	136	153	170	187	204	222	238	254
8⅜	3½	24	47	71	94	118	142	165	189	212	236	260	283	307	331	354
	4½	20	41	61	81	102	122	143	163	183	204	224	244	265	285	305
8½	3½	24	49	73	98	122	147	171	196	220	245	269	294	318	343	367
	4½	21	42	64	85	106	127	149	170	191	212	233	255	276	297	318
8⅝	3½	25	51	76	101	127	152	177	203	228	254	279	304	330	355	380
	4½	22	44	65	88	110	133	155	177	199	221	243	265	287	309	331
	5	20	40	60	81	101	121	141	161	181	202	222	242	262	282	302
8¾	3½	26	52	79	105	131	157	184	210	236	262	289	315	341	367	394
	4½	23	46	69	92	114	138	161	184	207	230	253	276	299	322	345
	5	21	42	63	84	105	126	147	168	189	210	231	252	273	295	316
9	4½	25	50	74	99	124	149	174	198	223	248	273	297	322	347	372
	5	23	46	69	91	114	137	160	183	206	229	251	274	297	320	343
9⅞	4½	32	63	95	126	158	189	221	252	284	315	347	378	410	441	473
	5	30	59	89	118	148	178	207	237	266	296	326	355	385	414	444
	5½	27	55	82	110	137	165	192	220	247	275	302	329	357	384	412
10⅝	4½	38	76	113	151	189	227	265	302	340	378	416	454	491	529	567
	5	36	72	108	143	179	215	251	287	323	359	395	430	466	502	538
	5½	34	67	101	135	169	202	236	270	303	337	371	405	438	472	506
11	4½	41	82	123	164	206	247	288	329	370	411	452	493	534	576	617
	5	39	78	118	157	196	235	274	313	353	392	431	470	509	549	588
	5½	37	74	111	148	185	222	259	296	333	370	407	444	481	518	555
12¼	4½	53	106	159	212	265	318	371	424	477	530	583	636	689	741	794
	5	51	102	153	204	255	306	357	408	459	510	561	612	663	714	765
	5½	49	98	147	196	245	293	342	391	440	489	538	587	636	684	733
	6⅝	43	87	130	173	217	260	303	347	390	433	477	520	563	606	650
13¾	4½	69	138	207	276	344	413	482	551	620	689	758	827	895	964	1,033
	5	67	134	201	268	335	402	469	536	603	670	736	803	870	937	1,004
	5½	65	130	194	259	324	389	454	518	583	648	713	778	842	907	972
	6⅝	59	118	178	237	296	355	415	474	533	592	652	711	770	829	888
15	4½	84	167	251	334	418	501	585	668	752	835	919	1,002	1,086	1,170	1,253
	5½	79	159	238	318	397	477	556	636	715	795	874	954	1,033	1,112	1,192
	6⅝	74	148	222	296	369	443	517	591	665	739	813	887	961	1,035	1,108
17½	4½	117	233	350	467	584	700	817	934	1,051	1,167	1,284	1,400	1,517	1,634	1,750
	5½	113	225	338	451	563	676	789	901	1,014	1,127	1,239	1,352	1,464	1,577	1,690
	6⅝	107	214	321	428	535	642	749	857	964	1,071	1,178	1,285	1,392	1,499	1,606
26	4½	268	535	803	1,071	1,338	1,605	1,873	2,141	2,408	2,676	2,944	3,211	3,479	3,746	4,014
	5½	264	527	791	1,054	1,318	1,581	1,845	2,108	2,372	2,636	2,899	3,163	3,427	3,690	3,954
	6⅝	258	516	774	1,032	1,290	1,548	1,806	2,064	2,321	2,579	2,837	3,095	3,353	3,611	3,869

Table 4-40 (continued)

1	2	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
		Annular Velocity, Ft/min														
Hole Size, In.	Drill-pipe Size, In.	160	170	180	190	200	210	220	230	240	250	260	270	280	290	300
4¾	2⅜	110	117	124	101	138	145	152	159	166	173	180	186	193	200	207
	2⅞	92	99	105	111	117	122	128	134	140	146	152	157	163	169	175
5⅝	2⅞	152	162	172	181	191	200	210	219	229	238	248	257	267	277	286
	3½	127	134	142	150	158	166	174	182	190	198	206	214	222	229	237
5⅞	2⅞	271	132	193	203	214	225	236	246	257	268	278	289	300	311	321
	3½	145	154	164	173	182	191	200	209	218	227	236	246	254	263	273
6	3½	155	165	174	184	194	203	213	222	232	242	252	262	271	281	291
6⅛	3½	165	175	186	196	206	216	227	237	247	258	268	278	289	299	309
6¼	3½	176	186	197	208	219	230	241	252	263	273	284	295	306	317	328
6¾	3½	217	231	245	258	272	285	299	313	326	340	353	367	381	394	408
7⅝	3½	300	318	337	356	374	393	412	431	449	468	487	506	524	543	562
	4½	247	263	278	294	309	325	340	356	371	386	402	417	433	448	464
7⅞	3½	326	345	365	386	406	426	447	467	487	508	528	548	569	589	609
	4½	275	290	307	324	341	358	375	392	409	426	443	460	477	494	511
8⅜	3½	378	402	425	449	472	496	520	543	567	590	614	638	661	685	709
	4½	326	346	366	387	407	427	448	468	489	509	529	550	570	590	611
8½	3½	392	416	441	465	490	514	539	563	588	612	636	661	685	710	734
	4½	339	361	382	403	424	446	467	488	509	530	552	573	594	616	636
8⅝	3½	406	431	456	482	507	532	558	583	608	634	659	685	710	735	761
	4½	353	374	396	420	442	464	484	508	530	552	574	596	618	641	663
	5	322	343	363	383	403	423	443	463	484	504	524	544	564	584	605
8¾	3½	420	445	472	499	525	551	577	604	630	656	682	708	735	761	787
	4½	368	391	414	437	460	482	505	528	551	574	597	620	643	666	689
	5	337	358	379	400	421	442	463	484	505	526	547	568	589	610	631
9	4½	397	421	446	471	496	521	545	570	595	620	644	669	694	719	744
	5	364	386	411	434	457	480	503	526	548	571	594	617	640	663	685
9⅞	4½	504	536	567	599	630	662	694	725	757	788	820	851	883	914	946
	5	472	502	533	562	592	621	651	680	710	740	769	799	828	858	888
	5½	439	467	494	521	549	576	604	631	659	686	714	741	768	796	823
10⅝	4½	605	643	680	718	756	794	832	869	907	945	983	1,021	1,058	1,096	1,134
	5	574	616	646	681	717	753	789	825	861	899	933	968	1,004	1,040	1,076
	5½	539	573	607	641	674	708	742	775	809	843	877	910	944	978	1,012
11	4½	662	699	740	781	822	863	904	945	987	1,028	1,069	1,110	1,151	1,192	1,232
	5	627	665	705	744	783	823	862	901	940	979	1,018	1,058	1,097	1,136	1,175
	5½	592	629	666	703	741	778	815	852	889	926	963	1,000	1,037	1,074	1,111
12¼	4½	847	900	953	1,006	1,059	1,112	1,165	1,218	1,271	1,324	1,377	1,430	1,483	1,536	1,589
	5	816	867	918	969	1,021	1,072	1,123	1,174	1,225	1,276	1,327	1,378	1,429	1,480	1,531
	5½	782	831	880	929	972	1,027	1,075	1,124	1,172	1,222	1,271	1,320	1,369	1,418	1,466
	6⅝	693	736	780	823	866	910	953	996	1,040	1,083	1,126	1,170	1,212	1,256	1,300
12¾	4½	1,103	1,171	1,240	1,309	1,377	1,446	1,515	1,584	1,652	1,722	1,791	1,860	1,928	1,997	2,066
	5	1,071	1,138	1,205	1,272	1,339	1,406	1,472	1,540	1,606	1,673	1,740	1,807	1,874	1,941	2,008
	5½	1,037	1,102	1,166	1,231	1,296	1,361	1,425	1,490	1,555	1,620	1,685	1,749	1,814	1,879	1,944
	6⅝	943	1,007	1,066	1,125	1,186	1,244	1,303	1,362	1,422	1,481	1,540	1,600	1,658	1,718	1,777
15	4½	1,337	1,420	1,504	1,587	1,671	1,754	1,838	1,921	2,006	2,088	2,172	2,256	2,339	2,422	2,506
	5½	1,272	1,351	1,430	1,510	1,589	1,669	1,748	1,828	1,907	1,986	2,066	2,145	2,225	2,304	2,384
	6⅝	1,182	1,256	1,330	1,404	1,478	1,552	1,626	1,700	1,773	1,847	1,921	1,996	2,069	2,143	2,217
17½	4½	1,867	1,984	2,100	2,217	2,334	2,450	2,567	2,684	2,800	2,917	3,034	3,151	3,267	3,384	3,501
	5½	1,803	1,914	2,027	2,140	2,252	2,365	2,477	2,590	2,702	2,815	2,928	3,040	3,153	3,266	3,378
	6⅝	1,715	1,820	1,927	2,034	2,141	2,248	2,355	2,462	2,569	2,676	2,783	2,890	2,997	3,104	3,211
20	4½	4,281	4,548	4,816	5,083	5,351	5,618	5,886	6,154	6,421	6,689	6,956	7,224	7,491	7,759	8,026
	5½	4,215	4,479	4,742	5,006	5,269	5,533	5,796	6,060	6,323	6,587	6,850	7,114	7,377	7,640	7,904
	6⅝	4,126	4,384	4,642	4,900	5,158	5,416	5,674	5,932	6,190	6,447	6,705	6,963	7,221	7,479	7,737

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Table 4-41
Approximate Input HP Required for Duplex Mud Pump Operation [15]

PWP, psi	200			300			400			500			600			700			800			900			1000		
SPM	40	50	60	40	50	60	40	50	60	40	50	60	40	50	60	40	50	60	40	50	60	40	50	60	40	50	60
Pump Size:																											
3 x 6	4.0	5.0	6.0	6.0	7.5	9.0	8.0	10.0	12.0	10.0	12.5	15.0	12.0	15.0	18.0	14.0	17.5	21.0	16.0	20.0	24.0	18.0	22.5	27.0	20.0	25.0	30.0
3½ x 6	5.47	6.84	8.2	8.2	10.3	12.3	10.9	13.7	16.4	13.7	17.1	20.5	16.4	20.5	24.6	19.1	23.9	28.7	21.9	27.4	32.8	24.6	30.8	36.9	27.4	34.2	41.0
4 x 6	7.17	8.97	10.7	10.7	13.5	16.1	14.4	17.9	21.5	17.9	22.4	26.9	21.5	26.9	32.3	25.1	31.4	37.7	28.7	35.9	43.0	32.3	40.4	48.4	35.9	44.9	53.8
4½ x 10	15	19	23	23	29	34	31	39	46	39	48	57	46	57	68	53	66	80	61	76	91	68	86	102	76	95	114
4¾ x 10	17	21	25	25	32	38	34	42	50	42	53	63	50	63	76	59	74	89	68	85	101	76	95	114	85	106	127
5 x 10	19	24	28	28	35	42	38	47	56	47	59	70	56	70	84	66	82	98	75	94	112	84	105	126	94	117	140
5¼ x 10	21	26	31	31	39	46	41	52	62	52	65	77	62	77	93	72	90	108	82	104	124	93	116	139	104	130	155
5½ x 10	23	28	34	34	43	51	46	57	68	57	71	85	68	85	102	79	99	118	91	113	136	102	127	152	113	141	170
5¾ x 10	25	31	37	37	47	56	50	62	74	62	77	93	74	93	111	87	108	130	99	123	148	111	139	167	123	154	185
6 x 10	27	34	41	41	51	61	54	68	81	68	84	101	81	101	121	94	118	141	108	135	161	121	151	181	135	168	202
6¼ x 10	29	37	44	44	55	66	59	73	88	73	91	110	88	110	131	102	128	153	117	146	175	131	164	197	146	182	219
6½ x 10	32	40	48	48	59	71	63	79	95	79	99	119	95	119	141	111	138	166	126	158	190	141	178	213	158	197	237
6¾ x 10	34	43	51	51	64	77	68	85	102	85	107	128	103	128	153	119	149	179	136	170	204	153	192	230	170	213	255
7 x 10	37	46	55	55	69	83	74	92	110	92	115	137	110	137	165	128	160	192	147	183	220	165	206	247	183	229	275
7¼ x 10	40	49	59	59	74	89	79	99	118	99	123	147	118	147	177	138	172	206	157	196	236	177	221	265	196	246	295
4½ x 12	18	23	27	27	34	41	36	46	55	46	57	68	55	68	82	64	80	96	73	91	109	82	102	123	91	114	136
4¾ x 12	20	25	31	31	38	46	41	51	61	51	63	76	61	76	91	71	89	106	81	101	121	91	114	136	101	126	152
5 x 12	22	28	34	34	43	51	45	56	67	56	70	84	67	84	101	79	99	118	90	112	135	101	127	152	112	140	168
5¼ x 12	25	31	37	37	47	56	50	62	74	62	78	93	74	93	111	87	109	130	100	124	148	111	139	167	124	155	186
5½ x 12	27	34	41	41	51	61	54	68	82	68	85	102	82	102	122	95	119	143	109	136	163	122	153	184	136	170	204
5¾ x 12	30	37	45	45	56	67	60	74	89	74	93	111	89	111	133	104	130	156	119	148	178	133	167	200	148	185	222
6 x 12	32	40	49	49	61	73	65	81	97	81	101	121	97	121	145	113	141	170	129	161	194	145	182	218	161	202	242
6¼ x 12	35	44	53	53	66	79	70	88	105	88	110	131	105	131	158	123	153	184	140	175	210	158	197	237	175	219	263
6½ x 12	38	47	57	57	71	85	76	95	114	95	118	142	114	142	171	133	166	199	151	190	227	171	213	256	190	237	284
6¾ x 12	41	51	61	61	77	92	82	102	123	102	128	153	123	153	184	143	179	215	164	204	245	184	230	276	204	256	307
7 x 12	44	55	66	66	83	99	88	110	132	110	137	165	132	165	198	154	192	231	176	220	264	198	247	297	220	275	330
7¼ x 12	47	59	71	71	88	106	94	118	141	118	147	177	141	177	212	165	206	248	188	235	283	212	265	318	236	294	344
5 x 14	26	33	39	39	49	59	53	66	79	66	82	98	79	98	118	92	114	137	105	131	157	118	147	177	131	164	196
5¼ x 14	29	36	43	43	54	65	58	72	87	72	90	108	87	108	130	101	126	151	115	144	173	130	162	195	144	180	216
5½ x 14	32	40	48	48	60	71	64	79	95	79	99	119	95	119	143	111	139	166	127	158	190	143	178	214	158	198	238
5¾ x 14	35	43	52	52	65	78	69	87	104	87	108	130	104	130	156	121	151	181	139	173	208	156	194	234	173	216	260
6 x 14	38	47	57	57	70	85	76	94	113	94	118	142	113	142	170	132	165	198	151	189	226	170	212	254	189	236	283
6¼ x 14	41	51	61	61	77	92	82	102	123	102	128	154	123	154	184	143	179	215	164	205	246	184	230	276	205	256	307
6½ x 14	44	55	67	67	83	100	89	111	133	111	139	166	133	166	199	155	194	232	177	221	266	199	249	298	221	277	332
6¾ x 14	48	60	72	72	90	107	95	119	143	119	149	179	143	179	216	167	209	251	191	239	286	214	269	322	239	298	358
7 x 14	51	64	77	77	96	115	103	128	154	128	161	193	154	193	231	180	225	270	205	257	308	231	289	346	257	321	385
7¼ x 14	55	68	83	83	103	124	110	138	165	138	172	207	165	207	248	193	241	289	220	275	330	248	310	371	275	344	413
5 x 15	28	35	42	42	53	63	57	70	84	70	88	105	84	105	126	98	123	147	114	140	168	126	158	189	140	176	210
5¼ x 15	31	39	47	47	58	70	62	77	93	77	97	116	93	116	140	108	135	162	124	155	186	140	174	209	155	193	232
5½ x 15	34	43	51	51	64	77	68	85	102	85	106	128	102	128	153	119	149	178	136	170	204	153	191	229	170	212	254
5¾ x 15	37	46	56	56	70	84	74	93	111	93	116	139	111	139	167	130	162	195	148	185	222	167	208	250	185	231	278
6 x 15	40	51	61	61	76	91	81	101	121	101	126	152	121	152	182	141	177	212	162	202	242	182	227	273	202	252	303
6¼ x 15	44	55	66	66	83	99	88	110	132	110	137	165	132	165	198	154	192	230	176	219	263	198	247	296	219	274	329
6½ x 15	48	59	71	71	89	107	95	119	142	119	149	178	142	178	213	166	208	249	190	237	284	213	267	320	237	296	356
6¾ x 15	51	64	77	77	96	115	102	128	153	128	160	192	153	192	230	179	224	268	204	256	306	230	288	345	256	320	384
7 x 15	55	69	83	83	103	124	110	137	165	137	172	206	165	206	249	192	240	288	220	275	330	249	310	372	275	344	412
7¼ x 15	59	74	88	88	111	133	118	147	177	147	184	221	177	221	265	206	258	309	236	295	353	265	332	398	295	369	442
5 x 16	30	38	45	45	56	67	60	75	90	75	93	112	90	112	135	105	131	157	120	150	179	135	168	202	150	187	224
5¼ x 16	33	41	50	50	62	74	66	82	99	82	103	124	99	124	148	115	144	173	132	165	198	148	185	223	165	206	247
5½ x 16	36	45	55	55	68	82	72	90	109	90	113	136	109	136	163	127	158	190	145	181	217	163	204	244	181	226	271
5¾ x 16	40	50	60	60	74	89	79	99	119	99	124	148	119	148	178	138	173	208	158	198	237	178	222	267	198	247	297
6 x 16	43	54	65	65	81	97	86	108	129	108	135	161	129	161	194	151	188	226	172	215	258	194	242	291	215	269	323
6¼ x 16	47	59	70	70	88	105	93	117	140	117	146	175	140	175	210	164	204	245	187	234	280	210	263	315	234	292	350
6½ x 16	51	63	76	76	95	114	101	126	152	126	158	190	152	190	227	177	221	265	202	253	303	227	284	341	253	316	379
6¾ x 16	55	68	82	82	102	123	109	136	164	136	170	204	164	204	245	191	238	286	218	273	327	245	307	368	273	341	409
7 x 16	59	74	88	88	110	132	117	147	176	147	183	220	176	220	264	205	256	307	234	293	352	264	330	396	293	366	440
7¼ x 16	63	79	95	95	118	141	126	157	189	157	196	236	189	236	283	220	275	330	251	314	377	283	354	424	314	393	472
7½ x 16	68	84	101	101	125	151	135	168	202	168	210	252	202	252	303	236	294	353	269	337	404	303	379	454	336	491	505
7¾ x 16	72	90	108	108	135	162	144	180	216	180	225	269	216	269	323	251	314	377	287	359	431	323	404	485	359	449	539
5 x 18	34	42	51	51	63	76	67	84	101	84	105	126	101	126	152	118	147	177	135	168	202	152	189	227	168	210	253
5¼ x 18	37	47	56	56	70	83	74	93	111	93	116	139	111	139	167	130	162	195	148	185	222	167	208	250	185	232	278
5½ x 18	41	51	61	61	77	91	82	102	122	102	127	153	122	153	183	143	178	214	163	204	244	183	229	276	204	255	306
5¾ x 18	44	56	67	67	84	100	89	111	133	111	139	167	133	167	200	156	195	233	178	222	267	200	250	300	222	278	334
6 x 18	49	61	73	73	91	109	97	121	145	121	151	182	145	182	218	170	212	254	194	242	291	218	273	327	242	303	364
6¼ x 18	53	66	79	79	99	117	105	131	157	131	164	197	157	197	236	184	230	276	210	262	315	236	296	354	262	328	394
6½ x 18	57	71	86	86	107	128	114	142	171	142	178	213	171	213	256	199	249	299	228	285	341	256	320	384	285	356	427
6¾ x 18	62	77	93	92	115	138	123	154	184	154	192	230	184	230	276	215	269	322	246	307	368	276	345	414	307	383	460
7 x 18	66	83	99	99	124	149	132	165	198	165	206	248	198	248	297	231	289	346	264	330	396	297	371	445	330	412	495
7¼ x 18	71	89	106	106	133	159	142	177	212	177	221	266	212	266	319	248	310	372	283	354	425	319	398	478	354	442	531
7½ x 18	76	95	114	114	142	171	152	190	227	190	237	284	227	284	341	265	331	398	303	379	454	341	426	511	378	473	568
7¾ x 18	81	101	121	121	152	182	162	202	243	202	253	303	243	303	364	283	354	425	324	404	485	364	455	546	404	506	606
5 x 20	37	47	56	56	70	84	75	93	112	93	117	140	112	140	168	131	164	196	150	187	224	168	210	252	187	234	280
5¼ x 20	41	52	62	62	77	93	82	103	124	103	129	155	124	155	185	144	180	216	165	206	247	185	232	278	206	258	309
5½ x 20	45	57	68	68	85	102	90	113	136	113	141	170	136	170	204	158	198	237	181	226	271	204	254	305	226	283	339
5¾ x 20	49	62	74	74	93	111	99	124	148	124	154	185	148	185	222	173	216	260	198	247	297	222	278	334	247	309	371
6 x 20	54	67	81	81	101	121	108	135	161	135	168	202	161	202	242	188	236	283	215	269	323	242	303	363	269	336	404
6¼ x 20	58	73	88	88	110	131	117	146	175	146	183	219	175	219	263	204	256	307	234	292	350	263	329	394	292	365	438
6½ x 20	63	79	95	95	118	142	126	158	190	158	197	237	190	237	284	221	276	332	253	316	379	284	355	426	316	395	474
6¾ x 20	68	85	102	102	128	153	136	170	204	170	213	255	204	255	307	238	298	358	273	341	409	307	383	460	341	426	511
7 x 20	73	92	110	110	137	165	147	183	220	183	229	275	220	275	330	256	321	385	293	366	440	330	412	495	366	458	550
7¼ x 20	79	98	118	118	147	177	157	196	236	196	246	295	236	295	354	275	344	413	314	393	472	354	442	531	393	491	589
7½ x 20	84	105	126	126	158	189	168	210	252	210	263	315	252	315	379	294	368	442	336	421	505	379	473	568	421	526	631
7¾ x 20	90	112	135	135	168	202	180	225	269	225	281	337	269	337	404	314	393	472	359	449	539	404	505	606	449	561	674

Note: For single action triplex pump, multiply table value by 0.75.
Note: Values in the table are calculated for a pump efficiency of 0.85.
Courtesy International Association of Drilling Contractors.

(text continued from page 631)

The required pump working pressure PWP (psi) can be calculated as

$$PWP = \Delta P_s + \Delta P_d + \Delta P_a + \Delta P_b \tag{4-35}$$

where ΔP_s = pressure loss through surface equipment in psi
ΔP_d = pressure loss through the inside of the drill string in psi

ΔP_a = pressure loss in annulus in psi
ΔP_b = pressure drop through bit nozzles in psi

Table 4-42 shows the jet velocity. Table 4-43 shows the diameters and areas of various nozzle sizes.

The required pump hydraulic horsepower (PHHP) can be calculated as

$$PHHP = HHP_{circ} + HHP_{bit} \tag{4-37}$$

where HHP_{circ} = total HHP loss due to pressure losses in the circulating system
HHP_{bit} = hydraulic horsepower required at the bit

The general hydraulic horsepower is

$$HHP = \frac{Q\ \Delta P}{1714} \tag{4-38}$$

where Q = flow rate in gal/min
ΔP = pressure difference in psi

The minimum bit HHP is shown in Figure 4-104. The maximum useful bit HHP is shown in Figure 4-105 and Figure 4-106 [18].

Useful Formulas

Theoretical output Q_t (gal/min) for a double action duplex pump is

$$Q_t = 0.0136NS\left(D_\ell^2 - \frac{d^2}{2}\right) \tag{4-39}$$

where N = strokes per minute
S = stroke length in in.
D_ℓ = liner diameter in m
d = piston rod diameter in in.

Theoretical output Q_t (gal/min) for a single action triplex pump is

$$Q_t = 0.0102\ NS\ D_\ell^2 \tag{4-40}$$

The volumetric efficiency η_v for duplex pumps or triplex pumps is

$$\eta_v = \frac{Q_a}{Q_t} \tag{4-41}$$

where Q_a = actual volumetric flow rate in gal

Input engine power IHP (hp) required for a given pump theoretical output Q_t and pump working pressure PWP is

(text continued on page 650)

Table 4-42
Jet Velocity [15]

	Nozzle Size									
GPM	5/16"	3/8"	13/32"	7/16"	1/2"	9/16"	5/8"	11/16"	3/4"	1"
100.	139	...	...	...	...	...	...	...	...	...
110.	153	...	...	...	...	...	...	...	...	...
120.	167	...	...	...	...	...	...	...	...	...
130.	181	...	...	...	...	...	...	...	...	...
140.	195	...	...	...	...	...	...	...	...	...
150.	209	145	...	...	...	...	...	...	...	...
160.	223	154	...	...	...	...	...	...	...	...
170.	237	164	...	...	...	...	...	...	...	...
180.	250	174	148	...	...	...	...	...	...	...
190.	264	183	156	...	...	...	...	...	...	...
200.	278	193	165	...	...	...	...	...	...	...
210.	292	203	173	149	...	...	...	...	...	...
220.	306	212	181	156	...	...	...	...	...	...
230.	320	222	189	163	...	99	...	...	...	...
240.	334	232	197	170	...	103	...	...	...	...
250.	348	241	206	177	...	107	...	...	...	...
260.	362	251	214	185	...	112	...	...	...	...
270.	376	261	222	192	147	116	...	...	...	...
280.	390	270	230	199	152	120	...	...	...	...
290.	403	280	239	206	156	124	101	...	...	...
300.	417	290	247	213	163	129	104	...	...	...
310.	431	299	255	220	168	133	108	...	...	...
320.	445	309	263	227	174	137	111	...	...	...
330.	459	318	271	234	179	142	115	...	...	...
340.	473	328	280	241	185	146	118	...	...	...
350.	487	338	288	248	190	150	122	101	...	...
360.	501	347	296	255	195	155	125	103	...	...
370.	515	357	304	263	201	159	129	106	...	...
380.	529	367	313	270	206	163	132	109	...	...
390.	543	376	321	277	212	167	136	112	...	...
400.	557	386	329	284	217	172	139	115	...	...
410.	...	396	337	291	223	176	143	118	99	...
420.	...	406	346	298	228	180	146	121	101	...
430.	...	416	354	305	234	185	150	124	104	...
440.	...	425	362	312	239	189	153	126	106	...
450.	...	435	370	319	245	193	156	129	109	...
460.	...	445	378	326	250	197	160	132	111	...
470.	...	454	387	334	256	202	163	135	114	...
480.	...	464	395	341	261	206	167	138	116	...
490.	...	474	403	348	266	210	170	141	118	...
500.	...	483	411	355	272	215	174	144	121	...
510.	...	493	420	362	277	219	177	146	123	...
520.	...	503	428	369	283	223	181	148	126	...
530.	...	512	436	376	288	227	184	152	128	...
540.	...	522	444	383	293	232	188	155	130	...
550.	...	532	452	391	299	236	191	158	133	...
560.	...	541	461	398	304	240	195	161	135	...
570.	...	551	469	404	310	245	198	164	138	...
580.	...	...	477	412	316	249	202	167	140	...
590.	...	...	485	419	321	253	205	170	142	...
600.	...	...	494	426	326	258	209	172	145	...
610.	...	...	502	433	331	262	212	175	147	...
620.	...	...	510	440	337	266	216	178	150	...
630.	...	...	518	447	342	270	219	181	152	...
640.	...	...	526	454	348	275	223	184	155	...
650.	...	...	535	461	353	278	226	187	157	...
660.	...	...	543	468	358	283	229	189	159	...
670.	...	...	551	475	364	288	233	192	162	...
680.	...	...	...	482	369	292	236	195	164	...
690.	...	...	...	490	376	296	240	198	167	...
700.	...	...	...	497	380	300	243	201	169	...
710.	...	...	...	504	386	305	247	204	171	...
720.	...	...	...	511	391	309	250	207	174	...
730.	...	...	...	518	396	313	254	210	176	...
740.	...	...	...	525	402	318	257	212	179	101
750.	...	...	...	532	407	322	261	215	181	102
760.	...	...	...	539	413	326	264	218	184	103
770.	...	...	...	546	418	330	268	221	186	105
780.	...	...	...	553	424	335	271	224	188	106
790.	...	...	...	...	429	339	275	227	191	107
800.	...	...	...	...	435	343	278	230	193	109
810.	...	...	...	...	440	348	282	232	196	110
820.	...	...	...	...	446	352	285	235	198	111
830.	...	...	...	...	451	356	289	238	200	113
840.	...	...	...	...	456	360	292	241	203	114
850.	...	...	...	...	462	365	296	244	205	115
860.	...	...	...	...	467	369	299	247	208	117
870.	...	...	...	...	473	373	302	250	210	118
880.	...	...	...	...	478	378	306	253	213	120
890.	...	...	...	...	484	382	309	255	215	121
900.	...	...	...	...	489	386	313	258	217	122
910.	...	...	...	...	494	390	316	261	220	124
920.	...	...	...	...	500	395	320	264	222	125
930.	...	...	...	...	505	399	324	267	225	126
940.	...	...	...	...	511	403	327	270	227	128
950.	...	...	...	...	516	408	331	273	229	129
960.	...	...	...	...	522	412	334	276	232	130
970.	...	...	...	...	527	416	338	278	234	132
980.	...	...	...	...	532	420	341	281	237	133
990.	...	...	...	...	538	425	345	284	239	134
1000.	...	...	...	...	543	429	348	287	242	136
1010.	...	...	...	...	549	433	351	290	244	137
1020.	...	...	...	...	554	438	355	293	246	138
1030.	...	...	...	...	...	442	358	296	249	140

Table 4-42 (continued)

	Nozzle Size									
GPM	5/16"	3/8"	11/32"	7/16"	1/2"	9/16"	5/8"	11/16"	3/4"	1"
1040.	...	...	...	...	...	446	362	298	251	141
1050.	...	...	...	...	...	450	365	301	254	143
1060.	...	...	...	...	...	455	369	304	256	144
1070.	...	...	...	...	...	459	372	307	258	145
1080.	...	...	...	...	...	463	376	310	261	147
1090.	...	...	...	...	...	468	379	313	263	148
1100.	...	...	...	...	...	472	383	316	266	149
1110.	...	...	...	...	...	476	386	319	268	151
1120.	...	...	...	...	...	480	390	322	270	152
1130.	...	...	...	...	...	485	393	325	273	153
1140.	...	...	...	...	...	489	397	327	275	155
1150.	...	...	...	...	...	493	400	330	278	156
1160.	...	...	...	...	...	498	403	333	280	158
1170.	...	...	...	...	...	502	407	336	283	159
1180.	...	...	...	...	...	506	410	339	285	160
1190.	...	...	...	...	...	510	414	342	287	162
1200.	...	...	...	...	...	515	417	345	290	163
1210.	...	...	...	...	...	519	421	348	292	164
1220.	...	...	...	...	...	523	424	350	295	166
1230.	...	...	...	...	...	528	428	353	297	167
1240.	...	...	...	...	...	532	431	356	299	168
1250.	...	...	...	...	...	536	435	359	302	170
1260.	...	...	...	...	...	540	438	362	304	171
1270.	...	...	...	...	...	545	442	365	307	172
1280.	...	...	...	...	...	549	445	368	309	174
1290.	...	...	...	...	...	...	449	371	312	175
1300.	...	...	...	...	...	...	452	373	314	177
1310.	...	...	...	...	...	...	455	376	316	178
1320.	...	...	...	...	...	...	459	379	319	179
1330.	...	...	...	...	...	...	463	382	321	181
1340.	...	...	...	...	...	...	466	385	324	182
1350.	...	...	...	...	...	...	470	388	326	183
1360.	...	...	...	...	...	...	473	391	328	185
1370.	...	...	...	...	...	...	477	394	331	186
1380.	...	...	...	...	...	...	480	396	333	187
1390.	...	...	...	...	...	...	483	400	336	189
1400.	...	...	...	...	...	...	487	402	338	190
1410.	...	...	...	...	...	...	490	405	341	192
1420.	...	...	...	...	...	...	494	408	343	193
1430.	...	...	...	...	...	...	497	411	345	194
1440.	...	...	...	...	...	...	501	414	348	196
1450.	...	...	...	...	...	...	504	417	350	197
1460.	...	...	...	...	...	...	508	419	353	198
1470.	...	...	...	...	...	...	511	422	355	200
1480.	...	...	...	...	...	...	515	425	357	201
1490.	...	...	...	...	...	...	518	428	360	202
1500.	...	...	...	...	...	...	522	431	362	204

Courtesy International Association of Drilling Contractors.

Table 4-43
Area of Nozzles [17]

D = Diameter. Area = .7853981634 D^2

Nozzle Size	Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area
	1/32	.000767	2	3.1416	6-7/8	37.122	11-3/4	108.434	16-5/8	217.08	21-1/2	363.05
	1/16	.003068	2-1/8	3.5466	7	38.485	11-7/8	110.753	16-3/4	220.35	21-5/8	367.28
	3/32	.006903	2-1/4	3.9761	7-1/8	39.871	12	113.10	16-7/8	223.65	21-3/4	371.54
	1/8	.01227	2-3/8	4.4301	7-1/4	41.282	12-1/8	115.47	17	226.98	21-7/8	375.83
	5/32	.01917	2-1/2	4.9088	7-3/8	42.718	12-1/4	117.86	17-1/8	230.33	22	380.13
	3/16	.02761	2-5/8	5.4119	7-1/2	44.179	12-3/8	120.28	17-1/4	233.71	22-1/8	384.46
7	7/32	.03758	2-3/4	5.9396	7-5/8	45.664	12-1/2	122.72	17-3/8	237.10	22-1/4	388.82
8	1/4	.04909	2-7/8	6.4918	7-3/4	47.173	12-5/8	125.19	17-1/2	240.53	22-3/8	393.20
9	9/32	.06213	3	7.0686	7-7/8	48.707	12-3/4	127.68	17-5/8	243.98	22-1/2	397.61
10	5/16	.07670	3-1/8	7.6699	8	50.266	12-7/8	130.19	17-3/4	247.45	22-5/8	402.04
11	11/32	.09281	3-1/4	8.2958	8-1/8	51.849	13	132.73	17-7/8	250.95	22-3/4	406.49
12	3/8	.1104	3-3/8	8.9462	8-1/4	53.456	13-1/8	135.30	18	254.47	22-7/8	410.97
13	13/32	.1296	3-1/2	9.6212	8-3/8	55.088	13-1/4	137.89	18-1/8	258.02	23	415.48
14	7/16	.1503	3-5/8	10.3206	8-1/2	56.745	13-3/8	140.50	18-1/4	261.59	23-1/8	420.00
15	15/32	.1726	3-3/4	11.0447	8-5/8	58.426	13-1/2	143.14	18-3/8	265.18	23-1/4	424.56
16	1/2	.1963	3-7/8	11.7933	8-3/4	60.132	13-5/8	145.80	18-1/2	268.80	23-3/8	429.13
17	17/32	.2217	4	12.566	8-7/8	61.862	13-3/4	148.49	18-5/8	272.45	23-1/2	433.74
18	9/16	.2485	4-1/8	13.364	9	63.817	13-7/8	151.20	18-3/4	276.12	23-5/8	438.36
—	19/32	.2769	4-1/4	14.186	9-1/8	65.397	14	153.94	18-7/8	279.81	23-3/4	443.01
20	5/8	.3068	4-3/8	15.033	9-1/4	67.201	14-1/8	156.70	19	283.53	23-7/8	447.69
—	21/32	.3382	4-1/2	15.904	9-3/8	69.029	14-1/4	159.48	19-1/8	287.27	24	452.39
22	11/16	.3712	4-5/8	16.800	9-1/2	70.882	14-3/8	162.30	19-1/4	291.04	24-1/8	457.11
—	23/32	.4057	4-3/4	17.721	9-5/8	72.760	14-1/2	165.13	19-3/8	294.83	24-1/4	461.86
24	3/4	.4418	4-7/8	18.665	9-3/4	74.662	14-5/8	167.99	19-1/2	298.65	24-3/8	466.64
—	25/32	.4794	5	19.635	9-7/8	76.589	14-3/4	170.87	19-5/8	302.49	24-1/2	471.44
26	13/16	.5185	5-1/8	20.629	10	78.540	14-7/8	173.76	19-3/4	306.35	24-5/8	476.26
—	27/32	.5591	5-1/4	21.648	10-1/8	80.516	15	176.71	19-7/8	310.24	24-3/4	481.11
28	7/8	.6013	5-3/8	22.691	10-1/4	82.516	15-1/8	179.67	20	314.16	24-7/8	485.98
	29/32	.6450	5-1/2	23.758	10-3/8	84.541	15-1/4	182.65	20-1/8	318.10	25	490.87
	15/16	.6903	5-5/8	24.850	10-1/2	86.590	15-3/8	185.66	20-1/4	322.06	25-1/8	495.79
	31/32	.7371	5-3/4	25.967	10-5/8	88.664	15-1/2	188.69	20-3/8	326.05	25-1/4	500.74
	1	.7854	5-7/8	27.109	10-3/4	90.763	15-5/8	191.75	20-1/2	330.06	25-3/8	505.71
	1-1/8	.9940	6	28.274	10-7/8	92.886	15-3/4	194.83	20-5/8	334.10	25-1/2	510.71
	1-1/4	1.2272	6-1/8	29.465	11	95.033	15-7/8	197.93	20-3/4	338.16	25-5/8	515.72
	1-3/8	1.4849	6-1/4	30.680	11-1/8	97.205	16	201.06	20-7/8	342.25	25-3/4	520.77
	1-1/2	1.7671	6-3/8	31.919	11-1/4	99.402	16-1/8	204.22	21	346.36	25-7/8	525.84
	1-5/8	2.0739	6-1/2	33.183	11-3/8	101.623	16-1/4	207.39	21-1/8	350.50	26	530.93
	1-3/4	2.4053	6-5/8	34.472	11-1/2	103.869	16-3/8	210.60	21-1/4	354.66		
	1-7/8	2.7612	6-3/4	35.785	11-5/8	106.139	16-1/2	213.82	21-3/8	358.84		

Courtesy Hughes Christensen.

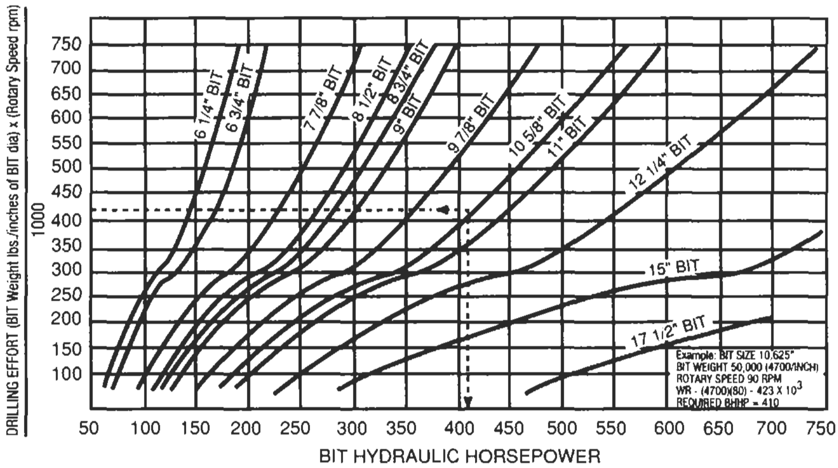
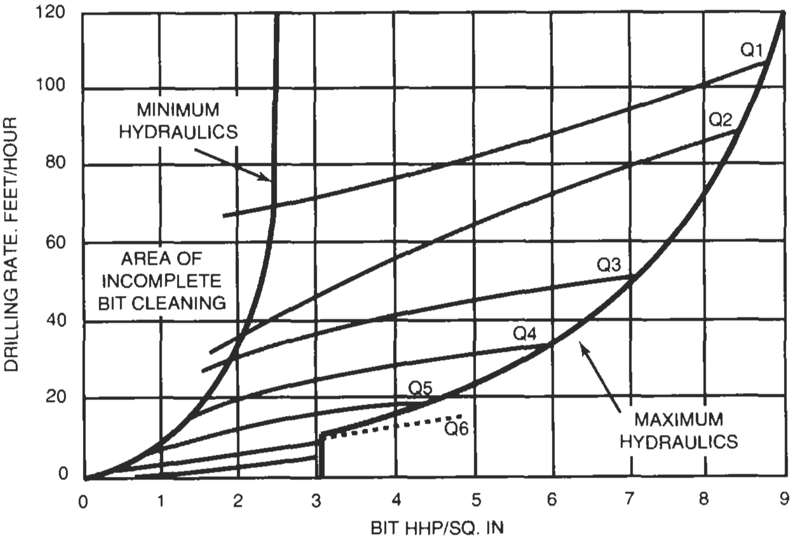


Figure 4-104. Minimum bit HHP to prevent hydraulic flounder [20]. *(Courtesy Smith International, Inc.)*



NOTE: PLACE POINT WHERE CONTROL BIT RUN IS LOCATED; USING SHIP CURVE, FIT Q-LINE BELOW POINT; MOVE SHIP CURVE *UP* TO POINT. EFFECT OF A CHANGE IN HYDRAULICS CAN THEN BE TRACED. POINTS OF COMPLETE BIT CLEANING ARE ON THE "MAXIMUM HYDRAULICS" CURVE.

Figure 4-105. Required bit hydraulic horsepower [21]. *(Courtesy Hant Publications. All rights reserved.)*

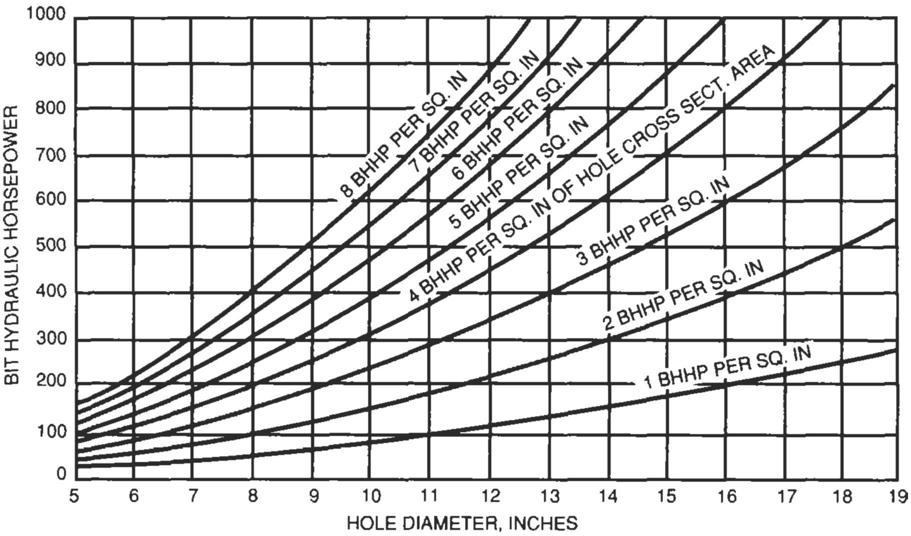


Figure 4-106. Bottomhole hydraulic horsepower chart [18]. *(Used by permission of the American Petroleum Institute, Production Department.)*

(text continued from page 645)

$$IHP = \frac{PWP\ (Q_t)}{1714\ \eta_m} \tag{4-42}$$

where η_m = mechanical efficiency of the pump

Pressure loss correction for mud weight change is

$$\Delta P_2 = \Delta P_1 \frac{\bar{\gamma}_2}{\bar{\gamma}_1} \tag{4-43}$$

where ΔP_1 = Pressure loss in system calculated using mud weight $\bar{\gamma}_1$ in psi
ΔP_2 = pressure loss in system calculated using mud weight $\bar{\gamma}_2$ in psi
$\bar{\gamma}_1, \bar{\gamma}_2$ = mud weight in lb/gal

DRILLING MUDS AND COMPLETION FLUIDS

Drilling Mud

Drilling muds are a special class of drilling fluid used to drill most deep wells. The term "mud" refers to the "thick" consistency of the fluid after the appropriate materials have been added to the water-liquid or oil-liquid base.

Functions

The functions of drilling fluid muds are:

a. To remove rock bit cuttings from the bottom of the hole and carry them to the surface.
b. To overcome formation fluid pressure.
c. To support and protect the walls of the hole.
d. To avoid damage to the producing formation.
e. To cool and lubricate the drill string and the bit.
f. To prevent drill pipe corrosion fatigue.
g. To allow the acquisition of information about the formation being drilled (e.g., electric logs, cuttings analysis).

Classification

The classification of drilling muds is based on their fluid phase, alkalinity, dispersion, and type of chemicals used.

Freshwater Muds—Dispersed Systems. The pH value of low-pH muds may range from 7.0 to 9.5. Low-pH muds include spud muds, bentonite-treated muds, natural muds, phosphate-treated muds, organic thinned muds (red muds, lignite muds, lignosulfate muds), and organic colloid-treated muds. The pH value of high pH muds, such as alkaline tannate-treated mud, is above 9.5.

Inhibited Muds—Dispersed Systems. These are water-base drilling muds that repress the hydration and dispersion of clays. There are essentially four types of inhibited muds: lime muds (high pH), gypsum muds (low pH), seawater muds (unsaturated saltwater muds, low pH), and saturated saltwater muds (low pH).

Low Solids Muds—Nondispersed Systems. These muds contain less than 3–6% solids by volume, weigh less than 9.5 lb/gal, and may be fresh or saltwater base. The typical low solids systems are flocculent, minimum solids muds, beneficiated clay muds, and low solids polymer muds. Most low solids drilling fluids are composed of water with varying quantities of bentonite and a polymer. The difference among low solids systems lies in the varying actions of different polymers.

Emulsions. Emulsions are formed when one liquid is dispersed as small droplets in another liquid with which the dispersed liquid is immiscible. Mutually immiscible fluids, such as water and oil, can be emulsified by stirring. The suspending liquid is called the *continuous phase,* and the droplets are called the *dispersed* (or *discontinuous*) *phase.* There are two types of emulsions used in drilling fluids: oil-in-water emulsions that have water as the continuous phase and oil as the dispersed phase, and water-in-oil emulsions that have oil as the continuous phase and water as the dispersed phase (invert emulsions).

Oil-Base Muds. Oil-base muds contain oil as the continuous phase and water as the dispersed phase. Oil-base muds contain less than 5% (by volume) water, while oil-base emulsion muds (invert emulsions) have more than 5% water in mud. Oil-base muds are usually a mixture of diesel fuel and asphalt; the filtrate is oil.

Testing of Drilling Fluids

Proper control of the properties of drilling mud is very important for their preparation and maintenance. Although oil-base muds are substantially different from water-base muds, several basic tests (such as specific weight, API funnel viscosity, API filtration, and retort analysis) are run in the same way. The test interpretations, however, are somewhat different. In addition, oil-base muds have several unique properties, such as temperature sensitivity, emulsion stability, aniline point, and oil coating–water wettability that require other tests. Therefore, testing of water and oil-base muds will be considered separately.

Water-Base Muds

Specific Weight of Mud. Often shortened to mud weight, this may be expressed as pounds per gallon (lb/gal), pounds per cubic foot (lb/ft^3), specific gravity (S_m), or pressure gradient (psi/ft) (see Table 4-44). Any instrument of sufficient accuracy within ±0.1 lb/gal or ±0.5 lb/ft^3 may be used. The mud balance is the instrument most commonly used [23]. The weight of a mud cup attached to one end of the beam is balanced on the other end by a fixed counterweight and a rider free to move along a graduated scale.

Viscosity. Mud viscosity is a measure of the mud's resistance to flow. The primary function of proper viscosity is to enable the mud to transport cuttings to the surface. Viscosity must be so high enough that the weighting material will remain suspended, but low enough to permit sand and cuttings to settle out and entrained gas to escape at the surface. Also, excessive viscosity creates high pump pressure and magnifies the swabbing or surging effect during tripping operations.

Gel Strength. This is a measure of the interparticle forces and indicates the gelling that will occur when circulation is stopped. This property prevents the cuttings from settling in the hole and sticking to the drill stem. High pump pressure is required to "break" circulation in a high gel mud. The following instruments are used to measure the viscosity and/or gel strength of drilling muds:

Marsh Funnel. The funnel is dimensioned so that, by following standard procedures, the outflow time of 1 qt (946 ml) of freshwater at a temperature of 70±5°F is 26±0.5 seconds [23]. A graduated cup or 1-qt bottle is used as a receiver.

Direct Indicating Viscometer. This is a rotational type instrument powered by an electric motor or by a hand crank. Mud is contained in the annular space between two cylinders. The outer cylinder or rotor sleeve is driven at a constant rotational velocity; its rotation in the mud produces a torque on the inner cylinder or bob. A torsion spring restrains the movement. A dial attached to the bob indicates its displacement. Instrument constants have been so adjusted that plastic viscosity, apparent viscosity, and yield point are obtained by using readings from rotor sleeve speeds of 300 and 600 rpm.

Plastic viscosity (PV) is centipoises equals the 600 rpm reading minus the 300 rpm reading. Yield point (YP) in pounds per 100 ft^2 equals the 300-rpm reading minus plastic viscosity. Apparent viscosity in centipoises equals the 600-rpm reading, divided by two. The interpretations of PV and YP measurements are presented in Figure 4-107.

Table 4-44
Specific Weight Conversion

1	2	3	4	5
			Gradient,	
lb/gal	lb/ft^3	g/cm^3 or specific gravity	psi/ft of depth	(kg/cm^2) m of depth
6.5	48.6	0.78	0.338	0.078
7.0	52.4	0.84	0.364	0.084
7.5	56.1	0.90	0.390	0.090
8.0	59.8	0.96	0.416	0.096
8.3	62.3	1.00	0.433	0.100
8.5	63.6	1.02	0.442	0.102
9.0	67.3	1.08	0.468	0.108
9.5	71.1	1.14	0.494	0.114
10.0	74.8	1.20	0.519	0.120
10.5	78.5	1.26	0.545	0.126
11.0	82.3	1.32	0.571	0.132
11.5	86.0	1.38	0.597	0.138
12.0	89.8	1.44	0.623	0.144
12.5	93.5	1.50	0.649	0.150
13.0	97.2	1.56	0.675	0.156
13.5	101.0	1.62	0.701	0.162
14.0	104.7	1.68	0.727	0.168
14.5	108.5	1.74	0.753	0.174
15.0	112.2	1.80	0.779	0.180
15.5	115.9	1.86	0.805	0.186
16.0	119.7	1.92	0.831	0.192
16.5	123.4	1.98	0.857	0.198
17.0	127.2	2.04	0.883	0.204
17.5	130.9	2.10	0.909	0.210
18.0	134.6	2.16	0.935	0.216
18.5	138.4	2.22	0.961	0.222
19.0	142.1	2.28	0.987	0.228
19.5	145.9	2.34	1.013	0.234
20.0	149.6	2.40	1.039	0.240
20.5	153.3	2.46	1.065	0.246
21.0	157.1	2.52	1.091	0.252
21.5	160.8	2.58	1.117	0.258
22.0	164.6	2.64	1.143	0.264
22.5	168.3	2.70	1.169	0.270
23.0	172.1	2.76	1.195	0.276
23.5	175.8	2.82	1.221	0.282
24.0	179.5	2.88	1.247	0.288

Gel strength, in units of lbf/100 ft^2, is obtained by noting the maximum dial deflection when the rotational viscometer is turned at a low rotor speed (usually 3 rpm) after the mud has remained static for some period of time. If the mud is allowed to remain static in the viscometer for a period of 10 s, the maximum dial deflection obtained when the viscometer is turned on is reported as the *initial gel* on the API mud report form. If the mud is allowed to remain static for 10 min, the maximum dial deflection is reported as the *10-min gel.*

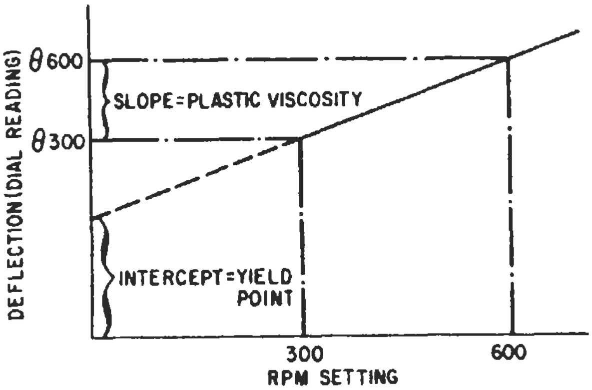


Figure 4-107. Typical flow curve of mud using a direct-indicating viscometer.

API Filtration. A filter press is used to determine the wall building characteristics of mud. The press consists of a cylindrical mud chamber made of materials resistant to strongly alkaline solutions. A filter paper is placed on the bottom of the chamber just above a suitable support. The filtration area is 7.1 (±0.1) in.^{2}. Below the support is a drain tube for discharging the filtrate into a graduate cylinder. The entire assembly is supported by a stand so that a 100-psi pressure can be applied to the mud sample in the chamber. At the end of the 30-min filtration time volume of filtrate is reported as API filtration in milliliters. To obtain correlative results, one thickness of the proper 9-cm filter paper, Whatman No. 50, S&S No. 5765, or the equivalent, must be used.

Thickness of the filter cake is measured and reported in $\frac{30}{60}$ of an inch. Also, the cake is visually examined and its consistency reported using such notations as "hard," "soft," tough," "rubbery," or "firm."

Sand Content. The sand content in mud is determined using a 200-mesh sieve screen $2\frac{1}{2}$ in. in diameter, a funnel to fit the screen, and a glass measuring tube. The measuring tube is marked for the volume of mud to be added to read directly the volume percent of sand on the bottom of the tube.

Sand content of the mud is reported in percent by volume. Also reported is the point of sampling, e.g., flowline, shaker, suction, pit, etc. Also, solids other than sand may be retained on the screen (lost circulation material, for example) and the presence of such solids should be noted.

Liquids and Solids Content. A mud retort is used to determine the liquids and solids content of the drilling fluid. Mud is placed in a steel container and heated until the liquid components have been vaporized. The vapors are passed through a condenser and collected in a graduated cylinder, and the volume of liquids (water and oil) is measured. Solids, both suspended and dissolved, are determined by volume as a difference between mud in container and distillate in graduated cylinder.

Specific gravity of the mud solids S_s is calculated as

$$S_s = \frac{100S_m - (v_w + 0.8v_o)}{v_s} \tag{4-44}$$

where v_w = volume percent water in %
v_o = volume percent oil in %
v_s = volume percent solids in %
S_m = specific gravity of the mud

(Oil is assumed to have a specific gravity of 0.8.)

For freshwater muds, a rough measure of the relative amounts of barite and clay in the solids can be made by using Table 4-45. As both suspended and dissolved solids are retained in the retort for muds containing substantial quantities of salt, corrections are made for the salt [23].

pH. Two methods for measuring the pH of drilling mud have been used: (1) a modified colorimetric method using paper test strips, and (2) the electrometric method using a glass electrode. The paper strip test may not be reliable if the salt concentration of the sample is too high. The electrometric method is subject to error in solutions containing high concentrations of sodium ions, unless a special glass electrode is used, or unless suitable correction factors are applied if an ordinary electrode is used. In addition, a temperature correction is required for the electrometric method of measuring pH.

The paper strips used in the colorimetric method are impregnated with such dyes that the color of the test paper is dependent upon the pH of the medium in which the paper is placed. A standard color chart is supplied for comparison with the test strip. Test papers are available in a wide range type, which permits estimating pH to 0.5 units, and in narrow range papers, with which the pH can be estimated to 0.2 units.

The glass electrode pH meter consists of a glass electrode system, an electronic amplifier, and a meter calibrated in pH units. The electrode system is composed of (1) the glass electrode, a thin walled bulb made of special glass within which is sealed a suitable electrolyte and an electrode; and (2) the reference electrode, which is a saturated calomel cell. Electrical connection with the mud is established through a saturated solution of potassium chloride contained in a tube surrounding the calomel cell. The electrical potential generated in the glass electrode system by the hydrogen ions in the drilling mud is amplified and operates the calibrated pH meter.

Table 4-45
Relative Amounts of Barite and Clay in Solids

Specific Gravity of Solids	Barite, Percent by Weight	Clay, Percent by Weight
2.6	0	100
2.8	18	82
3.0	34	66
3.2	48	52
3.4	60	40
3.6	71	29
3.8	81	19
4.0	89	11
4.3	100	0

Resistivity. Control of the resistivity of the mud and mud filtrate while drilling may be desirable to permit better evaluation of formation characteristics from electric logs. The determination of resistivity is essentially the measurement of the resistance to electrical current flow through a known sample configuration. Measured resistance is converted to resistivity by use of a cell constant. The cell constant is fixed by the configuration of the sample in the cell and is determined by calibration with standard solutions of known resistivity. The resistivity is expressed in ohm-meters.

Chemical Analysis. Standard chemical analyses have been developed for determining the concentration of various ions present in the mud [23]. Test for concentration of chloride, hydroxide and calcium ions are required to fill out the API drilling mud report. The tests are based on filtration, i.e., reaction of a known volume of mud filtrate sample with a standard solution of known volume and concentration. The end of chemical reaction is usually indicated by the change of color. The concentration of the ion being tested then can be determined from a knowledge of the chemical reaction taking place [7].

Chloride. The chloride concentration is determined by titration with silver nitrate solution. This causes the chloride to be removed from the solution as AgCl, a white precipitate. The endpoint of the titration is detected using a potassium chromate indicator. The excess Ag^+ present after all Cl^- has been removed from solution reacts with the chromate to form Ag_2CrO_4, an orange-red precipitate.

The mud contamination with chlorides results from salt intrusion. Salt can enter and contaminate the mud system when salt formations are drilled and when saline formation water enters the wellbore.

Alkalinity and Lime Content. *Alkalinity* is the ability of a solution or mixture to react with an acid. The *phenolphthalein alkalinity* refers to the amount of acid required to reduce the pH to 8.3, the phenolphthalein endpoint. The phenolphthalein alkalinity of the mud and mud filtrate is called the P_m and P_f, respectively. The P_f test includes the effect of only dissolved bases and salts while the P_m test includes the effect of both dissolved and suspended bases and salts. The *methyl orange alkalinity* refers to the amount of acid required to reduce the pH to 4.3, the methyl orange endpoint. The methyl orange alkalinity of the mud and mud filtrate is called the M_m and M_f, respectively. The API diagnostic tests include the determination of P_m, P_f, and M_f. All values are reported in cubic centimeters of 0.02 N (normality = 0.02) sulfuric acid per cubic centimeter of sample.

The P_f and M_f tests are designed to establish the concentration of hydroxyl, bicarbonate, and carbonate ions in the aqueous phase of the mud. At a pH of 8.3, the conversion of hydroxides to water and carbonates to bicarbonates is essentially complete. The bicarbonates originally present in solution do not enter the reactions.

As the pH is further reduced to 4.3, the acid then reacts with the bicarbonate ions to form carbon dioxide and water.

The P_f and P_m test results indicate the reserve alkalinity of the suspended solids. As the $[OH^-]$ in solution is reduced, the lime and limestone suspended in the mud will go into solution and tend to stabilize the pH. This reserve alkalinity generally is expressed as an equivalent lime concentration, in lb/bbl of mud.

Total Hardness. A combined concentration of calcium and magnesium in the mud water phase is defined as total hardness. These contaminants are often present in the water available for use in the drilling fluid. In addition, calcium can enter the mud when anhydrite ($CaSO_4$) or gypsum ($CaSO_4 \cdot 2H_2O$) formations are drilled. Cement also contains calcium and can contaminate the mud. The total hardness is determined by titration with a standard (0.02 N) Versenate (EDTA) solution. The standard Versenate solution contains Sodium Versenate, an organic compound capable of forming a chelate with Ca^{2+} and Mg^{2+}.

The hardness test sometimes is performed on the mud as well as the mud filtrate. The mud hardness indicates the amount of calcium suspended in the mud as well as the calcium in solution. This test usually is made on gypsum-treated muds to indicate the amount of excess $CaSO_4$ present in suspension. To perform the hardness test on mud, a small sample of mud is first diluted to 50 times its original volume with distilled water so that any undissolved calcium or magnesium compounds can go into solution. The mixture then is filtered through hardened filter paper to obtain a clear filtrate. The total hardness of this filtrate then is obtained using the same procedure used for the filtrate from the low-temperature low-pressure API filter press apparatus.

Methylene Blue. Frequently, it is desirable to know the cation exchange capacity of the drilling fluid. To some extent, this value can be correlated to the bentonite content of the mud.

The test is only qualitative because organic material and some other clays present in the mud also will absorb methylene blue. The mud sample usually is treated with hydrogen peroxide to oxidize most of the organic material. The cation exchange capacity is reported in milliequivalent weights (meq) of methylene blue per 100 ml of mud. The methylene blue solution used for titration is usually 0.01 N, so that the cation exchange capacity is numerically equal to the cubic centimeters of methylene blue solution per cubic centimeter of sample required to reach an endpoint. If other adsorptive materials are not present in significant quantities, the montmorillonite content of the mud in pounds per barrel is five times the cation exchange capacity.

The methylene blue test can also be used to determine cation exchange capacity of clays and shales. In the test a weighed amount of clay is dispersed into water by a high-speed stirrer. Titration is carried out as for drilling muds, except that hydrogen peroxide is not added. The cation exchange capacity of clays is expressed as milliequivalents of methylene blue per 100 g of clay.

Oil-Base Muds [23–25]

Specific Weight. Mud weight of oil muds is measured with a mud balance. The result obtained has the same significance as in water-base mud.

Viscosity. The measurement procedure for API funnel viscosity is the same as for water-base muds. Since temperature affects the viscosity, API procedure recommends that the mud temperature should always be recorded along with the viscosity.

Plastic Viscosity and Yield Point. Plastic viscosity and yield point measurements are obtained from a direct indicating viscometer. Due to the temperature effect on the flow properties of oil-base mud, the testing procedure is modified. The mud sample in the container is placed into a cup heater [23]. The heated viscometer cup provides flow property data under atmospheric pressure and bottomhole temperature.

Gel Strength. The gel strength of oil-base muds is measured with a direct indicating viscometer exactly like that of water-base muds.

Filtration. The API filtration test for oil-base muds usually gives an all-oil filtrate. The test may not indicate downhole filtration, especially in viscous oils. The alternative high-temperature–high-pressure (HT–HP) filtration test will generally indicate a pending mud problem by amount of fluid loss or water in the filtrate.

The instruments for the HT–HP filtration test consist essentially of a controlled pressure source, a cell designed to withstand a working pressure of at least 1000 psi, a system for heating the cell, and a suitable frame to hold the cell and the heating system. For filtration tests at temperatures above 200°F, a pressurized collection cell is attached to the delivery tube. The filter cell is equipped with a thermometer well, oil-resistant gaskets, and a support for the filter paper (Whatman No. 50 or the equivalent). A valve on the filtrate delivery tube controls flow from the cell. A nonhazardous gas such as nitrogen or carbon dioxide should be used for the pressure source.

The test is usually performed at a temperature of 300°F and a pressure of 600 psi over a 30-min period. When other temperatures, pressures, or times are used, their values should be reported together with test results. If the cake compressibility is desired, the test should be repeated with pressures of 200 psi on the filter cell and 100 psi back pressure on the collection cell.

Electrical Stability of Emulsions. The electrical stability test indicates the stability of emulsions of water in oil. The emulsion tester consists of a reliable circuit using a source of variable AC current (or DC current in portable units) connected to strip electrodes. The voltage imposed across the electrodes can be increased until a predetermined amount of current flows through the mud emulsion-breakdown point. Relative stability is indicated as the voltage at the breakdown point.

Sand Content. Sand content measurement is the same as for water-base muds except that diesel oil instead of water should be used for dilution.

Liquids and Solids Content. Oil, water, and solids volume percent is determined by retort analysis as in a water-base mud. More time is required to get a complete distillation of an oil mud than of a water mud. Then the corrected water phase volume, the volume percent of low gravity solids, and the oil–water ratio can be calculated; the chart in Figure 4-108 can be used for the calculations [24].

Example. Find the volume fraction of brine, the low gravity solids content, the adjusted mud weight, and the oil-to-water ratio from the test data below (use Figure 4-107).

Mud weight (specific weight) = 15.7 lb/gal
Volume % water (retort) = 20%
Volume % oil (retort) = 45%
Strong silver nitrate = 4.3 ml
(1 ml equivalent to 0.01 g Cl)

Step 1. To determine the percent by weight of calcium or of sodium chloride in the internal phase, locate the intersection of the line drawn horizontally from the cm^3 of strong silver nitrate required to titrate 1 cm^3 of whole mud with the line projected vertically from the volume percent of fresh water by retort.

Percent by weight brine in internal phase:

Strong silver nitrate = 4.3 ml
Volume % water (retort) = 20%
Read 25% by weight brine in internal phase

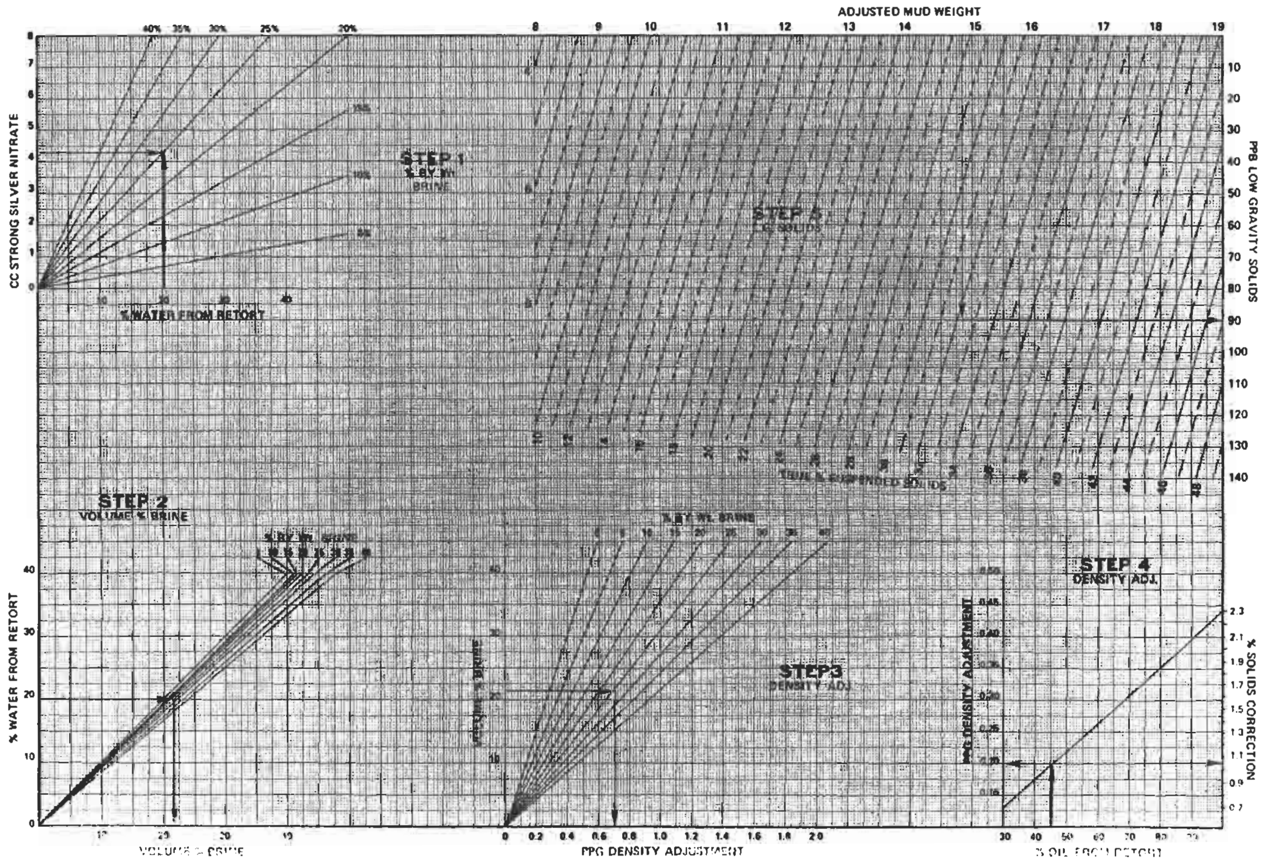


Figure 4-108. Liquids–solids analysis of oil mud. *(Courtesy M-I Drilling Fluids.)*

Step 2. Knowing the weight percent of brine and using the volume percent of freshwater by retort, the corrected volume fraction, which represents the true volume percent of brine in the solution, can be determined by running a line from the volume percent of water horizontally across until it meets the brine concentration, then dropping vertically to find the true volume percent of brine in the original mud. This number will always be greater than the volume percent of freshwater by retort.

Volume fraction brine in internal phase:

F_{vw} = 20%
Weight % brine = 25%
Read 21.5% volume fraction brine in internal phase

Step 3. To determine the gravity solids in the drilling mud, it is necessary to subtract from the mud weight all of the mud components except diesel oil and low gravity solids.

To do so, subtract from the measured mud weight the fraction contributed by brine and basic emulsifier (this step and Step 4). Knowing the volume fraction of brine in the internal phase (from Step 2) and the weight percent of brine (from Step 1), follow the appropriate value for the volume of brine horizontally to intersect the weight percent of brine. Extend that point vertically down to determine the weight in lb/gal and subtract that weight from the mud weight to correct for the weight of the internal phase.

Weight adjustment due to the internal phase:

Volume fraction of brine in internal phase = 21.5%
Weight percent of brine in internal phase = 25%
Read 0.69 lb/gal density adjustment

Step 4. This step corrects the mud weight and the volume percent of suspended solids as a function of the hydrocarbons distilled off by the mud still. Knowing the volume percent of oil from the mud still, follow this value vertically until it meets the line representing the system being run. Then extend this point horizontally to the left to determine the weight to subtract from the initial mud weight.

Weight adjustment due to distilled hydrocarbons:

Volume % oil = 45%
Read 0.20 lb/gal specific weight adjustment

The initial mud weight less the sum of the weight adjustments from Steps 3 and 4 is the corrected mud weight representing the weight of the diesel oil, low gravity solids, and barite only.

Calculation of adjusted mud weight:

Weight adjustment–internal phase = 0.69 lb/gal
Weight adjustment–emulsifier solids = 0.20 lb/gal
Mud weight = 15.7 lb/gal
Adjusted mud weight = 15.7 - 0.69 - 0.20 = 14.81 lb/gal

Step 5. After having found the adjusted mud weight, proceed horizontally from that point to the right to determine the volume percent of solids occupied by the basic emulsifier package. The volume percent of suspended solids is 100% less the sum of the volume-percent oil, the true volume-percent brine (Step 2), and the volume-percent emulsifier solids.

Calculation of volume-percent suspended solids:

Volume % emulsifier solids = 1.03%
Volume fraction of brine = 21.5%
Volume % oil = 45%
Volume % suspended solids = 100 - 21.5 - 45 - 1.03 = 32.47%

Find the adjusted mud weight value, extend that point downward until it meets the volume-percent suspended solids line. Proceed horizontally to find the ppg of low gravity solids.

Calculation of low gravity solids, lb/bbl

Adjusted mud specific weight = 14.81 lb/gal
Volume % suspended solids = 32.47%
Read 90 lb/bbl low gravity solids

Step 6. To find the oil-to-water ratio, divide the volume percent of oil in the liquid phase by (v_o) by the volume percent of water in the liquid phase (v_w).

Calculation of oil–water ratio:

$$v_o = 100\left[\frac{45}{45+20}\right] = 69\%$$

$$v_w = 100\left[\frac{20}{45+20}\right] = 31\%$$

$$\text{Oil-to-water ratio} = \frac{69}{31}$$

Aging. The aging test is used to determine how the bottomhole conditions affect oil-base mud properties. Aging cells were developed to aid in predicting the performance of drilling mud under static, high-temperature conditions. If the bottomhole temperature is greater than 212°F, the aging cells can be pressurized with nitrogen, carbon dioxide, or air to a desired pressure to prevent boiling and vaporization of the mud.

After aging period, three properties of the aged mud are determined before the mud is agitated: shear strength, free oil, and solids settling. Shear strength indicates whether the mud gels in the borehole. Second, the sample should be observed to determine if free oil is present. Separation of free oil is a measure of emulsion instability in the borehole, and is expressed in $\frac{32}{60}$ of an inch. Settling of mud solids indicates formation of a hard or soft layer of sediment in the borehole. After the unagitated sample has been examined, the usual tests for determining rheological and filtration properties are performed.

Alkalinity and Lime Content. The whole mud alkalinity test procedure is a titration method which measures the volume of standard acid required to react with the alkaline (basic) materials in an oil mud sample. The alkalinity value is used to calculate the pounds per barrel unreacted "excess" lime in an oil mud. Excess alkaline materials, such as lime, help to stabilize the emulsion and also neutralize carbon dioxide or hydrogen sulfide acidic gases.

To approximately 20 ml of a 1:1 mixture of toluene (xylene):isopropyl alcohol, add 1 ml of oil-base mud and 75 to 100 ml of distilled water. Add 8 to 10 drops of phenolphthalein indicator solution and stir vigorously with a stirring rod (the use of a Hamilton Beach mixer is suggested). Titrate slowly with H_2SO_4 (N/10) until red (or pink) color disappears permanently from the mixture. Report the alkalinity as the number of ml of H_2SO_4 (N/10) per ml of mud. Lime content may be calculated as

$$\text{Lime, ppb} = (1.5)(H_2SO_4\text{, ml})$$

Calcium Chloride [25]. Calcium chloride estimation is based on calcium titration. To 20 ml of 1:1 mixture of toluene (xylene):isopropyl alcohol, add a 1-ml (or 0.1-ml, if calcium is high) sample of oil-base mud, while stirring. Dilute the mixture with 75 to 100 ml of distilled water. Add 2 ml of hardness buffer solution and 10 to 15 drops of hardness indicator solution. Titrate mixture with standard versenate solution until the color changes from wine-red to blue. If common standard versenate solution (1 ml = 20 g calcium ions) is used, then

$$CaCl_2\text{, ppb} = (0.4)(\text{standard versenate, ml})$$

If strong standard versenate solution (1 ml = 2 g calcium ions) is used, then

$$CaCl_2\text{, ppb} = (4.0)(\text{strong standard versenate, ml})$$

Sodium Chloride [25]. Sodium chloride estimation is based on sodium titration. To 20 ml of a 1:1 mixture of toluene (xylene):isopropyl alcohol, add a 1-ml sample of oil-base mud, stirring constantly and 75 to 100 ml of distilled water. Add 8–10 drops of phenolphthalein indicator solution and titrate the mixture with H_2SO_4 (N/10) until the red (pink) color, if any, disappears. Add 1 ml of potassium chromate to the mixture and titrate with 0.282N $AgNO_3$ (silver nitrate, 1 ml = 0.001 g chloride ions) until the water portion color changes from yellow to orange. Then

$$NaCl\text{, ppb} = (0.58)(AgNO_3\text{, ml}) - (1.06)(CaCl_2\text{, ppb})$$

Some other procedures for $CaCl_2$ and NaCl content determination are used by mud service companies. Although probably more accurate, all of them are based on calcium filtration for $CaCl_2$ detection and on chlorides filtration for NaCl detection.

Total Salinity. The salinity control of oil-base mud is very important for stabilizing water-sensitive shales and clays. Depending upon the ionic concentration of the shale waters and of the mud water phase, an osmotic flow of pure water from the weaker salt concentration (in shale) to the stronger salt concentration (in mud) will occur. This may cause a dehydration of the shale and, consequently, affect its stabilization.

A standard procedure for estimating the salt content of oil-base muds consists of the following steps [26]:

1. Determination of calcium chloride concentration, lb/bbl.
2. Determination of sodium chloride concentration, lb/bbl.
3. Determination of soluble sodium chloride. By entering the graph in Figure 4-109 with the lb/bbl of calcium chloride at the correct volume percent of water (by retort) line, the maximum amount of soluble sodium chloride can be found. If the sodium chloride content determined in Step 2 is greater than the maximum soluble sodium chloride determined from Figure 4-108, only the soluble portion should be used for calculating the total soluble salts.
4. Determination of total mud salinity. The total pounds of soluble salts per barrel of mud are calculated as

   Total soluble salts (lb/bbl) = $CaCl_2$ (lb/bbl) - soluble NaCl (lb/bbl)

5. Determination of water phase salinity. By entering the graph in Figure 4-110 with total soluble salts, lb/bbl of mud, at the correct volume percent of water line, the water phase salinity can be read from the left-hand scale.

Example. Find the total salinity of the oil-base mud using the test data below and Figures 4-108 and 4-109.

Volume % water = 12%
From Ca titration, the $CaCl_2$ concentration is 18 lb/bbl mud
From Cl titration, the NaCl concentration is 9.9 lb/bbl mud

Step 1. The maximum soluble NaCl (from Figure 4-109) = 3 lb/bbl. (There is excess insoluble NaCl = 6.9 lb/bbl.)
Step 2. The total soluble salts in the mud = 18 + 3 = 21 lb/bbl.
Step 3. The water phase salinity (from Figure 4-110) = 330,000 ppm.

Water Wetting Solids. The water wetting solids test (oil-base mud coating test) indicates the severity of water wetting solids in oil-base mud [24]. The items needed are

1. Hamilton Beach mixer
2. Diesel oil
3. Xylene-isopropyl alcohol mixture
4. 16-oz. glass jar

Collect a 350-ml mud sample from the flowline and place the sample in the glass jar. Allow the sample to cool to room temperature before the test is conducted. Mix at 70 V with the mixer for 1 hr. Pour the mud out, add 100 ml diesel oil, and shake well. (Do not stir with mixer.) Pour the oil out, add 50 ml xylene-isopropyl alcohol (1:1) mixture, and shake well. Empty jar, turn upside down, and allow to dry. Observe the film on the wall of the jar and report the evaluation as

Opaque film–severe problem, probably settling of barite and plugging of the drill string.
Slight film, translucent–moderate problem, mud needs wetting agent immediately.
Very light film, highly translucent–slight wetting problem, mud needs some treatment.
No film–no water wetting problem.

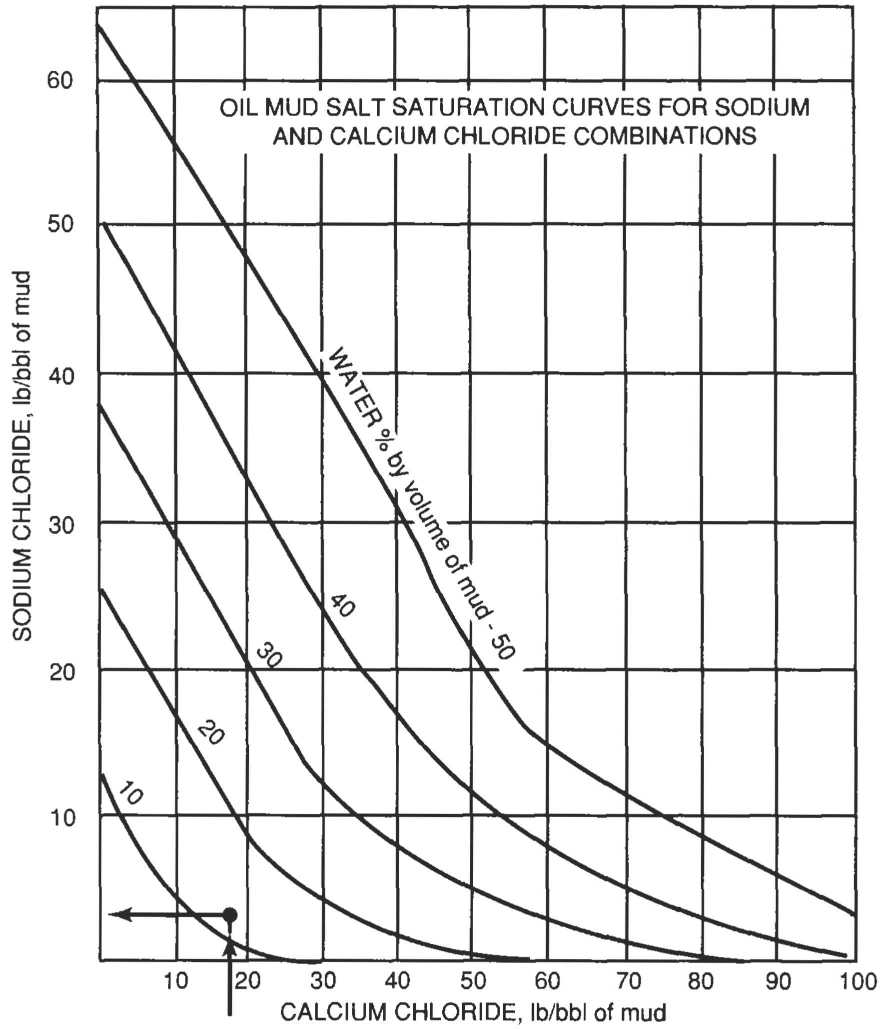


Figure 4-109. Solubility chart for calcium and sodium chloride brines [26]. *(Courtesy Baroid Drilling Fluids, Inc.)*

Drilling Fluids: Composition and Applications

Water-Base Mud Systems

Bentonite Mud

The bentonite muds include most types of freshwater muds. Bentonite is added to water-base muds to increase viscosity and gel strength, and also to improve

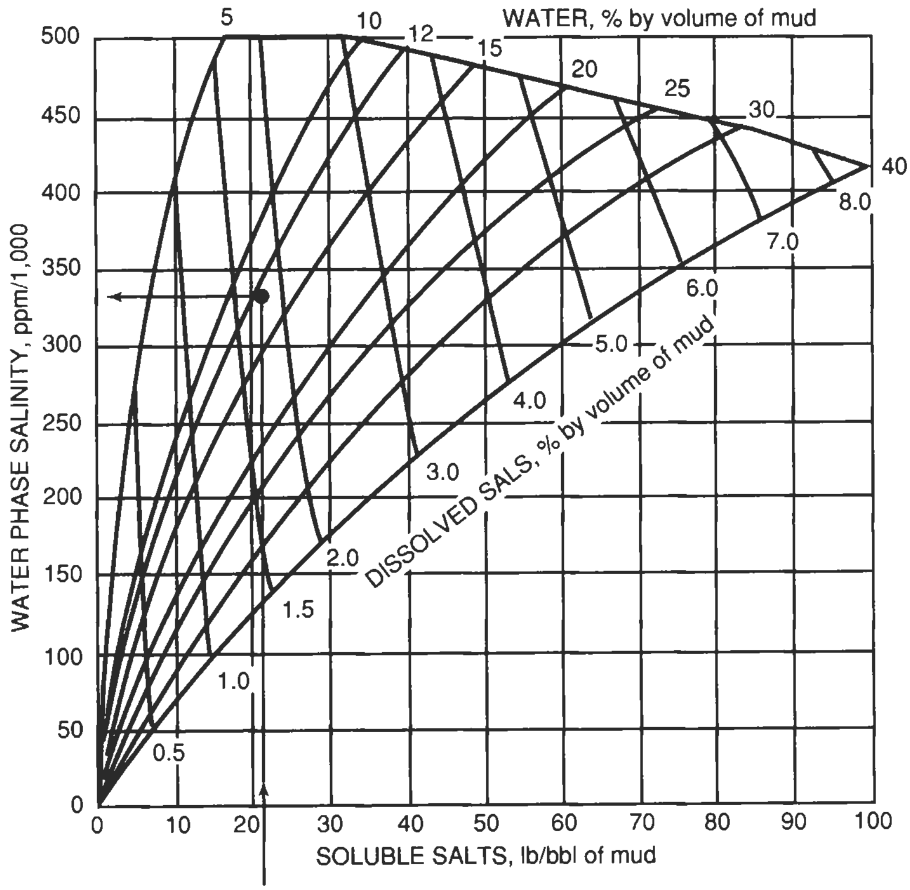


Figure 4-110. Total water phase salinity of oil mud [26]. *(Courtesy Baroid Drilling Fluids, Inc.)*

the filtration and filter cake properties of water-base muds. The comparison of the yield of commercial clays and active clays is shown in Table 4-46. The yield of clays is defined as the number of barrels of 15 cp mud that can be obtained from 1 ton of dry material. The API requirements for commercial drilling bentonite are as follows [27].

a. Bentonite concentration in distilled water—22.5 lb/bbl.
b. Sample preparation: 1. stir for 20 min. 2. Age overnight. 3. Stir for 5 min. 4. Test.
c. Apparent viscosity (viscometer dial reading at 600 rpm)—30 minimum.
d. Yield point, lb/100 ft^2—3 × plastic viscosity, maximum.
e. API filtrate, ml/30 min—15, maximum.
f. Yield, bbl mud/ton—91, minimum.

Classification of bentonite fluid systems is shown in Table 4-47 [28].

Table 4-46
Yield of Drilling Clays [28]

Drilling Clay	Yield, bbl/ton	Concentration % Volume	% by Weight	lb/bbl Mud	Mud Weight lb/bbl
Highest quality	100	2.5	6	20	8.6
Common	50	6.0	13	50	9.1
Lowest quality	25	10.0	20	75	9.6
Native	10	23.0	40	180	11.2

Courtesy International Drilling Fluids

Table 4-47
Classification of Bentonite Fluid Systems [28]

Solid-Solid Interactions	Inhibition Level	Drilling Fluid Type
Dispersed	Non-inhibited	1. *Fresh water clay based fluids.* Sodium chloride less than 1%, calcium ions less than 120 ppm a. Phosphate low pH (pH to 8.5) b. Tannin—high pH (pH 8.5–11+) c. Lignite d. Chrome lignosulphonate (pH 8.5–10)
Dispersed	Inhibited	2. *Saline (sodium chloride) fluids* a. Sea-water fluids b. Salt fluids c. Saturated salt fluids 3. *Calcium treated fluids* a. Lime b. Gypsum 4. *Low concentration lignosulphonate fluids*
Non-dispersed	Non-inhibited	*Fresh water—low solids* a. Extended bentonite systems b. Bentonite—polymer systems
Non-dispersed	Inhibited	*Salt—Polymer fluids*

Courtesy International Drilling Fluids

Dispersed Noninhibited Systems. Drilling fluid systems typically used to drill the upper hole sections are described as dispersed noninhibited systems. They would typically be formulated with freshwater and can often derive many of their properties from dispersed drilled solids or bentonite. They would not normally be weighted to above 12 lb/gal and the temperature limitation would be in the range of 176–194°F. The flow properties are controlled by a deflocculant, or thinner, and the fluid loss is controlled by the addition of bentonite and low viscosity CMC derivatives.

Phosphate-Treated Muds

The phosphates are effective only in small concentrations. Phosphate treated muds are subject to several limitations:

- Mud temperature should be lower than 130°F.
- Salt contamination should be lower than 5000 ppm chloride.
- Calcium concentration should be kept as low as possible.
- pH should be 8 to 9.5; in continuous use, the pH of some phosphates may decrease below the recommended limits so that pH maintenance with caustic soda is required.

Lignite Muds

Lignite muds are usually considered to be high-temperature-resistant since lignite is not affected by temperatures below 450°F. Lignite constitutes an inexpensive chemical for controlling apparent viscosity, yield point, gel strength, and fluid loss of a mud. Since lignite is refined humic acid (organic acid), caustic soda (sodium hydroxide) is usually necessary to adjust the pH of the mud to above 8; the treatment normally consists of adding 1 part of NaOH to 4 to 8 parts of lignite. If precausticized lignite (alkali + lignite) is being used, there is no need for the addition of caustic soda. The main limitations on lignite muds are

- hardness lower than 20 ppm
- pH of 8.5 to 10
- mud temperature below 450°F

Quebracho-Treated Muds

Quebracho-treated freshwater muds were used in drilling at shallow depths. The name of "red" mud comes from the deep red color imparted to the mud by quebracho. Muds treated with a mixture of lignite and quebracho, or a mixture of alkaline organic polyphosphate chemicals (alkaline-tannate treated muds), are also included in the quebracho treated muds. The quebracho thinners are very effective at low concentrations, and offer good viscosity and filtration control. The pH of "red" muds should be 8.5 to 10; mud temperature should be lower than 230°F.

Quebracho muds are used to increase the resistance to flocculation caused by contaminating salts, high pH (11 to 11.5). These muds can tolerate chloride contaminations up to 10,000 ppm.

Lignosulfonate Muds

Lignosulfonate freshwater muds contain ferrochrome lignosulfonate for viscosity and gel strength control. These muds are resistant to most types of drilling contamination due to the thinning efficiency of the lignosulfonate in the presence of large amounts of hardness and salt.

Lignosulfonate muds can be used efficiently at a pH of 9 to 10, and have a temperature limitation of about 350°F, above which lignosulfonates show severe thermal degradation. The recommended range of rheological properties of freshwater-base muds is shown in Figure 4-111 [29].

Dispersed Inhibited Systems. Dispersed inhibitive fluids attempt to combine the use of dispersed clays and deflocculants to derive the fundamental properties of viscosity and fluid loss with other features that will limit or inhibit the hydration of the formation and cuttings. It will be realized these functions are in opposition; therefore the ability of these systems to provide a high level of shale inhibition is limited. However, they have achieved a high level of success and in

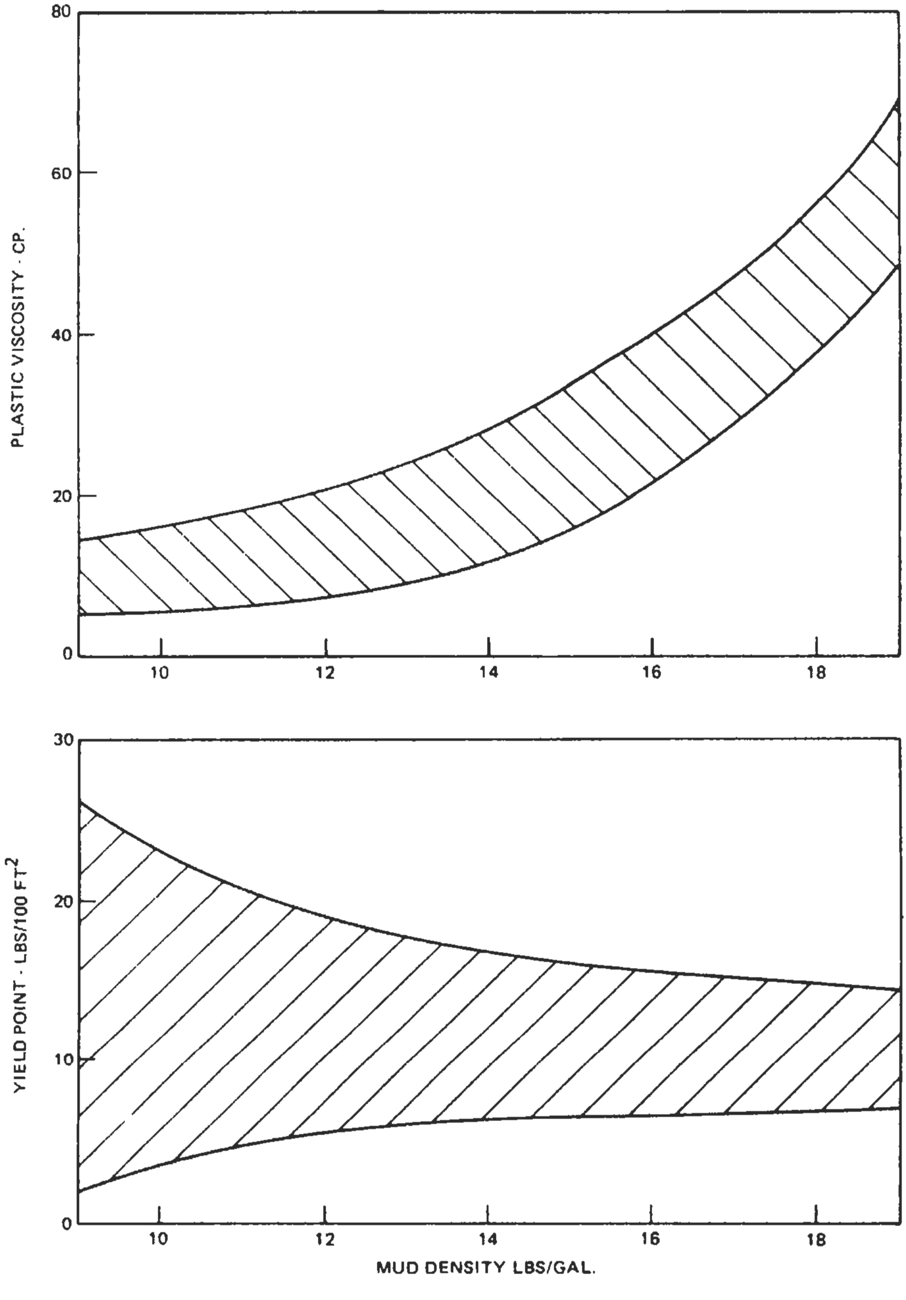


Figure 4-111. Suggested range of plastic viscosity and yield point for bentonite muds [29].

many formations represent a significant advance over dispersed non-inhibited types of fluids. Inhibition is sought through three mechanisms: addition of calcium (lime, gypsum), addition of salt, and addition of polymer.

Lime Muds

Lime muds are muds treated with caustic soda, an organic thinner, hydrated lime, and, for low filtrate loss, an organic colloid. This treatment results in muds having a pH of 11.8 or higher, with 3 to 20 ppm of calcium ions in the filtrate. Lime-treated muds exhibit low viscosity, low gels, good suspension of weighting material, ease of control at mud weights up to 20 lb/gal, tolerance to relatively large concentrations of flocculating salts, and easily maintained low filtration rates. One of the most important economic advantages of lime-treated mud is its ability to carry large concentrations of clay solids at lower viscosities than other types of mud. Except for a tendency to solidify under conditions of high bottomhole temperatures, lime-treated muds are well suited for deep drilling and for maintaining high weight muds. Pilot tests can be made on the mud to determine if the tendency to solidify exists; if so, solidification can be inhibited by chemical treatment for periods of time sufficient to allow normal drilling and testing activities. A lime-treated mud that exhibits a tendency to solidify should not be left in the casing-tubing annulus when the well is completed.

Lime-treated muds are prepared from freshwater drilling muds. The conversion should be made inside the basing. The initial step in conversion of freshwater mud to a lime mud involves dilution of the mud with water to reduce the clay solids content to avoid excessive mud viscosity (breakover). The recommended sequence of material addition is

a. Dilution water: 10–25% by volume
b. Thinner: 2 lb/bbl
c. Caustic soda: 2–3 lb/bbl
d. Lime: 4–8 lb/bbl
e. Thinner: 1 lb/bbl
f. Filtration control agent: 1–3 lb/bbl

The maintenance of lime-treated muds consists of monitoring the calcium content, i.e., the proper lime solubility. Since the lime solubility is controlled by the amount of caustic soda present in the mud, the proper alkalinity determination is of great importance. The recommended value of P_f is 5 to 8, and it is maintained with caustic soda; the recommended value of P_m is 25 to 40, and it is maintained with excess lime. The amount of excess lime should be from 5 to 8 lb/bbl.

The limitation of lime-treated mud is solidification at bottomhole temperatures higher than 250°F. Low lime mud was designed to minimize this tendency toward solidification and can be used at bottomhole temperatures as high as 350°F. In low lime mud, the total concentration of caustic soda and of lime is reduced. The recommended P_f is from 1 to 3, and the recommended P_m is from 10 to 15; the excess lime should be from 2 to 4 lb/bbl.

Gypsum-Treated Muds

Gypsum-treated muds have proved useful for drilling anhydride and gypsum, especially where these formations are interbedded with salt and shale. The treatment consists of conditioning the base mud with plaster (commercial calcium sulfate) before the anhydride or gypsum formation is penetrated. By adding the plaster at a controlled rate, the high viscosities and gels associated with this type of contaminant can be held within workable limits. After the clay in the base mud has reacted with the calcium ions in the plaster, no further thickening will occur upon drilling gypsum or salt formations. Gypsum-treated muds exhibit flat gels, and these flat gels depend in part upon the clay concentration in the mud. Filtration control is obtained by adding organic colloids; because the pH of these muds is low, preservatives are added to prevent the fermentation of starch.

Gypsum-treated muds are more resistant to contamination and more inhibitive (700 ppm of calcium ions) than lime-treated muds, and also have a greater temperature stability (350°F). A freshwater mud can be converted to a gypsum mud according to the following procedure:

a. Dilute with sufficient water to reduce API funnel viscosity to 35 s.
b. Add thinner (lignosulfonate) and caustic soda to avoid excessive viscosity build up (breakover).
c. Add gypsum at the mud hopper.

To control the stability of gypsum treated muds, the following mud properties should be maintained:

a. The mud pH should be 9.5 to 10.5; the alkalinity should be increased by adding lime rather than caustic soda.
b. The calcium ion concentration in the mud filtrate should be 600 to 1,000 ppm.
c. Addition of gypsum is necessary to maintain the amount of excess calcium sulfate ($CaSO_4$) between 2 and 6 lb/bbl; the relevant tests on excess calcium sulfate are subject to mud service on the rig.

Seawater Muds

Seawater muds or brackish water muds are saltwater muds. Saltwater muds are defined as those muds having salt (NaCl) concentrations above 10,000 ppm, or over 1%, salt; the salt concentration can vary from 10,000 to 315,000 ppm (saturation).

Seawater muds are commonly used on offshore locations, which eliminate the necessity of transporting large quantities of freshwater to the drilling location. The other advantage of seawater muds is their inhibition to the hydration and dispersion of clays, because of the salt concentration in seawater. The typical composition of seawater is presented in Table 4-48; most of the hardness of seawater is due to magnesium.

Calcium ions in seawater muds can be controlled and removed by forming insoluble precipitates accomplished by adding alkalis such as caustic soda, lime, or barium hydroxide. Soda ash or sodium bicarbonate is of no value in controlling the total hardness of sea water.

Seawater muds are composed of bentonite, thinner (lignosulfonate or lignosulfonate and lignite), and an organic filtration control agent. The typical formulation of a seawater mud is 3.5 lb/bbl of alkali (2 lb/bbl caustic soda and 1.5 lb/bbl lime), 8 to 12 lb/bbl of lignosulfonate, and 2 to 4 lb/bbl of bentonite to maintain viscosity and filtration. Another approach is to use bentonite/thinner (lignosulfonate)/freshwater premix, and mix it with seawater that has been treated for hardness. This technique will be discussed in the saturated saltwater muds section.

Chemical maintenance involves control of solids concentration, pH, alkalinity, and filtration, and pH control. Figure 4-112 shows the best operating range for

Table 4-48
Seawater Composition

Component	Concentration ppm	epm
Sodium	10440	454.0
Potassium	375	9.6
Magnesium	1270	104.6
Calcium	410	20.4
Chloride	18970	534.0
Sulfate	2720	57.8

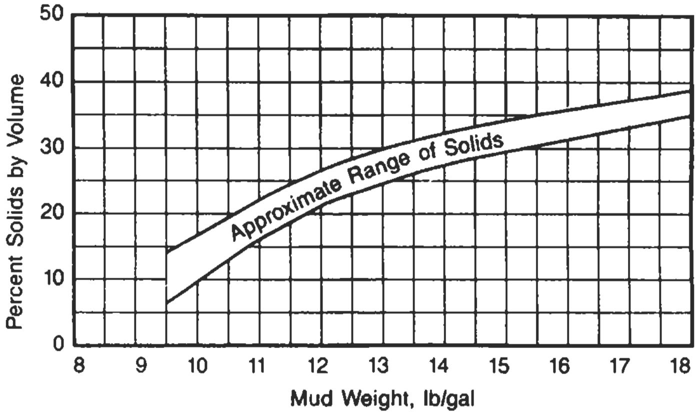


Figure 4-112. Approximate solids range in seawater muds [24]. *(Courtesy M-I Drilling Fluids.)*

solids in seawater muds [24]. The pH control is quite important, and pH should be maintained between 9 and 10. If the pH increases above 10, the magnesium will begin to precipitate. Caustic soda is used to control pH. Filtrate alkalinity P_f should be maintained at approximately 1.5 with caustic soda. Mud alkalinity P_m should be about 3.0 to 3.5; it is controlled with lime. If P_f is too low, the gel strength increases; if P_m is too low, mud aeration occurs; if P_f is too high, mud viscosity decreases. Filtration is controlled by addition of bentonite.

Saturated Saltwater Muds

The liquid phase of saturated saltwater muds is saturated with sodium chloride. Saturated saltwater muds are most frequently used as workover fluids or for drilling salt formations. These muds prevent solution cavities in the salt formations, making it unnecessary to set casing above the salt beds. If the salt formation is too close to the surface, a saturated saltwater mud may be mixed in the surface system as the spud mud. If the salt bed is deep, freshwater mud is converted to a saturated salt water mud.

Saturated saltwater muds can be weighted to more than 19 lb/gal. Saturated saltwater muds conditioned with organic colloids to control filtration can be used to drill below the salt beds, although high resistivity of the muds may result in unsatisfactory electric logs.

Conventional saturated salt muds are composed of attapulgite or "salt" clay and a starch, mixed with saturated brine water. The available make-up water or freshwater mud has to be saturated with salt (sodium chloride). Freshwater requires about 125 lb/bbl of salt to reach saturation; it then weighs approximately 10 lb/gal. Saltwater mud made up of 20 lb/bbl attapulgite clay has a funnel viscosity of about 40 s/qt, and a plastic viscosity of about 20 cp. Preparation of the saturated saltwater mud from freshwater mud requires the dumping of approximately half of the original mud, then saturation of the remaining original mud with salt and simultaneous extensive water dilution to avoid excessive viscosity buildup. Starch is used for filtration control in saturated saltwater muds at temperatures below 250°F. For higher temperatures (to 300°F), organic polymers must be used. Polymers and starches are not effective in the presence of cement or calcium concentration at high pH. If starches are used for filtration control, the salt concentration must be kept above 260,000 ppm or the pH above 11.5 to prevent fermentation. Alkalinity of the filtrate (P_f) should be kept at approximately 1 to control free calcium.

A modified saturated saltwater mud is prepared with bentonite clay by a special technique. First, bentonite is hydrated in freshwater, then treated with lignosulfonate and caustic soda. This premix is then mixed with saltwater (one-part premix to three-part saltwater). The mixture builds up a satisfactory viscosity and develops filtration control. Thinning of the mud is accomplished by saltwater dilutions; additional premix is required for viscosity and water loss control.

Nondispersed Noninhibited Systems. In nondispersed systems, no reagents are added to specifically deflocculate the solids in the fluid, whether they are formation clays or purposely added bentonite. The main advantage of these systems is to use the higher viscosities and, particularly, the higher yield point to plastic viscosity ratio. These altered flow properties provide better hole cleaning. They permit lower annular circulating rates and help prevent bore hole washouts.

Also, the higher degree of shear thinning provides for lower bit viscosities. This enables more effective use of hydraulic horsepower and faster penetration rates. In addition, shear thinning promotes more efficient operation of the solids removal equipment.

Low Solids–Clear (Fresh) Water Muds

It is a well-known fact in drilling practice that clear (fresh) water is the best drilling fluid as far as penetration rate is concerned. Therefore, whenever possible, drilling operators try to use minimum density and minimum solids drilling fluids to achieve the fastest drilling rate. Originally, the low solids–clear (fresh) water muds were used in hard formations, but now they are also applied to other areas.

Several types of flocculents can be added to clear water to promote the settling of drilled solids by flocculation. They are effective in low concentrations. The manufacturer's recommendations usually indicate lbs of flocculent per 100 ft of hole drilled. The typical application is prepared as follows:

a. Mix the polymer (flocculent) in a chemical barrel holding freshwater that has been treated for hardness with soda ash; the proportions are approximately 5 lb of polymer per 100 gal of water.

b. Inject the solution at the top of the flowline or below the shale shaker. The injection rate depends upon the hole size and the polymer efficiency (lb/100 ft of hole).
c. Let the mud circulate through all pits (tanks) available to increase the settling time; do not agitate the mud.
d. If additional flocculation is required, use lime or calcium chloride. The water at suction should be as clean as possible.
e. Slug the drill string prior to tripping with a high viscosity bentonite slurry (about 30 bbl) to remove excessive cuttings from the annulus.

Extended Bentonite Systems

To obtain a high viscosity at a much lower clay concentration, certain water-soluble vinyl polymers called *clay extenders* can be used. In addition to increasing the yield of sodium montmorillonite, clay extenders serve as flocculants for other clay solids. The flocculated solids are much easier to separate using solids control equipment.

The vinyl polymers increase viscosity by adsorbing on the clay particles and linking them together. The performance of commercially available polymers varies greatly as a result of differences in molecular weight and degree of hydrolysis. However, it is not uncommon to double the yield of commercial clays such as Wyoming bentonite using clay extenders in fresh water.

For low solids muds with bentonite extenders the API filtration rate is approximately twice that which would be obtained using a conventional clay/water mud having the same apparent viscosity.

However, good filtration characteristics often are not required when drilling hard, consolidated, low-permeability formations. In these formations the only concern is the effective viscosity in the annulus to improve the carrying capacity of the drilling fluid. The use of common grades of commercial clay to increase viscosity can cause a large decrease in the drilling rate. That is where bentonite extenders are mostly applicable. In addition to its viscosifying property bentonite extenders flocculate formation solids. A typical formulation of extended bentonite system is shown in Table 4-49 [28].

The bentonite should be specially selected for this type of system as being an untreated high yield Wyoming bentonite. The fluid has poor tolerance to calcium and salt, so the makeup water should be of good quality and pretreated with sodium carbonate, if any hardness exists. To increase viscosity bentonite extender is added through the hopper at the rate of one pound for every five sacks of bentonite. The extender is dissolved in water in the chemical barrel and added at a rate dependent on the drilling rate. Excessively high viscosities and gel strengths are normally the result of too high a solids content, which should be kept in the range 2–5% by dilution. Dispersants should not be added

Table 4-49
Extended Bentonite Mud System [28]

Fresh water	1 barrel
Bentonite extender	0.05 lb
Bentonite	11 lbs
Soda ash	0.25–0.5 lbs
Caustic soda	pH 8.5–9.0

as they compete too effectively with the extender for the adsorption sites on the clay.

A small excess of soda ash, of 0.57 kg/m^3 (0.2 lb/bbl), should be maintained to ensure the calcium level remains below 80 mg/l and to improve the efficiency of the extender. This level of soda ash will produce the required pH in most cases.

The system can be weighted to a maximum of 11 lb/gal provided the ratio of drill solids to clay solids is maintained at less than 2:1, by correct use of the solids removal equipment and careful dilution and makeup with bentonite from a premix tank.

Bentonite Substitute Systems

In this system, the high molecular weight polysaccharide polymer, is used to extend the rheological properties of bentonite.

A biopolymer produced by a particular strain of bacteria is becoming widely used as a substitute for clay in low-solids muds. Since the polymer is attacked readily by bacteria, a bactericide such as paraformaldehyde or a chlorinated phenol also must be used with the biopolymer. The system has more stable properties than the extended bentonite system, because biopolymer exhibits good rheological properties in its own right, and has a better tolerance to salt and calcium. The system can be formulated to include salt, such as potassium chloride. Such a system, however, would then be classed as a nondispersed inhibitive fluid.

Nondispersed Inhibited Systems. In these systems, the nondispersed character of the fluids is reinforced by some inhibition system, or combination of systems, such as (1) calcium ions, lime or gypsum; (2) salt-sodium chloride or potassium chloride; (3) polymers such as Polysaccharides, polyanionic cellulose, hydrolyzed polyacrylamide.

In these systems, particularly systems such as potassium chloride polymer, the role of bentonite is diminished because the chemical environment is designed to collapse and encapsulate the clays since this reaction is required to stabilize water-sensitive formations. The clay may have a role in the initial formulation of an inhibited fluid to provide the solids to create a filter cake.

Potassium Chloride–Polymer Muds

KCl-polymer (potassium chloride–polymer) muds can be classified as low solids–polymer muds or as inhibitive muds, due to their application to drilling in water-sensitive, sloughing shales. The use of polymers and the concentration of potassium chloride provide inhibition of shales and clays for maximum hole stability. The inverted flow properties (high yield point, low plastic viscosity) achieved with polymers and prehydrated bentonite provide good hole cleaning with minimum hole erosion.

The KCl–polymer muds are prepared by mixing potassium chloride (KCl) with fresh or saltwater. The desired KCl concentration depends upon the instability of the borehole and ranges from 3.5% by weight for drilling in shales containing illites and kaolinites to 10% by weight for drilling in bentonite shales. The polymer is then mixed in slowly through the hopper to the desired concentration (0.1 to 0.8 lb/gal depending upon the type of polymer). For additional viscosity, prehydrated bentonite (salt makeup water) can be added (0 to 12 lb/bbl) until satisfactory hole cleaning is achieved. The mud is adjusted to a pH of 9 to 10 with KOH or caustic soda. For filtration control, an organic filtration control agent should be used as recommended by the manufacturer.

Oil-Base Mud Systems

Oil-base muds are composed of oil as the continuous phase, water as the dispersed phase, emulsifiers, wetting agents, and gellants. There are other chemicals used for oil-base mud treatment such as degellants, filtrate reducers, weighting agents, etc.

The oil for an oil-base mud can be diesel oil, kerosene, fuel oil, selected crude oil, or mineral oil. There are several requirements for the oil: (1) API gravity = 36° - 37°, (2) flash point = 180°F or above, (3) fire point = 200°F or above, and (4) aniline point = 140°F or above. Emulsifiers are more important in oil-base mud than in water-base mud because contamination on the drilling rig is very likely, and it is very detrimental to oil mud. Thinners, on the other hand, are far more important in water-base mud than in oil-base mud; oil is dielectric, so there are no interparticle electric forces to be nullified.

The water phase of oil-base mud can be freshwater, or various solutions of calcium chloride ($CaCl_2$) or sodium chloride (NaCl). The concentration and composition of the water phase in oil-base mud determines its ability to solve the hydratable shale problem. Oil-base muds containing freshwater are very effective in most water-sensitive shales. The external phase of oil-base mud is oil and does not allow the water to contact the formation; the shales are thereby prevented from becoming water wet and dispersing into the mud or caving into the hole.

The stability of an emulsion mud is an important factor that has to be closely monitored while drilling. Poor stability results in coalescence of the dispersed phase, and the emulsion will separate into two distinct layers. Presence of oil in the emulsion mud filtrate is an indication of emulsion instability.

The advantages of drilling with emulsion muds rather than with water-base muds are (1) higher drilling rate, (2) reduction in drill pipe torque and drag, (3) less bit balling, and (4) reduction in differential sticking.

Oil-base muds are expensive and should be used when conditions justify their application. It is more economic to use oil base mud.

a. to drill troublesome shales that swell (hydrate) and disperse (slough) in water base muds,
b. to drill deep, high temperature holes in which water base muds solidify,
c. to drill water soluble formations such as salt, anhydride, carnallite, and potash zones,
d. to drill in producing zones.

For additional applications, oil muds can be used

a. as a completion and workover fluid,
b. as a spotting fluid to relieve stuck pipe,
c. as a packer fluid or a casing pack fluid.

There is one shale problem, however, that can be solved only by an oil-base mud with a $CaCl_2$ water solution. This shale problem is the "gumbo" or plastic flowing shale encountered in offshore Louisiana, the Oregon coast, Wyoming, and the Sahara desert. While drilling "gumbo" with water-base mud, the shale dispersion rate in the mud is so high that the drilling rate has to be slowed down or the mud will plug the annulus. All solids control problems are encountered, such as bit balling, collar balling, stuck pipe, shaker screens plugging, etc. An oil-base mud with a freshwater phase does not solve this problem, but only decreases the degree of severity. If the water phase of the oil mud is a solution of $CaCl_2$ (10 to 15 lb/bbl), dehydration of the wet (20 to 30% water) gumbo shale occurs; the shale becomes harder and it acts like a common water sensitive shale.

The general practice is to deliver the oil-base mud ready mixed to the rig, although some oil-base muds can be prepared at the rig. In the latter case, the most important principles are (1) to ensure that ample energy in the form of shear is applied to the fluid, and (2) to strictly follow a definite order of mixing. The following mixing procedure is recommended:

a. Pump the required amount of oil into the tank.
b. Add the calculated amounts of emulsifiers and wetting agent, stir, agitate, and shear these components until adequate dispersion is obtained.
c. Mix in all of the water, or the $CaCl_2$ water solution that has been premixed in the other mud tank. This requires shear energy. Add water slowly through the submerged guns; operation of a $\frac{1}{2}$-in. gun nozzle at 500 psi is considered satisfactory. After emulsifying all the water into the mud, the system should have a smooth, glossy, and shiny appearance. On close examination, there should be no visible droplets of water.
d. Add all the other oil-base mud products specified.
e. Add the weighting material last; make sure that there are no water additions while mixing in the weighting material.

When using an oil-base mud, certain rig equipment should be provided to control drilled solids in the mud and to reduce the loss of mud at the surfaces, i.e.,

a. Kelly valve—a valve installed between the kelly and the drill pipe will save about one barrel per connection.
b. Mud box—to prevent loss of mud while pulling wet string on trips and connections; it should have a drain to the flow line.
c. Wiper rubber—to keep the surface of the pipe dry and save mud.

Oil-base mud maintenance involves close monitoring of the mud properties along with the mud temperature, as well as the chemical treatment (in which the order of additions must be strictly followed). The following general guidelines should be considered:

a. The mud weight of an oil mud can be controlled within the interval from 7 lb/gal (aerated) to 22 lb/gal. A mud weight up to 10.5 lb/gal can be achieved with sodium chloride or with calcium chloride. For densities above 10.5 lb/gal, barite or ground limestone can be used. Limestone can weigh mud up to 14 lb/gal; it is used when an acid soluble solids fraction is desired, such as in drill-in fluids or in completion/workover fluids. Also, iron carbonate may be used to obtain weights up to 19.0 lb/gal when acid solubility is necessary.
b. Mud rheology of oil-base mud is strongly affected by temperature. API procedure recommends that the mud temperature be reported along with the funnel viscosity. The general rule for maintenance of the rheological properties of oil-base muds is that the API funnel viscosity, the plastic viscosity, and the yield point should be maintained in a range similar to that of comparable weight water muds. Estimated properties of two oil mud systems are shown in Figure 4-113 and Table 4-50. Excessive mud viscosity can be reduced by dilution with a diesel oil-emulsifier mixture that has been

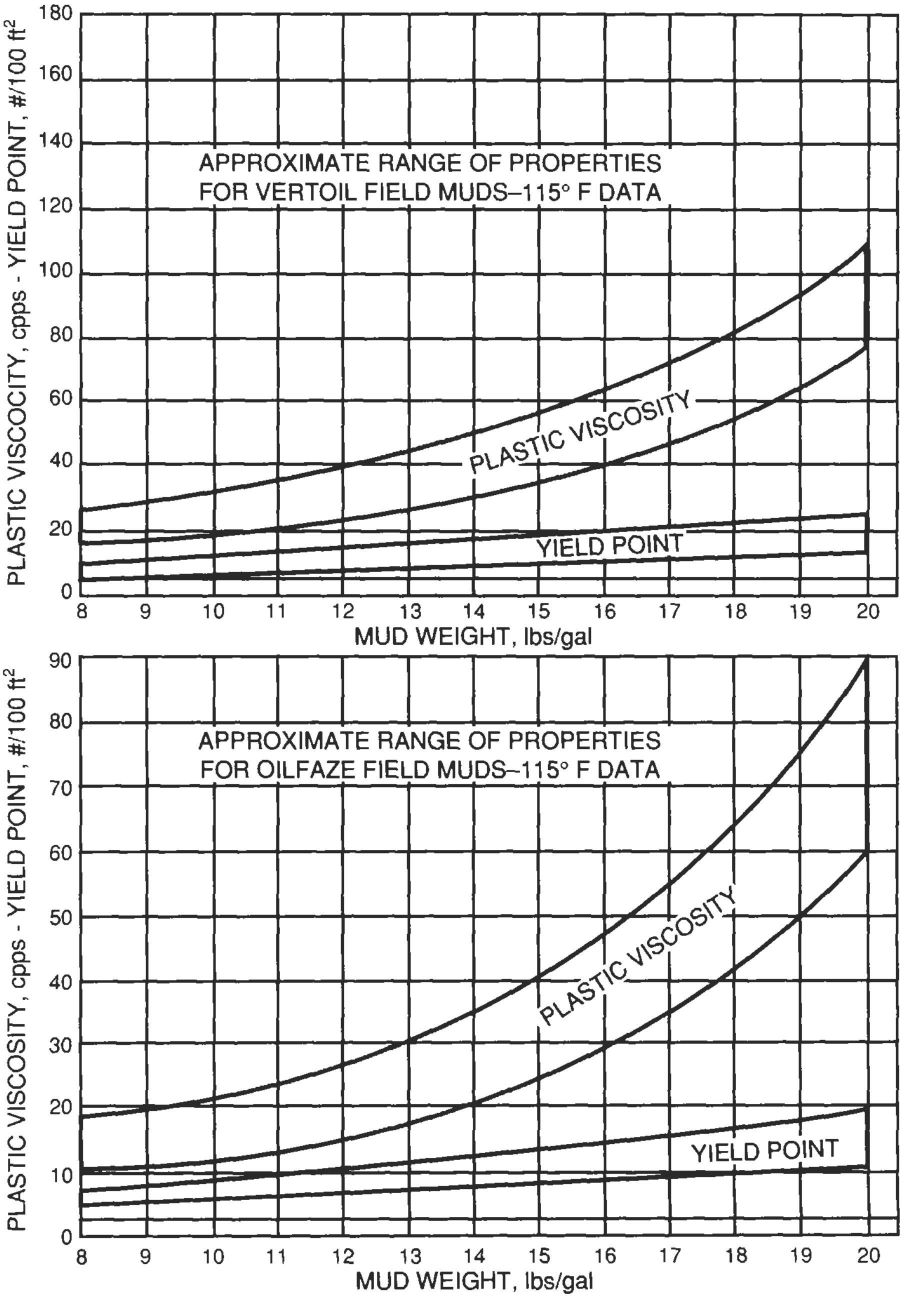


Figure 4-113. Approximate rheology of two oil-based mud systems. (VERTOIL = invert emulsion of oil and $CaCl_2$ brine; OILFAZE = invert emulsion of oil and freshwater)

Table 4-50
Estimated Requirements for Oil Mud Properties

Mud Weight ppg	Plastic Viscosity cP	Yield Point lbs/100 ft^2	Oil-Water Ratio	Electrical Stability
8–10	15–30	5–10	65/35–75/25	200–300
10–12	20–40	6–14	75/25–80/20	300–400
12–14	25–50	7–16	80/20–85/15	400–500
14–16	30–60	10–19	85/15–88/12	500–600
16–18	40–80	12–22	88/15–92/8	above 600

agitated in a separate tank. Insufficient viscosity can be corrected either by adding water (pilot testing required) or by treatment with a gellant.

c. There is no general upper limit on drilled solids concentration in oil muds, such as there is for water-base muds. However, a daily log of solids content enables the engineer to quickly determine a solids level at which the mud system performs properly.
d. Water wet solids is a very serious problem; in sever cases, uncontrollable barite settling may result. If there are any positive signs of water wet solids, a wetting agent should be added immediately. Tests for water wet solids should be run daily.
e. The dispersed water phase of an oil-base mud should be maintained in an alkaline pH range (i.e., pH above 7). Temperature stability as well as emulsion stability depends upon the proper alkalinity maintenance. If the concentration of lime is too low, the solubility of the emulsifier changes and the emulsion loses its stability. On the other hand, overtreatment with lime results in water wetting problems. Therefore, the daily lime maintenance has to be established and controlled by alkalinity testing. The recommended range of lime content for oil-base muds is from 2 to 4 lb/bbl.
f. $CaCl_2$ content should be checked daily and corrected.
g. The oil-water ratio influences viscosity and HT-HP (high-temperature-high-pressure) filtration of the oil-base mud. Retort analysis is used to detect any change in the oil-water ratio, giving the engineer a method for controlling the viscosity of the liquid phase by maintaining a relatively constant oil-water ratio.
h. Electrical stability is a measure of how well the water is emulsified in the continuous oil phase. Since many factors affect the electrical stability of oil-base muds, the test does not necessarily indicate that a particular oil-base mud is in good or in poor condition. For this reason, values are relative to the system for which they are being recorded. Stability measurements should be made routinely, and the values recorded and plotted so that trends may be noted. Any change in electrical stability indicates a change in the system.
i. HT-HP filtration should exhibit a low filtrate volume (about 3 ml). The filtrate should be water-free; water in the filtrate indicates a poor emulsion, probably caused by water wetting of solids.

Gaseous Drilling Mud Systems

The basic gaseous drilling fluids and their characteristics are presented in Table 4-51.

Tables 4-51
Gaseous Drilling Mud Systems

Type of Mud	Properties			Application Characteristics
	Density, ppg	pH	Temp. Limit °F	
Air/gas	0	—	500	High energy type system. Fastest drilling rate in dry, hard formations. Limited by water influx and hole size.
Mist	0.13–0.8	7–11	300	High energy system. Fast penetration rates. Can handle water intrusions. Stabilize unstable holes (mud misting).
Foam	0.4–0.8	4–10	400	Very low energy system. Good penetration rates. Excellent cleaning ability regardless of hole size. Tolerate large water influx.

Air–Gas Drilling Fluids

This system involves injecting air or gas downhole at the rates sufficient to attain annular velocity of 2,000 to 3,000 ft/min. Hard formations that are relatively free from water are most desirable for drilling with air-gas. Small quantities of water usually can be dried up or sealed off by various techniques.

Air-gas drilling usually increases drilling rate by three or four times over that when drilling with mud as well as one-half to one-fourth the number of bits are required. In some areas drilling with air is the only solution; these are (1) severe lost circulation, (2) sensitive producing formation that can be blocked by drilling fluid (skin effect), and (3) hard formations near the surface that require the use of an air hammer to drill.

There are two most important limitations on using air as a drilling fluid: large volumes of free water and size of the hole. Large water flow generally necessitates converting to another type of drilling fluid (mist or foam). Size of the hole determines a volume of air required for good cleaning. Lift ability of air is dependent upon annular velocity only (no viscosity or gel strength). Therefore, large holes require an enormous volume of air, which is not economical.

Mist Drilling Fluids

Misting involves the injection of air and mud or water and foamer. In case of "water mist" only enough water and foamer is injected into the airstream to clear the hole of produced fluids and cuttings. This unthickened water causes many problems due to wetting of the exposed formation which results in sloughing and caving. Mud misting, on the other hand, coats the walls of the hole with a thin film and has a stabilizing effect on water-sensitive formations. A mud slurry that has proved adequate for most purposes consists of 10 ppb of bentonite, 1 ppb of soda ash, and less than 0.5 ppb of foam stabilizing polymer such as high viscosity CMC. If additional foam stability is needed, additional organic filtration control agent should be added. One of the more important requisites for proper mud misting is foaming agent. The exact amount to be added depends on the particular foamer used, as most different brands have different amounts of active materials. Since air is the lifting medium in mist drilling fluid, the sufficient air velocity in the annulus should be from 2,000 to 3,000 ft/min. The approximate mud or water pumping rate is 10 bbl/hr.

Foam Drilling Fluids

Foam is gas-liquid dispersion in which the liquid is the continuous phase and the gas is the discontinuous phase. The first use of foam in drilling was reported in 1964.

Foam has been successfully used as a drilling fluid in several geological conditions.

1. In air drilling areas, the use of air drilling technique can be prolonged when formation water enters the hole by adding a small stream of liquid surfactant to the air stream. The addition of surfactant forms foam at the contact with formation water. The foam carries out cuttings and produced water. Considerable volumes of formation water can be held using this technique.
2. In hard rock drilling areas with loss of circulation, the application of preformed (mixed at the surface) stable foam shows four to ten times higher penetration rate than clay-based muds.
3. In oil-producing formations with high fluid loss, drilling in with foam and foam completion proves beneficial. Usually, these formations cannot stand a column of water—so it is impossible to establish returns with conventional mud. The use of foam for drilling in and completion results in substantial increases in production.

Stable foam systems consists of a detergent, freshwater, and compressed air. Gel-foam system includes bentonite added to the water-detergent mix. Additives may be included in the mixture for special purposes. To be used effectively as a circulating medium, foam must be preformed. That is, it must be generated without contact with the solid and liquid contaminants naturally encountered in the well. Once formed, foam systems have stabilizing characteristics that make them resistant to well-bore contaminants. Foam should have a gas-to-liquid volume ratio from 3–50 ft^3/gal depending on downhole requirements. The water-detergent solution that is mixed with gas to form foam can be prepared using a wide range of organic foaming agents (0.1–1.0 parts of foaming agent per 100 parts of solution). Foams can be prepared with densities as low as 0.26 lb/gal. Viscosity can be varied so high lifting capacities result when circulating at 300 fpm annular velocity. BHP measurements have indicated actual pressures of 15 psi at 1000 ft and 50 psi at 2,900 psi while circulating.

Drilling Fluid Additives

The classification of drilling fluid additives is based on the definitions of the International Association of Drilling Contractors [30].

a. Alkalinity or pH control additives are products designed to control the degree of acidity or alkalinity of a drilling fluid. These additives include lime, caustic soda, and bicarbonate of soda.

b. Bactericides reduce the bacteria count. Paraformaldehyde, caustic soda, lime, and starch are commonly used as preservatives.
c. Calcium removers are chemicals used to prevent and to overcome the contaminating effects of anhydride and gypsum, both forms of calcium sulfatc, which can wreck the effectiveness of nearly any chemically treated mud. The most common calcium removers are caustic soda, soda ash, bicarbonate of soda, and certain polyphosphates.
d. Corrosion inhibitors such as hydrated lime and amine salts are often added to mud and to air-gas systems. Mud containing an adequate percentage of colloids, certain emulsion muds, and oil muds exhibit, in themselves, excellent corrosion inhibiting properties.
e. Defoamers are products designed to reduce foaming action, particularly that occurring in brackish water and saturated saltwater muds.
f. Emulsifiers are used for creating a heterogenous mixture of two liquids. These include modified lignosulfonates, certain surface-active agents, anionic and noionic (negatively charged and noncharged) products.
g. Filtrate, or fluid loss, reducers such as bentonite clays, CMC (sodium carboxymethyl cellulose), and pregelatinized starch serve to cut filter loss, a measure of the tendency of the liquid phase of a drilling fluid to pass into the formation.
h. Flocculents are used sometimes to increase gel strength. Salt (or brine), hydrated lime, gypsum, and sodium tetraphosphates may be used to cause the colloidal particles of a suspension to group into bunches or "floos," causing solids to settle out.
i. Foaming agents are most often chemicals that also act as surfactants (surface-active agents) to foam in the presence of water. These foamers permit air or gas drilling through water-producing formations.
j. Lost circulation materials (LCM) include nearly every possible product used to stop or slow the loss of circulating fluids into the formation. This loss must be differentiated from the normal loss of filtration liquid, and from the loss of drilling mud solids to the filter cake (which is a continuous process in an open hole).
k. Extreme pressure lubricants are designed to reduce torque by reducing the coefficient of friction, and thereby increase horsepower at the bit. Certain oils, graphite powder, and soaps are used for this purpose.
l. Shale control inhibitors such as gypsum, sodium silicate, chrome lignosulfonates, as well as lime and salt are used to control caving by swelling or hydrous disintegration of shales.
m. Surface-active agents (surfactants) reduce the interfacial tension between contacting surfaces (e.g., water–oil, water–solid, water–air, etc.); these may be emulsifiers, deemulsifiers, flocculents, or deflocculents, depending upon the surfaces involved.
n. Thinners and dispersants modify the relationship between the viscosity and the percentage of solids in a drilling mud, and may further be used to vary the gel strength, improve "pumpability," etc. Tannins (quebracho), various polyphosphates, and lignitic materials are chosen as thinners or as dispersants, since most of these chemicals also remove solids by precipitation or sequestering, and by deflocculation reactions.
o. Viscosifers such as bentonite, CMC, attapulgite clays, subbentonites, and asbestos fibers (all colloids) are employed in drilling fluids to assure a high viscosity–solids ratio.
p. Weighting materials, including barite, lead compounds, iron oxides, and similar products possessing extraordinarily high specific gravities, are used

to control formation pressures, check caving, facilitate pulling dry drill pipe on round trips, and aid in combatting some types of circulation loss.

The most common, commercially available drilling mud additives are published annually by *World Oil.* The listing includes names and description of over 2,000 mud additives.

Environmental Aspects of Drilling Fluids

Much attention has been given in recent years to the environmental aspects of both the drilling operation and the drilling fluid components. Well-deserved concern with the possibility of polluting underground water supplies and of damaging marine organisms, as well as with the more readily observed effects on soil productivity and surface water quality, has stimulated widespread studies on this subject.

Drilling Fluid Toxicity

Sources of Toxicity. There are three contributing mechanisms of toxicity in drilling fluids, chemistry of mud mixing and treatment, storage/disposal practices, and drilled rock. The first group conventionally has been known the best because it includes products deliberately added to the system to build and maintain the rheology and stability of drilling fluids.

Petroleum, whether crude or refined products, need no longer be added to water-based muds. Adequate substitutes exist and are, for most situations, economically viable. Levels of 1% or more of crude oil may be present in drilled rock cuttings, some of which will be in the mud.

Common salt, or sodium chloride, is also present in dissolved form in drilling fluids. Levels up to 3,000 mg/L chloride and sometimes higher are naturally present in freshwater muds as a consequence of the salinity of subterranean brines in drilled formations. Seawater is the natural source of water for offshore drilling muds. Saturated brine drilling fluids become a necessity when drilling with water-based muds through salt zones to get to oil and gas reservoirs below the salt.

In onshore drilling there is no need for chlorides above these "background" levels. Potassium chloride has been added to some drilling fluids as an aid to controlling problem shale formations drilled. Potassium acetate or potassium carbonate are acceptable substitutes in most of these situations.

Heavy metals are present in drilled formation solids and in naturally occurring materials used as mud additives. The latter include barite, bentonite, lignite, and mica (sometimes used to stop mud losses downhole). There are background levels of heavy metals in trees that carry through into lignosulfonate made from them.

Recently attention has focused on the heavy metal impurities in barite. Proposed U.S. regulations would exclude many sources of barite ore. European and other countries are contemplating regulations of their own.

Chromium lignosulfonates are the biggest contributions to heavy metals in drilling fluids. Although studies have shown minimal environmental impact, substitutes exist that can result in lower chromium levels in muds. The less used chromium lignites (trivalent chromium complexes) are similar in character and performance with less chromium. Nonchromium substitutes are effective in many situations. Typical total chromium levels in muds are 100–1000 mg/l.

Zinc compounds such as zinc oxide and basic zinc carbonate are used in some drilling fluids. Their function is to react out swiftly sulfide and bisulfide ions originating with hydrogen sulfide in drilled formations. Because human safety is at stake, there can be no compromising effectiveness, and substitutes for zinc have not seemed to be effective. Fortunately, most drilling situations do not require the addition of sulfide scavengers.

Indiscriminate storage/disposal practices using drilling mud reserve pits can contribute toxicity to the spent drilling fluid as shown in Table 4-52 [31]. The data in Table 4-52 is from the EPA survey of the most important toxicants in spent drilling fluids. The survey included sampling active drilling mud (in circulating system) and spent drilling mud (in the reserve pit). The data show that the storage disposal practices became a source of the benzene, lead, arsenic, and fluoride toxicities in the reserve pits because these components had not been detected in the active mud systems.

The third source of toxicity in drilling discharges is drilled rocks. A recent study [32] of 36 cores collected from three areas (Gulf of Mexico, California, and Oklahoma) at various drilling depths (ranging from 300 to 18,000 ft) revealed that the total concentration of cadmium in drilled rocks was over five times greater than cadmium concentration in commercial barites. It was also estimated, using a 10,000-ft model well discharge volumes, that 74.9% of all cadmium in drilling waste may be contributed by cuttings while only 25.1% originate from the barite and the pipe dope.

Mud Toxicity Test. Presently, the only toxicity test for drilling fluids having an EPA approval is the Mysid shrimp bioassay. The test was developed in the mid-1970s as a joint effort of the EPA and the oil industry.

A bioassay is a test designed to measure the effect of a chemical on a test population of organisms. The effect may be a physiological or biochemical parameter, such as growth rate, respiration, or enzyme activity. In the case of drilling fluids, bioassays lethality is the measured effect.

To quantify the effect of a chemical on a population, groups of organisms are exposed to different concentrations of the chemical for a predetermined interval. The concentration at which 50% of the test population responds is known as the EC_{50} (effective concentration 50%); when death is the measured response, it is called the LC_{50} (lethal concentration 50%).

The LC_{50} concept is visualized in the dose-response curve presented in Figure 4-114 [32A]. The dose or concentration is plotted on the abscissa, and

Table 4-52
Toxicity Difference between Active and Waste Drilling Fluids [31]

TOXICANT	ACTIVE MUD	DETECTION RATE, %	RESERVE PIT	DETECTION RATE, %
Benzene	NO	-	YES	39
Lead	NO	-	YES	100
Barium	YES	100	YES	100
Arsenic	NO	-	YES	52
Fluoride	NO	-	YES	100

Courtesy SPE.

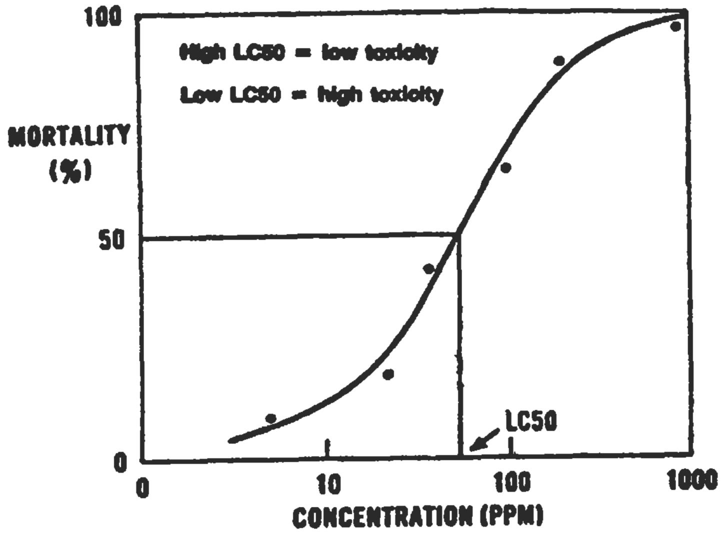


Figure 4-114. Determination of lethal toxicity LC_{50} from the dose–response curve [32A]. *(Courtesy SPE.)*

the corresponding response is plotted on the ordinate. The 50% value is interpolated from the resulting curve.

A high LC_{50} value indicates low toxicity, and a low LC_{50} value indicates a high degree of toxicity.

The 50% value is generally chosen because it represents the response of the average organism to the toxic exposure, thus providing the greatest predictive ability.

The vast majority of bioassays on marine organisms have been conducted on toxicants that are soluble in seawater. Because drilling mud contains solid particles, a special procedure had to be developed.

The test divides the drilling fluid into three phases: the liquid phase, the suspended particulate phase, and the solid phase. These phases are designed to represent the anticipated conditions that organisms would be exposed to when drilling mud is discharged into the ocean. Certain drilling fluid components are water column, others are fine particulates which would stay suspended, and still water soluble and will dissolve in the other material would settle rapidly to the bottom.

The procedure for phase separation follows the schematic in Figure 4-115 [32A]. To prepare the three test phases, a 1:9 ratio by volume of mud to seawater is mixed for 30 min. The pH is adjusted to that near seawater (pH = 7.8–9.0) by the addition of acetic acid. The slurry is allowed to settle for one hour. A portion of the supernatant is filtered through a 0.45-μm filter. The filtrate is designated as the "liquid phase." The remaining unfiltered supernatant of the slurry is the "suspended particulate phase," while the "solid phase" is the settled solid material at the bottom of the mixing vessel.

The filtered phase and suspended particulate phase of the 1:9 slurry represent the 100% concentration or 1,000,000 ppm. Serial dilutions of these two phases of drilling fluids are used in the test procedure to expose mysid shrimp (Mysidopsis bahia) for 96 hr and determine the LC_{50}.

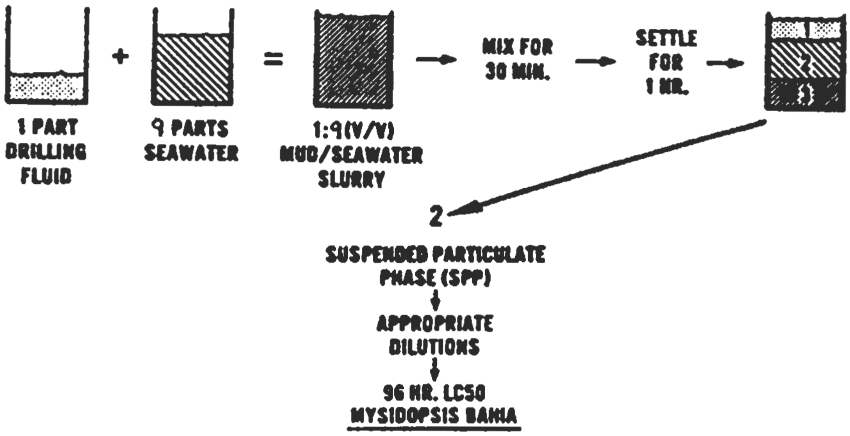


Figure 4-115. Schematic of toxicity test for drilling fluids [32A]. *(Courtesy SPE.)*

Results of these experiments usually give LC_{50}s ranging from 25,000 ppm to greater than 1,000,000 ppm of the phase for a variety of muds.

The mysid shrimp, *Mysidopsis bahia,* is the test organism for the liquid and suspended particulate phases. This species has been shown to be exceptionally sensitive to toxic substances and is considered to be a representative marine organism for bioassay testing by EPA. An LC_{50} is determined the suspended particulate phase (SPP) bioassay tests.

Low-Toxicity Drilling Fluids

The large number of existing oilfield facilities operation in or discharging produced water into surface waters of the United States has prompted EPA to issue general NPDES permits. The general permit allows discharge of low-toxicity drilling fluid directly to the sea.

These "generic" muds were identified by reviewing the permit requests and selecting the minimum number of mud systems that would cover all those named by the prospective permittees. Eight different mud systems were identified that encompass virtually all water-based muds used on the OCS (Table 4-53) [32A]. Instead of naming a set concentration for each component in each mud system, concentration ranges were specified to allow the operators sufficient flexibility to drill safely.

There are several significant permit conditions. As with all other OCS permits, the discharge of oil-based muds is prohibited. Similarly, the permit does not unconditionally authorize the discharge of any of the eight generic muds. Their discharge is subject to limitations on additives. To monitor the use of mud additives, the permit requires the additive not to drop or to decrease the 96-hr median lethal concentration (LC_{50}) test below 7,400 ppm on the basis of the suspended particulate phase or 740 ppm for the whole mud. This parameter is based on a test of Generic Mud 8, which is formulated with 5% mineral oil.

There is a mud-discharge-rate limitation of 1,000 bbl/hr, with reduced rates near areas of biological concern. The discharge of mud containing diesel for lubricity purposes is prohibited.

Table 4-53
Low-Toxicity "Generic" Drilling Fluids [32A]

1. Potassium/Polymer Mud		5. Spud Mud (Slugged Intermittently with Seawater)	
KCl	5 to 50	Attapulgite or bentonite	10 to 50
Starch	2 to 12	Lime	0.5 to 1
Cellulose polymer	0.25 to 5	Soda ash/sodium bicarbonate	0 to 2
XC polymer	0.25 to 2	Caustic	0 to 2
Drilled, solids	20 to 100	Barite	0 to 50
Caustic	0.5 to 3	Seawater	as needed
Barite	0 to 450		
Seawater or freshwater	as needed	6. Seawater/Freshwater Gel Mud	
2. Seawater/Lignosulfate Mud		Attapulgite or bentonite	10 to 50
Attapulgite or bentonite	10 to 50	Caustic	0.5 to 3
Lignosulfonate	2 to 15	Cellulose polymer	0 to 2
Lignite	1 to 10	Drilled solids	20 to 100
Caustic	1 to 5	Barite	0 to 50
Barite	25 to 450	Soda ash/sodium bicarbonate	0 to 2
Drilled solids	20 to 100	Lime	0 to 2
Soda ash/sodium bicarbonate	0 to 2	Seawater or freshwater	as needed
Cellulose polymer	0.25 to 5		
Seawater	as needed	7. Lightly Treated Lignosulfonate Freshwater/Seawater Mud	
3. Lime Mud		Bentonite	10 to 50
Lime	2 to 20	Barite	0 to 180
Bentonite	10 to 50	Caustic	1 to 3
Lignosulfonate	2 to 15	Lignosulfonate	2 to 6
Lignite	0 to 10	Lignite	0 to 4
Barite	25 to 180	Cellulose polymer	0 to 2
Caustic	1 to 5	Drilled solids	20 to 100
Drilled Solids	20 to 100	Soda ash/sodium bicarbonate	0 to 2
Soda ash/sodium bicarbonate	0 to 2	Lime	0 to
Freshwater or seawater	as needed	Seawater to freshwater ratio	=1:1
4. Nondispersed Mud		8. Lignosulfonate Freshwater Mud	
Bentonite	5 to 15	Bentonite	10 to 50
Acrylic polymer	0.5 to 2	Barite	0 to 450
Barite	25 to 180	Caustic	2 to 5
Drilled solids	20 to 70	Lignosulfonate	4 to 15
Freshwater or seawater	as needed	Lignite	2 to 10
		Drilled solids	20 to 100
		Cellulose polymer	0 to 2
		Soda ash/sodium bicarbonate	0 to 2
		Lime	0 to 2
		Freshwater	as needed

Courtesy SPE

Typical Calculations in Mud Engineering

Weighing Mud Up—Unlimited Volume

It is desired to increase the specific weight of 300 bbl of 10.5-lb/gal mud to 11.4-lb/gal using barite. The final volume is not limited. Determine the new volume of the mud. Also determine the weight of the barite to be added [7].

The new volume, V_2 (bbl), is

$$V_2 = V_1 \frac{(\bar{\gamma}_b - \bar{\gamma}_1)}{(\bar{\gamma}_b - \bar{\gamma}_2)} \tag{4-45}$$

where V_1 = the initial volume in bbl
$\bar{\gamma}_1$ = the specific weight of the initial mud in lb/gal
$\bar{\gamma}_2$ = the specific weight of the final mud in lb/gal
$\bar{\gamma}_b$ = the specific weight of barite (35.0 lb/gal).

Therefore, the final volume is

$$V_2 = (300)\frac{(35.0 - 10.5)}{(35.0 - 11.4)}$$

$$= 311.44 \text{ bbl}$$

The weight of the barite to be added is

$$W_b = (311.44 - 300.00)(35.0)(42)$$

$$= 16{,}817 \text{ lb}$$

Weighing Mud Up—Limited Volume

Example. It is desired to increase the specific weight of 700 bbl of 12.0-lb/gal mud to 14.0-lb/gal mud. To keep the new mixture from becoming too viscous, 1 gal of water is to be added with each 100-lb sack of barite. A final mud volume of 700 bbl is required. Determine the volume of initial mud that should be discarded and the weight of barite to be added [7].

The initial and final volumes are related by

$$V_1 = V_2 \left[\frac{\bar{\gamma}_b \left(\dfrac{1 + \bar{\gamma}_w v_{wb}}{1 + \bar{\gamma}_b v_{wb}} \right) - \bar{\gamma}_2}{\bar{\gamma}_b \left(\dfrac{1 + \bar{\gamma}_w v_{wb}}{1 + \bar{\gamma}_b v_{wb}} \right) - \bar{\gamma}_1} \right] \tag{4-46}$$

and the weight of barite added is

$$W_b = \frac{\bar{\gamma}_b}{1 + \bar{\gamma}_b v_{wb}} (V_2 - V_1) \tag{4-47}$$

where V_{bw} is the water requirement for the added barite (gal/lb).
Therefore, the initial volume is

$$V_1 = 700\left[\frac{35.0\left(\dfrac{1+8.33(0.01)}{1+35.0(0.01)}\right)-14.0}{35.0\left(\dfrac{1+8.33(0.01)}{1+35.0(0.01)}\right)-12.0}\right]$$

$$= 612.99 \text{ bbl}$$

Thus

$$700 - 612.99 = 87.01 \text{ bbl}$$

is the volume of initial mud that should be discarded before adding barite. The weight of barite needed is

$$W_b = \frac{35.0}{1+35.0(0.01)}(87.01)(42)$$

$$= 94{,}744 \text{ lb}$$

The total volume of water to be added with the barite, V_w (gal), often called dilution water, is

$$V_w = vb_w\ W_b \tag{4-48}$$

$$= 0.01\ (94744)$$

$$= 947.4 \text{ gal}$$

Determination of Oil/Water Ratio from Retort Data

To determine the O/W ratio, it is first necessary to measure oil and water percent by volume in the mud by retort analysis. From the data obtained the oil/water ratio is calculated as follows:

$$\%\text{ oil in the liquid phase} = \frac{\%\text{ oil by vol}}{\%\text{ oil by vol} + \%\text{ water by vol}} \times 100$$

$$\%\text{ water in the liquid phase} = \frac{\%\text{ water by vol}}{\%\text{ water of vol} + \%\text{ oil by vol}} \times 100$$

The oil/water ratio or O/W = % oil in liquid phase/% water in liquid phase.
For example, retort analysis:

51% oil by vol
17% water by vol
32% solids by vol

$$\% \text{ oil in liquid phase} = \frac{51}{51+17} \times 100 = 75\%$$

$$\% \text{ water in liquid phase} = \frac{17}{17+51} \times 100 = 25\%$$

Change of Oil/Water Ratio

It may become necessary to change the oil/water ratio of an oil mud while drilling. If the oil/water ratio is to be increased add oil, if it is to be decreased, add water. To determine how much oil or water is to be added to change the oil/water ratio, the following calculations are made:

1. Determine present oil/water ratio.
2. Decide whether oil or water is to be added.
3. Calculate how much oil or water is to be added for each hundred barrels of mud as follows:

Example A. Retort analysis:

51% oil by volume
17% water by volume
32% solids by volume

O/W ratio is 75/25 (from previous example). Change oil/water ratio to 80/20. Use basis of 100 bbl of mud.

Table 4-54
Comparison of Diesel Oil and Mineral Oil Muds [33]

Formulation or Property	Diesel-Oil Mud		Mineral Oil Mud	
Oil, bbl	0.59		0.59	
Primary Emulsifier, lb	9		9	
Secondary Emulsifier, lb	2		2	
Lime, lb	5		5	
High-Temperature Stabilizer, lb	8		8	
Water, bbl	0.2		0.2	
Organophilic Bentonite, lb	3		3	
Barite, lb	214		214	
Calcium Chloride, lb	37.2		37.2	
Aged at 300°F, hour	—	16	—	16
Plastic Viscosity, cp	55	39	47	32
Yield Point, lb/100 sq. ft.	30	26	27	20
10-Min. Gel, lb/100 sq. ft.	14	14	13	13
Electrical Stability, volts	960	1030	880	930
API Filtrate, ml	0.6	1.6	1.4	2.0
300°F Filtrate, ml	6.6	6.8	8.4	12.4

Courtesy SPE.

In 100 bbl of this mud there are 68 bbl of liquid (oil and water). To get to the new oil/water ratio we must add oil. The total liquid volume will be increased by the volume of oil added but the water volume will not change. The 17 bbl of water now in the mud represents 25% of the liquid volume, but it will represent only 20% of the final or new liquid volume. Therefore, let

x = final liquid volume; then 0.2x = 17

= 85 bbl

This is the new liquid volume. New liquid volume - original liquid vol = bbl of liquid (oil in this case) to be added, or 85 - 68 = 17. Add 17 bbl of oil/100 bbl of mud.

Check the calculation as follows: If the calculated amount of liquid is added, what will be the resulting oil/water ratio?

$$\% \text{ oil in liquid phase} = \frac{\text{original vol of oil} + \text{new oil added}}{\text{original vol} + \text{new oil added}} \times 100$$

$$= \frac{51 + 17}{68 + 17} \times 100$$

$$= \frac{68}{85} \times 100$$

$$= 80\%$$

100 - 80 = 20% water in liquid phase. New oil/water ratio is 80/20.

Example B. Retort analysis:

51% oil by volume
17% water by volume
32% solids by volume
oil/water ratio = 75/25

Change oil/water ratio to 70/30. Use basis of 100 bbl of mud.

As in Example A, there are 68 bbl of liquid in 100 bbl of mud. In this case, however, water will be added and the oil volume will remain constant. The 51 bbl of oil represents 75% of the original liquid volume and 70% of the final liquid volume. Therefore, let

x = final liquid volume

then

0.7x = 51

= 73 (new liquid volume)

New liquid vol - original liquid vol = amount of liquid (water in this case) to be added. 73 - 68 = 5 bbl of water to be added Check:

$$\% \text{ water in liquid phase} = \frac{\text{original water vol} + \text{water added}}{\text{original liquid vol} + \text{water added}} \times 100$$

$$\frac{17+5}{68+5} \times 100 = \frac{22}{73} \times 100 = 30\% \text{ water in liquid phase}$$

100 - 30 = 70% oil in liquid phase. New oil/water ratio is 70/30.

Solids Control

A mud system consists of the subsurface mud system and the surface mud system. The subsurface mud system consists only of the borehole and drill string, and its volume increases with the rate of drilling plus the rate of caving or sloughing. The surface mud system includes the equipment and the tanks through which the drilling mud passes after it flows out of the hole and before it is pumped back into the hole. The low-pressure surface mud system tends to decrease in volume as the hole is drilled due to increasing hole volume, rate of filtration, and cuttings removal. A rapid temporary change in surface mud system volume may occur because of formation fluids influx (kick), the addition of mud chemicals, or loss of circulation.

The unavoidable addition of solids comes from the continual influx of drilled cuttings into the active mud system. Undesirable solids increase drilling cost because they reduce penetration rate through their effect on mud specific weight and mud viscosity.

The surface mud system is designed to restore the mud to the required properties before it is pumped downhole. Most of the equipment is used for solids removal; only a small part of the surface mud system is designed to treat chemical contamination of the mud. There are three basic means of removing drilled solids from the mud: dilution-discard, chemical treatment, and mechanical removal.

The dilution-discard method is the traditional (sometimes the only) way to control the constant increase of colloidal size cuttings in weighted water-base muds. It is effective but also expensive, due to the high cost of barites used to replace the total weighting material in the discard. The daily mud dilutions amount to an average of 5 to 10% of the total mud system.

The chemical treatment methods reduce dispersability property, of drilling fluids through the increase of size of cuttings which improves separation and prevents the buildup of colloidal solids in the mud. These methods include ionic inhibition, cuttings encapsulation, oil phase inhibition (with oil-base muds), and flocculation. The mechanical solids removal methods are based on the principles presented in Table 4-55.

The surface mud system consists of solids removal equipment, mud agitating equipment, mixing equipment, and additional equipment. Solids removal equipment includes pits or tanks, shale shakers, sand traps, desanders, desilters, mud cleaners, and centrifuges. Mud-agitating equipment includes mud guns and mixers, mud-mixing equipment, and mud hoppers. Additional equipment includes the degasser, centrifugal pumps, suction lines, and discharge lines.

Solids Classification

Solids can be classified as those required for drilling and those detrimental to the drilling operation. Required solids are viscosifers (bentonite), filtration control agents, and weighting materials (barite). Viscosifers and filtration control agents are usually colloidal in size, i.e., smaller than 2 μm—Table 4-56 [29].

Table 4-55
Drill Cuttings Separation Principles

Method	Sorting Mechanisms	Characteristics	Devices
Screening	Size exclusion	Adhesion of fines to coarse solids; High throuput; Dry underflow	Shakers. Mud cleaners
Settling	Gravity forces	No shear; Low throuput; Liquidous underflow	Settling tanks
	Combination of drag and centrifugal forces	High shear High throuput Liquidous underflow	Desanders Desilters
	Centrifugal forces	Low shear; Low throuput underflow	Decanting centrifuge
		Low shear; Low throuput Liquidous underflow	Peforated rotor centrifuge

Table 4-56
Particle Size Terminology

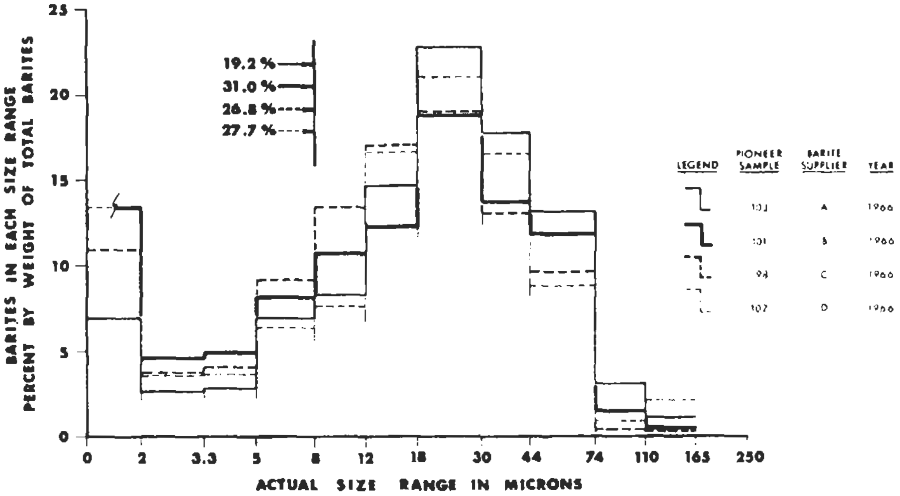
Solids size, microns		2	44	74	200 250	2000	
Geological	Sediment	Clay	Silt		Sand		Gravel
	Rock	Shale	Siltstone		Sandstone		Conglomerate
API Bullentin RP 13C		Colloidal	Ultra Fine	Fine	Medium	Intermediate	Coarse
Practical		Clay	Silt		API Sand or cuttings		

Barites range in size from 2 to 74 μm; its typical size distribution is shown in Figure 4-116 [34]. Also, the API-approved barite should have a minimum specific gravity of 4.2.

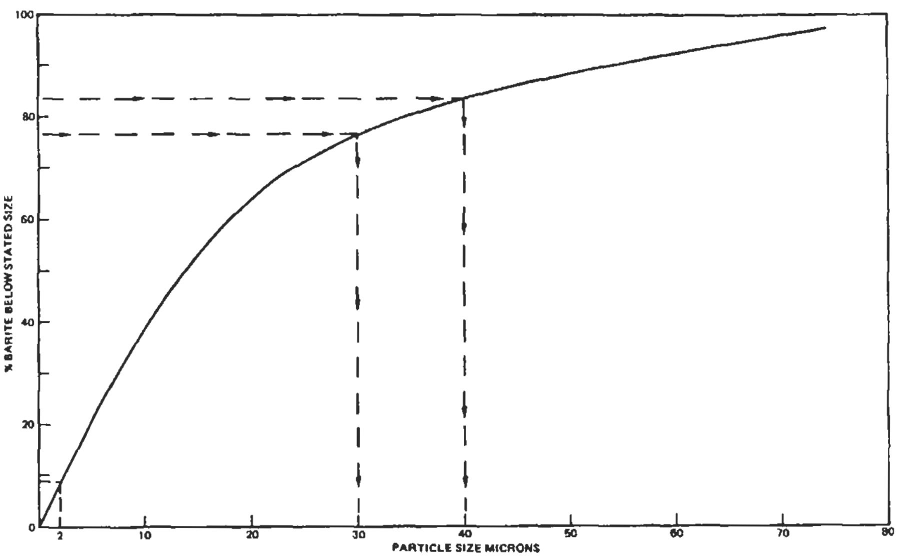
Undesirable solids are drilled cuttings and those solids sloughed into the borehole. They usually occur in all size ranges from colloidal to coarse. The specific gravity of commonly encountered drilled solids ranges from 2.35 (shale), through 2.65 (sand), 2.69 (limestone), to 2.85 (dolomite); see Table 4-57 [29].

Drilled solids include active drilled solids and inactive drilled solids. Clays and shales are considered to be active drilled solids; they disperse into colloidal size readily and become detrimental to drilling by increasing the apparent viscosity and gel strength of the mud. Inactive drilled solids are sand, dolomite, limestone, etc.; if they occur in colloidal size, these solids may increase plastic viscosity of the drilling mud.

For all practical purposes, solids in drilling mud are considered to be either low-gravity solids (drilled solids and gel, SG = 2.5 or 2.6) or high gravity solids (barite, SG = 4.2).



A. Distribution histogram



B. Cumulative distribution curve

Figure 4-116. Particle size distribution of commercial barites [34]. *(Copyright PennWell Books, 1986)*

Table 4-57
Specific Gravity of Liquids and Solids in Mud [29]

Material	Specific Gravity*	Composition
Anhydrite	2.95	$CaSo_4$
Barite**	4.45	$BaSo_4$
Calcite	2.71	Cao_3
Chlorite	2.71	$(Mg,Al,Fe)_{12}$ $(Si,Al)_8O_{20}(OH)_{16}$
Dolomite	2.85	$CaMg\ (CO_3)_2$
Galena	7.50	Pb S
Gypsum	2.32	$CaSo_42H_{20}$
Hematite	5.26	Fe_2O_3
Illite	2.84	$KAL_5Si_7\ O_{20}\ (OH)_4$
Lignite	1.10	
Limestone	2.69	
Montmorillonite	2.35	$(OH)_4Si_8AL_4\ O_{20}(H_2O)$
Pyrite	5.06	FeS_2
Quartz	2.65	
Sodium Chloride	2.165	NaCl
Sulfur	1.96	
Water	1.00	

*Chemically pure

Solids in Unweighted Muds

Solids in unweighted muds include viscosifers and drilled solids. The most expensive portion of unweighted muds are the liquids and colloidals. The main concern in unweighted muds is to keep mud weight as low as possible and to maintain flow properties. Thus, viscosifers are added as needed to unweighted muds to control filtration, to suspend solids, and to provide the properties necessary to clean the borehole. The detrimental solids in unweighted muds are those of ultrafine size and larger, produced by the bit. It is essential to have a good solids removal system to prevent solids dispersion and mud density buildup. Size and range of solids in unweighted muds are shown in Figure 4-117 [29].

Solids in Weighted Muds

Solids in weighted muds consist of viscosifers, weighting material, and drilled cuttings. The most expensive portion of weighted mud is the weighting material. The main problem related to solids control is the prevention of viscosity increase caused by accumulation of colloidal drilled solids. Chemical treatment can be used initially to control this viscosity, but it becomes ineffective as colloidal solids in the mud increases. Eventually, mud dilution and mechanical removal of solids are needed. The size range of solids in a weighted mud is illustrated in Figure 4-118 [29].

Figure 4-119 through 4-122 can be used to evaluate the extent of drilled solids contamination [25].

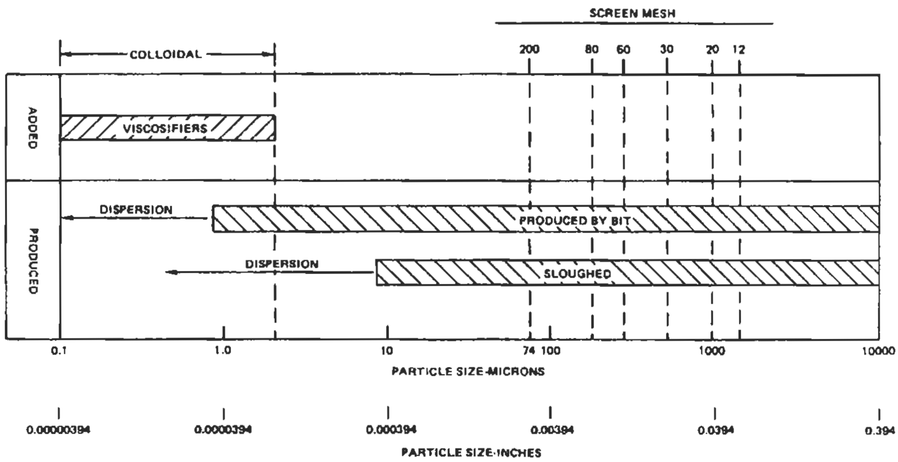


Figure 4-117. Size range of solids in unweighted muds [29].

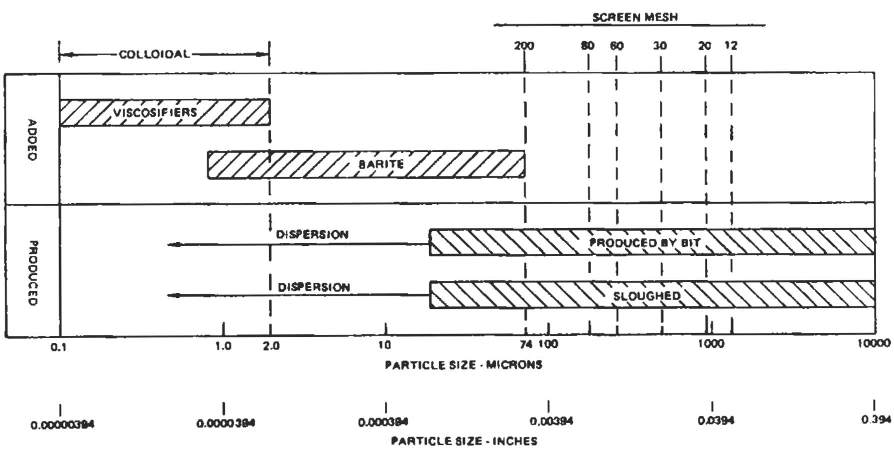


Figure 4-118. Size range of solids in weighted muds [29].

Mud-Related Hole Problems

Table 4-58 summarizes general hole problems related to the use of their drilling mud [25].

Tables 4-59 through 4-61 summarize formation related hole problems in surface, intermediate, and production drilling, respectively.

(text continued on page 701)

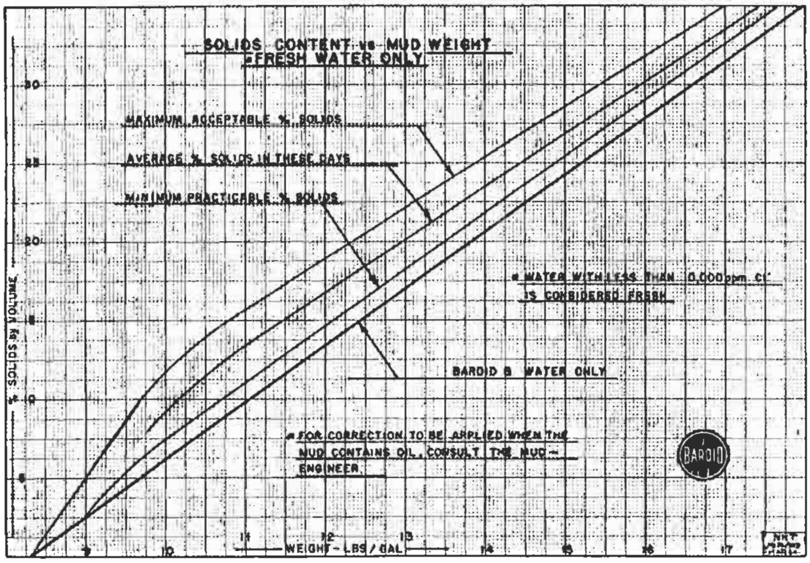


Figure 4-119. Practical limits on solids content in freshwater-base mud [26]. *(Courtesy Baroid Drilling Fluids, Inc.)*

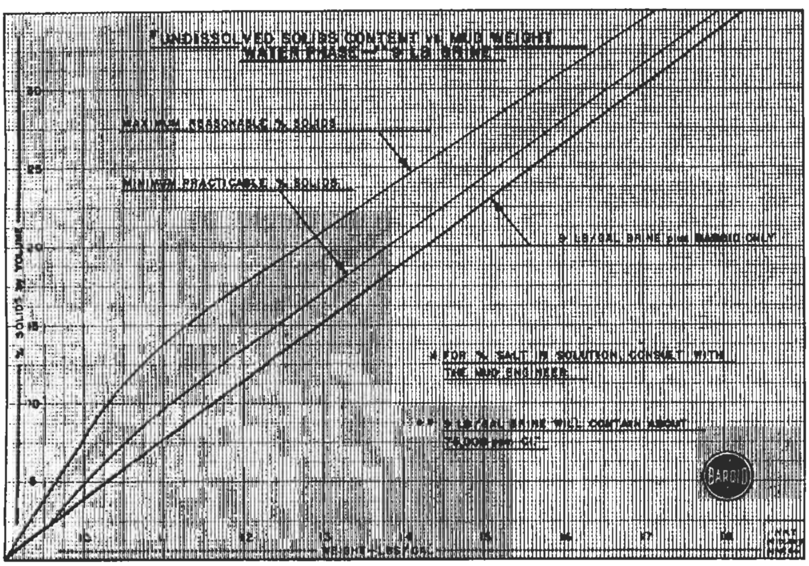


Figure 4-120. Practical limits on solids content in saltwater mud (75,000 ppm chlorides) [26]. *(Courtesy Baroid Drilling Fluids, Inc.)*

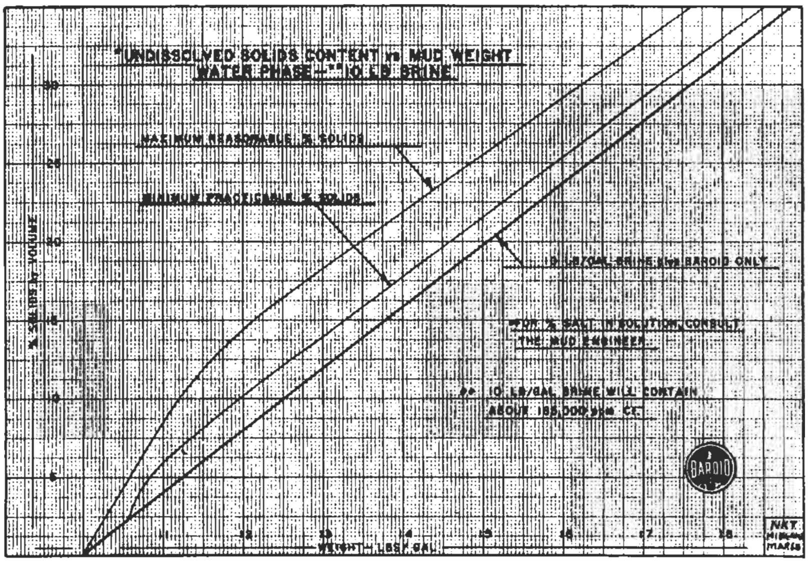


Figure 4-121. Practical limits on solids content in saltwater mud (185,000 ppm chlorides) [26]. *(Courtesy Baroid Drilling Fluids, Inc.)*

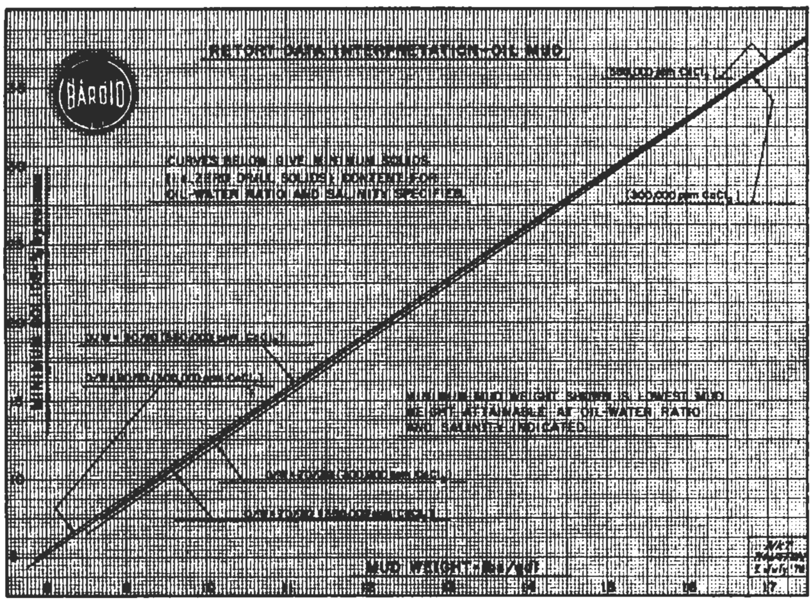


Figure 4-122. Minimum solids content in oil mud [26]. *(Courtesy Baroid Drilling Fluids, Inc.)*

Table 4-58
Drilling Fluids: General Trouble Shooting [25]

Problem	Symptoms	Treatment
1. *Mud Properties* Mud weight	Mud weight too low; low viscosity	Add weighting material and bentonite
	Mud weight too low; viscosity controlled	Add weighting material, bentonite and thinner
	Mud weight too high	Run mechanical solids removal equipment. Dilute with water or thin with thinner
Viscosity	Funnel viscosity too high; high PV; high gels and high solids	Run mechanical solids removal equipment, dilute with water
	High funnel viscosity, high YP, high gels; normal PV and solids	Add thinner
Filtration	Fluid loss too high, viscosity can be increased	Treatment with filtration control agent
	Fluid loss too high, viscosity controlled	Treatment with thinner and filtration control agent
	Fluid loss too high; thick, soft filter cake; low Methylene Blue Test. Treatment with filtration agent does not help	Add bentonite to the system
Mud foaming	Foam on surface at mud pits. No reduction in mud weight	Spray water or diesel over the pit surface, use defoamer. Check surface system for air entraintment.
	Reduction in mud weight. Increased funnel viscosity. Pump pressure drops. Internal foam	Use defoamer. Thin the mud to reduce yield point. Consult mud service engineer.
2. *Contaminations* Drilled solids	High viscosity and gel strength, slow drilling rate, chemical treatment ineffective	Dilute with water, make use of solids removal equipment. Displace mud.
Cement	High viscosity, gel strength, increase in pH, water loss and filtrate calcium.	Chemical treatment: (1) Bi-carb $NaHCO_3$, (2) Thinner. If large concentration of Ca ions-change to an inhibitive mud
Gipsum or anhydrite	High viscosity, high flat gel strength; increased water loss, filtrate calcium and sulfate	Chemical treatment; soda ash Na_2CO_3 (0.02 ppb at soda ash for every epm of hardness) for drilling massive anhydrite change to gip mud.

Problem	Symptoms	Treatment
Salt rock	High viscosity, high gel strength increased water loss, filtrate chlorides	Chemical treatment: (1) thinners—to reduce apparent viscosity, gel strength and yield point; (2) caustic soda—to adjust pH; (3) CMC, filtration control agent. If massive salt is to be drilled—convert to saturated salt water mud.
Salt water	Same as salt rock; increase in pit volume; reduction in mud	Weight-up to overcome salt water flow. Chemical treatment as for salt rock.
Overtreatment with soda ash or calcium bicarbonate	Excessive viscosity and yield point. YP cannot be lowered with thinner. Increase in 10 min gel strength. Methyl Orange alkalinity $M_f > 5$	Run alkalinity test and calculate CO_3 and HCO_3 ions concentration. Calculate lime required: • to remove CO lime, ppb = 0.0013 × epm CO_3 • to remove HCO_3 lime, ppb = 0.026 × epm HCO_3
3. *High Pressure Zones*		
Gas/water influx	Increase in pit volume. Gas or salt water cut mud. Mud flows when pumps are shut off.	Shut in well. Record drill pipe and casing pressure. Circulate out gas or water influx and separate on surface. Calculate mud weight necessary to balance formation pressure. Kill the well.
Gas cutting	Normally shows up as gas-cut mud after trips. If encountered while drilling, it is usually accompanied by rapid change in filtrate chlorides.	Add weighting material to rise density. Thin mud with water and thinners. Use degasser to clear gas from mud. Continue to circulate and avoid use of blowout preventers if possible.
4. *Lost circulation*		Keep mud weight as low as possible. Maintain minimum flow resistance of mud. Consider air drilling or foam drilling.

Table 4-58 (continued)

Problem	Symptoms	Treatment
• to porous formations	A very rapid seepage loss	"Gunk" squeeze or a high filtration squeeze
• to permeable sandstones	Gradual seepage loss	1. Increase colloidal content of mud 2. Add granular or flake LCM If the loss zone is located—a "soft plug" to be spotted.
• to cavernous formations	Immediate and complete loss	"Blind" drilling (no returns); then set casing or cement.
• to fractured formations	Sudden but not total loss of fluid. It occurs often after trips (induced fractures) when heavy mud is in use. The loss zone is frequently below the deepest casing shoe.	Apply cement squeeze or attapulgite gel-barite squeeze.
5. *Pipe stuck*		
Differential sticking	The drilling string got stuck after remaining motionless in the hole circulation can be broken and continued at normal pressure. Permeable formation is exposed above the bit. The borehole is clean and in good condition.	 1. Try spotting fluids 2. Wash over
Key-seat	see "Diagnosis of stuck pipe"	
Undergauge hole	as above	
Particles in the hole	as above	
6. *Hole instability*	While drilling: tourgue, drag, difficulty with making connections, bridging, fill on bottom, stuck pipe, presence of slaughing material in cuttings. After drilling: caliper log shows caving.	Control mud weight to counter balance pore pressure. Keep fluid loss as low as possible. Keep viscosity and gel strength low to prevent swabbing. Convert to: 1) salt-polymer mud, or 2) potassium system, or 3) salinity controlled oil mud.
7. *Corrosion*		
• in water base muds	Internal and external pitting	1. Keep pH above 10. 2. Use cationic type inhibition. 3. Identify type of corrosion. 4. Add specific corrosion inhibition.

Problem	Symptoms	Treatment
• in aerated muds	Severe pitting, black to red rust	1. Keep pH above 11 with caustic soda on line; 2. Use cationic-type inhibition. 3. Identify type of corrosion contaminant. 4. Treat with specific corrosion inhibition.
8. *High temperature*		
High temperature gelatin	Difficult to start circulation. High viscosity and gel strength of mud off bottom. Deceased alkalinity and increased water loss.	Dilute with water and add bentonite. Treat with thinner. Spot a slurry of mud treated with 1–2 ppb sodium chromate in the high-temperature section of the hole.

Adapted from IADC Drilling Manual, 10th edition, 1982; Courtesy IADC.

(text continued from page 695)

Completion and Workover Fluids

Completion and workover fluids are those placed against the formation while killing well, cleaning out, plugging back, stimulating, or perforating. Their primary functions are (1) to transfer treating fluid to a particular zone in the borehole, (2) to protect the producing formation from damage, (3) to control the well pressure during servicing operations, (4) to clean the well, and (5) to displace other fluids or cement.

Design Considerations for Completion/Workover Fluids

While designing completion/workover fluids the main consideration is given to the effect of the fluids on well's productivity. Low production rates can be due to factors that are unrelated to the fluids introduced to the production zone. These would include poor or shallow perforations, cement filtrate invasion, paraffin wax deposition from crude oil, or movement of formation sand to block the well-bore.

Productivity damage attributable to drilling or completion fluids results from three mechanisms:

- Particulate invasion which blocks the formation pores.
- Filtercake can fill up and plug large cracks, fractures or perforations. This is difficult to remove by flowing the well or acidisation.
- Filtrate invasion can interact in various ways with solids or liquids in the pores to cause a reduction in flow.

Table 4-59
Drilling Fluids: Problems in Surface Drilling

FORMATIONS	PROBLEMS	CONTROL
UNCONSOLIDATED SANDS AND GRAVELS	Caving.	Adequate gel strength to consolidate loose sand. Maintain filtration rate below 15-cc API to prevent hydrous disintegration of shales. Prevent erosion by adequate viscosity and gel strength. Maintain sufficient viscosity for lifting cuttings and cavings.
	Lost circulation.	Maintain colloidal content to provide good filtration properties, low density, and thick mud. Viscosity pill and lost circulation materials. Care in use of mechanical equipment.
	Retention of sand in mud. (Weight build up. Sand retained in mud will increase density, which may aggravate tendency towards lost circulation).	Low enough viscosity and gels to allow sand to drop out in ditch and pits by reducing viscosity and gels with water dilution and chemical treatment. Ditch and pit arrangement may be improved to promote sand settling. Mechanical sand and shale separators or centrifugation may aid.
CLAYS AND SHALES	Excessive viscosity and gels, tight hole, sloughing, hydrous disintegration.	Reduce viscosity and gels; reduce filtration rate to prevent hydrous disintegration or sloughing.
HARD ROCK	Removal of cuttings from hole.	Maintain colloidal content high enough to provide gels and viscosities adequate to remove cuttings or prevent settling. Maintain sufficient annular velocity.
CHARGED SAND	Pressure control.	Weight mud to give hydrostatic pressure above formation pressure. Maintain low viscosity and gels for gas removal. Keep hole full at all times.
	Cement contamination.	If saving mud, thin with water and chemicals. If not saving mud, discard contaminated mud and mix fresh mud.

Table 4-60
Drilling Fluids: Problems in Intermediate Drilling

FORMATIONS	PROBLEMS	CONTROL
SHALES	Sloughing, caving, heaving.	Employ mud with low filtration rate, low viscosity and gels. Utilize high concentration of colloidal material and thin with chemicals. Use inhibited mud.
	Hydration, swelling, and dispersion in presence of conventional filtrates.	Maintain minimum filtration rate. If possible use chemicals instead of water to thin mud. Repress hydration by presence of calcium ion. Use oil-base or oil-base emulsion.
	Internal high pressure.	Maintain adequate mud density.
	Swabbing action of bit withdrawal.	Maintain thin filter cake, low gels. Withdraw bit slowly. Check to see that hole is taking mud through the annulus.
	Lubricating effect of filtrate or mud on shale plane.	Maintain minimum filtration rate. Utilize lost circulation materials to keep mud in hole.
	Balled bit.	Employ chemical thinners and water dilution if clay solids are high.
	Overtreatment with chemical thinners. Mud does not respond.	Change type of chemical thinners. Change type of mud. Add water. Add new mud.
	Weight build up with formation solids.	Dilute with water. Centrifuge mud. Convert to inhibited mud.
	Chemically reactive shales, making use of high pH muds difficult.	Employ low or medium pH muds. Convert to inhibited mud.
	Increase in filtration rate.	Add colloidal material. Check for contaminants and if present chemically treat.
	Mud-making resulting in Viscosity and gel build up.	Dilute with water. Use chemical thinners. Use inhibited mud.
SALT OR SALT WATER BEARING FORMATIONS	Flocculation of drilling mud. Corrosion of drill pipe. Pressure from salt water intrusion.	Prevent or overcome flocculation by use of salt-resistant materials and chemical treatments resulting in salt tolerant muds. Employ salt-resistant muds. Maintain adequate mud density. Add corrosion inhibitor.
SILTSTONE	Weight build up.	Thin with water or chemicals to release sand by reduction in viscosity. Maintain adequate colloids. Centrifuge mud.
SAND	Low-pressure circulation losses.	Employ as light a mud as consistent with safety. Use lost circulation materials if necessary. Check operation of mechanical equipment: spudding, balling, pump surges, etc. Spot-treat with lost circulation materials.
	Cake build up-tight hole. Differential sticking (wall sticking).	Add colloidal materiel to reduce cake thickness. Add surfactants.
	High filtration rate.	Add colloidal material to reduce filtrate loss.
ANHYDRITE AND GYPSUM	Flocculation of drilling mud. High flat gels. High filtration rate. High viscosity.	Chemically treat to counteract gypsum and anhydrite flocculation. Employ mud tolerant to gypsum and anhydrite contamination.
LIMESTONE	Lost circulation.	Add colloidal material. Employ lost circulation materials. Drilling with water may be feasible. Drilling with air or gas may be feasible. Check operation of mechanical equipment. Do not generate fluid pressure surges by spudding, going in hole rapidly, or starting pump rapidly.
HARD ROCK (Schist, Igneous, etc.)	Transportation of cuttings.	Increase viscosity and gels by addition of colloidal materials.
FRACTURED FORMATIONS AND CONGLOMERATES	Lost circulation.	Increase viscosity and gels. Employ lost circulation material. Use lowest density mud possible. Drilling with air or gas may be feasible. Check operation of mechanical equipment. Prevent fluid pressure surges.
MISCELLANEOUS SALT-BEARING FORMATIONS Magnesium and calcium chloride. Magnesium and calcium sulfate and oil.	Flocculation of drilling mud. Corrosion of drill pipe.	Employ drilling mud not susceptible to flocculation. Chemical treatments resulting in salt tolerant muds. Add corrosion inhibitor such as X-COR.
GAS AND OIL BEARING FORMATIONS	Blowouts. Gas-cut mud.	Add weight materials to raise density of drilling mud. Thin mud with water and thinners to permit release of entrained gas. Increase gas removal efficiency in ditches and pits: by changing design; efficient stirring, degassing riffles and vacuum separators.
	Protection of formations.	Lower filtration rate and cake thickness. Use inverted emulsion mud. Use oil base mud.

The invasion of particles can be eliminated either by using solids-free systems or by formation of a competent filter cake on the rock surface. If the components forming the filter cake are correctly chosen and blended, they will form a very effective "downhole" filter element. This ensures that colloidal sized clays or polymeric materials are retained within the filter cake and do not enter the formation. Further protection is provided by ensuring that a thin filter cake is formed due to low dynamic and static filtrate losses. Thus, the cake may be easily removed when the well is brought into production. Additionally, the filter cake can be soluble in acid or oil.

Table 4-61
Drilling Fluids: Problems in Production Drilling

FORMATIONS	PROBLEMS	CONTROL
SANDS Low pressure.	Water or mud blocking.	Minimum filtration rate water-base muds. Minimum filtration rate water-base emulsions. Miminum filtration rate oil-base emulsions. Oil-base muds. Inhibited muds. Minimum weight muds. Crude oil or diesel oil.
	Loss of crude or diesel oil used as completion fluid.	Add oil-soluble lost circulation material.
Normal Pressure.	Water or mud blocking. Screen plugging.	Minimum filtration rate muds. Thin friable filter cakes.
High Pressure.	Blowout prevention.	Maintain adequate mud density. Maintain hole full of mud to prevent reduced hydrostatic head resulting from short column of mud. Withdraw tools slowly to prevent swabbing action.
Limestones. Coral reef. Dolomite. Fractured shales. Conglomerates and schists. Shales.	Lost Circulation. Formation Protection.	 Maintain low gels and thin filter cake.

SURFACE DRILLING—Includes muds used before setting casing. Starting muds should maintain a clean, competent hole at reasonable cost. The pump volume should be controlled to remove cuttings without washing out formations.
INTERMEDIATE DRILLING—Includes muds used for drilling between surface casing and completion. Accounts for greatest footage and fastest drilling time. Mud should speed drilling by removing cuttings rapidly and by preventing troubles, thus extending the "straight drilling" footage and allowing bit to stay on bottom for maximum periods. Mechanical considerations include proper drilling weights for drill stem design; avoidance of drill pipe compression sections to reduce hole enlargements; shortening of bit runs where drill pipe in tension might cut wall of long shale sections. COMPLETING—Includes muds to provide a clean, full-gauge hole into the producing formation, with a maximum of safety and a minimum of interference with production.

The filtercake plugging of perforations or fractures is usually difficult to remove through acidizing or backflowing. The solutions are:

- use of solids-free brine
- use of bridging solids that are acid/oil soluble
- use of commercial bridging materials (large fractures or perforations)

The compatibility of invading fluids with pay zone rocks may relate to swelling clays, water blocking, or emulsion blocking. In many sandstone reservoirs there are agglomerations of clay minerals and other fine formation particles are in equilibrium with the pore fluids. If the existing brine is displaced with a lower salinity fluid from the completion fluid, swelling clays such as montmorillonite or some illites can expand, and non-swelling clays such as kaolinite can disperse. The swelling and disaggregation can lead to a blocking of the pores.

In the water-blocking mechanism large volumes of invaded liquid may be retained by low permeability or low-pressure formations. The blocking may occur for an oil wet and a water wet sandstones.

The design factors to prevent blocking involve the use of low-viscosity fluids with minimum interfacial tension, minimum capillary pressure, and minimal fluid loss.

The emulsion blocking mechanism involves formation of emulsion in the pores either by self-emulsification of water-based filtrate with the crude oil, or oil filtrate from an oil-based fluid emulsifying formation water. The emulsions are viscous and can block the pores. The remedial design is to prevent emulsification either by eliminating oil from completion fluid or by the use of demulsifiers.

Components in the invading water-based filtrate and in the formation waters may react to form insoluble precipitates which can block the pores and give rise to skin damage. The scale can be formed by interaction of calcium-based brines with carbon dioxide or sulfate ions in the formation water. Alternatively sulfate ions in the invading fluid may react with calcium or barium ions in the formation water. Analysis of the formation water can identify whether such a problem may arise.

Table 4-62 contains a checklist for proper selection of completion/workover fluids.

Completion/Workover Fluid Systems

Selection of completion/workover fluid system is entirely dependent upon its function, which, in turn, depends on the completion method. The method may involve underreaming, gravel packing, perforation, or workover. Completion fluids used for underreaming have to display formation bridging and low spurt loss and filtrate loss to support the sand and prevent sloughing. Because the filter cake will be trapped between the gravel pack and the formation, the fluid should be composed of particles, soluble in acid or oil, and small enough not to bridge off the gravel pack when the well is flowed.

Gravel packing completion fluids should exhibit sufficient viscosity to carry and place the gravel efficiently. However, high gel strengths for prolonged suspension are not necessary. Thus the polymer solution can easily flow out of the pack on production. Also, the solution can be formulated with a breaker (enzyme or oxidizer) such that the viscosity is completely broken allowing complete cleanup. Normally, filtrate loss control is not employed in the gravel carrying fluid.

Low-density perforating completion fluids for underbalanced perforation greatly reduce the possibilities of plugging. If overbalance perforation is needed,

Table 4-62
Checklist of Completion and Workover Fluids Considerations [26]

Factor Considered	Completion and Workover Fluid Considered
1. Mechanical	
Annular velocity	Higher annular velocity-low viscosity system or low annular velocity-higher viscosity systems can be selected. Annular velocity can be substituted for viscosity in lifting particle. Annular velocity at 150 ft/min should be sufficient for borehole cleaning with 1 cp viscosity clear salt water.
Mixing facilities	If mixing facilities are poor to produce adequate shear, the completion/workover fluid should be prepared and maintained with very small amounts of material.
Annular space	The size of bottom hole equipment (liners, packers, etc.) reduces annular space and increases pressure losses. The fluid must maintain rheological properties which reduce pressure losses.
Circulation frequency	In the completion and workover operations, there are long periods when fluid in the hole is not circulated. Fluid suspension and thermal stability should be determined in order to evaluate the necessary circulation frequency.
Corrosion	Some workover fluids can produce high corrosion rates. Corrosion control can be accomplished through H control, inhibitors or bactericides. The practical corrosivity limit is 0.05 lb/ft^2 per operation.
Fluid components	Solubility at fluid components at the well bore conditions (pressure and temperature) should be considered. Glazing at jet and bullet tracks should not occur while perforating.
2. Formation	
Permeability damage	Fluid solids should be kept as low as possible. Fluid should not contain solids larger than two microns in size unless bridging material.
Formation pressure	Density control with calcium carbonate, iron carbonate, barium carbonate, ferric oxide.
Clay content	Fluid inhibition with electrolyte additive.
Vugular formation	In order to prevent "seepage loss" of circulation to the vugular formation, bridging the formation—by properly sized, acid-soluble on oil-soluble resin particles as well as colloidal particles—should be considered.
Formation sensitivity	Formations can be oil wet or water wet. The fluid filtrate depends on what is the continuous phase of the completion fluid. Thus the formation wettability can be reduced by wettability charge. This effect can be controlled either by proper fluid selection or by treatment with water wetting additives.
Temperature	In high temperature wells, the temperature degradation of polymens should be considered.

Factor Considered	Completion and Workover Fluid Considered
3. Fluid properties	
Density	Ideal requirements: Fluid density should not be greater than that which balances formation pressure. Practical recommendation: Differential pressure should not exceed 100–200 psi.
Solids content	Ideal requirements: no solids in completion and workover fluids. Practical recommendation: Solids smaller than two microns can be tolerated as well as the bridging solids. The bridging solids should be: (1) greater than one-half of the average fracture diameter; (2) readily flushed from the hole; acid or solvent soluble.
Fluid loss	Ideal requirements: no fluid loss. Practical recommendations: Fluid loss to the formation can be controlled by: (1) fluid-loss agents or vixcositiers such as polymens, calcium carbonate, gilsonit, asphalt etc., (2) bridging materials.
Rheology	Ideal requirements: low viscosity with the yield point and gels necessary for hole cleaning and solids suspension. Practical recommendation: A compromise should be found to minimize pressure losses and bring sand or cutting to the surface at reasonable circulating rate.

Courtesy Baroid Drilling Fluids, Inc.

low damaging fluids are recommended and often a solids-free fluid is preferred. This is because filter cake from a high solids fluid can completely fill a perforation and be difficult to remove on back flowing or acidizing.

Perforating under diesel is sometimes employed. In this case, care is necessary to ensure that good displacement of the previous denser fluid occurs, and the completed zone remains in contact with the diesel without density swapping.

Perforating fluids used may be filtered clear brine or $CaCO_3$ type completion fluids, oil, seawater, acetic acid, gas or mud.

Where large losses to the formation are probable, perforation under slugs containing degradable bridging and loss control materials is advised. At least, such materials should be on hand should the need arise. Under these conditions, it is far better to fill perforations with good, degradable, bridging material than the common mixture of iron and rust particles, mud solids, and excess pipe dope. These foreign solids may be also not exhibit bridging and be injected into the rock around the perforations, causing irreparable damage.

Clear Brines. Brine solutions are made from formation saltwater, seawater, or bay water, as well as from prepared saltwater. They do not contain viscosifers or weighting materials. Formation water-base fluids should be treated for emulsion formation and for wettability problems. They should be checked on location to ensure that they do not form a stable emulsion with the reservoir oil, and that they do not oil or wet the reservoir rock. The usual treatment includes a small amount (0.1%) of the proper surfactant.

Seawater or bay water base completion fluids should be treated with bactericides to inhibit bacterial growth. Since these fluids usually contain clays, inhibition with NaCl or KCl may be necessary to prevent plugging of the producing formation.

Prepared saltwater completion fluids are made of fresh surface water, with sufficient salts added to produce the proper salt concentration. Usually, the addition of 5 to 10% NaCl, 2% $CaCl_2$, or 2% KCl is considered satisfactory for clay inhibition in most formations. Sodium chloride solutions have been extensively used for many years as completion fluids; these brines have densities up to 10 lb/gal. Calcium chloride solutions may have densities up to 11.7 lb/gal. The limitations of $CaCl_2$ solutions are (1) flocculation of certain clays, causing permeability reduction, and (2) high pH (10 to 10.5) that may accelerate formation clays dispersion. In such cases, CaCl2-based completion fluids should be replaced with potassium chloride solutions. Other clear brines can be formulated using various salts over wide range of densities, as shown in Figure 4-123 [28].

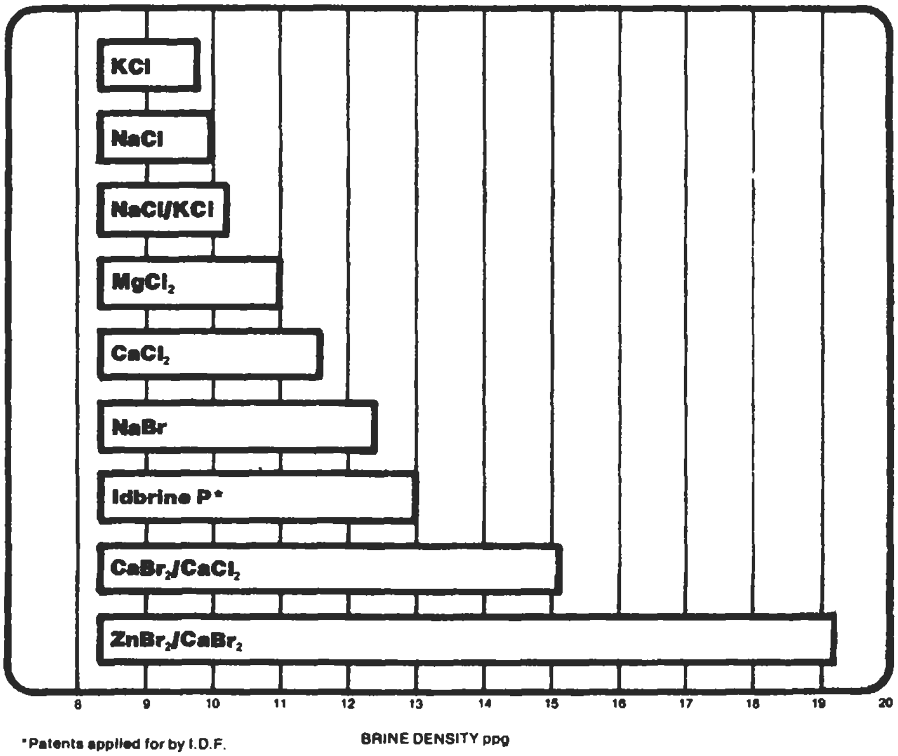


Figure 4-123. Salts used in clear brine completion fluids of various densities [28]. *(Courtesy International Drilling Fluids, Inc.)*

Material requirements for brine solutions are given in Tables 4-63 through 4-65.

Brine-polymer systems are composed of water-salt solutions with polymers added as viscosifers or filtration control agents. If fluid loss control is desired, bridging material must be added to build a stable, low permeability bridge that will prevent colloidal partial movement into the formation.

The polymers used for completion and workover fluids may be either natural or synthetic polymers. Guar gum is a natural polymer that swells on contact with water and thus provides viscosity and filtration control; it is used in concentrations of 1 to 3 lb/bbl. Guar gum forms a filter cake that may create

Table 4-63
Material Requirements for Preparing Sodium Chloride Salt Solutions (60°F)

Density lb/gal	Fresh Water (gal/final bbl)	Sodium Chloride (lb/final bbl)
8.33	42	0
8.6	41.2	16
8.8	40.5	28
9.0	40.0	41
9.2	39.5	54
9.4	39.0	68
9.6	38.5	82
9.8	38.0	95
10.0	37.5	110

Based on 100% purity.

Table 4-64
Material Requirements for Preparing Calcium Chloride Solutions (60°F)

Density lb/gal	Fresh Water (gal/final bbl)	Calcium Chloride (lb/final bbl)
10.0	39.0	95
10.2	38.5	107
10.4	38.0	120
10.6	37.5	132
10.8	37.0	145
11.0	36.5	157
11.2	36.0	170
11.4	35.5	185
11.6	35.0	197
11.8	34.0	210

Based on 95% chloride.

Table 4-65
Material Requirements for KCl Solutions (60°F)

Density ppg	Fresh Water gal/bbl final	Potassium Chloride lb/bbl final
8.42	41.7	7.0
8.64	41.0	21.1
8.86	40.2	35.2
9.09	39.4	53.6
9.32	38.6	70.5
9.56	37.6	88.2
9.78	36.7	105.0

problems for squeeze cementing, but is removed with production and increasing temperatures.

Starch is also used for fluid loss control. It does not provide carrying capacity; therefore other polymers are required. Although starch is relatively cheap, it has two serious limitations: (1) starch is subject to fermentation, and (2) it causes significant permeability reduction due to plugging.

The synthetic polymers commonly used in completion fluids are HEC and Xanthan gum (XC Polymer). Xanthan gum is a biopolymer that provides good rheological properties and that is completely soluble in HCl. HEC-hydroxyethyl cellulose is currently the best viscosifer. It gives good carrying capacity, fluid loss control, and rheology; it is completely removable with hydrochloric acid. The effect of HCl on the restored permeability for HEC completion fluid is shown in Figure 4-124 and Table 4-68 [36]. It can be noticed that 100% of the original core permeability was restored by displacing acid-broken HEC with brine. The comparison of permeability damage caused by different polymers is given in Table 4-69 [36].

The bridging materials commonly used in completion and workover fluids are ground calcium carbonate, gilsonite, and asphalt. These materials should demonstrate uniform particle size distribution and be removable by acid or by backflow. Their mesh size should enable them to flush through the gravel pack; a mesh size of 200 is considered satisfactory for most completions. Calcium carbonate bridging materials are completely soluble in hydrochloric acid. Resins give effective bridging; they are soluble in oil solutions (2% by volume oil).

A typical formulation of a brine-polymer completion fluid might include 8.5 to 11 lb/gal salt water solution (NaCl, $CaCl_2$, KCl, or a mixture), 0.25 to 1.0 lb/bbl polymer and 5 to 15% calcium carbonate.

Density control in brine-polymer systems can be achieved with salt solutions or with weighting materials. When mixing heavy brine completion fluids, the following factors should be considered:

1. Cost–heavy brines are very expensive.
2. Downhole temperature effect on the brine density–Table 4-69 [26].
3. Crystallization temperature–Figure 4-125 [37].
4. Corrosion–various salts have different acidities (pH of brine can be controlled with lime, caustic soda, or calcium bicarbonate).
5. Safety–burns from heat generated while mixing and skin damage should be prevented.
6. Toxicity–dispersal cost depends on type of salt and concentration.

Table 4-66
Mixing Chart for Zinc Bromide/Calcium Bromide Solution Blend

13.7 lb/gal $CaBr_2/CaCl_2$ + 19.2 lb/gal $ZnBr_2/CaBr_2$

Desired Brine Density lb/gal	Barrels 13.7 lb/gal $CaBr_2/CaCl_2$	Barrels 19.2 lb/gal $ZnBr_2/CaBr_2$	Crystallization Point (F)	(C)
15.0	0.7636	0.2364	46	7
15.1	0.7454	0.2546	43	6
15.2	0.7273	0.2727	40	4
15.3	0.7091	0.2909	38	3
15.4	0.6909	0.3091	36	2
15.5	0.6727	0.3273	34	1
15.6	0.6545	0.3455	32	± 0
15.7	0.6364	0.3636	30	− 1
15.8	0.6182	0.3818	28	− 2
15.9	0.6000	0.4000	25	− 3
16.0	0.5818	0.4182	22	− 5
16.1	0.5636	0.4364	19	− 7
16.2	0.5454	0.4546	16	− 8
16.3	0.5273	0.4727	13	−10
16.4	0.5091	0.4909	9	−12
16.5	0.4909	0.5091	3	−16
16.6	0.4727	0.5273	4	−15
16.7	0.4546	0.5454	9	−12
16.8	0.4364	0.5636	14	−10
16.9	0.4182	0.5818	19	− 7
17.0	0.4000	0.6000	23	− 5
17.1	0.3818	0.6182	23	− 5
17.2	0.3636	0.6364	23	− 5
17.3	0.3455	0.6545	24	− 4
17.4	0.3273	0.6727	24	− 4
17.5	0.3091	0.6909	25	− 3
17.6	0.2909	0.7091	25	− 3
17.7	0.2727	0.7273	25	− 3
17.8	0.2546	0.7454	26	− 3
17.9	0.2364	0.7636	26	− 3
18.0	0.2182	0.7818	27	− 2
18.1	0.2000	0.8000	27	− 2
18.2	0.1818	0.8182	27	− 2
18.3	0.1636	0.8364	25	− 3
18.4	0.1455	0.8546	25	− 3
18.5	0.1273	0.8727	25	− 3
18.6	0.1091	0.8909	22	− 5
18.7	0.0909	0.9091	21	− 6
18.8	0.0727	0.9273	21	− 6
18.9	0.0545	0.9455	20	− 6
19.0	0.0364	0.9636	19	− 7
19.1	0.0182	0.9818	17	− 8
19.2	0.0000	0.1000	16	− 8

Courtesy Halliburton Co.

Table 4-67
Mixing Chart for Heavy Brines Using Calcium Bromide and Calcium Chloride Brines and Calcium Chloride Pellets

Brine Density Desired	Barrels 14.2 lb/gal $CaBr_2$ Brine	Barrels 11.6 lb/gal 38% $CaCl_2$ Brine	Pounds Calcium Chloride Pellets	Crystallization Point (F)	Crystallization Point (C)
11.7	.0254	.9714	2.86	50	10
11.8	.0507	.9429	6.06	52	11.1
11.9	.0762	.9143	9.09	53	11.6
12.0	.1016	.8857	12.13	54	12.2
12.1	.1269	.8572	15.15	55	12.7
12.2	.1524	.8286	18.18	56	13.3
12.3	.1778	.8000	21.22	56.5	13.6
12.4	.2032	.7715	24.24	57	13.8
12.5	.2286	.7429	27.28	57.5	14.1
12.6	.2540	.7143	30.31	58	14.4
12.7	.2794	.6857	33.34	58.5	14.7
12.8	.3048	.6572	36.37	59	15.0
12.9	.3302	.6286	39.41	59.5	15.2
13.0	.3556	.6000	42.44	60	15.5
13.1	.3810	.5714	45.47	60	15.5
13.2	.4064	.5429	48.49	60.5	15.8
13.3	.4318	.5143	51.53	61	16.1
13.4	.4572	.4857	54.56	61	16.1
13.5	.4826	.4572	57.59	61.5	16.3
13.6	.5080	.4286	60.62	61.5	16.3
13.7	.5334	.4000	63.66	61.5	16.3
13.8	.5589	.3714	66.69	61.5	16.3
13.9	5842	.3429	69.72	61.5	16.3
14.0	.6069	.3143	72.75	62	16.6
14.1	.6351	.2857	75.78	62	16.6
14.2	.6604	.2572	78.81	62	16.6
14.3	.6858	.2286	81.84	62	16.6
14.4	.7113	.2000	84.88	62.5	16.9
14.5	.7366	.1715	87.90	63	17.0
14.6	.7620	.1429	90.94	63.5	17.5
14.7	.7875	.1143	93.97	64	17.7
14.8	.8128	.0858	96.99	65	18.3
14.9	.8382	.0572	100.03	66	18.8
15.0	.8637	.0286	103.06	67	19.4
15.1	.8891	.0000	106.10	68	20.0

Courtesy Halliburton Co.

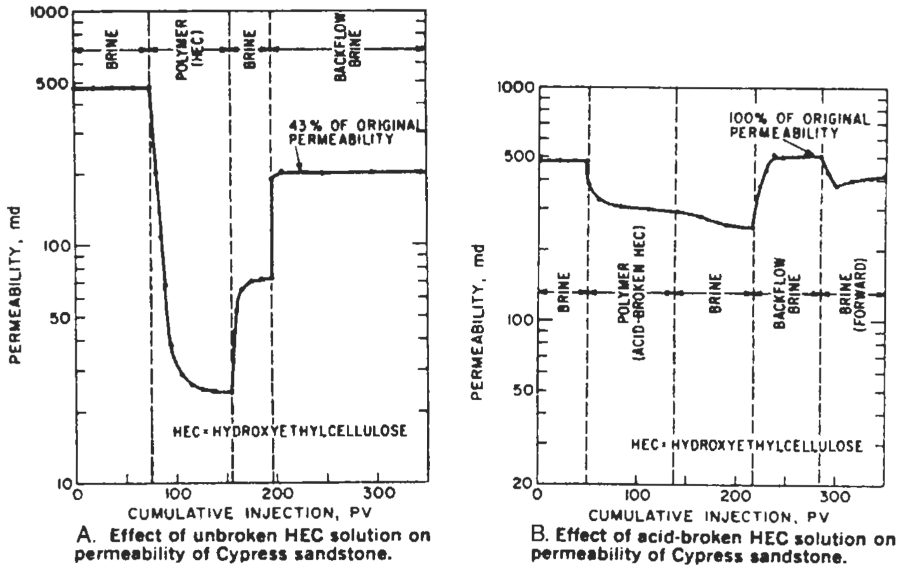


Figure 4-124. Effect of HCl on HEC on permeability damage caused by completion fluid [36]. *(Courtesy SPE.)*

Table 4-68
Effect of Polymers on Core Permeability [36]

Core Type	Polymer Solution (in 50-percent brine)	Percent of Original Perme-ability to Brine* (forward)	Percent of Original Perme-ability to Brine* (reverse)
2,400-md Cypress sand	0.3% Polyoxyethylene	100	100
740-md Cypress sand	0.3% Polyoxyethylene	100	100
230-md Berea sand	0.3% Polyoxyethylene	100	100
4-md Bedford lime	0.3% Polyoxyethylene	100	100
740-md Cypress sand	0.4% HEC	15	43
740-md Cypress sand	0.4% HEC acid	76	92
740-md Cypress sand	0.4% guar gum	1	25
230-md Berea sand	0.2% guar gum	17	30
4-md Bedford lime	0.1% guar gum	6	15
740-md Cypress sand	0.4% guar gum + enzyme breaker	10	54

*Permeability to 5-percent brine measured after resaturating core with brine for a period ranging from 2 to 24 hours following the polymer flood.
Courtesy SPE.

Table 4-69
Fluid Density Adjustment for Downhole Temperature Effect [26]

Surface-measured density		Loss in density per 100°F rise in average circulating temperature above surface-measured temperature	
lb/gal	sp gr	lb/gal	sp gr
8.5	1.020	0.35	0.042
9	1.080	0.29	0.035
10	1.201	0.26	0.031
11	1.321	0.23	0.028
12	1.441	0.20	0.024
13	1.561	0.16	0.019
14	1.681	0.13	0.016
15	1.801	0.12	0.014

Courtesy Baroid Drilling Fluids, Inc.

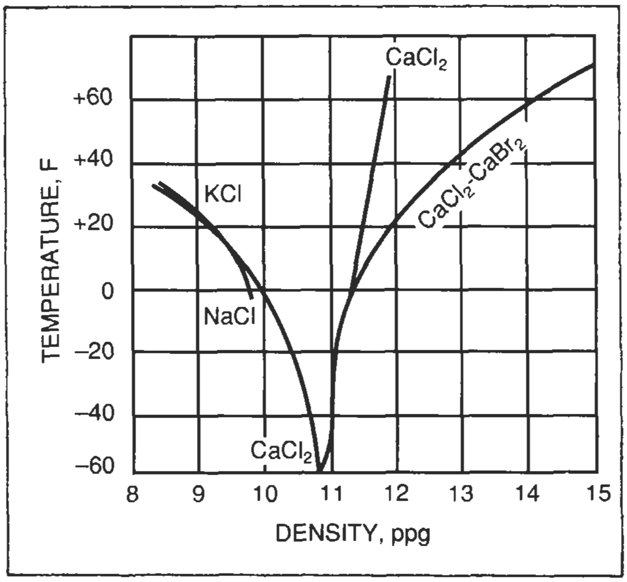


Figure 4-125. Crystallization temperature of various brines [37].

Weight materials commonly used in completion fluids are given in Table 4-70. Although they add solid particles to the fluid, their use can be economical where densities exceed 11.0 lb/bbl.

Oil-Base Systems. Oil-base completion and workover fluids contain oil as the continuous phase. Their application is limited by their density to formations with

Table 4-70
Density Increase with Weighting Materials

Material	Specific Gravity	Practical Weight Increase, lb/gal
$CaCO_3$	2.7	3.5
$FeCO_3$	3.85	6.5
$BaCO_3$	4.43	8.0
Fe_2CO_3	5.24	10.0

pressure gradients lower than 0.433 psi/ft (freshwater gradient). Weighted oil muds cannot be used for completion operations since they form mud plugs while perforating. Most of the oil-base systems contain asphalt that can plug the formation and reverse its wetting characteristic (from water-wet to oil-wet). Moreover, oil-based fluids are expensive. Their use can be justified. However, in oil-producing formations where water base fluids would cause serious permeability damage due to clay problems, or high-condensate gas wells oil-base fluids are feasible. In these formations the produced fluids will clean up the oil filtrate.

Foam Systems. The preparation, composition, and maintenance of foam completion and workover fluids is similar to that of foam drilling fluids. The advantage of foam is the combination of low density and high lifting capacity at moderate flow rates. The use of foam as a completion fluid can be justified by:

1. Low hydrostatic pressure (0.3 to 0.6 psi).
2. Circulation with returns where other fluids have no returns to the surface.
3. Easy identification of formation fluids.
4. No inorganic solids; other solids are discarded with the foam at the surface.
5. In wells with sand problems, faster operation and more complete sand removal.
6. The ability to clean out low pressure wells without killing them.

The limitations of foam are (1) operational complexity, (2) high cost, and (3) the pressure effect on foam consistency, i.e., below about 3,000 ft, foam com-presses to a near liquid form.

DRILL STRING: COMPOSITION AND DESIGN

The drill string is defined here as a drill pipe with tool joints and drill collars. The drill stem consists of the drill string and other components of the drilling assembly that includes the kelly, subs, stabilizers, reamers as well as shock absorbers, and junk baskets or drilling jars used in certain drilling conditions. The drill stem (1) transmits power by rotary motion from the surface to a rock bit, (2) conveys drilling fluid to the rock bit, (3) produces the weight on bit for efficient rock destruction by the bit, and (4) provides control of borehole direction.

The drill pipe itself can be used for formation evaluation (Drill Stem Testing—DST), well stimulation (fracturing, acidizing), and fishing operations.

Therefore, the drill string is a fundamental part, perhaps one of the most important parts, of any drilling activity.

The schematic, typical arrangement of a drill stem is shown in Figure 4-126.

Drill Collar

The term "drill collar" originally derives from the short sub originally used to connect the bit to the drill pipe. Modern drill collars are each about 30 ft in length, and the total length of the string of drill collars may range from about 100 to 700 ft and more.

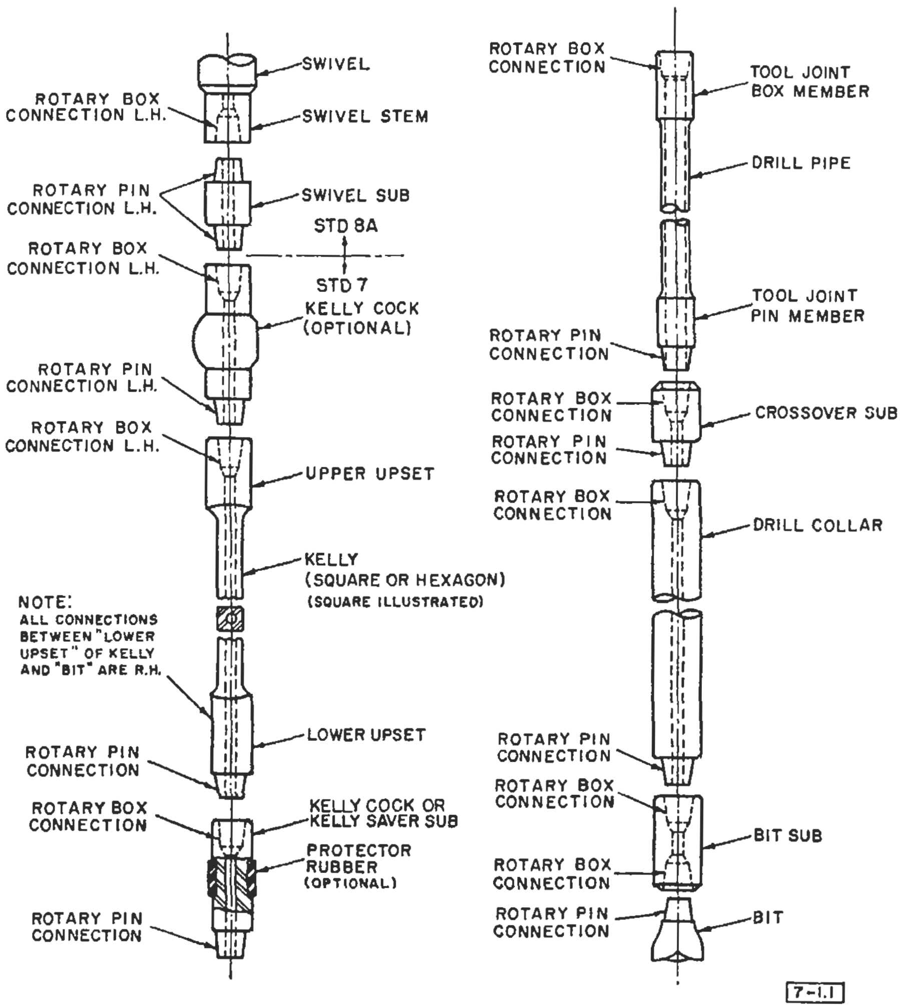


Figure 4-126. Typical drill-stem assembly [13].
Requirements on swivel and swivel sub connections are included in API spec 8A.

Basically, the purpose of drill collars is to furnish weight on bit. However, both size and length of drill collars have an effect on bit performance, hole deviation, and drill pipe service life. Drill collars may be classified according to the shape of their cross-sections as round drill collars (conventional drill collars), square drill collars, or spiral drill collars (drill collars with spiral grooves).

Square drill collars are used to increase the stiffness of the drill string and are recommended for drilling in crooked hole areas. The spiral type of drill collar is used for drilling formations in which the differential pressure can cause sticking of drill collars. The spiral grooves on the drill collar side reduce the area of contact between drill collar and wall, which considerably reduces the sticking force.

Conventional drill collars are made with uniform outside diameter and with slip and elevator recesses. Slip and elevator recesses are designed to reduce drill collar handling time while tripping by eliminating lift subs and safety clamps. However, the risk of drill collar failure for such a design is increased. The slip and elevator recesses may be used together or separately.

Dimensions, physical properties, and unit weight of new, conventional drill collars are specified in Tables 4-71, 4-72, and 4-73, respectively. Technical data on square and spiral drill collars are available from manufacturers.

Selecting Drill Collar Size

Selection of the proper outside and inside diameter of drill collars is usually a difficult task. Perhaps the best way to select drill collar size is to study results obtained from offset wells previously drilled under similar conditions.

The most important factors in selecting drill collar size are:

1. bit size
2. coupling diameter of the casing to be set in a hole
3. formation's tendency to produce sharp changes in hole deviation and direction
4. hydraulic program
5. possibility of washing over if the drill collar fails and is lost in the hole

To avoid an abrupt change in hole deviation (which may make it difficult or even impossible to run casing) when drilling in crooked hole areas with an unstabilized bit and drill collars, the required outside diameter of the drill collar placed right above the bit can be found from the following formula [38]:

$$D_{dc} = 2(\text{casing coupling OD}) - \text{bit OD} \tag{4-49}$$

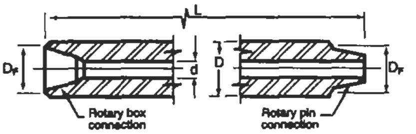
Example

The casing string for a certain well is to consist of $13\frac{3}{8}$-in. casing with coupling outside diameter of 14.375 in. Determine the required outside diameter of the drill collar in order to avoid possible problems with running casing if the borehole diameter is assumed to be $17\frac{1}{2}$ in.

$$D_{dc} = 2(14.375) - 17.5 = 11.15 \text{ in.}$$

Being aware of standardized drill collar sizes, an 11 or 12-in. drill collar should be selected. To avoid such large drill collar OD, a stabilizer or a proper-sized square drill collar (or a combination of the two) should be placed above the

Table 4-71
Drill Collars [13] (all dimensions in inches)



1 Drill Collar Number*	2 Outside Dia, D	3 Bore, $+\frac{1}{16}$ -0 d	4 Length, ft, ±6 in. L	5 Bevel Dia, $\pm\frac{1}{64}$ D_F	6 Bending Strength Ratio
NC23-31 (tentative)	*3 1/8*	*1 1/4*	*30*	*3*	*2.57:1*
NC26-35(2 3/8 IF)	3 1/2	1 1/2	30	3 17/64	2.42:1
NC31-41(2 7/8 IF)	4 1/8	2	30	3 61/64	2.43:1
NC35-47	4 3/4	2	30	4 33/64	2.58:1
NC38-50(3 1/2 IF)	5	2 1/4	30	4 49/64	2.38:1
NC44-60	6	2 1/4	30 or 31	5 11/16	2.49:1
NC44-60	6	2 13/16	30 or 31	5 11/16	2.84:1
NC44-62	6 1/4	2 1/4	30 or 31	5 7/8	2.91:1
NC46-62(4IF)	6 1/4	2 13/16	30 or 31	5 29/32	2.63:1
NC46-65(4IF)	6 1/2	2 1/4	30 or 31	6 3/32	2.76:1
NC46-65(4IF)	6 1/2	2 13/16	30 or 31	6 3/32	3.05:1
NC46-67(4IF)	6 3/4	2 1/4	30 or 31	6 9/32	3.18:1
NC50-70(4 1/2 IF)	7	2 1/4	30 or 31	6 23/32	2.54:1
NC50-70(4 1/2 IF)	7	2 13/16	30 or 31	6 23/32	2.73:1
NC50-72(4 1/2 IF)	7 1/4	2 13/16	30 or 31	6 15/16	3.12:1
NC56-77	7 3/4	2 13/16	30 or 31	7 19/32	2.70:1
NC56-80	8	2 13/16	30 or 31	7 45/64	3.02:1
6 5/8 REG	8 1/4	2 13/16	30 or 31	7 45/64	2.93:1
NC61-90	9	2 13/16	30 or 31	8 3/8	3.17:1
7 5/8 REG	9 1/2	3	30 or 31	8 27/32	2.81:1
NC70-97	9 3/4	3	30 or 31	9 5/32	2.57:1
NC70-100	10	3	30 or 31	9 31/64	2.81:1
NC77-110 (tentative)	*11*	*3*	*30 or 13*	*10 [illegible]*	*2.78:1*

* The drill collar number (Col. 1) consists of two parts separated by a hyphen. The first part is the connection number in the NC style. The second part, consisting of 2 (or 3) digits, indicates the drill collar outside diameter in units and tenths of inches. The connections shown in parentheses in Col. 1 are not a part of the drill collar number; they indicate interchangeability of drill collars made with the standard (NC) connections as shown. If the connections shown in parentheses in column 1 are made with the V-0.038R thread from the connections and drill collars are identical with those in the NC style. Drill collars with 8 1/4 and 9 1/2 inches outside diameters are shown with 6-5/8 and 7-5/8 REG connections, since there are no NC connections in the recommended bending strength ratio range.

Table 4-72
Physical Properties and Tests—New Drill Collars [13]

1 Drill Collar OD Range, inches	2 Minimum Yield Strength, psi	3 Minimum Tensile Strength, psi	4 Elongation, Minimum, With Gage Length Four Times Diameter, percent
3 1/8 thru 6 7/8	110,000	140,000	13
7 thru 10	100,000	135,000	13

NOTE 1: Tensile properties shall be determined by tests on cylindrical specimens conforming to the requirements of ASTM A-370, 0.2% offset method.

NOTE 2: Tensile specimens from drill collars shall be taken within 3 feet of the end of the drill collar in a longitudinal direction, having the centerline of the tensile specimen 1 inch from the outside surface or midwall, whichever is less.

Table 4-73
Drill Collar Weight (Steel) [51] (pounds per foot)

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill Collar OD, in.	Drill Collar ID, in.												
	1	1¼	1½	1¾	2	2¼	2½	2 13/16	3	3¼	3½	3¾	4
2⅞	19	18	16										
3	21	20	18										
3⅛	22	22	20										
3¼	26	24	22										
3½	30	29	27										
3¾	35	33	32										
4	40	39	37	35	32	29							
4⅛	43	41	39	37	35	32							
4¼	46	44	42	40	38	35							
4½	51	50	48	46	43	41							
4¾			54	52	50	47	44						
5			61	59	56	53	50						
5¼			68	65	63	60	57						
5½			75	73	70	67	64	60					
5¾			82	80	78	75	72	67	64	60			
6			90	88	85	83	79	75	72	68			
6¼			98	96	94	91	88	83	80	76	72		
6½			107	105	102	99	96	91	89	85	80		
6¾			116	114	111	108	105	100	98	93	89		
7			125	123	120	117	114	110	107	103	98	93	84
7¼			134	132	130	127	124	119	116	112	108	103	93
7½			144	142	139	137	133	129	126	122	117	113	102
7¾			154	152	150	147	144	139	136	132	128	123	112
8			165	163	160	157	154	150	147	143	138	133	122
8¼			176	174	171	168	165	160	158	154	149	144	133
8½			187	185	182	179	176	172	169	165	160	155	150
9			210	208	206	203	200	195	192	188	184	179	174
9½			234	232	230	227	224	220	216	212	209	206	198
9¾			248	245	243	240	237	232	229	225	221	216	211
10			261	259	257	254	251	246	243	239	235	230	225
11			317	315	313	310	307	302	299	295	291	286	281
12			379	377	374	371	368	364	361	357	352	347	342

NOTE 1: Refer to API Spec 7, Table 6.1 for API standard drill collar dimensions.
NOTE 2: For special configurations of drill collars, consult manufacturer for reduction in weight.

rock bit. If there is no tendency to cause an undersized hole, the largest drill collars that can be washed over are usually selected. The current tendency, i.e., not to run large drill collars that cannot be washed over, seems to be obsolete. Due to considerably improved technology in drill collar manufacturing, the possibility of losing the drill collar in the hole is greatly reduced; perhaps the gain in penetration rate by applying higher weight on the bit can overcome the risk of drill collar failure.

In general, if the optimal drilling programs require large drill collars, the operator should not hesitate to use them.

Typical hole and drill collar sizes used in soft and hard formations are listed in Table 4-74.

Length of Drill Collars

The length of the drill collar string should be as short as possible, but adequate to create the desired weight on bit. Ordinary drill pipe must never be used for exerting bit weight.

Table 4-74
Popular Hole and Drill Collar Sizes [39]

Hole size, inches	Drill collar sizes and connections	
	Soft formation	Hard formations
[1]4¾	3⅛″ OD x 1¼″ ID with 2⅞″ PAC or 2⅜″ Reg.	3½″ OD x 1½″ ID with 2⅞″ PAC or 2⅜″ Reg.
5⅞-6⅛	4⅛″ OD x 2″ ID with 2⅞″ IF	4¾″ OD x 2″ ID with 3½″ XH or 2⅞″ IF
[2]6½-6¾	4¾″ OD x 2¼″ ID with 3½″ IF	5″-5¼″ OD x 2″ ID with 3½″ IF
7⅝-7⅞	6″ OD x 2$\frac{13}{16}$″ ID with 4″ IF or 4″ H-90	6¼″ or 6½″ OD x 2″ or 2¼″ ID with 4½″ H-90, 4″ IF or 4″ H-90
8½-8¾	6¼″ OD x 2$\frac{13}{16}$″ ID with 4″ IF 6½″ OD x 2$\frac{13}{16}$″ ID with 4″ IF or 4½″ IF	6¾″ or 7″ OD x 2¼″ ID with 5″ H-90 or 4½″ IF
[3]9½-9⅞	7″ OD x 2$\frac{13}{16}$″ ID with 4½″ IF or 5″ H-90 8″ OD x 2$\frac{13}{16}$″ ID with 6⅝″ Reg.	7″ OD x 2¼″ ID with 4½″ IF or 5″ H-90 8″ OD x 2$\frac{13}{16}$″ ID with 6⅝″ H-90 or 6⅝″ Reg.
10⅝-11	7″ OD x 2$\frac{13}{16}$″ ID with 4½″ IF or 5″ H-90 8″ OD x 2$\frac{13}{16}$″ ID with 6⅝″ Reg.	8″ OD x 2$\frac{13}{16}$″ ID with 6⅝″ H-90 or 6⅝″ Reg. 9″ OD x 2$\frac{13}{16}$″ ID with 7⅝″ Reg.
12¼	8″ OD x 2$\frac{13}{16}$″ ID with 6⅝″ H-90 or 6⅝″ Reg.	8″ OD x 2$\frac{13}{16}$″ ID with 6⅝″ H-90 or 6⅝″ Reg. 9″ OD x 2$\frac{13}{16}$″ ID with 7⅝″ Reg. 10″ OD x 2$\frac{13}{16}$″ or 3″ ID with 7⅝″ H-90 or 7⅝″ Reg.
17½	8″ OD x 2$\frac{13}{16}$″ ID with 6⅝″ H-90 or 6⅝″ Reg.	8″ OD x 2$\frac{13}{16}$″ ID with 6⅝″ H-90 or 6⅝″ Reg. 9″ OD x 2$\frac{13}{16}$″ ID with 7⅝″ Reg. 10″ OD x 2$\frac{13}{16}$″ or 3″ ID with 7⅝″ H-90 or 7⅝″ Reg. 11″ OD x 3″ ID with 8⅝″ Reg.
18½-26	Drill collar programs are the same as for the next reduced hole size.	

Abbreviations: Reg. — API Regular
IF — API Internal Flush
XH — Hughes Xtra Hole
H-90 — Hughes H-90
PAC — Phil A. Carnell

[1] Range of hole sizes commonly used in deepening, workovers and drilling below small liners.
[2] Most planned drilling will fall within this range of hole sizes.
[3] Range of hole sizes is gaining popularity because of large number of deep wells being drilled and because of large production strings needed for high volume wells.

In highly deviated holes, an excessive torque is encountered with conventional drill collars; therefore, a heavy wall drill pipe can be used to supply part of the required weight.

The required length of drill collars can be obtained from

$$L_{dc} = \frac{(DF)W}{W_{dc}K_b \cos\alpha} \tag{4-50}$$

where DF = design factor (DF = 1.1–1.2)
W = weight on bit in lb
W_{dc} = unit weight of drill collar in air in lb/ft
K_b = buoyancy factor
$K_b = 1 - \gamma_m/\gamma_{st}$
γ_m = drilling fluid density, e.g., in lb/gal
γ_{st} = drill collar density, e.g., in lb/gal (for steel, γ_{st} = 65.5 lb/gal)
α = hole inclination from vertical in degrees (°)

Design factor (DF) is needed to place the neutral point below the top of the drill collar string. Some excess of drill collar weight is required to take care of inaccurate handling of the brake by the driller. For an "ideal driller," the design factor should be equal to 1. The excess of drill collars also helps to prevent transverse movement of drill pipe due to the effect of centrifugal force. While the drill string rotates, a centrifugal force is generated that may produce a lateral movement of drill pipe and, in turn, bending stress and excessive torque. The centrifugal force also contributes to vibration of the drill pie. Hence, some excess of drill collars is suggested. The magnitude of the design factor can be determined by field experiments in any particular set of drilling conditions.

The pressure area method (PAM), occasionally used for evaluation of drill collar string length, is wrong because it does not consider the triaxial state of stresses that actually occurs. It must be remembered that hydrostatic forces cannot cause any buckling of the drill string as long as the density of the string is greater than the density of the drilling fluid.

Example

Determine the required length of 7 by $2\frac{1}{4}$-in. drill collars if desired weight on bit is W = 40,000 lb, drilling fluid density γ_m = 100 lb/gal, and hole deviation from vertical α = 20°. From offset wells, it is known that a design factor DF of 1.1 is satisfactory.

Solution

From Table 4-75 the unit weight of drill collar W_{dc} = 117 lb/ft. The buoyant factor is

$$K_b = 1 - \frac{10}{65.5} = 0.847$$

Applying Equation 4-50 gives

$$L_{dc} = \frac{(1.1)(40,000)}{(147)(0.847)(\cos 20)} = 376\,\text{ft}$$

The closed length, based on 30-ft collars, is 390 ft or 13 joints of drill collars.

Actually, drill collar sizes and lengths should be considered simultaneously. The optimal selection should result in the maximum penetration rate. Such an approach, however complex, is particularly important when drilling formations sensitive to the effect of differential pressure and also in cases where the amount of hydraulic energy delivered at the rock bit is a controlling factor of drilling efficiency.

Drill Collar Connections

It is current practice to select the rotary shoulder connection that provides the balanced bending fatigue resistance for the pin and the box. The pin and the box are equally strong in bending if the cross-section module of the box in its critical zone is 2.5 times greater than the cross-section module of the pin at its critical zone. These critical zones are shown in Figure 4-127. Section modulus ratios from 2.25 to 2.75 are considered to be very good and satisfactory performance has been experienced with ratios from 2.0 to 3.2 [39].

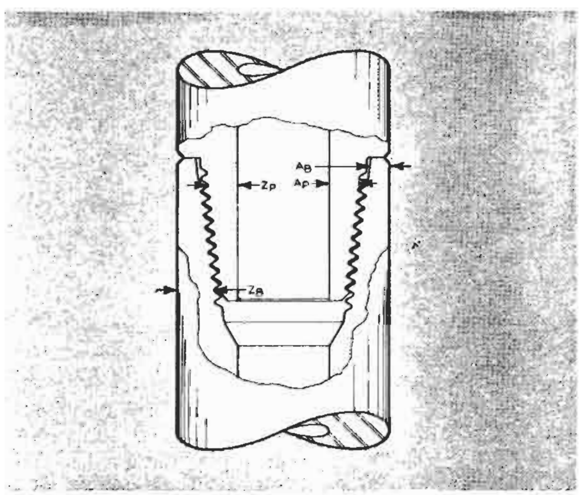
The above statements are valid if the connection is made up with the recommended makeup torque. For practical purposes, a set of charts is available from DRILCO, Division of Smith International, Inc. Some of these charts are presented in Figures 4-128 to 4-132. The method to use the connection selection charts is as follows [38]:

The best group of connections is defined as those that appear in the **shaded** sections of the charts. Also, the nearer the connection lies to the reference line, the more desirable is its selection.

The second best group of connections is defined as those that lie in the unshaded section of the charts on the **left**. The nearer the connection lies to the reference line, the more desirable is its selection.

The third best group of connections is defined as those that lie in the unshaded section of the charts on the **right**. The nearer the connection lies to the reference line, the more desirable is its selection.

(text continued on page 731)



The section modulus, Z_b, of the box should be $2\frac{1}{2}$ times greater than the section modulus, z_p, of the pin in a drill collar connection. On the right side of the connection are the spots at which the critical area of both the pin (A_p) and box (A_b) should be measured for calculating torsional strength.

Figure 4-127. The drill collar connection. *(After C. E. Wilson and W. R. Garrett.)*

2″ ID

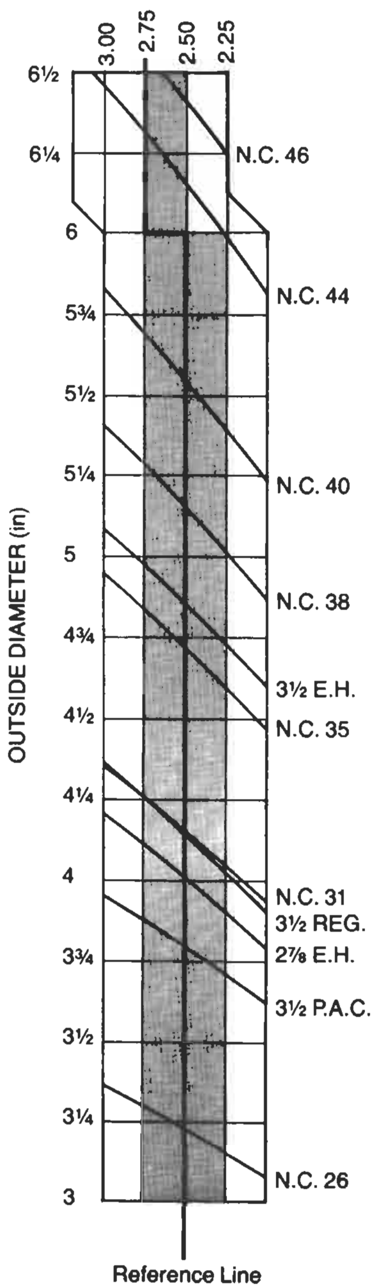


Figure 4-128. Practical chart for drill collar selection—2-in. ID. *(From Drilco, Division of Smith International, Inc.)*

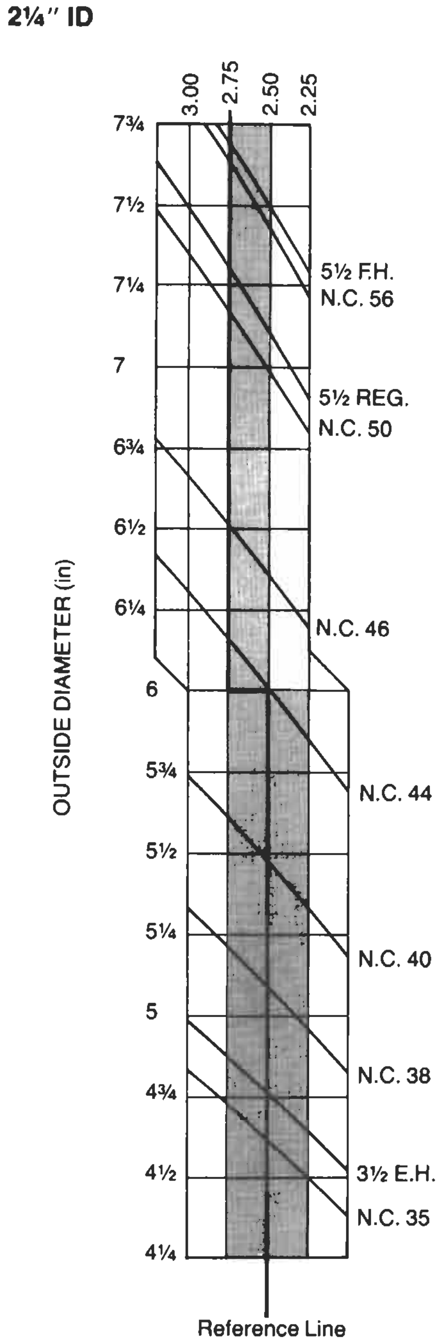


Figure 4-129. Practical chart for drill collar selection—2$\frac{1}{4}$-in. ID. *(From Drilco, Division of Smith International, Inc.)*

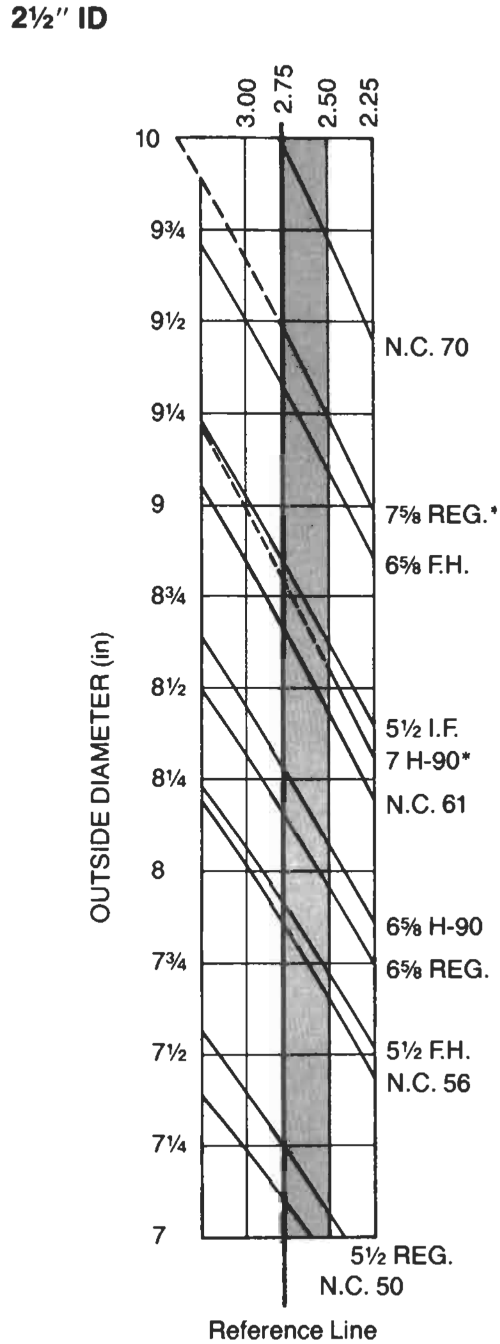


Figure 4-130a. Practical chart for drill collar selection—2½-in. ID. *(From Drilco, Division of Smith International, Inc.)*

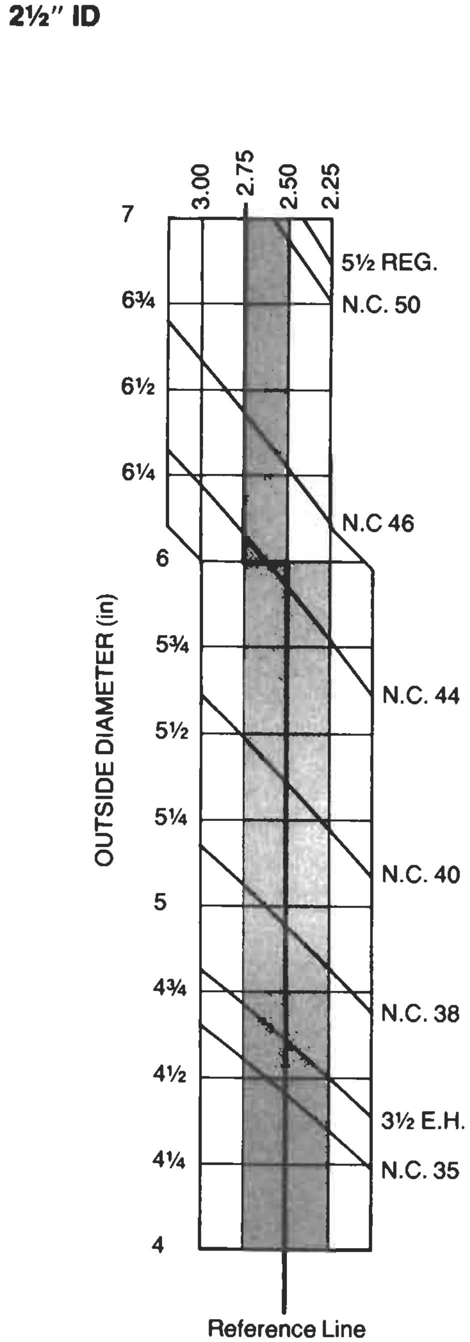


Figure 4-130b. Practical chart for drill collar selection—2 $\frac{1}{2}$-in. ID. *(From Drilco, Division of Smith International, Inc.)*

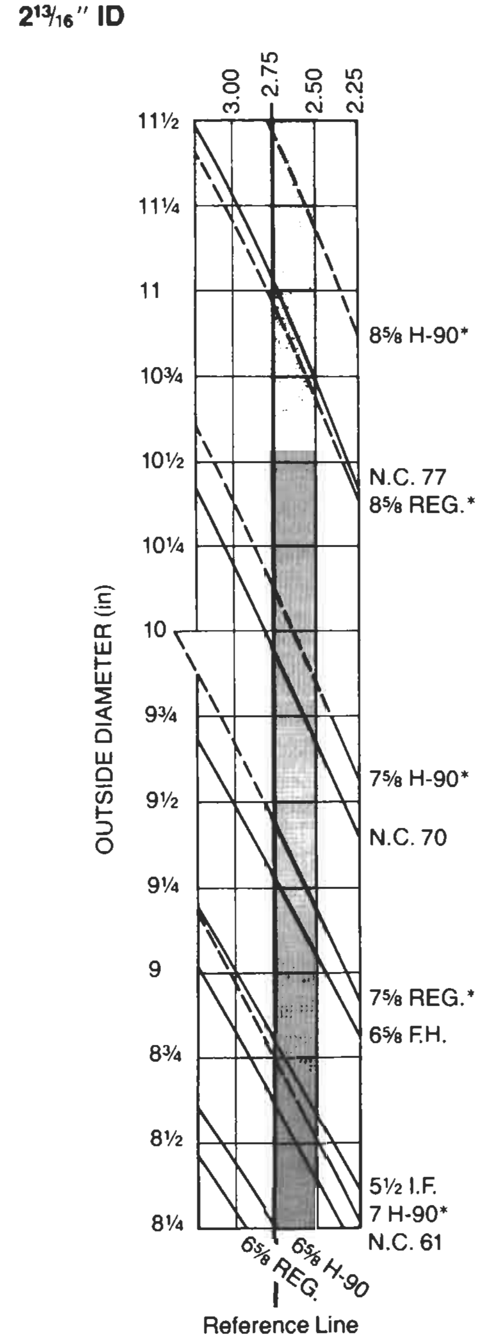


Figure 4-131a. Practical chart for drill collar selection—2 $\frac{13}{16}$-in. ID. *(From Drilco, Division of Smith International, Inc.)*

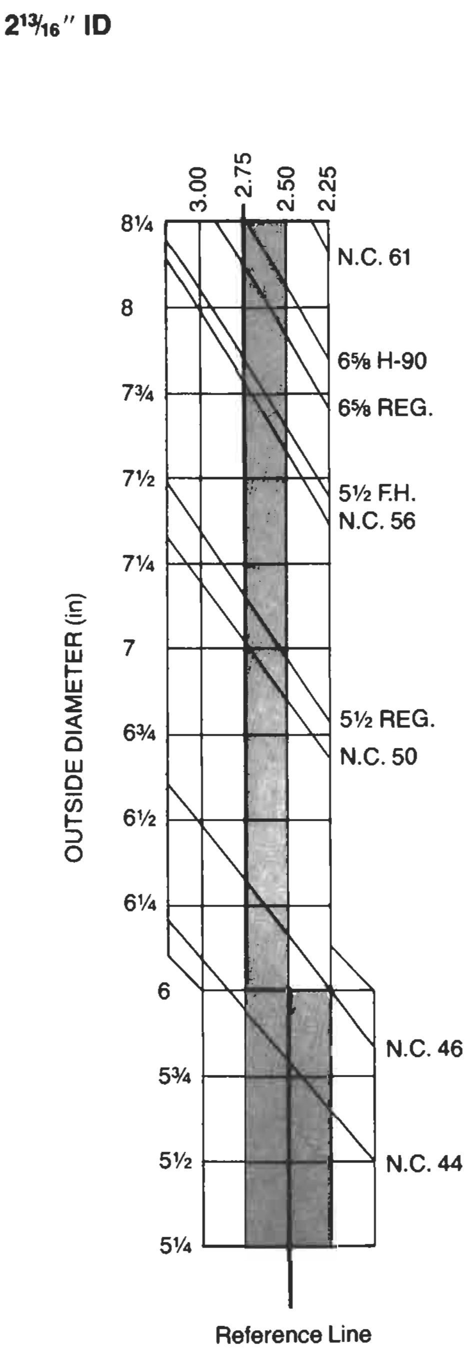


Figure 4-131b. Practical chart for drill collar selection—2 $\frac{13}{16}$-in. ID. *(From Drilco, Division of Smith International, Inc.)*

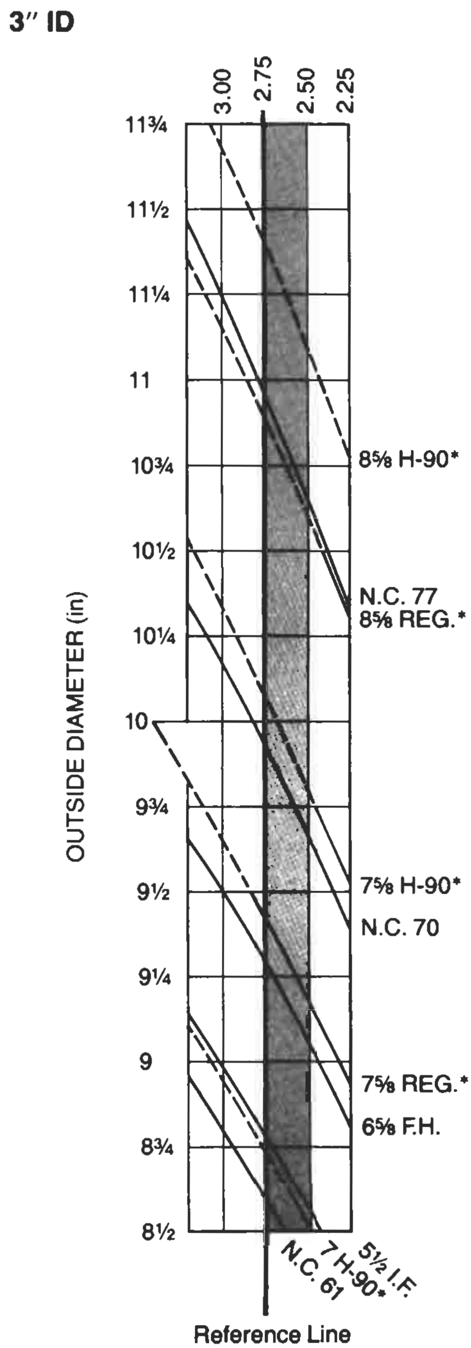


Figure 4-132a. Practical chart for drill collar selection—3-in. ID. *(From Drilco, Division of Smith International, Inc.)*

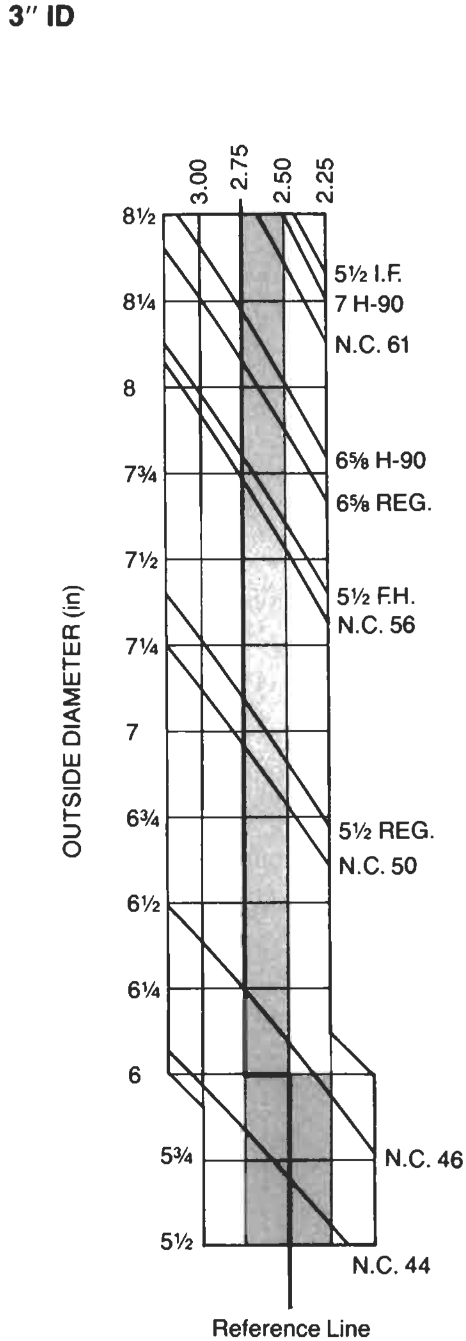


Figure 4-132b. Practical chart for drill collar selection—3-in. ID. *(From Drilco, Division of Smith International, Inc.)*

(text continued from page 722)

Example

Suppose you want to select the best connections for $9\frac{3}{4} \times 2\frac{3}{16}$-in. ID drill collars.
For average conditions, you should select in this order of preference (see Figure 4-131):

1. Best = N.C. 70 (shaded area and nearest reference line.)
2. Second best = $7\frac{5}{8}$ in. REG. (low torque). (Light area to left and nearest to reference line.)
3. Third best = $7\frac{5}{8}$ in. H-90. (Light area to right and nearest to reference line.)

But in extremely abrasive and/or corrosive conditions, you might want to select in this order of preference:

1. Best = $7\frac{5}{8}$ in. REG. (Low torque) = strongest box.*
2. Second best = N.C. 70 = second strongest box.
3. Third best = $7\frac{5}{8}$ in. H-90 = weakest box.

Recommended Makeup Torque for Drill Collars

The rotary shoulder connections must be made up with such torque that the shoulders will not separate under downhole conditions. This is of critical importance because the shoulder is the only area of seal in a rotary shoulder connection. Threads are designed to provide a clearance between crest and root that acts as a channel for lubricant and also accommodates the small solid particles.

To keep the shoulders together, the shoulder load must be high enough to create a compressive stress at the shoulder face capable of offsetting the bending that occurs due to drill collar buckling. This backup load is generated by a makeup torque. Field observations indicate that an average stress of 62,500 psi in pin or box, whichever is weaker (cross-sectional area), should be created by the makeup torque to prevent shoulder separation in most drilling conditions. It should be pointed out that the makeup torque creates the tensile stress in the pin and, consequently, the number of cycles for fatigue failure of the pin is decreased. Therefore, too high a makeup torque has a detrimental effect on the drill collar service life.

The recommended makeup torque for drill collars is given in Tables 4-75 and 4-76.

Drill Collars Buckling

In a vertical straight hole with no weight on the bit, a string of drill collars remains straight. As the weight for which the straight form of the string is not stable is reached, the drill string buckles and contacts the wall. If weight on the bit is further increased, the string buckles a second time and contacts the borehole wall at two points. With still further increased weight on the bit, the third and higher order of buckling occurs. The problem of drill collars buckling

(text continued on page 734)

*The connection furthest to the left on the chart has the strongest box. This connection should be considered as possible first choice for very abrasive formations or corrosive conditions.

Table 4-75
Recommended Makeup Torque [38]

RECOMMENDED MAKEUP TORQUE (ft-lb) |See Note 2|

Size and Type of Connection	OD (in)	Bore of Drill Collars (in): 1	1¼	1½	1¾	2	2¼	2½	2 13/16	3	3¼
API N.C. 23	3	2500†	2500†	2500†							
	3⅛	3300†	3300†	2600							
	3¼	4000	3400	2600							
2⅜" P.A.C. (See Note 4)	3		3800†	3800†	2900						
	3⅛		4900†	4200	2900						
	3¼		5200	4200	2900						
2⅜" API I.F. or API N.C. 26 or 2⅞" Slim Hole	3½		4600†	4600†	3700						
	3¾		5500	4700	3700						
2⅞" Extra Hole or 3½" Dbl. Streamline or 2⅞" Mod Open	3¾		4100†	4100†	4100†						
	3⅞		5300†	5300†	5300†						
	4⅛		8000†	8000†	7400						
2⅞" API I.F. or API N.C. 31 or 3½" Slim Hole	3⅞		4600†	4600†	4600†	4600†					
	4⅛		7300†	7300†	7300†	6800					
	4¼		8800†	8800†	8100	6800					
	4½		10800	9300	8100	6800					
API N.C. 35	4½				8900†	8900†	8900†	7400			
	4¾				12100	10800	9200	7400			
	5				12100	10800	9200	7400			
3½" Extra Hole or 4" Slim Hole or 3½" Mod Open	4¼				5100†	5100†	5100†	5100†			
	4½				8400†	8400†	8400†	8200			
	4¾				11900†	11700	10000	8200			
	5				13200	11700	10000	8200			
	5¼				13200	11700	10000	8200			
3½" API I.F. or API N.C. 38 or 4½" Slim Hole	4¾				9900†	9900†	9900†	9900†	8300		
	5				13800†	13800†	12800	10900	8300		
	5¼				16000	14600	12800	10900	8300		
	5½				16000	14600	12800	10900	8300		
3½" H-90 (See Note 3)	4¾				8700†	8700†	8700†	8700†	8700†		
	5				12700†	12700†	12700†	12700†	10400		
	5¼				16900†	16700	15000	13100	10400		
	5½				18500	16700	15000	13100	10400		
4" Full Hole or API N.C. 40 or 4" Mod Open or 4½" Dbl. Streamline	5				10800†	10800†	10800†	10800†	10800†		
	5¼				15100†	15100†	15100†	14800	12100		
	5½				19700†	18600	16900	14800	12100		
	5¾				20400	18600	16900	14800	12100		
	6				20400	18600	16900	14800	12100		
4" H-90 (See Note 3)	5¼					12500†	12500†	12500†	12500†		
	5½					17300†	17300†	17300†	16500		
	5¾					22300†	21500	19400	16500		
	6					23500	21500	19400	16500		
	6¼					23500	21500	19400	16500		
4½" API Regular	5½					15400†	15400†	15400†	15400†		
	5¾					20300†	20300†	19400	16200		
	6					23400	21600	19400	16200		
	6¼					23400	21600	19400	16200		
API N.C. 44	5¾					20600†	20600†	20600†	18000		
	6					25000	23300	21200	18000		
	6¼					25000	23300	21200	18000		
	6½					25000	23300	21200	18000		
4½" API Full Hole	5½					12900†	12900†	12900†	12900†	12900†	
	5¾					17900†	17900†	17900†	17900†	17700	
	6					23300†	23300†	22800	19800	17700	
	6¼					27000	25000	22800	19800	17700	
	6½					27000	25000	22800	19800	17700	
4½" Extra Hole or API N.C. 46 or 4" API I.F. or 5" Dbl. Streamline or 4½" Mod Open	5¾						17600†	17600†	17600†	17600†	
	6						23200†	23200†	22200	20200	
	6¼						28000	25500	22200	20200	
	6½						28000	25500	22200	20200	
	6¾						28000	25500	22200	20200	
4½" H-90 (See Note 3)	5¾						17600†	17600†	17600†	17600†	
	6						23400†	23400†	23000	21000	
	6¼						28500	26000	23000	21000	
	6½						28500	26000	23000	21000	
	6¾						28500	26000	23000	21000	
5" H-90 (See Note 3)	6¼						25000†	25000†	25000†	25000†	
	6½						31500†	31500†	29500	27000	
	6¾						35000	33000	29500	27000	
	7						35000	33000	29500	27000	
5½" H-90 (See Note 3)	6¾						34000†	34000†	34000†	34000	
	7						41500†	40000	36500	34000	
	7¼						42500	40000	36500	34000	
	7½						42500	40000	36500	34000	
5½" API Regular	6¾						31500†	31500†	31500†	31500†	
	7						39000†	39000†	36000	33500	
	7¼						42000	39500	36000	33500	
	7½						42000	39500	36000	33500	
4½" API I.F. or API N.C. 50 or 5" Extra Hole or 5" Mod Open or 5½" Dbl. Streamline	6¼						22800†	22800†	22800†	22800†	22800†
	6½						29500†	29500†	29500†	29500†	26500
	6¾						36000†	35500	32000	30000	26500
	7						38000	35500	32000	30000	26500
	7¼						38000	35500	32000	30000	26500

NOTE:

1. The calculations for recommended makeup torque assume the use of a thread compound containing 40% to 60% by weight of finely powdered metallic zinc, or 60% by weight of finely powdered metallic lead, applied thoroughly to all threads and shoulders; the use of the modified jackscrew formula as shown in the IADC Drilling Manual and the API Spec RP 7G (latest addition); and a unit stress of 62,500 psi in the box or pin, whichever is weaker.
2. Normal torque range — tabulated minimum value to 10% greater. Largest diameter shown for each connection is the maximum recommended for that connection. If the connections are used on drill collars larger than the maximum shown, increase the torque values shown by 10% for a minimum value.

Table 4-76
Recommended Makeup Torque [38]

RECOMMENDED MAKEUP TORQUE (ft-lb) [See Note 2]

Size and Type of Connection	OD (in)	Bore of Drill Collars (in) 2½	2 13/16	3	3¼	3½	3¾
5½" API Full Hole	7	32500†	32500†	32500†	32500†		
	7¼	40500†	40500†	40500†	40500†		
	7½	49000†	47000	45000	41500		
	7¾	51000	47000	45000	41500		
API N.C. 56	7¼	40000†	40000†	40000†	40000†		
	7½	48500†	48000	45000	42000		
	7¾	51000	48000	45000	42000		
	8	51000	48000	45000	42000		
6⅝" API Regular	7½	46000†	46000†	46000†	46000†		
	7¾	55000†	53000	50000	47000		
	8	57000	53000	50000	47000		
	8¼	57000	53000	50000	47000		
6⅝" H-90 (See Note 3)	7½	46000†	46000†	46000†	46000†		
	7¾	55000†	55000†	53000	49500		
	8	59500	56000	53000	49500		
	8¼	59500	56000	53000	49500		
API N.C. 61	8	54000†	54000†	54000†	54000†		
	8¼	64000†	64000†	64000†	61000		
	8½	72000	68000	65000	61000		
	8¾	72000	68000	65000	61000		
	9	72000	68000	65000	61000		
5½" API I.F.	8	56000†	56000†	56000†	56000†	56000†	
	8¼	66000†	66000†	66000†	63000	59000	
	8½	74000	70000	67000	63000	59000	
	8¾	74000	70000	67000	63000	59000	
	9	74000	70000	67000	63000	59000	
	9¼	74000	70000	67000	63000	59000	
6⅝" API Full Hole	8½		67000†	67000†	67000†	67000†	66500
	8¾		78000†	78000†	76000	72000	66500
	9		83000	80000	76000	72000	66500
	9¼		83000	80000	76000	72000	66500
	9½		83000	80000	76000	72000	66500
API N.C. 70	9		75000†	75000†	75000†	75000†	75000†
	9¼		88000†	88000†	88000†	88000†	88000†
	9½		101000†	101000†	100000	95000	90000
	9¾		107000	105000	100000	95000	90000
	10		107000	105000	100000	95000	90000
	10¼		107000	105000	100000	95000	90000
API N.C. 77	10			107000†	107000†	107000†	107000†
	10¼			122000†	122000†	122000†	122000†
	10½			138000†	138000†	133000	128000
	10¾			143000	138000	133000	128000
	11			143000	138000	133000	128000
Connections with Full Face							
7" H-90 (See Note 3)	8*		53000†	53000†	53000†	53000†	
	8¼*		63000†	63000†	63000†	60500	
	8½*		71500	68500	65000	60500	
7⅝" API Regular	8½*			60000†	60000†	60000†	60000†
	8¾*			71000†	71000†	71000†	71000†
	9*			83000†	83000†	79000	74000
	9¼*			88000	83000	79000	74000
	9½*			88000	83000	79000	74000
7⅝" H-90 (See Note 3)	9*			72000†	72000†	72000†	72000†
	9¼*			85500†	85500†	85500†	85500†
	9½*			98000†	98000†	98000†	95500
8⅝" API Regular	10*			108000†	108000†	108000†	108000†
	10¼*			123000†	123000†	123000†	123000
	10½*			139000	134000	129000	123000
8⅝" H-90 (See Note 3)	10¼*			112500†	112500†	112500†	112500†
	10½*			128500†	128500†	128500†	128500†
Connections with Low Torque Face							
7" H-90 (See Note 3)	8¾		67500†	67500†	66500	62000	
	9		74000	71000	66500	62000	
7⅝" API Regular	9¼			72000†	72000†	72000†	72000†
	9½			85000†	85000†	82000	77000
	9¾			91000	87000	82000	77000
	10			91000	87000	82000	77000
7⅝" H-90 (See Note 3)	9¾			91000†	91000†	91000†	91000†
	10			105000†	105000†	103500	98000
	10¼			112500	108000	103500	98000
	10½			112500	108000	103500	98000
8⅝" API Regular	10¾			112000†	112000†	112000†	112000†
	11			129000†	129000†	129000†	129000†
8⅝" H-90 (See Note 3)	10¾			92500†	92500†	92500†	92500†
	11			110000†	110000†	110000†	110000†
	11¼			128000†	128000†	128000†	128000†

In addition to the increased minimum torque value, it is also recommended that a fishing neck be machined to the maximum diameter shown.

3. The H-90 connection makeup torque is based on 56,250 psi stress and other factors as stated in note 1.
4. The 2⅞" P.A.C. makeup torque is based on 87,500 psi stress and other factors as stated in note 1.

*5. The largest diameter shown is the maximum recommended for those full-face connections. If larger diameters are used, machine connections with low torque faces and use the torque values shown under the low torque face table. If low torque faces are not used, see note 2 for increased torque values.

†6. Torque figures succeeded by a (†) indicate that the weaker member in torsion for the corresponding outside diameter and bore is the BOX. For all other torque values, the weaker member in torsion is the PIN.

(text continued from page 731)

in vertical holes has been studied by A. Lubinski [171] and the weight on the bit that results in first and second order buckling can be calculated as

$$W_{crI} = 1.94(EIp^2)^{1/3} \tag{4-51}$$

$$W_{crII} = 3.75(EIp^2)^{1/3} \tag{4-52}$$

where E = module of elasticity for drill collars in lb/ft^2 (for steel, $E = 4320 \times 10^6\ lb/ft^2$)
p = unit weight of drill collar in drilling fluid in lb/ft
I = moment of inertia of the drill collar cross-section with respect to its diameter, in ft
$I = (\pi/64)(D^4_{dc} - d^4_{dc})$
D_{dc} = outside diameter of drill collars in ft
d_{dc} = inside diameter of drill collars in ft

Example

Find the magnitude of the weight on bit and corresponding length of drill collars that result in second order of buckling. Drill collars: $6\frac{3}{4}$ in. × $2\frac{1}{4}$ ft, mud density = 12 lb/gal.

Solution

Moment of inertia:

$$I = \frac{\pi}{64}\left[\left(\frac{6.75}{12}\right)^4 - \left(\frac{2.25}{12}\right)^4\right] = 4.853 \times 10^{-3}\ ft^4$$

Unit weight of drill collar in drilling fluid:

$$p = 108\left(1 - \frac{12}{65.4}\right) = 88.18\ lb/ft$$

For weight on the bit that results in the second order of buckling, use Equation 4-61:

$$W_{cr,II} = 3.75(4320 \times 10^6 \times 4.853 \times 10^{-3} \times 88.18^2)^{1/3} = 20{,}468\ lb$$

Corresponding length of drill collars:

$$L_{dc} = \frac{20{,}468}{88.18} = 232\ ft$$

If the total length of drill collar string would be, for example, 330 ft, then the number "232 ft" would indicate the distance from the bit to the neutral point.

A. Lubinski also found [171] that to drill a vertical hole in homogeneous formations, it is best to carry less weight on the bit than the critical value of the first order at which the drill string buckles. However, if such weight is not sufficient, it is advisable to avoid the weight that falls between the first and second buckling order and to carry a weight close to the critical value of the third order.

For practical purposes, in many instances, the above statement holds trues if formations being drilled are horizontal. When drilling in dipping formations, a proper drill collar stabilization is required for vertical or nearly vertical hole drilling. In an inclined hole, a critical value of weight on the bit that produces buckling may be calculated from the formula given by R. Dawson and P. R. Paslay [40]:

$$W_{crit} = 2\left(\frac{EIp \sin \alpha}{r}\right)^{1/2} \tag{4-53}$$

where α = hole inclination measured from vertical in degrees (°)
r = radial clearance between drill collar and borehole wall in ft
E,I,p = as for Equations 4-51 and 4-52

Few straightforward computations can reveal that, in regular drilling conditions, the critical weight is very high. The reason why drill collars in an inclined hole are very resistant to buckling is that the hole is supporting the drill collar along its contact with the borehole wall.

This explains why heavy-weight drill pipe is successfully used for creating weight on the bit in highly deviated holes. However, in drilling a vertical or nearly vertical hole, a drill pipe must never be run in effective compression or, in other words, the neutral point must always reside in the drill collar string.

Rig Maintenance of Drill Collars [38]

It is recommended practice to break a different joint on each trip, giving the crew an opportunity to look at each pin and box every third trip. Inspect the shoulders for signs of loose connections, galls, and possible washouts.

Thread protectors should be used on pin and box when picking up or laying down the drill collars.

Periodically, based on drilling conditions and experience, a magnetic particle inspection should be performed using a wet fluorescent and black light method.

Before storing, the drill collars should be cleaned. If necessary, reface the shoulders with a shoulder refacing tool, and remove the fins on the shoulders by beveling. A good rust preventative or drill collar compound should be applied to the connections liberally, and thread protectors should be installed.

Drill Pipe

The major portion of drill string is composed of drill pipe. The drill pipe is manufactured by the seamless process. According to API Specification 5A (Thirty-fifth Edition, March 1981), seamless pipe is defined as a wrought steel tabular product made without a welded seam. It is manufactured by hot working steel or, if necessary, by subsequently cold finishing the hot worked tabular product to produce the desired shape, dimensions and properties.

Classification of Drill Pipe

Drill pipe is classified according to:

- type of ends upset
- sizes (outside diameter)
- wall thickness (nominal weight)
- steel grade
- length range.

Standardized pipe upsets are:

- internal upset (IU)
- external upset (EU)
- internal and external upset (IEU).

Geometrical data of upset drill pipe for weld-on tool joints are specified in Table 4-77 (steel grades D and E) and Table 4-78 (steel grades X, G and S).

API standardized new drill pipe sizes and unit weights are given in Table 4-79.

Drill pipe is manufactured in the following random length ranges:

Range 1—18 to 22 ft
Range 2—27 to 30 ft
Range 3—38 to 45 ft

The drill pipe most commonly used is Range 2 pipe.

To meet specific downhole requirements, seamless drill pipe is available in five steel grades, namely D, E, X, G and S*. (Grades X, G and S are considered to be high-strength pipe grades.) The mechanical properties of these steel grades are as follows:

	API Steel Grade				
Property	**D**	**E**	**X(95)**	**105(G)**	**135(S)**
Minimum yield strength, psi	55,000	75,000	95,000	105,000	135,000
Maximum yield strength, psi	85,000	105,000	125,000	135,000	165,000
Minimum tensile strength, psi	95,000	100,000	105,000	115,000	145,000

For practical engineering calculations, the minimum yield strength is usually used; however, for some calculations, the average yield strength is used.

Minimum Performance Properties of Drill Pipe

The torsion, tension, collapse and internal pressure resistance for new, premium class 2 and class 3 drill pipe are specified in Tables 4-80, 4-81, 4-82 and 4-83, respectively.

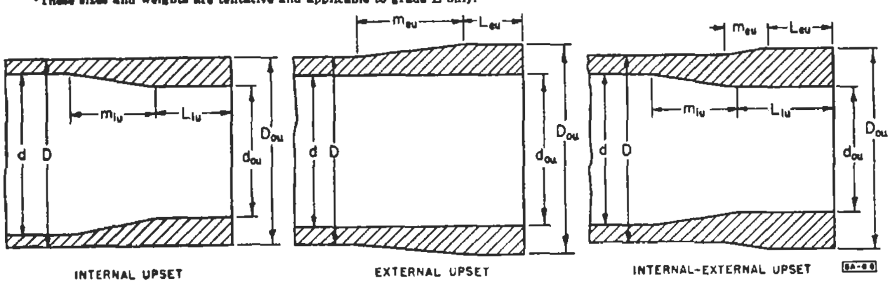
Calculations for the minimum performance properties of drill pipe are based on formulas given in Appendix A of API RP 7G. It must be remembered that numbers in Tables 4-80–4-83 have been obtained for the uniaxial state of stress, e.g., torsion only or tension only, etc. The tensile stress resistance is decreased when the drill string is subjected to both axial tension and torque; a collapse

*The above data are obtained from the IADC Drilling Manual, Section B, p. 1, revised January 1975.

Table 4-77
Upset Drill Pipe for Weld-on Tool Joints (Grades D and E) [30]

1	2	3	4	5	6	7	8	9	10	11	12		13
				Calculated Weight		[5]Upset Dimensions, in.							
Pipe Size: Outside Dia., in. D	Nominal Wt.,[1] lb/ft	Wall Thickness, in. t	Inside Diameter, in. d	Plain End lb/ft w_{pe}	Upset[4] lb e_w	Outside Diameter,[2] $+\frac{1}{8}$, $-\frac{1}{32}$ D_{ou}	Inside Diameter at End of Pipe,[3] $\pm\frac{1}{16}$ d_{ou}	Length of Internal Upset $+1\frac{1}{2}$ $-\frac{1}{2}$ L_{iu}	Length of Internal Taper, min. m_{iu}	Length of External Upset, min. L_{eu}	Length of External Taper, min. m_{eu}	max.	Length End of Pipe to Taper Fadeout, max. $L_{eu}+m_{eu}$
						INTERNAL-UPSET DRILL PIPE							
2⅞	10.40	0.362	2.151	9.72	3.20	2.875	1 5/16	1¾	1½	...	...	...	....
3½	9.50	0.254	2.992	8.81	4.40	3.500	2¼	1¾	...	...	...	...	....
3½	13.30	0.368	2.764	12.31	4.40	3.500	1 15/16	1¾	1½	...	...	...	....
3½	15.50	0.449	2.602	14.63	3.40	3.500	1 15/16	1¾	1½	...	...	...	....
[6]4	11.85	0.262	3.476	10.46	4.20	4.000	2 15/16	1¾	...	...	...	...	....
4	14.00	0.330	3.340	12.93	4.60	4.000	2¾	1¾	2	...	...	...	....
[6]4½	13.75	0.271	3.958	12.24	5.20	4.500	3⅜	1¾	...	...	...	...	....
4½	16.60	0.337	3.826	14.98	5.80	4.500	3 5/32	1¾	2	...	...	...	....
[6]5	16.25	0.296	4.408	14.87	6.60	5.000	3¾	1¾	...	...	...	...	....
						EXTERNAL-UPSET DRILL PIPE							
2⅜	6.65	0.280	1.815	6.26	1.80	2.656	1.815	...	...	1½	1½	...	4
2⅞	10.40	0.362	2.151	9.72	2.40	3.219	2.151	...	...	1½	1½	...	4
3½	9.50	0.254	2.992	8.81	2.60	3.824	2.992	...	...	1½	1½	...	4
3½	13.30	0.368	2.764	12.31	4.00	3.824	2.602	2¼	2	1½	1½	...	4
3½	15.50	0.449	2.602	14.63	2.80	3.824	2.602	...	...	1½	1½	...	4
[6]4	11.85	0.262	3.476	10.46	5.00	4.500	3.476	...	...	1½	1½	...	4
4	14.00	0.330	3.340	12.93	5.00	4.500	3.340	...	...	1½	1½	...	4
[6]4½	13.75	0.271	3.958	12.24	5.60	5.000	3.958	...	...	1½	1½	...	4
4½	16.60	0.337	3.826	14.98	5.60	5.000	3.826	...	...	1½	1½	...	4
4½	20.00	0.430	3.640	18.69	5.60	5.000	3.640	...	...	1½	1½	...	4
						INTERNAL-EXTERNAL-UPSET DRILL PIPE							
4½	20.00	0.430	3.640	18.69	8.60	4.781	3	2¼	2	1½	1	1½	....
5	19.50	0.362	4.276	17.93	8.60	5.188	3 11/16	2¼	2	1½	1	1½	....
5	25.60	0.500	4.000	24.03	7.80	5.188	3 7/16	2¼	2	1½	1	1½	....
5½	21.90	0.361	4.778	19.81	10.60	5.563	4	2¼	2	1½	1	1½	....
5½	24.70	0.415	4.670	22.54	9.00	5.563	4	2¼	2	1½	1	1½	....

[1]Nominal weights (Col. 2), are shown for the purpose of identification in ordering.
[2]The ends of internal-upset drill pipe shall not be smaller in outside diameter than the values shown in Col. 7, including the minus tolerance. They may be furnished with slight external upset, within the tolerance specified.
[3]Maximum taper on inside diameter of internal upset and internal-external upset is ¼ in. per ft. on diameter.
[4]Weight gain or loss due to end finishing.
[5]The specified upset dimensions do not necessarily agree with the bore and OD dimensions of finished weld-on assemblies. Upset dimensions were chosen to accommodate the various bores of tool joints and to maintain a satisfactory cross section in the weld zone after final machining of the assembly.
[6]These sizes and weights are tentative and applicable to grade E only.



pressure resistance is also decreased when the drill pipe is simultaneously affected by collapse and tensile loads.

Load Capacity of Drill Pipe

In normal drilling operations, as well as in such operations as DST or washover, drill pipe is subjected to combined effects of stresses.

To evaluate the load capacity of drill pipe (e.g., allowable tensile load while simultaneously a torque is applied), the maximum distortion energy theory is

Table 4-78
Upset Drill Pipe for Weld-on Tool Joints (Grades X, G and S) [30]

1	2	3	4	5	6	7	8	9	10	11
				Calculated Weight		Upset Dimensions, in. [5]				
Pipe Size: Outside Dia., in. D	Nominal Wt.:[1] lb/ft	Wall Thickness, in. t	Inside Diameter, in. d	Plain End lb/ft w_{pe}	Upset[4] lb e_w	Outside Diameter,[2] +1/8, −1/32 D_{ou}	Inside Diameter at End of Pipe,[3] ±1/16 d_{ou}	Length of Internal Upset +1 1/2 − 1/2 L_{iu}	Length of External Upset, min. L_{eu}	Length End of Pipe to Taper Fadeout, Ext. Upset max. $L_{eu} + m_{eu}$
				INTERNAL-UPSET DRILL PIPE						
2 7/8	10.40	0.362	2.151	9.72	5.40	2.875	1 9/16	3 1/2	...	...
3 1/2	13.30	0.368	2.764	12.31	7.40	3.500	1 15/16	3 1/2	...	...
4	14.00	0.330	3.340	12.93	8.80	4.000	2 5/8	3 1/2	...	...
4 1/2	16.60	0.337	3.826	14.98	13.60	4.500	2 13/16	3 1/2	...	...
5	16.25	0.296	4.408	14.87	13.60	5.000	3 9/16	3 1/2	...	...
				EXTERNAL-UPSET DRILL PIPE						
2 3/8	6.65	0.280	1.815	6.26	4.60	2.656	1 9/16	4 1/4	3	5 1/2
2 7/8	10.40	0.362	2.151	9.72	6.20	3.250	1 15/16	4 1/4	3	5 1/2
3 1/2	13.30	0.368	2.764	12.31	10.20	4.000	2 1/2	4 1/4	3	5 1/2
3 1/2	15.50	0.449	2.602	14.63	8.20	4.000	2 1/2	4 1/4	3	5 1/2
4	14.00	0.330	3.340	12.93	14.40	4.625	3 1/16	4 1/4	3	5 1/2
4 1/2	16.60	0.337	3.826	14.98	17.20	5.188	3 9/16	4 1/4	3	5 1/2
4 1/2	20.00	0.430	3.640	18.69	16.00	5.188	3 7/16	4 1/4	3	5 1/2
5	19.50	0.362	4.276	17.93	21.60	5.750	3 15/16	4 1/4	3	5 1/2
5	25.60	0.500	4.000	24.03	21.20	5.875	3 13/16	4 1/4	3	5 1/2
				INTERNAL-EXTERNAL-UPSET DRILL PIPE						
3 1/2	15.50	0.449	2.602	14.63	11.00	3.781	1 15/16	4 1/4	3	5 1/2
4 1/2	20.00	0.430	3.640	18.69	17.60	4.781	2 13/16	4 1/4	3	5 1/2
5	19.50	0.362	4.276	17.93	16.80	5.188	3 9/16	4 1/4	3	5 1/2
5	25.60	0.500	4.000	24.03	15.40	5.188	3 5/16	4 1/4	3	5 1/2
5 1/2	21.90	0.361	4.778	19.81	21.00	5.563	3 13/16	4 1/4	3	5 1/2
5 1/2	24.70	0.415	4.670	22.54	18.40	5.563	3 13/16	4 1/4	3	5 1/2

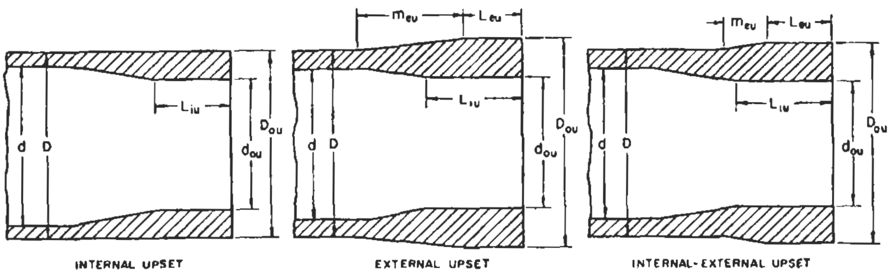
[1] Nominal weights (Col. 2), are shown for the purpose of identification in ordering.

[2] The ends of internal-upset drill pipe shall not be smaller in outside diameter than the values shown in Col. 7, including the minus tolerance. They may be furnished with slight external upset, within the tolerance specified.

[3] Maximum taper on inside diameter of internal upset and internal-external upset is 1/4 in. per ft. on diameter.

[4] Weight gain or loss due to end finishing.

[5] The specified upset dimensions do not necessarily agree with the bore and OD dimensions of finished weld-on assemblies. Upset dimensions were chosen to accommodate the various bores of tool joints and to maintain a satisfactory cross section in the weld zone after final machining of the assembly.



Note: Permissible internal taper within length L_{iu} shall not exceed 1/4 in. per ft (21 mm per m) on diameter.

usually applied. This theory is in good agreement with experiments on ductile materials such as steel. According to this theory, the equivalent stress may be calculated from the following formula [42A]:

$$2\sigma_e^2 = (\sigma_z - \sigma_t)^2 + (\sigma_t - \sigma_r)^2 + (\sigma_r - \sigma_z)^2 + 6\tau^2 \tag{4-54}$$

where σ_e = equivalent stress in psi
σ_z = axial stress in psi ($\sigma_z > 0$ for tension, $\sigma_z < 0$ for compression)

Table 4-79
New Drill Pipe—Dimensional Data [51]

1	2	3	4	5	6	7
Size OD in. D	Nominal Weight Threads & Couplings lb/ft	Plain End Weight[1] lb/ft	Wall Thickness in.	ID in. d	Section Area Body of Pipe[2] sq. in. A	Polar Sectional Modulus[3] cu. in. Z
2⅜	4.85	4.43	.190	1.995	1.3042	1.321
	6.65	6.26	.280	1.815	1.8429	1.733
2⅞	6.85	6.16	.217	2.441	1.8120	2.241
	10.40	9.72	.362	2.151	2.8579	3.204
3½	9.50	8.81	.254	2.992	2.5902	3.923
	13.30	12.31	.368	2.764	3.6209	5.144
	15.50	14.63	.449	2.602	4.3037	5.847
4	11.85	10.46	.262	3.476	3.0767	5.400
	14.00	12.93	.330	3.340	3.8048	6.458
	15.70	14.69	.380	3.240	4.3216	7.157
4½	13.75	12.24	.271	3.958	3.6004	7.184
	16.60	14.98	.337	3.826	4.4074	8.543
	20.00	18.69	.430	3.640	5.4981	10.232
	22.82	21.36	.500	3.500	6.2832	11.345
5	16.25	14.87	.296	4.408	4.3743	9.718
	19.50	17.93	.362	4.276	5.2746	11.415
	25.60	24.03	.500	4.000	7.0686	14.491
5½	19.20	18.87	.304	4.892	4.9624	12.221
	21.90	19.81	.361	4.778	5.8282	14.062
	24.70	22.54	.415	4.670	6.6296	15.688
6⅝	25.20	22.19	.330	5.965	6.5262	19.572

[1] lb/ft = 3.3996 x A (col. 6)
[2] $A = 0.7854 (D^2 - d^2)$
[3] $Z = 0.19635 \left(\frac{D^4 - d^4}{D}\right)$

σ_t = tangential stress in psi ($\sigma_z > 0$ for burst pressure, $\sigma_z < 0$ for collapse pressure)
σ_r = radial stress (usually neglected for the drill pipe strength analysis)
τ = shear stress in psi

The yielding of pipe does not occur provided that the equivalent stress is less than the yield strength of the drill pipe. For practical calculations, the equivalent stress is taken to be equal to the minimum yield strength of the pipe as specified by API. *It must be remembered that the stresses being considered in Equation 4-54 are the effective stresses that exist beyond any isotropic stresses caused by hydrostatic pressure of the drilling fluid.*

(text continued on page 744)

Table 4-80
New Drill Pipe—Torsional, Tensile, Collapse and Internal Pressure Data [30]

Size OD in.	Nom. Wt. New Pipe w/T.J.	Torsional Data* Torsional Yield Strength, ft-lb					Tensile Data Based on Minimum Values Load at the Minimum Yield Strength, lb.					Collapse Pressure Based On Minimum Values, psi.					Internal Pressure At Minimum Yield Strength, psi.				
		D	E	95	105	135	D	E	95	105	135	D	E	95	105	135	D	E	95	105	135
2 3/8	4.85	....	4760	6020	6660	8560	....	97820	123900	136940	176060	8100	11040	13980	15460	19070	–	10500	13300	14700	18900
	6.65	4580	6240	7900	8740	11240	101360	138220	175080	193500	248780	11440	15600	19760	21840	28080	11350	15470	19600	21660	27850
2 7/8	6.85	....	8070	10220	11300	14530	....	135900	172140	190260	244620	7680	10470	12930	14010	17060	–	9910	12550	13870	17830
	10.40	8460	11530	14610	16150	20760	157190	214340	271500	300080	385820	12110	16510	20910	23110	29720	12120	16530	20930	23140	29750
3 1/2	9.50	....	14120	17890	19770	25420	....	194260	246070	271970	349680	7400	10040	12060	13050	15780	–	9520	12070	13340	17150
	13.30	13580	18520	23460	25930	33330	199160	271570	343990	380190	488820	10350	14110	17880	19760	25400	10120	13800	17480	19320	24840
	15.50	15440	21050	26660	29470	37890	236720	322780	408850	451890	581000	12300	16770	21250	23480	30190	12350	16840	21330	23570	30310
4	11.85	....	19440	24620	27220	34990	....	230750	292290	323050	415360	6590	8410	9960	10700	12650	–	8600	10890	12040	15480
	14.00	17050	23250	29450	32550	41840	209280	285360	361460	399500	513650	8330	11350	14380	15900	20170	7940	10830	13720	15160	19490
	15.70	18890	25760	32630	36070	46380	237710	324150	410590	453810	583420	9460	12900	16340	18050	23210	9140	12470	15790	17460	22440
4 1/2	13.75	....	25860	32760	36210	46550	....	270030	342040	378050	486060	5720	7200	8400	8950	10310	–	7900	10010	11070	14230
	16.60	22550	30750	38950	43050	55350	242380	330560	418700	462780	595000	7620	10390	12750	13820	16800	7210	9830	12450	13760	17690
	20.00	27010	36840	46660	51570	66300	302390	412360	522320	577300	742240	9510	12960	16420	18150	23330	9200	12540	15890	17560	22580
	22.82	30000	40910	51820	57280	73640	345580	471240	596900	659740	848230	10860	14810	18770	20740	26670	10690	14580	18470	20420	26250
5	16.25	....	34980	44310	48970	62970	....	328070	415560	459300	590530	5560	6970	8090	8610	9860	–	7770	9840	10880	13990
	19.50	30135	41090	52050	57530	73970	290100	395600	501090	553830	712070	7390	10000	12010	12990	15700	6970	9500	12040	13300	17110
	25.60	38250	52160	66070	73030	93900	388770	530140	671520	742200	954260	9900	13500	17100	18900	24300	9620	13120	16620	18380	23620
5 1/2	19.20	....	44180	55960	61850	79520	....	372180	471430	521050	669920	4910	6070	6930	7300	8120	–	7250	9190	10160	13060
	21.90	37120	50620	64120	70870	91120	320550	437120	553680	611960	786810	6610	8440	10000	10740	12710	6320	8610	10910	12060	15510
	24.70	41410	56470	71530	79060	101650	364630	497220	629810	696110	895000	7670	10460	12920	14000	17050	7260	9900	12540	13860	17830
6 5/8	25.20	51740	70550	89360	98770	....	358930	489460	619990	685250	....	4010	4810	5300	5480	6040	4790	6540	8280	9150	–

*Based on the shear strength equal to 57.7% of minimum yeidl strength and nominal wall thickness.
NOTE: Calculations are based on formulas in Appendix A, API RP7G
Table is based on API RP7G. Tables 2.1 and 2.2.

Table 4-81
Premium (Used) Drill Pipe—Torsional, Tensile, Collapse and Internal Pressure Data [30]

Size OD in.	Nom. Wt. New Pipe w/T.J. lb/ft	1,2Torsional Yield Strength Based On Uniform Wear, ft-lb					2Tensile Data Based On Uniform Wear Load At Minimum Yield Strength, lb.					3Collapse Pressure Based On Minimum Values, psi.					3Internal Pressure At Minimum Yield Strength, psi.				
		D	E	95	105	135	D	E	95	105	135	D	E	95	105	135	D	E	95	105	135
2 3/8	4.85	2730	3720	4710	5210	6690	56400	76910	97420	107680	138440	6690	8550	10150	10900	12920	7040	9600	12160	13440	17280
	6.65	3520	4800	6080	6720	8640	78925	107620	136320	150670	193720	9810	13380	16950	18730	24080	10370	14150	17920	19810	25470
2 7/8	6.85	4640	6320	8010	8850	11370	78450	106970	135500	149760	192550	6060	7670	9000	9620	11210	6640	9060	11470	12680	16300
	10.40	6480	8840	11200	12380	15920	122100	166500	210990	233100	299700	10430	14220	18020	19910	25600	11080	15110	19140	21150	27200
3 1/2	9.50	8120	11070	14030	15500	19930	112220	153020	193830	214230	275440	5650	7100	8270	8800	10120	6390	8710	11030	12190	15680
	13.30	10510	14340	18160	20070	25800	155650	212250	268850	297150	382050	8810	12020	15220	16820	21630	9250	12620	15980	17660	22710
	15.50	11820	16120	20420	22560	29010	183700	250500	317300	305700	450900	10610	14470	18330	20260	26050	11290	15390	19500	21550	27710
4	11.85	11210	15280	19360	21400	27510	133510	182060	230610	254890	327710	4670	5730	6490	6820	7470	5760	7860	9960	11000	14150
	14.00	13320	18160	23010	25430	32690	165715	225980	286240	316360	406760	7000	9040	10780	11610	13870	7260	9900	12540	13860	17820
	15.70	14690	20030	25370	28040	36050	186160	253860	321560	355400	456950	8000	10910	13820	15180	18630	8340	11380	14410	15930	20480
4 1/2	13.75	14940	20370	25800	28510	36660	156420	213310	270190	298630	383950	3940	4710	5170	5340	5910	5300	7230	9150	10120	13010
	16.60	17670	24100	30520	33740	43370	190740	260100	329460	364140	468180	5980	7550	8850	9460	10990	6590	8990	11380	12580	16180
	20.00	21000	38630	36270	40090	51540	236830	322950	409070	452130	581310	8050	10980	13900	15340	18840	8410	11470	14520	16050	20640
	22.82	23150	31570	39990	44200	56820	269550	367570	465590	514590	661620	9280	12660	16030	17120	22780	9820	13400	16970	18750	24112
5	16.25	20210	27560	34910	38580	49610	190100	259220	328350	362910	466600	3800	4510	4920	5060	5670	5210	7100	9000	9950	12790
	19.50	23630	32230	40820	45120	58010	228360	311400	394440	435960	560520	5630	7070	8230	8760	10050	6370	8690	11000	12160	15640
	25.60	29680	40470	51270	56660	72850	306000	417500	535000	585000	750000	8400	11460	14510	16040	20540	8800	12000	15200	16800	21600
5 1/2	19.20						215790	294260	372730	411970	529670	3260	3760	4140	4340	4720	4860	6630	8400	9290	11940
	21.90	29180	39790	50400	55710	71630	252910	344880	436840	482820	620770	4690	5760	6530	6860	7520	5780	7880	9980	11030	14180
	24.70	32450	44250	56050	61950	79650	304095	391282	525260	580540	746420	6060	7670	9000	9620	11200	6640	9050	11470	12680	16300
6 5/8	25.20	40870	55740	70600	78030	100320	284150	387480	490810	452470	697460	2510	2930	3250	3350	3430	4290	5850	7420	8200	10540

1 Based on the shear strength equal to 57.7% of minimum yield strength.
2 Torsional and Tensile data based on 20% uniform wear.
3 Collapse and Internal pressure data based on minimum wall of 80% of nominal (new) wall.
NOTE: Calculations for Premium Class drill pipe are based on formulas in Appendix A, API RP7G.
Table is based on API RP7G, tables 2.3 and 2.4.

Table 4-82
Class 2 (Used) Drill Pipe—Torsional, Tensile, Collapse and Internal Pressure Data [30]

Size OD in.	Nom.Wt. New Pipe w/T.J.	1,2 Torsional Yield Strength Based On Uniform Wear, ft-lb					2 Tensile Data Based On Uniform Wear Load At Minimum Yield Strength, lb.					3 Collapse Pressure Based On Minimum Values, psi					3 Internal Pressure At Minimum Yield Strength, psi.				
		D	E	95	105	135	D	E	95	105	135	D	E	95	105	135	D	E	95	105	135
2 3/8	4.85	2230	3040	3850	4250	5470	56400	76910	97420	107680	138440	4880	6020	6870	7240	8030					
	6.65	2880	3920	4970	5490	7060	78925	107620	136320	150670	193720	8420	11480	14540	16080	20630	8430	11490	14560	16090	20680
2 7/8	6.85	3790	5160	6540	7230	9290	78450	106970	135500	149760	192550	4340	5270	5900	6150	6610	5390	7360	9320	10300	13240
	10.40	5300	7220	9150	10110	13000	122100	166500	210900	233100	299700	8990	12260	15520	17160	22060	8990	12260	15530	17170	22070
3 1/2	9.50	6630	9040	11450	12660	16280	112220	153020	193830	214230	275440	3990	4790	5270	5450	6010	5185	7070	8960	9900	12730
	13.30	8590	11710	14830	16390	21070	155650	212250	268850	297150	382050	7520	10250	12420	13450	16310	7510	10240	12970	14340	18440
	15.50	9650	13160	16670	18430	23690	183700	250500	317300	305700	450900	9150	12480	15810	17480	22470	9170	12510	15850	17520	22530
4	11.85	9150	12480	15810	17470	22460	133510	182060	230610	254890	327710	3160	3620	4020	4210	4550	4670	6370	8070	8920	11470
	14.00	10880	14830	18790	20770	26700	165715	225980	286240	316360	406760	5180	6440	7410	7850	8840	5880	8020	10165	11240	14440
	15.70	12000	16360	20720	22900	29450	186160	253860	321560	355400	456950	6700	8560	10150	10910	12930	6970	9260	11730	12970	16670
4 1/2	13.75	12190	16630	21060	23280	29930	156430	213310	270190	298630	383950	2540	2960	3290	3400	3480					
	16.60	14430	19680	24920	27550	35420	190740	260100	329460	364140	468180	4270	5170	5770	6010	6490	5350	7300	9250	10220	13140
	20.00	17150	23380	29620	32740	42090	236830	322950	409070	452130	581310	6770	8660	10280	11050	13120	6840	9330	11820	13070	16800
	22.82						269550	367570	465590	514390	661620	9940	10830	13710	14950	18320	7940	10830	13720	15170	19500
5	16.25	16500	22500	28500	31500	40500	190100	259220	328350	362910	466600	2420	2850	3150	3240	3300	4220	5760	7300	8060	10370
	19.50	19300	26320	33330	36840	47370	228360	311400	394440	435960	560520	3970	4760	5230	5410	5970	5190	7080	9070	9910	12740
	25.60	24240	33050	41870	46270	59490	306000	417500	535000	585000	750000	7150	9420	11270	12160	14590	7150	9750	12350	13650	17550
5 1/2	19.20						215790	294260	372730	411970	529670	2110	2440	2610	2650	2650	3960	5400	6840	7560	9720
	21.90	23830	32490	41150	45490	58480	252910	344870	436840	482820	620770	3170	3640	4040	4230	4580	4700	6410	8120	8970	11540
	24.70	26490	36130	45760	50580	65030	304095	391282	525260	580540	746420	4340	5260	5890	6140	6610	5400	7360	9330	10310	13250
6 5/8	25.20						284150	387480	490810	542470	697460	1690	1870	1900	1900	1900	3550	4840	6137	6783	8721

[1] Based on the shear strength equal to 57.7% of minimum yield strength.
[2] Torsional data based on 35% eccentric wear and tensile data based on 20% uniform wear.
[3] Data is based on minimum wall of 65% nominal wall.
NOTE: Calculations for Class II drill pipe are based on formulas in Appendix A, API RP7G.
Table is based on API RP7G, tables 2.5 and 2.6.

Table 4-83
Class 3 (Used) Drill Pipe—Torsional, Tensile, Collapse and Internal Pressure Data [30]

Size OD in.	Nom.Wt. New Pipe w/T.J. lb/ft	1,2Torsional Yield Strength Based On Eccentric Wear, ft-lb					2Tensile Data Based On Uniform Wear Load at Minimum Yield Strength, lb.					1Collapse Pressure Based On Minimum Values, psi.					1Internal Pressure At Minimum Yield Strength, psi.				
		D	E	95	105	135	D	E	95	105	135	D	E	95	105	135	D	E	95	105	135
2 3/8	4.85	1870	2550	3230	3570	4590	43380	59160	74940	82820	106490	3620	4260	4590	4810	5350	4860	6631	8400	9280	11940
	6.65	2390	3260	4130	4570	5870	60170	82050	103930	114870	147690	7400	10030	12050	13040	15760	7132	9730	12320	13620	17510
2 7/8	6.85	3180	4340	5490	6070	7810	60380	82340	104300	115280	148220	3140	3600	4010	4190	4530	4630	6320	8000	8840	11370
	10.40	4400	6000	7590	8390	10790	93060	126900	160740	177660	228420	7920	10800	13680	14880	18230	7610	10380	13150	14540	18690
3 1/2	9.50	5580	7600	9630	10640	13680	86450	117880	148320	165040	212190	2840	3230	3650	3790	4000	4400	6000	7600	8400	10800
	13.30	7170	9770	12380	13680	17590	118965	162220	205480	227120	292000	6320	8040	9480	10160	11930	6350	8660	10970	12120	15580
	15.50	8010	10920	13830	15290	19660	139700	190500	241300	266700	342900	8070	11010	13950	15420	18960	7770	10590	13410	14820	19050
4	11.85	7700	10500	13310	14710	18910	103010	140470	177930	196650	252840	2210	2570	2790	2840	2850	3960	5400	6840	7560	9720
	14.00	9130	12440	15760	17420	22400	126555	172580	218600	241600	310640	3880	4630	5070	5230	5810	5000	6820	8640	9560	12280
	15.70	10040	13690	17340	19160	24640	142700	194600	246490	272430	350270	5210	6490	7480	7920	8940	5750	7840	9930	10970	14110
4 1/2	13.75	10280	14010	17750	19620	25220	120800	164730	208660	230620	296510	1850	2090	2170	2170	2170	3630	4950	6270	6930	8910
	16.60	12130	16530	20940	23150	29760	146800	200180	253560	280240	360320	3080	3520	3930	4110	4420	4520	6170	7810	8630	11100
	20.00	14350	19560	24780	27390	35210	181665	247720	313780	346820	445900	5280	6580	7590	8040	9100	5770	7870	9960	11010	14160
	22.82						205860	280720	355380	393000	505290	6960	8960	10680	11500	13710	6720	9170	11610	12830	16500
5	16.25	13910	18960	24020	26550	34130	146860	200260	253660	280360	360460	1780	1990	2050	2050	2050	3590	4890	6190	6850	8800
	19.50	16220	22120	28020	30970	39820	176220	240300	304380	336420	432540	2820	3210	3630	3770	3960	4380	5970	7560	8360	10750
	25.60	20260	27620	34990	38670	49720	232870	317550	402230	444570	571590	5760	7250	8460	9020	10410	6050	8250	10450	11550	14850
5 1/2	19.20						166840	227510	288180	318520	409520	1520	1640	1640	1640	1640	3340	4550	5770	6380	8200
	21.90	20060	27350	34640	38290	49230	195700	266040	336980	372460	478870	2220	2580	2810	2860	2870	3980	5430	6870	7600	9770
	24.70	22260	30350	38450	42490	54640	221045	301420	381800	422000	542560	3140	3600	4000	4190	4520	4560	6220	7880	8710	11190
6 5/8	25.20						219960	299950	379930	419930	539900	1160	1170	1170	1170	1170	3020	4120	5220	5770	7420

1 [1]Based on the shear strength equal to 57.7% ofminimum yield strength.
[2]Torsional data based on 45% eccentric wear and Tensile data based on 37.5% uniform wear.
[3]Data is based on minimum wall of 55% nominal wall.
NOTE: Calculations for Class III drill pipe are based on formulas in Appendix A, API RP7G.
Table is based on API RP7G, tables 2.7 and 2.8

(text continued from page 739)

Consider a case in which the drill pipe is exposed to an axial load (P) and a torque (T). The axial stress (σ_z) and the shear stress (τ) are given by the following formulas:

$$\sigma_z = \frac{P}{A} \tag{4-55}$$

$$\tau = \frac{T}{Z} \tag{4-56}$$

where P = axial load in lb
A = cross-sectional area of drill pipe in.^{2}
T = torque in in-lb
Z = polar section modulus of drill pipe, in.^{3}
Z = $2J/D_{dp}$
J = $(\pi/32)(D^4_{dp} - d^4_{dp})$ polar moment of inertia, in.^{4}
D_{dp} = outside diameter of drill pipe in in.
d_{dp} = inside diameter of drill pipe in in.

Substituting Equation 4-55 and Equation 4-56 into Equation 4-54 and putting $\sigma_e = Y_m$, $\sigma_t = 0$ (tangential stress equals zero in this case), the following formulas are obtained:

$$P = A\left[Y_m^2 - 3\left(\frac{T}{Z}\right)^2\right]^{1/2} \tag{4-57}$$

$$P = \left[P_t^2 - 3\left(\frac{AT}{Z}\right)^2\right]^{1/2} \tag{4-58}$$

where $P_t = Y_m A$ = tensile load capacity of drill pipe in uniaxial tensile stress in lb

Equation 4-58 permits calculation of the tensile load capacity when the pipe is subjected to rotary torque (T).

Example

Determine the tensile load capacity of a $4\frac{1}{2}$-in., 16.6-lb/ft, steel grade X-95 new drill pipe subjected to a rotary torque of 12,000 ft-lb if the required safety factor is 2.0.

Solution

From Table 4-71, cross-sectional area body of pipe A = 4.4074 in.^{2} and Polar section modulus Z = 8.542 in.^{3}

From Table 4-80, tensile load capacity of drill pipe at the minimum yield strength P_t = 418,700 lb (P_t = 4.4074 × 95,000 = 418,703 lb).

Using Equation 4-58,

$$P = \left[(418700)^2 - 3\left(\frac{4.4079 \times 144{,}000}{8.542}\right)^2\right]^{1/2} = 398{,}432\,\text{lb}$$

Due to the safety factor of 2.0 the tensile load capacity of the drill pipe is 398432/2 = 199,216 lb.

Example

Calculate the maximum value of a rotary torque that may be applied to the drill pipe as specified in Example 5 if the actual working tension load P = 300,000 lb. (For instance, pulling and trying to rotate a differentially stuck drill string.)

Solution

From Equation 4-58, the magnitude of rotary torque is

$$T = \frac{Z}{A}\left(\frac{P_t^2 - P^2}{3}\right)^{1/2}$$

so

$$T = \frac{8.542}{4.4074}\left[\frac{(418{,}700^2 - 300{,}000)^2}{3}\right]^{1/2} = 353{,}571 \text{ in-lb or } 29{,}464 \text{ ft-lb}$$

Caution: No safety factor is included in this example calculation. Additional checkup must be done if the obtained value of the torque is not greater than the recommended makeup torque for tool joints.

During normal rotary drilling processes, due to frictional pressure losses, the pressure inside the drill string is greater than that of the outside drill string. The greatest difference between these pressures is at the surface.

If the drill string is thought to be a thin wall cylinder with closed ends, then the drill pipe pressure produces the axial stress and tangential stress given by the following formulas:

(For stress calculations, the pressure loss in the annulus may be ignored.)

$$\sigma_a = \frac{P_{dp} D_{dp}}{4t} \tag{4-59}$$

$$\sigma_t = \frac{P_{dp} D_{dp}}{2t} \tag{4-60}$$

where σ_a = axial stress in psi
σ_t = tangential stress in psi
P_{dp} = internal drill pipe pressure in psi
t = wall thickness of drill pipe in psi
D_{dp} = outside diameter of drill pipe in in.

Substituting Equations 4-59, 4-60, 4-56, and 4-55 into Equation 4-54 and solving for the tensile load capacity of drill pipe yields

$$P = \left[P_t^2 - \frac{3}{16}\left(\frac{P_{dp}D_{dp}A}{t}\right)^2 - 3\left(\frac{AT}{Z}\right)^2 \right]^{1/2} \tag{4-61}$$

Example

Find the tensile load capacity of 5-in., nominal weight 19.5-lb/ft, steel grade E, premium class drill pipe exposed to internal drill pipe pressure P_{dp} = 3,000 psi and rotary torque T = 15,000 ft-lb.

Solution

From Table 4-79 Nominal D_{dp} = 5 in., nominal d_{dp} = 4.276 in., nominal wall thickness t = 0.362. Reduced wall thickness for premium class drill pipe = (0.8)(0.362) = 0.2896 in. Reduced D_{dp} for premium class = 4.276 + (2)(0.2896) = 4.8552 in. Cross-sectional area for premium class = Area based on reduced D_{dp} – Area based on nominal d_{dp}:

$$d_{dp} = \frac{\pi}{4}(4.8552)^2 - \frac{\pi}{4}(4.276)^2 = 4.1538\,\text{in.}^2$$

Section modulus for premium class:

$$Z = \frac{\pi}{16}\left(\frac{D_{dp}^4 - d_{dp}^4}{D_{dp}}\right) = \frac{\pi}{16}\left(\frac{4.8552^4 - 4.276^4}{4.8552}\right) = 8.9526\,\text{in.}^3$$

From Table 4-81, P_t = 311,400 lb (using Equation 4-61),

$$P = \left\{(311,400) - \frac{3}{16}\left[\frac{(3,000)(4.8552)(4.1538)}{0.2896}\right]^2 - 3\left[\frac{(4.1538)(180,000)}{8.9526}\right]^2\right\}^{1/2}$$

$$= 260,500\,\text{lb}$$

The reduction in the tensile load capacity of the drill pipe is 311,400 – 260,500 = 50,900 lb. That is about 17% of the tensile drill pipe resistance calculated at the minimum yield strength in uniaxial state of stress. For practical purposes, depending upon drilling conditions, a reasonable value of safety factor should be applied.

During DST operations, the drill pipe may be affected by a combined effect of collapse pressure and tensile load. For such a case,

$$\frac{\sigma_t}{Y_m} = \frac{P_{cc}}{P_c} \tag{4-62}$$

or

$$\sigma_t = Y_m \frac{P_{cc}}{P_c} \tag{4-63}$$

where P_c = minimum collapse pressure resistance as specified by API in psi
P_{cc} = corrected collapse pressure resistance for effect of tension in psi
Y_m = minimal yield strength of pipe in psi

Substituting Equation 4-63 and Equation 4-55 into Equation 4-54 (note: $\sigma_r = 0$, $\tau = 0$, $\sigma_e = Y_m$) and solving P_{cc} yields

$$P_{cc} = P_c\left\{\left[1-3\left(\frac{P}{2AY_m}\right)^2\right]^{1/2} - \frac{P}{2AY_m}\right\} \tag{4-64}$$

or

$$P_{cc} = P_c\left\{\left[1-0.75\left(\frac{\sigma_z}{Y_m}\right)^2\right]^{1/2} - 0.5\frac{\sigma_z}{Y_m}\right\} \tag{4-65}$$

Equation 4-65 indicates that increased tensile load results in decreased collapse pressure resistance. The decrement of collapse pressure resistance during normal DST operations is relatively small; nevertheless, under certain conditions, it may be quite considerable.

Example

Determine if the drill pipe is strong enough to satisfy the safety factor on collapse of 1.1 for the DST conditions as below:

- Drill pipe: $4\frac{1}{2}$-in., 16.6-lb/ft nominal weight, G-105 steel grade, class 2
- Drilling fluid with a density of 12 lb/gal and drill pipe empty inside
- Packer set at the depth of 8,500 ft
- Tension load of 45,000 lb, applied to the drill pipe

From Table 4-84, the collapse pressure resistance in uniaxial state of stress, P_c - 6,010 psi. Reduced wall thickness for class 2 drill pipe = (0.65)(0.337) = 0.219 in. Reduced D_{dp} for class 2 drill pipe = 3.826 + (2)(0.219) = 4.264 in. Reduced cross-sectional area of class 2 drill pipe equals:

$$\frac{\pi}{4}(4.264)^2 - \frac{\pi}{4}(3.826)^2 = 2.783\,\text{in.}^2$$

The axial tensile stress at packer level is

$$\sigma_z = \frac{45,000}{2.783} = 16,170\,\text{psi}$$

The corrected collapse pressure resistance according to Equation 4-65 is

$$P_{cc} = 6,010\left\{\left[1 - 0.75\left(\frac{16,170}{105,000}\right)^2\right] - 0.5\frac{16,170}{105,000}\right\} = 5,493 \text{ psi}$$

Hydrostatic pressure of the drilling fluid behind the drill string at the packer level is

$$P_h = (0.052)(12)(8,500) = 5,304 \text{ psi}$$

Obtained safety factor = 5,493/5,304 = 1.0356.

Since the obtained magnitude of safety factor (1.03) is less than desired (1.1), the drill pipe must not be run empty inside.

Tool Joints

The heart of any drill pipe string is the threaded rotary shoulder connection (Figure 4-133), known as the tool joint. Today, the only API standard tool joint is the weld-on joint shown at the bottom of Figure 4-133.

Tool joint dimensions for drill pipe grades E, X, G and S (recommended by API) are given in Table 4-84. Selection of tool joints should be discussed with the manufacturer. This is due to the fact that, up to the present time, there are no fully reliable formulas for calculating load capacity of tool joints. It is recommended that a tool joint be selected in such a manner that the torsional load capacity of the tool joint and the drill pipe would be comparable. The decision can be based on data specified in Tables 4-85 through 4-88.

Makeup Torque of Tool Joints

The tool joint holds drill pipe together, and the shoulders (similar to drill collars) form a metal-to-metal seal to avoid leakage. The tool joint threads are designed to be made up with drilling fluid containing solids. Clearance must be provided at the crest and root of threads in order to accommodate these solids. Therefore, the shoulder is the only seal. To keep the shoulders together, proper makeup torque is required.

However, makeup torque applied to the tool joint produces as axial preloading in the pin and the box as well as a torsional stress.

In particular, makeup torque induces a tensile state of stress within the pin and compression stress in the box. Thus, when the tool joint is exposed to the additional axial load due to the weight of the drill string suspended below the joint, the load capacity of the tool joint is determined by the tensile strength of the pin.

The magnitude of the makeup torque corresponding to the maximum load capacity of the tool joint is called the recommended makeup torque.

Therefore, the actual torque applied to the drill string should not exceed the recommended makeup torque; otherwise, the load capacity of the tool joint is reduced.

The API recommended makeup torque for different types of tool joints and classes of drill pipe is given in Table 4-89.

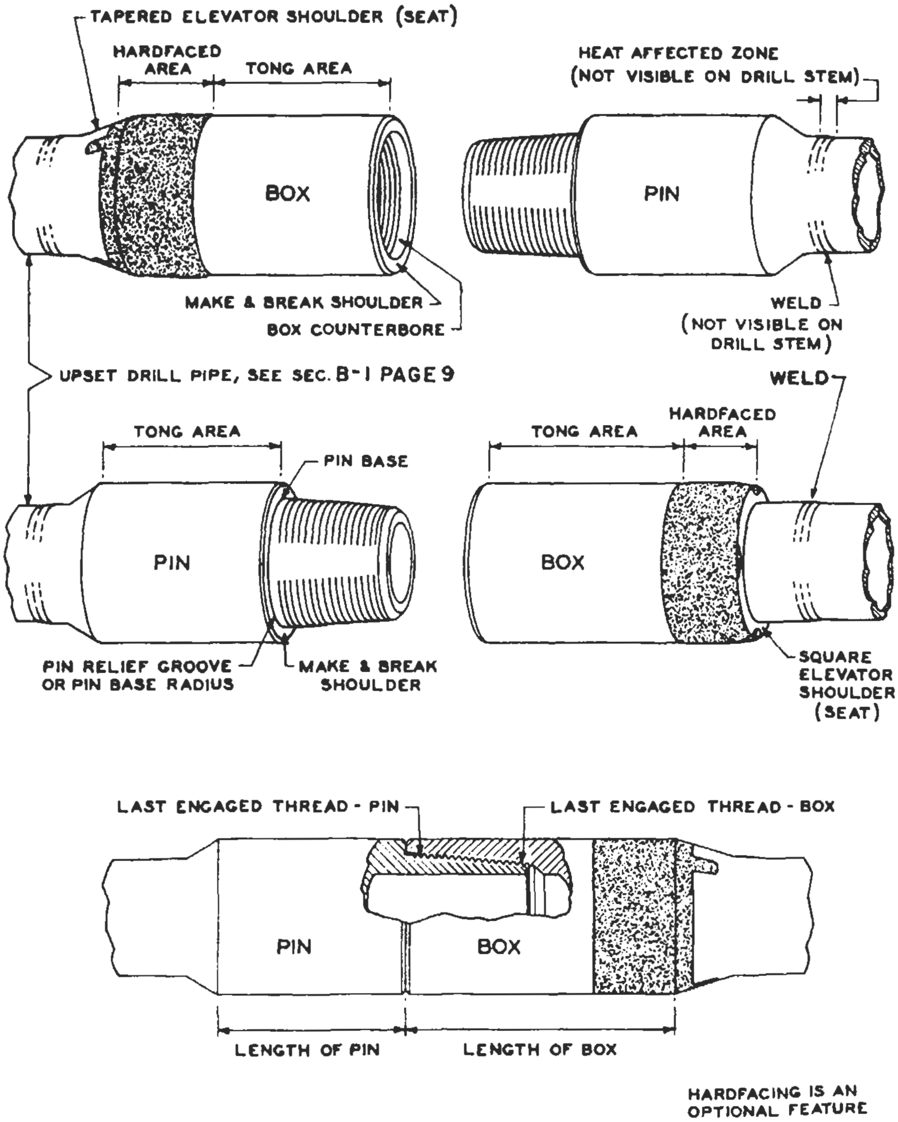


Figure 4-133. Tool joint nomenclature [30].

Heavy-Weight Drill Pipe

Heavy-weight drill pipe (with wall thicknesses of approximately 1 in.) is frequently used for drilling vertical and directional holes (Figure 4-134). So far, there is no sound, consistent, engineering theory of drill string behavior while

(text continued on page 760)

Table 4-84
Tool Joint Dimensions for Grade E, X, G and S Drill Pipe [30]

1	2	3	4	5	6	7	8	9	10	11	12	13	14
	DRILL PIPE			TOOL JOINT									
Tool Joint Designation[1]	Size and Style	Nom. Wt.[2] lb per ft	Grade	Outside Dia of Pin and Box ±1/32 D	Inside Dia of Pin[3] +1/64 −1/32 d	Bevel Dia of Pin and Box Shoulder ±1/64 D_F	Total Length Tool Joint Pin +1/4 −1/8 L_P	Pin Tong Space ±1/4 L_{PB}	Box Tong Space ±1/4 L_B	Combined Length of Pin and Box ±1/2 L	Dia of Pin at Elevator Upset, Max D_{PE}	Dia of Box at Elevator Upset, Max D_{TE}	Torsional Ratio, Pin to Drill Pipe
NC26(2 3/8 IF)	2 3/8 EU	6.65	E75	3 3/8°	1 3/4°	3 17/64	9	6	7	13	2 9/16	2 9/16	1.10
			X95	3 3/8°	1 3/4°	3 17/64	9	6	7	13	2 9/16	2 9/16	.87
			G105	3 3/8°	1 3/4°	3 17/64	9	6	7	13	2 9/16	2 9/16	.79
NC31(2 7/8 IF)	2 7/8 EU	10.40	E75	4 1/8°	2 1/8°	3 61/64	9 1/2	6	8	14	3 3/16	3 3/16	1.03
			X95	4 1/8°	2	3 61/64	9 1/2	6	8	14	3 3/16	3 3/16	.90
			G105	4 1/8°	2	3 61/64	9 1/2	6	8	14	3 3/16	3 3/16	.82
			S135	4 3/8	1 5/8	3 61/64	9 1/2	6	8	14	3 3/16	3 3/16	.82
NC38[4]	3 1/2 EU	9.50	E75	4 3/4°	3	4 37/64	10 1/2[4]	7	9 1/2	16 1/2	3 7/8	3 7/8	.91
NC38(3 1/2 IF)	3 1/2 EU	13.30	E75	4 3/4°	2 11/16°	4 37/64	11	7	9 1/2	16 1/2	3 7/8	3 7/8	.98
			X95	5	2 9/16	4 49/64	11	7	9 1/2	16 1/2	3 7/8	3 7/8	.87
			G105	5	2 7/16	4 49/64	11	7	9 1/2	16 1/2	3 7/8	3 7/8	.86
			S135	5	2 1/8	4 49/64	11	7	9 1/2	16 1/2	3 7/8	3 7/8	.80
		15.50	E75	5	2 9/16	4 49/64	11	7	9 1/2	16 1/2	3 7/8	3 7/8	.97
			X95	5	2 7/16	4 49/64	11	7	9 1/2	16 1/2	3 7/8	3 7/8	.83
			G105	5	2 1/8	4 49/64	11	7	9 1/2	16 1/2	3 7/8	3 7/8	.90
NC40(4FH)	3 1/2 EU	15.50	S135	5 1/2	2 1/4	5 1/64	11 1/2	7	10	17	3 7/8	3 7/8	.87
	4 IU	14.00	E75	5 1/4°	2 13/16°	5 1/64	11 1/2	7	10	17	4 3/16	4 3/16	1.01
			X95	5 1/4°	2 11/16	5 1/64	11 1/2	7	10	17	4 3/16	4 3/16	.86
			G105	5 1/2	2 7/16	5 1/64	11 1/2	7	10	17	4 3/16	4 3/16	.93
			S135	5 1/2	2	5 1/64	11 1/2	7	10	17	4 3/16	4 3/16	.87
NC46(4IF)	4 EU	14.00	E75	6°	3 1/4°	5 23/32	11 1/2	7	10	17	4 1/2	4 1/2	1.43
			X95	6°	3 1/4°	5 23/32	11 1/2	7	10	17	4 1/2	4 1/2	1.13
			G105	6°	3 1/4°	5 23/32	11 1/2	7	10	17	4 1/2	4 1/2	1.02
			S135	6°	3	5 23/32	11 1/2	7	10	17	4 1/2	4 1/2	.94
	4 1/2 IU	16.60	E75	6°	3 1/4°	5 23/32	11 1/2	7	10	17	4 11/16	4 11/16	1.09
			X95	6°	3	5 23/32	11 1/2	7	10	17	4 11/16	4 11/16	1.01
			G105	6°	3	5 23/32	11 1/2	7	10	17	4 11/16	4 11/16	.91
			S135	6 1/4	2 3/4	5 23/32	11 1/2	7	10	17	4 11/16	4 11/16	.81
	4 1/2 IEU	20.00	E75	6°	3	5 23/32	11 1/2	7	10	17	4 11/16	4 11/16	1.07
			X95	6 1/4	2 3/4	5 23/32	11 1/2	7	10	17	4 11/16	4 11/16	.96
			G105	6 1/4	2 1/2	5 23/32	11 1/2	7	10	17	4 11/16	4 11/16	.96
			S135	6 1/4	2 1/4	5 23/32	11 1/2	7	10	17	4 11/16	4 11/16	.81
4 1/2 FH°°	4 1/2 IU	16.60	E75	6°	3°	5 23/32	11	7	10	17	4 11/16	4 11/16	1.12
			X95	6°	2 3/4	5 23/32	11	7	10	17	4 11/16	4 11/16	1.02
			G105	6°	2 3/4	5 23/32	11	7	10	17	4 11/16	4 11/16	.92
			S135	6 1/4	2 1/2	5 23/32	11	7	10	17	4 11/16	4 11/16	.81
	4 1/2 IEU	20.00	E75	6°	3°	5 23/32	11	7	10	17	4 11/16	4 11/16	.95
			X95	6°	2 1/2	5 23/32	11	7	10	17	4 11/16	4 11/16	.95
			G105	6°	2 1/2	5 23/32	11	7	10	17	4 11/16	4 11/16	.86

°Denotes standard OD or standard ID.
°°Obsolescent connection.
[1]The tool joint designation (Col. 1) indicates the size and style of the applicable connection.
[2]Nominal weights, threads and couplings, (Col. 3) are shown for the purpose of identification in ordering.
[3]The inside diameter (Col. 6) does not apply to box members, which are optional with the manufacturer.
[4]Length of pin thread reduced to 3 1/2 inches (1/2 inch short) to accommodate 3 inch ID.
NOTE 1: Neck diameters (D_{PE} & D_{TE}) and inside diameters (d) of tool joints *prior* to welding are at manufacturer's option. The above table specifies finished dimensions after final machining of the assembly.
NOTE 2: Appendix H contains more dimensions of obsolescent connections and for square elevator shoulders.
NOTE 3: No torsional ratio (tool joint pin to drill pipe) less than 0.80 is shown. In particular areas, tool joints having much smaller torsional yields may prove to be adequate.

Table 4-84 (continued)

1	2	3	4	5	6	7	8	9	10	11	12	13	14
	DRILL PIPE			TOOL JOINT									
Tool Joint Designation[1]	Size and Style	Nom. Wt.[2] lb per ft	Grade	Outside Dia of Pin and Box ±1/32	Inside Dia of Pin[3] +1/64 −1/32	Bevel Dia of Pin and Box Shoulder ±1/64	Total Length Tool Joint Pin +¼ −⅛	Pin Tong Space ±¼	Box Space ±¼	Combined Length of Pin and Box ±½	Dia of Pin at Elevator Upset, Max	Dia of Box at Elevator Upset, Max	Torsional Ratio, Pin to Drill Pipe
				D	d	D_F	L_P	L_{PB}	L_B	L	D_{PE}	D_{TE}	
NC50(4½IF)	4½ EU	16.60	E75	6⅝°	3¾°	5[illegible]	11½	7	10	17	5	5	1.23
			X95	6⅝°	3¾°	5[illegible]	11½	7	10	17	5	5	.97
			G105	6⅝°	3¾°	5[illegible]	11½	7	10	17	5	5	.88
			S135	6⅝°	3½	5[illegible]	11½	7	10	17	5	5	.81
	4½ EU	20.00	E75	6⅝°	3¾°	5[illegible]	11½	7	10	17	5	5	1.02
			X95	6⅝°	3½	5[illegible]	11½	7	10	17	5	5	.96
			G105	6⅝°	3½	5[illegible]	11½	7	10	17	5	5	.86
			S135	6⅝	3	5[illegible]	11½	7	10	17	5	5	.87
	5 IEU	19.50	E75	6⅝°	3¾°	5[illegible]	11½	7	10	17	5⅛	5⅛	.92
			X95	6⅝°	3½	5[illegible]	11½	7	10	17	5⅛	5⅛	.86
			G105	6½	3¼	5[illegible]	11½	7	10	17	5⅛	5⅛	.89
			S135	6⅝	2¾	5[illegible]	11½	7	10	17	5⅛	5⅛	.86
	5 IEU	25.60	E75	6⅝°	3½	5[illegible]	11½	7	10	17	5⅛	5⅛	.86
			X95	6½	3	5[illegible]	11½	7	10	17	5⅛	5⅛	.86
			G105	6⅝	2¾	5[illegible]	11½	7	10	17	5⅛	5⅛	.87
5½ FH°°	5 IEU	19.50	E75	7°	3¾	6[illegible]	13	8	10	18	5⅛	5⅛	1.53
			X95	7°	3¾	6[illegible]	13	8	10	18	5½	5⅛	1.21
			G105	7°	3¾	6[illegible]	13	8	10	18	5⅛	5⅛	1.09
			S135	7¼	3½	6[illegible]	13	8	10	18	5⅛	5⅛	.98
	5 IEU	25.60	E75	7°	3½	6[illegible]	13	8	10	18	5⅛	5⅛	1.21
			X95	7°	3½	6[illegible]	13	8	10	18	5⅛	5⅛	.95
			G105	7¼	3½	6[illegible]	13	8	10	18	5½	5⅛	.99
			S135	7¼	3¼	6[illegible]	13	8	10	18	5⅛	5⅛	.83
	5½ IEU	21.90	E75	7°	4	6[illegible]	13	8	10	18	5[illegible]	5[illegible]	1.11
			X95	7°	3¾	6[illegible]	13	8	10	18	5[illegible]	5[illegible]	.98
			G105	7¼	3½	6[illegible]	13	8	10	18	5[illegible]	5[illegible]	1.02
			S135	7½	3	7[illegible]	13	8	10	18	5[illegible]	5[illegible]	.96
	5½ IEU	24.70	E75	7°	4°	6[illegible]	13	8	10	18	5[illegible]	5[illegible]	.99
			X95	7¼	3½	6[illegible]	13	8	10	18	5[illegible]	5[illegible]	1.01
			G105	7¼	3½	6[illegible]	13	8	10	18	5[illegible]	5[illegible]	.92
			S135	7½	3	7[illegible]	13	8	10	18	5[illegible]	5[illegible]	.86

°Denotes standard OD or standard ID.
°°Obsolescent connection.
[1]The tool joint designation (Col. 1) indicates the size and style of the applicable connection.
[2]Nominal weights, threads and couplings, (Col. 3) are shown for the purpose of identification in ordering.
[3]The inside diameter (Col. 6) does not apply to box members, which are optional with the manufacturer.
[4]Length of pin thread reduced to 3¼ inches (⅛ inch short) to accommodate 3 inch ID.
NOTE 1: Neck diameters (D_{PE} & D_{TE}) and inside diameters (d) of tool joints *prior* to welding are at manufacturer's option. The above table specifies finished dimensions after final machining of the assembly.
NOTE 2: Appendix H contains more dimensions of obsolescent connections and for square elevator shoulders.
NOTE 3: No torsional ratio (tool joint pin to drill pipe) less than 0.80 is shown. In particular areas, tool joints having much smaller torsional yields may prove to be adequate.

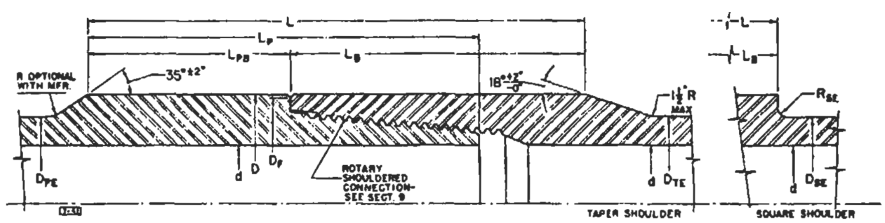


Table 4-85
Selection Chart—Tool Joints Applied to Standard Weight Drill Pipe—Grade E [30]

DRILL PIPE DATA – TOOL JOINT ATTACHED							TOOL JOINT DATA					MECHANICAL PROPERTIES			
				Upset						Tong Space in.		Tensile Yield lb		Torsional Yield ft–lb	
Nom. Size	Nom. Wt. lb/ft	Adj. Wt. lb/ft (1)	I.D. in.	Type	Max. O.D. in.	Drift Dia	Conn.	O.D. in.	I.D. in.	Box	Pin	Pipe (2)	Tool Joint (3)	Pipe (4)	Tool Joint (5)
2 3/8	6.65	6.75	1.815	IU	2 1/2	1.312	PAC	2 7/8	1 3/8	7	6	138220	262000	6240	5200
		6.87		EU	2 9/16	1.627	O.H.	3 1/4	1 3/4	7	6	138220	287280	6240	6400
		7.00		EU	2 9/16	1.627	NC26(I.F.)	3 3/8	1 3/4	7	6	138220	323760	6240	6800
2 7/8	10.40	10.29	2.151	IU	3	1.438	PAC	3 1/8	1 1/2	8	6	214340	269470	11530	5800
		11.20		IU	3	1.812	E.H.	4 1/4	1 7/8	8	6	214340	516840	11530	13400
		10.37		IU	3	1.688	NC26(S.H.)	3 3/8	1 3/4	8	6	214340	323760	11530	6800
		10.82		EU	3 3/16	1.963	NC31(I.F.)	4 1/8	2 1/8	8	6	214340	459120	11530	12000
		10.57		EU	3 3/16	1.963	O.H.	3 7/8	2 5/32	8	6	214340	345840	11530	8600
		10.53		EU	3 3/16	2.006	SL-H90	3 7/8	2 5/32	8	6	214340	384600	11530	11800
3 1/2	13.30	14.06	2.764	IU	3 11/16	2.313	E.H.	4 3/4	2 7/16	9 1/2	7	271570	584880	18520	17300
		13.51		IU	3 11/16	2.000	NC31(S.H.)	4 1/8	2 1/8	8 1/2	6 1/2	271570	459120	18520	12000
		13.86		EU	3 7/8	2.446	NC38(I.F.)	4 3/4	2 11/16	9 1/2	7	271570	602160	18520	18700
		13.86		EU	3 7/8	2.446	O.H.	4 3/4	2 11/16	9 1/2	7	271570	560040	18520	14900
4	14.00	15.13	3.340	IU	4 3/16	2.688	NC40(F.H.)	5 1/4	2 13/16	10	7	285360	727440	23250	24000
		14.29		IU	4 3/16	2.438	S.H.	4 5/8	2 9/16	9 1/2	6 1/2	285360	525840	23250	15400
		15.50		IU	4 3/16	2.688	H90	5 1/2	2 13/16	10	7	285360	913680	23250	35520
		15.56		EU	4 1/2	3.125	NC46(I.F.)	6	3 1/4	10	7	285360	918960	23250	34100
		15.07		EU	4 1/2	3.125	O.H.	5 1/2	3 1/4	10	7	285360	760080	23250	26800
4 1/2	16.60	17.94	3.826	IU	4 11/16	3.125	NC46(E.H.)	6	3 1/4	10	7	330560	918960	30750	34100
		17.70		IU	4 11/16	2.875	R.H.	6	3	10	7	330560	976440	30750	34600
		16.66		IU	4 11/16	2.562	NC38(S.H.)	5	2 11/16	9 1/2	6 1/2	330560	601880	30750	18700
		17.94		IU	4 11/16	3.125	H90	6	3 1/4	10	7	330560	938280	30750	37400
		17.64		EU	5	3.625	NC50(I.F.)	6 3/8	3 3/4	10	7	330560	958800	30750	37900
		17.22		EU	5	3.125	O.H.	5 7/8	3 3/4	10	7	330560	718080	30750	27400
5	19.50	20.99	4.276	IEU	5 1/8	3.625	NC50(E.H.)	6 3/8	3 3/4	10	7	395600	958800	41090	37900
5 1/2	21.90	23.94	4.778	IEU	5 11/16	3.875	F.H.	7	4	10	8	437120	1265880	50620	55700
6 5/8	25.20	27.14	5.965	IU	6 3/4	4.875	F.H.	8	5	14 1/2	9 1/2	489460	1448880	70550	73000

[1] Tool Joint Plus 29.4' of Drill Pipe.
[2] Tensile Yield Strength of Drill Pipe Based on 75,000 psi.
[3] Tensile Yield Strength of the Tool Joint Pin is based on 120,000 psi Yield and the Cross Sectional Area at the Root of the Thread 5/8 inch from the Shoulder.
[4] Torsional Yield Strength of the Drill Pipe is Based on a Shear Strength of 57.7% of the Minimum Yield Strength.
[5] Torsional Yield Strength of the Tool Joint Based on Tensile Yield Strength of the Pin and Compressive Yield Strength of the Box – Lowest Value Prevailing.

Table 4-86
Selection Chart—Tool Joints Applied to Lightweight Drill Pipe—Grade E [30]

DRILL PIPE DATA - TOOL JOINT ATTACHED							TOOL JOINT DATA					MECHANICAL PROPERTIES			
				Upset						Tong Space in.		Tensile Yield lb		Torsional Yield ft-lb	
Nom. Size	Nom. Wt. lb/ft	Adj. Wt. lb/ft (1)	I.D. in.	Type	Max. O.D. in.	Drift Dia	Conn.	O.D. in.	I.D. in.	Box	Pin	Pipe (2)	Tool Joint (3)	Pipe (4)	Tool Joint (5)
2 3/8	4.85	5.15	1.995	EU	2 9/16	1.688	NC26(I.F.)	3 3/8	1 3/4	7	6	97820	323760	4760	6800
		4.94		EU	2 9/16	1.850	S.L.-H90	3 1/4	1.995	7	6	97820	204550	4760	5500
		4.87		EU	2 9/16	1.807	O.H.	3 1/8	2	7	6	97820	206400	4760	4400
		5.09		EU	2 9/16	1.807	W.O.	3 3/8	2	7	6*	97820	205920	4760	4500
2 7/8	6.85	7.33	2.441	EU	3 3/16	2.063	NC31(I.F.)	4 1/8	2 1/8	8	6*	135900	459120	8070	12000
		6.97		EU	3 3/16	2.296	S.L.-H90	3 7/8	2.441	8	6	135900	259180	8070	7600
		6.91		EU	3 3/16	2.253	O.H.	3 3/4	2 7/16	8	6	135900	224040	8070	5800
		7.25		EU	3 3/16	2.253	W.O.	3 7/8	2 7/16	8	6*	135900	289080	8070	7500
3 1/2	9.50	10.39	2.992	EU	3 7/8	2.563	NC38(I.F.)	4 3/4	2 11/16	9 1/2	7*	194260	602160	14120	18700
		10.12		EU	3 7/8	2.847	S.L.-H90	4 5/8	2.992	9 1/2	6 1/2	194260	370970	14120	13200
		9.95		EU	3 7/8	2.804	O.H.	4 1/2	3	9 1/2	7	194260	392280	14120	11800
		10.25		EU	3 7/8	2.804	W.O.	4 3/4	3	9 1/2	7*	194260	434400	14120	13400
	11.85	13.09	3.476	IU	4 3/16	2.688	H90	5 1/2	2 13/16	10	7	230750	913680	19440	35500
		13.13		EU	4 1/2	3.125	NC46(I.F.)	5 3/4	3 1/4	10	7*	230750	918960	19440	34100
		12.16		EU	4 1/2	3.287	O.H.	5 1/4	3 15/32	10	7	230750	621960	19440	22100
		13.03		EU	4 1/2	3.287	W.O.	5 3/4	3 7/16	10	7*	230750	801120	19440	29300
4 1/2	13.75	15.24	3.958	IU	4 11/16	3.125	H90	6	3 1/4	10	7	270030	938280	25860	37400
		14.98		EU	5	3.625	NC50(I.F.)	6 1/8	3 3/4	10	7*	270030	958920	25860	37900
		14.10		EU	5	3.770	O.H.	5 3/4	3 31/32	10	7	270030	559440	25860	21100
		14.90		EU	5	3.770	W.O.	6 1/8	3 7/8	10	7*	270030	868920	25860	34100

*If weight is of paramount importance, these connections can be supplied with a smaller O.D. or shorter tong space on box and pin, or a combination of the two to afford maximum weight reduction without sacrificing safety in the joint. However, where possible, the manufacturers recommend ordering the joints shown to obtain the maximum economical tool joint service.

Other tool joint sizes to accommodate special situations can be furnished on request.

[1] Tool Joint Plus 29.4' of Drill Pipe.
[2] Tensile Yield Strength of Drill Pipe Based on 75,000 psi.
[3] Tensile Yield Strength of the Tool Joint Pin is based on 120,000 psi Yield and the Cross Sectional Area at the Root of the Thread 5/8 inch from the Shoulder.
[4] Torsional Yield Strength of the Drill Pipe is Based on a Shear Strength of 57.7% of the Minimum Yield Strength.
[5] Torsional Yield Strength of the Tool Joint Based on Tensile Yield Strength of the Pin and Compressive Yield Strength of the Box – Lowest Value Prevailing.

Table 4-87
Selection Chart—Tool Joints Applied to Heavy-Weight Drill Pipe—Grade E [30]

DRILL PIPE DATA - TOOL JOINT ATTACHED							TOOL JOINT DATA					MECHANICAL PROPERTIES			
				Upset						Tong Space in.		Tensile Yield lb		Torsional Yield ft–lb	
Nom. Size	Nom. Wt. lb/ft	Adj. Wt. lb/ft (1)	I.D. in.	Type	Max. O.D. in.	Drift Dia	Conn.	O.D. in.	I.D. in.	Box	Pin	Pipe (2)	Tool Joint (3)	Pipe (4)	Tool Joint (5)
3 1/2	15.50	16.42	2.602	EU	3 7/8	2.414	NC38(I.F.)	5	2 9/16	9 1/2	7	322780	663550	21050	20600
4	15.70†	16.99	3.240	IU	4 3/16	2.562	NC40(F.H.)	5 1/4	2 11/16	10	7	324150	791620	25760	25900
		17.30		IU	4 3/16	2.688	H90	5 1/2	2 13/16	10	7	324150	913680	25760	35500
		17.43		EU	4 1/2	3.052	NC46(I.F.)	6	3 1/4	10	7	324150	918960	25760	34100
4 1/2	20.00	21.73	3.640	IEU	4 11/16	2.875	NC46(E.H.)	6	3	10	7	412360	1066030	36840	39400
		21.73		IEU	4 11/16	2.875	F.H.	6	3	10	7	412360	976130	36840	34600
		21.73		IEU	4 11/16	2.875	H90	6	3	10	7	412360	1085380	36840	44600
		22.33		EU	5	3.452	NC50(I.F.)	6 3/8	3 3/4	10	7	412360	95800	36840	37900
	22.82	24.53	3.500	IEU	4 11/16	2.875	NC46 (E.H.)	6	3	10	7	471240	1066030	40910	39400
		24.53		IEU	4 11/16	2.875	F.H.	6	3	10	7	471240	976130	40910	34600
		24.53		IEU	4 11/16	2.875	H-90	6	3	10	7	471240	1085380	40910	44600
		24.54		EU	5 1/8	3.312	NC50 (I.F.)	6 3/8	3 1/2	10	7	471240	958800	40910	37900
5	25.60	27.17	4.00	IEU	5 1/8	3.375	NC50(E.H.)	6 3/8	3 1/2	10	7	53180	1128960	52160	44600
		28.08		IEU	5 1/8	3.375	5 1/2F.H.	7	3 1/2	10	8	530180	1268540	52160	59000
5 1/2	24.70	26.86	4.670	IEU	5 11/16	3.875	F.H.	7	4	10	8	497220	1265880	56470	55700

†Not API weight

[1] Tool Joint Plus 29.4' of Drill Pipe.
[2] Tensile Yield Strength of Drill Pipe Based on 75,000 psi.
[3] Tensile Yield Strength of the Tool Joint Pin is based on 120,000 psi Yield and the Cross Sectional Area at the Root of the Thread 5/8 inch from the Shoulder.
[4] Torsional Yield Strength of the Drill Pipe is Based on a Shear Strength of 57.7% of the Minimum Yield Strength.
[5] Torsional Yield Strength of the Tool Joint Based on Tensile Yield Strength of the Pin and Compressive Yield Strength of the Box – Lowest Value Prevailing.

Table 4-88
Selection Chart—Tool Joints Applied to High Strength Drill Pipe [30]

DRILL PIPE DATA – TOOL JOINT ATTACHED								TOOL JOINT DATA					MECHANICAL PROPERTIES			
					Upset				Recommended		Tong Space in.		Tensile Yield lb		Torsional Yield ft–lb	
Nom. Size	Nom. Wt. lb/ft	Adj. Wt. lb/ft (1)	I.D. in.	Min. Tensile Yield Strength p.s.i.	Type	Max. O.D. in.	Drift Dia in.	Conn.	O.D. in.	I.D. in.	Box	Pin	Pipe (2)	Tool Joint (3)	Pipe (4)	Tool Joint (5)
2 7/8	10.40	9.94	2.151	95,000	EU	3 3/16	1.781	NC31(I.F.)	4 1/8	2	8	6	271330	507670	14610	13400
		10.53		95,000	EU	3 3/16	2.006	SL-H90	3 7/8	2 5/32	8	6	271330	384600	14610	11500
		11.09		105,000	EU	3 3/16	1.875	NC31(I.F.)	4 1/8	2	8	6	300080	507670	16150	13400
		11.45		135,000	EU	3 3/16	1.781	3 1/2E.H.	4 3/4	2	8	6	370080	767520	19968	23040
3 1/2	13.30	14.32	2.764	95,000	EU	3 7/8	2.313	NC38(I.F.)	5	2 9/16	9 1/2	7	343990	667300	23460	20600
		13.72		95,000	EU	3 7/8	2.563	SL-H90	4 5/8	2 11/16	9 1/2	6 1/2	343990	536520	23460	19200
		14.38		105,000	EU	3 7/8	2.313	NC38(I.F.)	5	2 7/16	9 1/2	7	380190	726220	25930	22100
		14.95		135,000	EU	3 7/8	2.125	NC40(4FH)	5 1/2	2 1/4	10	7	488820	995460	33330	32500
	15.50	16.54	2.602	95,000	EU	3 7/8	2.313	NC38(I.F.)	5	2 7/16	9 1/2	7	408850	726220	26660	22100
		16.88		105,000	EU	3 7/8	2.313	NC40(4FH)	5 1/2	2 7/16	10	7	451890	912580	29470	30200
4	14.00	15.29	3.340	95,000	IU	4 3/16	2.562	NC40(F.H.)	5 1/4	2 11/16	10	7	361460	791620	29450	25400
		15.45		95,000	IU	4 3/16	2.688	H90	5 1/2	2 13/16	10	7	361460	913680	29450	35500
		15.56		95,000	EU	4 1/2	2.875	NC46(I.F.)	6	3 1/4	10	7	361460	918720	29450	34100
		15.55		105,000	IU	4 3/16	2.313	NC40(F.H.)	5 1/2	2 7/16	10	7	399500	912580	32550	30200
		15.45		105,000	IU	4 3/16	2.688	H90	5 1/2	2 13/16	10	7	399500	913680	32550	35500
		15.56		105,000	EU	4 1/2	3.125	NC46(I.F.)	6	3 1/4	10	7	399500	918720	32550	34100
		16.17		135,000	EU	4 1/2	2.875	NC46(I.F.)	6	3	10	7	513650	1066030	41840	39400
	15.70†	17.41	3.240	95,000	IU	4 3/16	2.562	NC40(F.H.)	5 1/4	2 11/16	10	7	410590	791620	32630	25400
		17.30		95,000	IU	4 3/16	2.688	H90	5 1/2	2 13/16	10	7	410590	913680	32630	35500
		18.18		135,000	EU	4 1/2	2.625	NC46(I.F.)	6	2 3/4	10	7	583420	1201540	46380	44200
4 1/2	16.60	18.31	3.826	95,000	IU	4 11/16	2.625	F.H.	6	2 3/4	10	7	418700	1111680	38950	38900
		18.19		95,000	IU	4 11/16	2.875	NC46(E.H.)	6	3	10	7	418700	1066080	38950	39400
		18.19		95,000	IU	4 11/16	2.875	H90	6	3	10	7	418700	1085380	38950	44600
		17.99		95,000	EU	5 1/8	3.375	NC50(I.F.)	6 3/8	3 3/4	10	7	418700	958800	38950	37900
		18.38		105,000	IU	4 11/16	2.625	F.H.	6	2 3/4	10	7	462780	1111680	43050	38900
		18.31		105,000	IU	4 11/16	2.875	NC46(E.H.)	6	3	10	7	462780	1066030	43050	39400
		18.19		105,000	IU	4 11/16	2.875	H90	6	3	10	7	462780	1085380	43050	44600
		17.99		105,000	EU	5 1/8	3.375	NC50(I.F.)	6 3/8	3 3/4	10	7	462780	958800	43050	37900
		18.92		135,000	IU	4 11/16	2.375	NC46(E.H.)	6 1/4	2 3/4	10	7	595000	1201540	55350	44200
		18.80		135,000	EU	5 1/8	3.375	NC50(I.F.)	6 3/8	3 1/2	10	7	595000	1128460	55350	44600
	20.000	21.69	3.640	95,000	IEU	4 11/16	2.375	F.H.	6	2 1/2	10	7	522320	1235040	46660	43400
		21.84		95,000	IEU	4 11/16	2.625	NC46(E.H.)	6 1/4	2 3/4	10	7	522320	1201440	46660	44600
		21.73		95,000	IEU	4 11/16	2.875	H90	6	3	10	7	522320	1085380	46660	44600
		22.11		95,000	EU	5 1/8	3.375	NC50(I.F.)	6 3/8	3 1/2	10	7	522320	1128460	46660	44600
		22.06		105,000	IEU	4 11/16	2.375	NC46(E.H.)	6 1/4	2 1/2	10	7	577300	1325280	51570	49400
		22.26		105,000	EU	5 1/8	3.375	NC50(I.F.)	6 3/8	3 1/2	10	7	577300	1128460	51570	44600
		22.47		135,000	EU	5 1/8	2.875	NC50(I.F.)	6 5/8	3	10	7	742240	1416510	66300	57120

High Strength table continued on next page.

Table 4-88
(continued)

DRILL PIPE DATA – TOOL JOINT ATTACHED								TOOL JOINT DATA					MECHANICAL PROPERTIES			
				Min. Tensile Yield Strength	Upset				Recommended*		Tong Space in.		Tensile Yield lb		Torsional Yield ft–lb	
Nom. Size	Nom. Wt. lb/ft	Adj. Wt. lb/ft (1)	I.D. in.	in.	Type	Max. O.D. in.	Drift Dia in.	Conn.	O.D. in.	I.D. in.	Box	Pin	Pipe (2)	Tool Joint (3)	Pipe (4)	Tool Joint (5)
4 1/2	22.82	24.60	3.500	95,000	IEU	4 11/16	2.625	NC46(E.H.)	6 1/4	2 3/4	10	7	596,900	1201440	51800	44600
		25.08	3.500	95,000	IEU	4 11/16	2.125	F.H.	6 1/4	2 1/4	10	7	596,900	1235040	51800	43400
		24.55	3.500	95,000	EU	5	3.375	NC50(I.F.)	6 3/8	3 1/2	10	7	596.900	1128960	51800	44600
		24.71	3.500	105,000	IEU	4 11/16	2.375	NC46(E.H.)	6 1/4	2 1/2	10	7	659,740	1325280	57280	49400
		24.71	3.500	105,000	EU	5	3.125	NC50(I.F.)	6 3/8	3 1/4	10	7	659,740	1287840	57280	49900
		25.60	3.500	135,000	IEU	4 11/16	2.875	NC50(I.F.)	6 5/8	3	10	7	848,230	1436050	73640	59400
		24.83	3.500	135.000	EU	5	2.875	NC50(I.F.)	6 5/8	3	10	7	848.230	1436050	73640	59400
5	19.50	21.34	4.276	95,000	IEU	5 1/8	3.375	NC50(E.H.)	6 3/8	3 1/2	10	7	501090	1128460	52050	44600
		21.55		95.000	IEU	5 1/8	3.125	H90	6 1/2	3 1/4	10	7	501090	1176000	52050	51100
		21.60		105,000	IEU	5 1/8	3.125	NC50(E.H.)	6 1/2	3 1/4	10	7	553830	1287840	57530	49900
		23.24		135,000	IEU	5 1/8	3.375	5 1/2F.H.	7 1/4	3 1/2	10	8	712070	1619520	73970	71000
	25.60	28.62	4.000	95,000	IEU	5 1/8	3.375	5 1/2F.H.	7	3 1/2	10	8	671000	1619520	66070	62400
		28.99		105,000	IEU	5 1/8	3.375	5 1/2F.H.	7 1/4	3 1/2	10	8	742000	1619520	73030	71000
5 1/2	21.90	24.24	4.778	95,000	IEU	5 11/16	3.625	F.H.	7	3 3/4	10	8	553680	1448640	64120	62400
		24.36		95,000	IEU	5 11/16	3.375	H90	7	3 1/2	10	8	553680	1268540	64120	59000
		24.98		105,000	IEU	5 11/16	3.375	F.H.	7 1/4	3 1/2	10	8	611960	1619520	70870	71000
		25.86		135,000	IEU	5 11/16	2.874	F.H.	7 1/2	3	10	8	786810	1925760	91120	85000
		25.66		135,000	IEU	5 11/16	3.375	6 5/8Reg.	7 1/2	3 1/2	10	8	786810	1802320	91120	82560
	24.70	27.89	4.670	95,000	IEU	5 11/16	3.375	F.H.	7 1/4	3 1/2	10	8	629810	1619520	71530	71000
		27.89		105,000	IEU	5 11/16	3.375	F.H.	7 1/4	3 1/2	10	8	696110	1619520	79060	71000
		28.91		135,000	IEU	5 11/16	2.874	F.H.	7 1/2	3	10	8	895000	125760	101650	85000

[1] Tool Joint Plus 29.4' of Drill Pipe.
[2] Tensile Yield Strength of Drill Pipe Based on 75,000 psi.
[3] Tensile Yield Strength of the Tool Joint Pin is based on 120,000 psi Yield and the Cross Sectional Area at the Root of the Thread 5/8 Inch from the Shoulder.
[4] Torsional Yield Strength of the Drill Pipe is Based on a Shear Strength of 57.7% of the Minimum Yield Strength.
[5] Torsional Yield Strength of the Tool Joint Based on Tensile Yield Strength of the Pin and Compressive Yield Strength of the Box – Lowest Value Prevailing.

Note – Tool joints with outside diameters shown for Grade E on Tables B1-1, 2 and 3 are adequate for high strength drill pipe when used in combination strings with Grade E drill pipe. Thus, sizes shown for NC joints are designed so that *torsional yield of the joint is a minimum of 80% of that of the pipe* to which it is attached. Others have balanced torsional strength between pipe and joint.

Table 4-89
Recommended Minimum OD and Makeup Torque of Weld-on Type Tool Joints by Class [30]

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
DRILL PIPE DATA			NEW TOOL JOINT DATA				①PREMIUM CLASS			CLASS 2			CLASS 3		
Nom Size in.	Nom Wt lb/ft	Type Upset and Grade	Conn	New OD in	New ID in	①Make-Up Torque ft-lb	①Min OD Tool Joint in.	①Min. Box Shoulder With Eccentric Wear in	①Make-Up Torque For Min OD Tool Joint ft-lb	①Min OD Tool Joint in.	①Min Box Shoulder With Eccentric Wear in	①Make-Up Torque For Min OD Tool Joint ft-lb	①Min. OD Tool Joint in	①Min. Box Shoulder With Concentric Wear in	①Make-Up Torque For Min OD Tool Joint ft-lb
2 3/8	4.85	E.U. 75	W.O.	3 3/8	2	2200	3 3/32	3/64	2200	3 1/16	1/16	2100	3	1/16	1600
	4.85	E.U. 75	NC26 (I.F.)	3 3/8	1 3/4	3600	3 3/32	3/32	2300	3 3/32	1/16	1800	3 1/16	1/16	1500
	4.85	E.U. 75	O.H.	3 1/4	2	2400	3 1/16	7/64	2300	3	3/64	1800	2 31/32	3/64	1500
	4.85	E.U. 75	SL-H90	3 1/4	2	2800	3	3/32	2300	2 15/16	3/64	1700	2 29/32	3/64	1500
	6.65	E.U. 55	NC26 (I.F.)	3 3/8	1 3/4	3600	3 1/8	3/64	2000	3 3/32	1/16	1600	3 1/16	1/16	1500
	6.65	I.U. 75	P.A.C.	①2 7/8	1 3/8	2500	2 7/8	15/64	2500	2 7/8	15/64	2500	2 23/32	5/32	2200
	6.65	E.U. 75	NC26(I.F.)	3 3/8	1 3/4	3600	3 5/32	1/8	2800	3 3/32	3/32	2300	3 1/8	3/32	2000
	6.65	E.U. 75	SL-H90	3 1/4	2	2800	3 1/16	5/64	2800	3	7/64	2300	2 31/32	7/64	2000
	6.65	E.U. 75	O.H.	3 1/4	1 3/4	3300	3 3/32	13/64	3100	3 3/32	1/8	2600	3 1/32	1/8	2300
2 7/8	6.85	E.U. 75	NC31 (I.F.)	4 1/8	2 1/8	6300	3 27/32	1/8	3600	3 21/32	3/32	3000	3 5/8	3/32	2500
	8.85	E.U. 75	W.O.	4 1/8	2 7/16	3800	3 21/32	7/64	3700	3 5/8	3/32	3400	3 9/16	3/32	2600
	6.85	E.U. 75	O.H.	①3 3/4	2 7/16	2900	3 17/32	1/8	2900	3 1/2	7/64	2900	3 7/16	7/64	2600
	6.85	E.U. 75	SL-H90	3 7/8	2 7/16	3900	3 17/32	5/64	3900	3 15/32	7/64	3200	3 7/16	7/64	2800
	10.40	E.U. 55	NC31 (I.F.)	4 1/8	2 1/8	6300	3 3/4	1/8	4000	3 11/16	3/32	3200	3 5/8	3/32	2500
	10.40	I.U. 55	E.H.	4 1/4	1 7/8	7000	3 21/32	9/64	3700	3 19/32	7/64	2900	3 9/16	7/64	2500
	10.40	I.U. 55	NC26 (①S.H.)	①3 3/8	1 3/4	3600	3 11/32	11/64	3600	3 1/4	1/8	3100	3 3/16	1/8	2500
	10.40	E.U. 55	O.H.	3 7/8	2 5/32	4500	3 17/32	1/8	3700	3 1/2	7/64	3400	3 7/16	7/64	2600
	10.40	E.U. 75	NC31 (I.F.)	4 1/8	2 1/8	6300	3 27/32	5/32	5200	3 25/32	1/8	4300	3 23/32	1/8	3600
	10.40	I.U. 75	E.H.	4 1/4	1 7/8	7000	3 25/32	3/16	5100	3 11/16	9/64	4100	3 5/8	9/64	3300
	10.40	I.U. 75	NC26 (①S.H.)	①3 3/8	1 3/4	3600	3 3/8	3/16	3600	3 3/8	3/16	3600	3 5/16	3/16	3600
	10.40	E.U. 75	O.H.	①3 7/8	2 5/32	4500	3 21/32	3/16	4500	3 19/32	5/32	4500	3 17/32	5/32	3700
	10.40	E.U. 75	SL-H90	3 7/8	2 5/32	6000	3 5/8	11/64	5200	3 9/16	9/64	4300	3 1/2	9/64	3600
	10.40	E.U. 75	P.A.C.	①3 1/8	1 1/2	3100	3 1/8	17/64	3100	3 1/8	17/64	3100	3 1/8	17/64	3100
	10.40	E.U. 95	NC31 (I.F.)	4 1/8	2	7000	3 31/32	7/32	6800	3 7/8	11/64	5600	3 25/32	11/64	4400
	10.40	E.U. 95	SL-H90	①3 7/8	2 5/32	6000	3 3/4	15/64	6000	3 27/32	3/16	5500	3 19/32	3/16	4700
	10.40	E.U. 105	NC31 (I.F.)	①4 1/8	2	7000	4 1/32	17/64	7000	3 29/32	13/64	6200	3 27/32	13/64	5200
	10.40	E.U. 135	NC31 (I.F.)	①4 1/8	2	7000	4 1/8	9/32	7000	4 1/16	1/4	7000	3 15/16	1/4	6400
3 1/2	9.50	E.U. 75	NC38 (W.O.)	①4 3/4	3	6800	4 13/32	3/16	6800	4 3/8	9/64	5500	4 11/32	9/64	4900
	9.50	E.U. 75	NC38 (I.F.)	4 3/4	2 11/16	9800	4 7/16	5/32	6500	4 3/8	1/8	5400	4 5/16	1/8	4300
	9.50	E.U. 75	O.H.	①4 3/4	3	6000	4 11/32	11/64	6000	4 9/32	9/64	5800	4 7/32	9/64	4800
	9.50	E.U. 75	SL-H90	4 5/8	3	6500	4 1/4	11/64	6500	4 5/32	1/8	5200	4 1/8	1/8	4700
	13.30	E.U. 75	NC38 (I.F.)	4 3/4	2 11/16	9800	4 9/16	7/32	8600	4 15/32	11/64	7000	4 13/32	11/64	5900
	13.30	I.U. 75	NC31 (①S.H.)	4 1/8	①2 1/8	6300	4 1/8	19/64	6300	4	15/64	6300	3 29/32	15/64	6300
	13.30	E.U. 75	O.H.	①4 3/4	2 11/16	8800	4 15/32	7/32	8600	4 3/8	11/64	7300	4 9/32	11/64	5800
	13.30	E.U. 75	H90	5 1/4	2 3/4	12000	4 19/32	11/64	8700	4 1/2	1/8	8900	4 7/16	1/8	5600
	13.30	E.U. 95	NC38 (I.F.)	5	2 9/16	11500	4 11/16	9/32	10700	4 9/16	7/32	8600	4 1/2	7/32	7600
	13.30	E.U. 95	SL-H90	4 5/8	①2 11/16	9700	4 15/32	1/4	9700	4 3/8	13/64	9100	4 9/32	13/64	7400
	13.30	E.U. 95	H 90	5 1/4	2 3/4	12000	4 11/16	13/64	10500	4 5/8	11/64	9200	4 17/32	11/64	7500
	13.30	E.U. 105	NC38 (I.F.)	5	①2 1/16	11500	4 3/4	19/64	11500	4 5/8	15/64	9700	4 17/32	15/64	8100
	13.30	E.U. 135	NC40 (4 F.H.)	5 3/4	2 7/16	16000	5 1/8	11/32	15700	4 31/32	17/64	12500	4 7/8	17/64	10600
	13.30	E.U. 135	NC38(3 1/2 I.F.)	5	①2 1/16	11500	4 13/16	3/8	11500	4 25/32	19/64	11500	4 21/32	19/64	10200
	15.50	E.U. 75	NC38 (I.F.)	5	2 9/16	11000	4 5/8	15/64	9700	4 17/32	3/16	8100	4 7/16	3/16	6500
	15.50	E.U. 95	NC38 (I.F.)	5	2 7/16	11500	4 3/4	19/64	11500	4 5/8	15/64	9700	4 17/32	15/64	8100
	15.50	E.U. 105	NC40 (4 F.H.)	5 1/4	2 9/16	15000	5 1/32	19/64	13200	4 29/32	15/64	11300	4 13/16	15/64	9400

Table 4-89 (continued)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
DRILL PIPE DATA			NEW TOOL JOINT DATA				⑤PREMIUM CLASS			CLASS 2			CLASS 3		
Nom Size in.	Nom. Wt. lb/ft	Type Upset and Grade	Conn.	New OD in.	New ID in.	①Make-Up Torque ft-lb	②Min. OD Tool Joint in.	③Min. Box Shoulder With Eccentric Wear in.	①Make-Up Torque For Min. OD Tool Joint ft-lb	②Min. OD Tool Joint in.	③Min. Box Shoulder With Eccentric Wear in.	①Make-Up Torque For Min. OD Tool Joint ft-lb	②Min. OD Tool Joint in.	③Min. Box Shoulder With Concentric Wear in.	①Make-Up Torque For Min. OD Tool Joint ft-lb
4	11.85	E.U. 75	NC46 (I.F.)	5 3/4	3 1/4	18000	5 1/4	3/64	8800	5 3/16	7/64	7300	5 1/8	7/64	5800
	11.85	E.U. 75	NC46 (W.O.)	5 3/4	3 7/16	15400	5 5/32	3/32	9500	5 7/32	1/8	8000	5 5/32	1/8	6500
	11.85	E.U. 75	O.H.	5 1/4	3 15/32	11300	5 3/32	1/16	9300	4 31/32	3/32	7400	4 15/16	3/32	6600
	11.85	E.U. 75	H 90	5 1/2	2 13/16	18500	4 15/16	11/64	9000	4 27/32	1/8	6900	4 13/16	1/8	6000
	14.00	I.U. 75	NC40 (F.H.)	5 1/4	2 13/16	12500	4 7/8	15/64	10600	4 23/32	3/16	8800	4 21/32	3/16	7600
	14.00	E.U. 75	NC46 (I.F.)	5 3/4	3 1/4	18000	5 11/32	3/16	11000	5 1/4	5/64	8800	5 3/16	5/64	7200
	14.00	I.U. 75	⑥S.H.	4 5/8	⑦2 9/16	8000	4 9/16	19/64	8000	4 7/16	15/64	8000	4 11/32	19/64	6000
	14.00	E.U. 75	O.H.	5 1/2	3 1/4	14000	5 1/8	7/32	11100	5 3/32	11/64	8900	4 31/32	11/64	7400
	14.00	E.U. 75	H 90	5 1/2	2 13/16	18500	5	3/16	10800	4 29/32	5/64	8300	4 27/32	5/64	6900
	14.00	I.U. 95	NC40 (F.H.)	5 1/4	2 9/16	15000	5 1/32	19/64	13200	4 25/32	15/64	11300	4 9/16	15/64	9400
	14.00	E.U. 95	NC46 (I.F.)	5 3/4	3 1/4	18000	5 15/32	1/4	14000	5 11/32	3/16	11000	5 5/32	3/16	9500
	14.00	I.U. 95	H 90	5 1/2	2 13/16	18500	5 1/8	15/64	13800	5 1/32	3/16	11400	4 15/16	3/16	9000
	14.00	I.U. 105	NC40(F.H.)	5 3/8	2 7/16	16000	5 3/32	21/64	15000	4 31/32	17/64	12600	4 7/8	17/64	10600
	14.00	E.U. 105	NC46 (I.F.)	5 3/4	3 1/4	18000	5 1/2	1/4	14800	5 13/32	13/64	12500	5 5/16	13/64	10300
	14.00	I.U. 105	H 90	5 1/2	2 13/16	18500	5 3/16	9/32	15000	5 1/16	7/32	12300	5	7/32	10800
	14.00	E.U. 135	NC46 (I.F.)	6	2 7/8	22000	5 11/16	11/32	19600	5 17/32	17/64	15600	5 7/16	17/64	13300
	15.70	I.U. 55	NC40 (F.H.)	5 1/4	2 11/16	13500	4 23/32	3/16	8800	4 11/16	5/64	7000	4 5/8	5/64	5800
	15.70	E.U. 55	NC46 (I.F.)	5 3/4	3 1/4	18000	5 1/4	5/64	8800	5 3/16	7/64	7200	5 1/8	7/64	5800
	15.70	I.U. 75	NC40 (F.H.)	5 1/4	2 11/16	13500	4 15/16	1/4	11900	4 27/32	13/64	10000	4 3/4	13/64	8200
	15.70	E.U. 75	NC46 (I.F.)	5 3/4	3 1/4	18000	5 3/8	13/64	11700	5 9/32	5/32	9500	5 7/32	5/32	8000
	15.70	E.U. 75	H 90	5 1/2	2 13/16	18500	5 1/16	13/64	12300	4 31/32	5/32	9900	4 7/8	5/32	7500
	15.70	I.U. 95	NC40 (F.H.)	5 3/8	2 7/16	15700	5 3/32	21/64	15000	4 31/32	17/64	12600	4 7/8	17/64	10600
	15.70	E.U. 95	NC46 (I.F.)	5 3/4	3	18000	5 1/2	1/4	14800	5 13/32	13/64	12500	5 5/16	13/64	10300
	15.70	I.U. 95	H 90	5 1/2	2 13/16	18500	5 3/16	17/64	15000	5 1/16	13/64	12300	4 31/32	13/64	9900
	15.70	E.U. 105	NC46 (I.F.)	5 7/8	3	20200	5 9/16	19/64	16400	5 7/16	15/64	13300	5 3/8	15/64	11700
	15.70	I.U. 105	H 90	5 1/2	2 13/16	18500	5 1/4	19/64	18800	5 1/8	15/64	13800	5 1/32	15/64	11400
	15.70	I.U. 135	NC46 (I.F.)	6 1/4	2 5/8	24900	5 3/4	23/64	21200	5 5/8	19/64	18000	5 1/2	19/64	14800
	15.70	E.U. 135	NC46 (I.F.)	6 1/4	2 3/4	23500	5 3/4	23/64	21200	5 5/8	19/64	18000	5 1/2	19/64	14800
4 1/2	13.75	E.U. 75	NC50 (W.O.)	6 3/8	3 7/8	17500	5 23/32	11/64	12100	5 21/32	9/64	10400	5 19/32	9/64	8800
	13.75	E.U. 75	NC50 (I.F.)	6 3/8	3 3/4	19700	5 23/32	11/64	12100	5 21/32	9/64	10400	5 19/32	9/64	8800
	13.75	E.U. 75	O.H.	5 3/4	⑦3 31/32	10800	5 1/2	15/64	10800	5 7/16	13/64	10400	5 3/8	13/64	8500
	13.75	E.U. 75	H 90	6	3 1/4	20000	5 5/16	3/16	12000	5 1/4	5/32	10200	5 3/16	5/32	6400
	16.60	I.U. 55	F.H.	5 3/4	3	18000	5 3/32	11/64	10300	5 3/16	1/8	8200	5 1/8	1/8	8800
	16.60	I.U. 55	NC46 (E.H.)	6	3 1/4	18000	5 5/16	11/64	10300	5 1/4	5/64	8800	5 3/16	5/64	7300
	16.60	E.U. 55	NC50 (I.F.)	6 3/8	3 3/4	19700	5 21/32	5/64	10400	5 19/32	7/64	8800	5 17/32	7/64	7100
	16.80	I.U. 75	O.H.	⑦5 7/8	3 3/4	19600	5 19/32	1/4	15700	5 15/32	13/64	12500	5 3/8	13/64	10200
	16.60	I.U. 75	F.H.	5 3/4	3	18000	5 7/16	15/64	14000	5 11/32	3/16	11800	5 1/4	3/16	9600
	16.60	I.U. 75	NC46 (E.H.)	6	3 1/4	18000	5 15/32	15/64	14000	5 3/8	3/16	11800	5 9/32	3/16	9500
	16.60	E.U. 75	NC50 (I.F.)	6 3/8	3 3/4	19700	5 13/16	7/32	14800	5 11/16	5/32	11300	5 5/8	5/32	9600
	16.60	E.U. 75	H 90	6	3 1/4	20000	5 15/32	15/64	14400	5 5/16	3/16	12000	5 1/4	3/16	10200
	16.60	I.U. 95	F.H.	6	2 7/8	19000	5 5/8	21/64	18400	5 15/32	1/4	14800	5 3/8	1/4	12500
	16.60	I.U. 95	NC46 (E.H.)	6	3	21000	5 5/8	5/16	18000	5 1/2	1/4	14800	5 13/32	1/4	12500
	16.60	E.U. 95	NC50 (I.F.)	6 1/4	3 5/8	21500	5 15/16	9/32	18300	5 13/16	7/32	14800	5 3/4	7/32	13000
	16.60	I.U. 95	H 90	6	3	23500	5 9/16	19/64	18700	5 7/16	15/64	15300	5 11/32	15/64	12700
	16.60	I.U. 105	F.H.	6	2 3/4	20300	5 11/16	23/64	19900	5 17/32	9/32	16200	5 7/16	9/32	14100
	16.60	I.U. 105	NC46 (E.H.)	6	2 7/8	22000	5 23/32	23/64	20400	5 9/16	9/32	16400	5 15/32	9/32	14000
	16.60	E.U. 105	NC50 (I.F.)	6 1/4	3 5/8	22000	6	19/64	20000	5 7/8	15/64	16500	5 25/32	15/64	13900
	16.60	I.U. 105	H 90	6	3	23500	5 5/8	5/16	20400	5 1/2	1/4	17000	5 3/8	1/4	13600

Table 4-89
(continued)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
DRILL PIPE DATA			NEW TOOL JOINT DATA				②PREMIUM CLASS			CLASS 2			CLASS 3		
Nom Size in	Nom Wt lb/ft	Type Upset and Grade	Conn	New OD in	New ID in	①Make-Up Torque ft-lb	②Min. OD Tool Joint in	③Min Box Shoulder With Eccentric Wear in	①Make-Up Torque For Min OD Tool Joint ft-lb	④Min OD Tool Joint in.	③Min Box Shoulder With Eccentric Wear in.	①Make-Up Torque For Min. OD Tool Joint ft-lb	⑤Min. OD Tool Joint in	③Min Box Shoulder With Concentric Wear in	①Make-Up Torque For Min OD Tool Joint ft-lb
4½	16.60	I.U. 135	NC46 (E.H.)	6¼	2½	25600	5 13/16	29/64	25600	5¾	23/64	21200	5¾	23/64	18000
	16.60	E.U. 135	NC50 (I.F.)	6½	3	29000	6 3/16	23/64	25400	6 1/32	3/16	20900	5 15/16	3/16	18300
	20.00	I.E.U. 55	NC46 (E.H.)	6	3	21000	5 13/32	7/32	12500	5 3/16	11/64	10300	5¼	11/64	8800
	20.00	I.E.U. 55	F.H.	6	3	18000	5⅜	13/64	12500	5 5/32	5/32	10300	5 3/16	5/32	8200
	20.00	E.U. 55	NC50 (I.F.)	6¼	3⅝	22000	5¾	3/16	13000	5 21/32	9/64	10400	5 19/32	9/64	8800
	20.00	I.E.U. 75	F.H.	6	3	18000	5 5/16	19/64	17000	5 7/16	15/64	14100	5 11/32	15/64	11800
	20.00	I.E.U. 75	NC46 (E.H.)	6	3	21000	5 15/32	19/64	17200	5 15/32	15/64	14000	5⅜	15/64	11800
	20.00	E.U. 75	NC50 (I.F.)	6¼	3⅝	22000	5 25/32	9/32	17400	5 23/32	7/32	13900	5¾	7/32	13000
	20.00	I.E.U. 75	H 90	6	3	23500	5½	17/64	17000	5 13/32	7/32	14400	5 5/16	7/32	12000
	20.00	I.E.U. 95	F.H.	6	2½	23000	5¾	⅜	21400	5 19/32	19/64	17700	5 15/32	19/64	14800
	20.00	I.E.U. 95	NC46 (E.H.)	6⅜	2¾	23500	5 25/32	⅜	22000	5⅝	19/64	18000	5½	19/64	14800
	20.00	E.U. 95	NC50 (I.F.)	6⅜	3½	23200	6 1/16	21/64	21800	5 29/32	¼	17400	5 13/16	¼	14800
	20.00	I.E.U. 95	H 90	6	3	23500	5 21/32	11/32	21400	5 17/32	9/32	17800	5 7/16	9/32	15300
	20.00	I.E.U. 105	NC46 (E.H.)	6¼	2½	25600	5 27/32	13/32	23700	5 11/16	21/64	19600	5 9/16	21/64	16000
	20.00	E.U. 105	NC50 (I.F.)	6⅜	3¼	26000	6⅛	11/32	23600	6	9/32	20000	5⅞	9/32	16500
	20.00	E.U. 135	NC50 (I.F.)	6⅜	2¾	31100	6 11/32	7/16	30100	6 3/16	23/64	25400	6 1/32	23/64	20900
5	19.50	I.E.U. 75	NC50 (E.H.)	6⅜	3¾	20000	5 31/32	9/32	19200	5 27/32	7/32	15600	5¾	7/32	13000
	19.50	I.E.U. 95	NC50 (E.H.)	6⅜	3¼	28000	6 5/32	23/64	24500	6	9/32	20000	5⅞	9/32	16500
	19.50	I.E.U. 95	H 90	6½	3¼	26700	5 31/32	23/64	24200	5 15/16	9/32	19900	5 27/32	9/32	18900
	19.50	I.E.U. 105	NC50 (E.H.)	6½	3	29000	6¼	13/32	27300	6 1/16	5/16	21800	5 15/16	5/16	18300
	19.50	I.E.U. 105	H 90	6½	3	30200	6 3/32	25/64	26500	5⅞	5/16	21700	5 27/32	5/16	18900
	19.50	I.E.U. 135	5½FH	7¼	3½	37000	6⅞	27/64	34300	6 23/32	11/32	28700	6 19/32	11/32	24200
	25.60	I.E.U. 75	NC50 (E.H.)⑦	6½	3½	23200	6 3/32	23/64	23200	6	9/32	20000	5⅞	9/32	16500
	25.60	I.E.U. 75	5½FH	7	3¾	32300	6 19/32	19/64	24200	6 15/32	15/64	19900	6⅜	15/64	16600
	25.60	I.E.U. 95	5½FH	7	3½	32300	6 23/32	⅜	30900	8⅝	19/64	25300	6½	19/64	20900
	25.60	I.E.U. 105	5½FH	7¼	3½	37000	6 27/32	13/32	33700	6 11/16	21/64	27500	6 9/16	21/64	23100
5½	21.90	I.U. 75	F.H.	7	4	28800	6 19/32	3/16	24200	6 7/16	15/64	18800	6⅜	15/64	16600
	21.90	I.E.U. 95	F.H.	7	3¾	32300	6⅝	⅜	29800	6 19/32	19/64	24200	6½	19/64	20900
	21.90	I.E.U. 95	H 90	7	3½	30300	6 9/16	23/64	30000	6 15/32	5/16	24300	6 7/16	5/16	20800
	21.90	I.E.U. 105	F.H.	7¼	3½	37000	6 27/32	13/32	33200	6 11/16	21/64	27500	6 9/16	21/64	23100
	21.90	I.E.U. 135	F.H.	7½	3	44100	7 3/32	17/32	42300	6⅞	27/64	34300	6¾	27/64	29800
	24.70	I.U. 75	F.H.	7	4	28800	6 23/32	3/16	26400	6 17/32	¼	22000	6 13/32	¼	17700
	24.70	I.E.U. 95	F.H.	7¼	3½	37000	6 27/32	13/32	33200	6 11/16	21/64	27500	6 9/16	21/64	23100
	24.70	I.E.U. 105	F.H.	7¼	3½	37000	6 15/16	29/64	36500	6¾	23/64	29800	6⅝	23/64	25300

①Basis of calculations for recommended tool joint make-up torque assumed the use of a thread compound containing 40-60% by weight of finely powdered metallic zinc applied thoroughly to all threads and shoulders, and a tensile stress of 62,500 psi for column 7 and 75,000 psi for columns 10, 13, and 16.

②80% minimum wall drill pipe. O.D. of tool joints listed for Premium Class drill pipe are based on drill pipe having uniform wear and a minimum wall thickness of 80%.

③Minimum box shoulder disregards bevel.

④O.D. of tool joints shown for class 2 drill pipe are based on drill pipe having all the wear on one side and a minimum wall thickness of 65%.

⑤O.D. of tool joints shown for class 3 drill pipe are based on drill pipe having all the wear on one side and a minimum wall thickness of 55%.

⑥The use of O.D.'s smaller than those listed in the table may be acceptable on Slim Hole Tool Joints due to special service requirements.

⑦Tool Joint with dimensions shown has a lower torsional yield strength than the drill pipe to which it is attached.

★*Tool joint outside diameters (OD) specified are required* to retain torsional strength in the tool joint comparable to the torsional strength of the attached drill pipe. The use of tool joints with outside diameters smaller than those listed may be acceptable in special service requirements to provide sufficient clearance in casing programs. *In such cases, the torsional strength of the joint may be considerably below that of the drill pipe body to which it is attached.*

Tool joints with outside diameters shown from Grade E drill pipe are adequate for high strength drill pipe when used in combination strings with Grade E drill pipe.

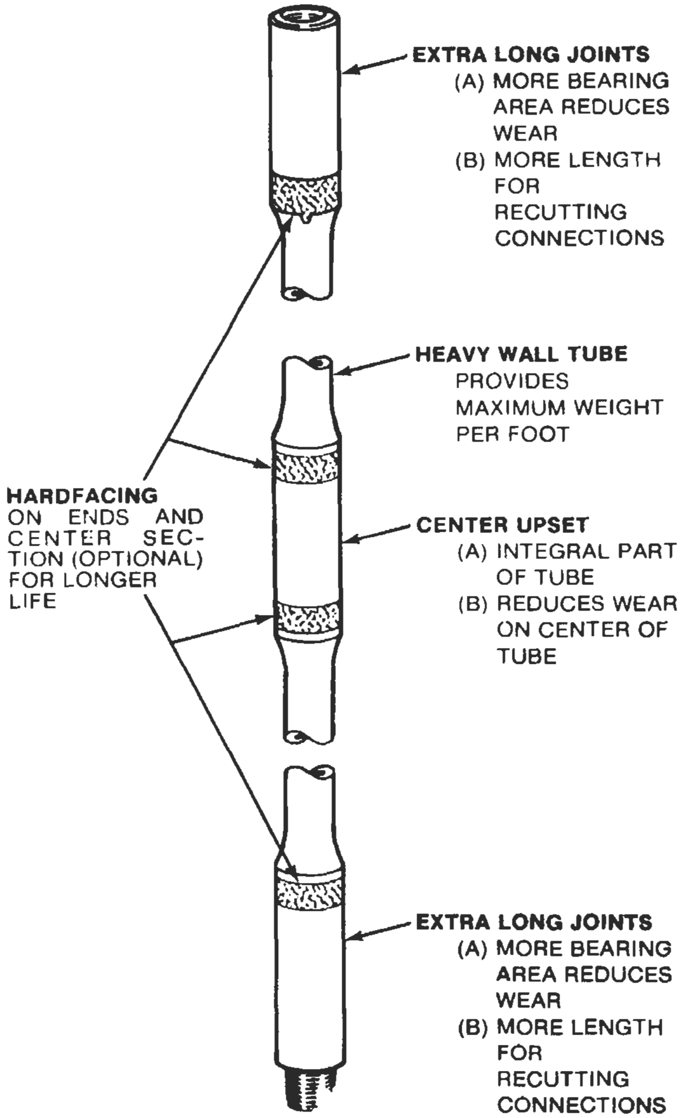


Figure 4-134. Drilco's Hevi-Wate® drill pipe [42].

(text continued from page 749)

drilling. However, based on field experience, if a heavy-weight drill pipe above the drill collars is used, it reduces drill pipe failure. The failure of regular drill pipe and tool joints is influenced by several factors. Probably the most important one is a cyclic bending stress reversal resulting from the accidental running of drill pipe in compression, the centrifugal force effect or passing through short and sharp dog-legs, or a combination of these factors. In directional drilling, a heavy-weight drill pipe is used to create weight on the bit if, for some reason (e.g., excessive torque and drag or differential problem sticking), a long string of drill collars cannot be run.

The best performance of the individual members of the drill string is obtained when the bending stress ratio of subsequent members is less than 5.5 [38]. Bending stress ratio (BSR) is defined as a ratio of the bending section moduli of two subsequent members, e.g., between the drill collar and the pipe right above it.

To maintain the BSR at less than 5.5, the string of drill collars must frequently be composed of different sizes. For severe drilling conditions (hole enlargement, corrosive environment, hard formations), reduction of the BSR to 3.5 helps to reduce frequency of drill pipe failure.

Geometrical and mechanical properties of heavy-weight drill pipe (Hevi-Wate®) manufactured by Drilco are given in Table 4-90.

Example

Calculate the required length of $4\frac{1}{2}$ in. Hevi-Wate® drill pipe for the following conditions:

- Hole size: $9\frac{1}{2}$ in.
- Hole angle: 40°
- Desired weight on bit: 40,000 lb
- Drill collars: $7 \times 2\frac{13}{16}$ in.
- Length of drill collars: 330 ft
- Drilling fluid specific gravity: 1.2
- Desired safety factor for neutral point: 1.15

Solution

Check to see if the BSR of drill collar and Hevi-Wate(c) drill pipe is less than 5.5.

Bending section modulus of drill collar is

$$\frac{\pi}{16}\left[\frac{(7)^4-(2.8125)^4}{7}\right]=65.592\,\text{in.}^3$$

Bending section modulus of Hevi-Wate® drill pipe is

$$\frac{\pi}{16}\left[\frac{(4.5)^4-(2.75)^4}{4.5}\right]=15.397\,\text{in.}^3$$

$$\text{BSR}=\frac{65.592}{15.397}=4.26<5.5$$

Unit weight of drill collar in drilling fluid is

$$110\left(1-\frac{1.2}{7.85}\right)=93.18\,\text{lb/ft}$$

Table 4-90
Properties of Hevi-Wate® Drill Pipe
(® Drilco Trademark) [38]

DIMENSIONAL DATA RANGE II

	TUBE					Mech Properties Tube Section	
	Nom Tube Dimension						
Nom Size (in)	ID (in)	Wall Thick-ness (in)	Area (in²)	Center Upset (in)	Elevator Upset (in)	Tensile Yield (lb)	Tor-sional Yield (ft-lb)
3½	2 1/16	.719	6.280	4	3⅝	345,400	19,575
4	2 9/16	719	7.409	4½	4⅛	407,550	27,635
4½	2¾	.875	9.965	5	4⅝	548,075	40,715
5	3	1.000	12.568	5½	5⅛	691,185	56,495

	TOOL JOINT					WEIGHT Approx. Wt Incl Tube & Joints (lb)		
				Mech Properties				
Nom Size (in)	Conn Size (in)	OD (in)	ID (in)	Tensile Yield (lb)	Torsional Yield (ft-lb)	Wt/ ft	Wt/Jt. 30 ft	Make-Up Torque (ft-lb)
3½	N C 38(3½ I.F.)	4¾	2 3/16	748,750	17,575	25.3	760	9,900
4	N.C.40 (4 F.H.)	5¼	2 11/16	711,475	23,525	29.7	890	13,250
4½	N.C.46 (4 I.F.)	6¼	2⅞	1,024,500	38,800	41.0	1230	21,800
5	N.C.50 (4½ I.F.)	6½	3 1/16	1,266,000	51,375	49.3	1480	29,400

DIMENSIONAL DATA RANGE III

	TUBE					Mech Properties Tube Section	
	Nom Tube Dimension						
Nom Size (in)	ID (in)	Wall Thick-ness (in)	Area (in²)	Center Upset (in)	Elevator Upset (in)	Tensile Yield (lb)	Tor-sional Yield (ft-lb)
4½	2¾	.875	9,965	5	4⅝	548,075	40,715
5	3	1.000	12,656	5½	5⅛	691,185	56,495

	TOOL JOINT					WEIGHT Approx. Wt Incl Tube & Joints (lb)		
				Mech Properties				
Nom Size (in)	Conn Size (in)	OD (in)	ID (in)	Tensile Yield (lb)	Torsional Yield (ft-lb)	Wt/ ft	Wt/Jt. 44 ft	Make-Up Torque (ft-lb)
4½	N.C.46 (4 I.F.)	6¼	2⅞	1,024,500	38,800	39.9	1795	21,800
5	N.C 50 (4½ I.F.)	6½	3 1/16	1,266,000	51,375	48.5	2180	29,400

Unit weight of Hevi-Wate[c] drill pipe = (41)(0.847) = 34.72 lb/ft. Part of weight on bit that may be created using drill collars = (93.18)(330)(cos 40) = 23,555 lb. Required length of Hevi-Wate[c] drill pipe is

$$\frac{(40000-23555)(1.5)}{(34.72)(\cos 40)} = 711\,\text{ft}$$

Assuming an average length of one joint of Hevi-Wate[c] drill pipe to be 30 ft, 24 joints are required.

Fatigue Damage of Drill Pipe

It should be understood that the majority of drill pipe and tool joint failures occur as a result of a fatigue damage. The problem of fatigue failure is not adequately researched; however, it is basically agreed that tension and bending (reversing tension and compression of the same drill pipe fiber), magnified by vibrations, contribute the most to such type of failure. Cycling stress results in a crack that spreads across the cross-section and causes ultimate failure. A fracture begins at a point on or near the surface that is weaker than any other due to a number of reasons (e.g., surface imperfections or stress raisers). Fatigue cracks can also initiate at points below the surface of the drill string if the proper conditions exist. It should also be remembered that drill pipe fatigue is cumulative in nature, so the changes that affect failure are usually long delayed and require a certain amount of time to be detected.

The drill collars and particularly their connections are also exposed to cyclic stresses. Subsequently, these are susceptible to fatigue damage, but the changes that may influence failure are more quickly discovered.

Based on work done by A. Lubinski, J. E. Hansford and R. W. Nicholson (API RP 7G, Section 6), gives the formula for the maximum permissible hole curvature in order to avoid fatigue damage to drill pipe.

$$C = \frac{432{,}000}{\pi}\frac{\sigma_b}{ED_{dp}}\frac{\tanh(KL)}{KL} \tag{4-66}$$

$$K = \left(\frac{T}{EI}\right)^{1/2} \tag{4-67}$$

For Grade E drill pipe

$$\sigma_b = 19{,}500 - (0.149)\sigma_t = 1.34\ 10^{-6}\ (\sigma_t - 33{,}500)^2 \tag{4-68}$$

For Grade S-135 drill pipe

$$\sigma_b = 20{,}000\left(1 - \frac{\sigma_t}{145{,}000}\right) \tag{4-69}$$

$$\sigma_t = \frac{T}{A} \tag{4-70}$$

where c = maximum permissible dog-leg severity in °/100 ft
σ_b = maximum permissible bending strength in psi
σ_t = tensile stress due to the weight of the drill string suspended below a dog-leg in psi
E = Young's modulus, E = 30 × 10^6 psi
D_{dp} = outside diameter of drill pipe in in.
L = half the distance between tool joints, L = 180 in. for Range 2 drill pipe. Equation 4-75 does not hold true for Range 3 drill pipe.
T = weight of drill pipe suspended below the dog-leg in lb
I = drill pipe moment of inertia with respect to its diameter in $in.^4$
A = cross-sectional area of drill pipe in $in.^2$

By intelligent application of these formulas, several practical questions can be answered, both at the borehole design state and while drilling.

Example

Calculate the maximum permissible hole curvature for data as below:

- New EU $4\frac{1}{2}$-in. Range 2 drill pipe, nominal weight 16.6 lb/ft, steel grade S-135, with NC50 (IF) tool joint
- Drill collars, 7 × $2\frac{1}{4}$ in., unit weight 117 lb/ft
- Length of drill collars, 550 ft
- Drilling fluid density, 12 lb/gal
- Anticipated length of the hole below the dog-leg, 8,000 ft
- Assume the hole is vertical below the dog-leg

Solution

From Table 4-79: D_{dp} = 4.5 in., d_{dp} - 3.826 in., A = 4.4074 in.; and from Table 4-100: unit weight of drill pipe adjusted for tool joint, W_{dp} = 18.8 lb/ft.
Weight of drill collar string is

$$(550)(117)\left(1 - \frac{12}{65.4}\right) = 52,543\,\text{lb}$$

Weight of drill pipe is

$$(8,000 - 550)(18.8)\left(1 - \frac{12}{65.4}\right) = 114,361\,\text{lb}$$

Weight suspended below the dog-leg, T = 166,904 lb,

$$\text{Tensile stress } \sigma_t = \frac{166,904}{4.4074} = 37,869 \text{ psi}$$

Maximum permissible bending stress,

$$\sigma_b = 20,000\left(1 - \frac{37,869}{145,000}\right) = 14,776\,\text{lb}$$

Drill pipe moment of inertia,

$$I = \frac{\pi}{64}[(4.54)^4 - (3.826)^4] = 9.61\ \text{in.}^4$$

$$K = \sqrt{\frac{166,904}{(30)(10)^6(9.61)}} = 2.406110^{-2}\ \text{in.}^{-1}$$

Maximum permissible hole curvature,

$$c = \frac{42,000}{\pi}\ \frac{14,776}{(30)(10)^6(4.5)}\ \frac{\tanh(2.406110^{-2})(180)}{(2.4061)(10)^{-2}(180)} = 3.47°/100\ \text{ft}$$

The calculations, although based on reasonable theory, must be approached with caution. For practical purposes, some safety factor is recommended.

Drill Pipe Inspection Procedure

To avoid costly fishing operations, loss of material and time, the drill pipe must be carefully inspected according to the following procedure [30]:

1. Determine the pipe and joint cross-sectional area.
2. Determine tool joint outside diameter. Tool joint box should have sufficient OD and tool joint pin sufficient ID to withstand the same torsional loading as the pipe body. When tool joints are eccentrically worn, determine the minimum shoulder width acceptable for tool joint class in Table 4-101.
3. Check the inside and outside surfaces for presence of cracks, notches and severe pitting.
4. Check slip areas for longitudinal and transverse cracks and sharp notches.
5. Check tool joints for wear, galls, nicks, washes, fins, fatigue cracks at root of threads, or other items that would affect the pressure holding capacity or stability of the joint.
6. Ascertain if joint has proper bevel diameter.
7. Random check 10% of the joints for manufacturer markings and date of tool joint installation to determine if tool joint has been reworked.

Optional:

1. Using data in Table 4-89, determine minimum shoulder width acceptable for tool joint in class.
2. Check for box swell and/or pin stretch. These are indications of overtorquing, and their presence greatly affects the future performance of the joint.
3. Use thread profile gauge for indications of overtorque, lapped, or galled threads and stretching.
4. Magnetic particle inspection for cracks should be made if there is evidence of stretching or swelling. Check box and pin threaded area, especially last engaged thread.

Drill String Design

The drill string design is to determine an optimum combination of drill pipe sizes and steel grades for the lowest cost of string or the lowest total load (in very deep drilling) that has sufficient strength to successfully accomplish expected goals. Having in mind that the drill string is subjected to many loads that may exist as static loads, cycling loads and dynamic loads, the problem of drill string design is complex. Due to the complexity of the problems, some simplifications are always made and, therefore, several decisions are left up to the person responsible for the design.

In general, a reasonably bad working condition should be assumed and, for that reason, a good knowledge of expected problems much as hole drag, torquing, risk of becoming stuck, tendency to drill a crooked hole, vibrations, etc., is of critical importance.

The person responsible for the design must know drill string performance properties, data from wells already drilled in the nearest vicinity and current prices of the drill string elements.

The designer should simultaneously consider the following main conditions:

1. The working load at any part of the string must be less or equal to the load capacity of the drill string member under consideration divided by the safety factor.
2. Ratio of section moduli of individual string members should be less than 5.5.
3. To minimize pressure losses, the ratio of drill pipe outside diameter to borehole diameter, whenever possible, should be about 0.6.

Normally, based on hole diameter, the designer can select drill collar diameter and drill pipe diameter. Next, specific pipe is chosen; the maximum length of that pipe must be determined based on condition 1. For this purpose, the following equation is used:

$$(L_{dc}W_{dp} + L_{hw}W_{hw} + L_{dp1}W_{dp1})K_b = \frac{P_1}{SF} \tag{4-71}$$

where L_{dc} = length of drill collar string in ft
W_{dp} = unit weight of drill collar in air in lb/ft
L_{hw} = length of heavy-weight drill pipe (if used in the string) in ft
W_{hw} = unit weight of heavy-weight drill pipe in lb/ft
L_{dp1} = length of drill pipe under consideration above the heavy-weight drill pipe in ft
W_{dp1} = unit weight of drill pipe (section 1) in lb/ft
K_b = buoyant factor
P_1 = tension load capacity of drill pipe (section 1) in lb
SF = safety factor

Solving Equation 4-71 for L_{dpl} yields

$$L_{dp1} = \frac{P_1}{SF\,K_b W_{dp1}} - \frac{L_{dc}W_{dc}}{W_{dp1}} - \frac{L_{hw}W_{hw}}{W_{dp1}} - \frac{L_{dp1}W_{dp1}}{W_{dp1}} \tag{4-72}$$

If the sum of $L_{dc} + L_{hw} + L_{dpl}$ is less than the planned borehole depth, the stronger pipe must be selected or a heavier pipe must be used in the upper part of the hole.

The maximum length of the upper part in a tapered string may be calculated from Equation 4-73:

$$L_{dp2} = \frac{P_2}{SF\,K_b W_{dp1}} - \frac{L_{dc} W_{dc}}{W_{dp2}} - \frac{L_{hw} W_{hw}}{W_{dp2}} - \frac{L_{dp1} W_{dp1}}{W_{dp2}} \tag{4-73}$$

where P_2 = tension load capacity of next (upper) section of drill pipe in lb
L_{dp2} = length of section (2) in ft
W_{dp2} = unit weight of drill pipe (section 2) in lb/ft

Normally, not more than two sections are designed but, if absolutely necessary, even three sections can be used. To calculate the tensile load capacity of drill pipe, it is suggested to apply Equation 4-58 and use the recommended makeup torque of the weakest tool joint for the rotary torque.

The magnitude of the safety factor is very important and usually ranges from 1.4 to 2.8 depending upon downhole conditions, drill pipe quality and acceptable degree of risk. It is recommended that a value of safety factor be selected to produce a margin of overpull of at least about 70,000 lb.

Additional checkup, especially in deep drilling, should be done to avoid drill pipe crushing in the slip area. The maximum load that can be suspended in the slips can be found from Equation 4-74:

$$W_{max} = \frac{P_t}{SF\left[1 + \frac{D_{dp}}{2}\frac{K}{L_s} + \left(\frac{D_{dp}}{2}\frac{K}{L_s}\right)^2\right]^{1/2}} \tag{4-74}$$

where W_{max} = maximum allowable drill string load that can be suspended in the slips in lb
P_t = load capacity of drill pipe based on minimum yield strength in lb
D_{dp} = outside diameter of drill pipe in in
L_s = length of slips (L_s = 12–16 in.)
K = lateral load factor of slip, $K = (1 - f \tan \alpha)/(f + \tan \alpha)$
f = friction coefficient between slips and bushing
α = slip taper ($\alpha = 9° 27'45''$)
SF = safety factor to account for dynamic loads when slips are set on moving drill pipe (SF = 1.1)

Normally, if the drill pipe is sufficiently strong for tension, it will have satisfactory strength in torsion, collapse and burst; however, if there is any doubt, additional checkup calculations must be performed.

Example

Design a drill string for conditions as specified below:

- Hole depth: 10,000 ft
- Hole size: $9\frac{7}{8}$ in.
- Mud weight: 12 lb/gal
- Maximum weight on bit: 60,000 lb
- Neutral point design factor: 1.15
- No crooked hole tendency
- Safety factor for tension, SF = 1.4
- Required margin of overpull: 100,000 lb

- From offset wells, it is known that six joints of heavy-weight drill pipe are desirable
- Assume vertical hole.

Solution

Selection drill collar size, Table 4-73, $7\frac{3}{4} \times 2\frac{13}{16}$ in., unit weight = 139 lb/ft. Such drill collars can be caught with overshot or washed over with washpipe.

$$\text{Length of drill collars} = \frac{(60,000)(1.15)}{(139)(0.816)} = 608\,\text{ft}$$

Note: Buoyant factor K_b = 0.816.

Select 21 joints of $7\frac{3}{4} \times 2\frac{13}{16}$ in. drill collars that give the length of 630 ft.

Section modulus of drill collars calculate to be 89.6 in.^{3}.

Determine size of heavy-weight drill pipe.

To maintain BSR of less than 5.5, selection 5-in. heavy-weight drill pipe with unit weight of 49.3 lb/ft (see Table 4-91) and section modulus of 21.4 in^3.

Length of heavy-weight drill pipe L_{hw} = (5)(30) = 180 ft.

Selection 5 in. IEU new drill pipe with unit nominal weight 19.5 lb/ft (see Table 4-79), steel grade X-95, with NC 50 tool joint (see Table 4-89).

Unit weight of drill pipe corrected for tool joint is 21.34 lb/ft. Section modulus of this pipe can be calculated to be 5.7 in^3.

From Table 4-80, the minimum tensile load capacity of selected drill pipe P_t = 501,090 lb.

From Table 4-89, the recommended makeup torque T = 26,000 ft/lb.

The tensile load capacity of drill pipe corrected for the effect of the maximum allowable torque, according to Equation 4-58 is

$$P = \left\{(501090)^2 - 3\left[\frac{(6.2832)(312,000)}{11.416}\right]^2\right\}^{1/2} = 403,271\,\text{lb}$$

Determine the maximum allowable length of the selected drill pipe from Equation 4-72:

$$L_{dp} = \frac{403,271}{(1.4)(0.816)(21.34)} - \frac{(630)(139)}{21.34} - \frac{(180)(49.3)}{21.34} = 12,022\,\text{ft}$$

Required length of drill pipe L_{dp} = 10000 - (630+180) = 9,190 ft.

Since the required length of drill pipe (9,190 ft) is less than the maximum allowable length (12,022 ft), it is apparent that the selected drill pipe satisfies tensile load requirements.

Obtained margin of overpull:

$$\text{MOP} = (0.9)(501090) - [(630)(139) + (180)(49.3) + (9190)(21.34)](0.816)$$

$$= 212,253 \text{ lb (greater than required 100,000 lb).}$$

In the above example, the cost of drill string is not considered. From a practical standpoint, the calculations outlined above should be performed for various drill pipe unit weights and steel grades and, finally, the design that produces the lowest cost should be selected.

The maximum load that can be suspended in the slips, from Equation 4-83 (assume K = 2.36, L_s = 12 in.) is

$$W_{max} = \frac{501090}{1.1\left[1 + \frac{5}{2}\frac{2.36}{12} + \left(\frac{5}{2}\frac{2.36}{12}\right)^2\right]^{1/2}} = 346,056\,\text{lb}$$

Total weight of string = 238,727 lb.

The drill pipe will not be crushed in the slips. The drill string design satisfies the specified criteria.

DRILLING BITS AND DOWNHOLE TOOLS

Classification of Drilling Bits

Numerous individual rotary bit designs are available from a number of manufacturers. All of them are designed to give optimum performance in various formation types. There is no universal agreement on this subject; variations in operating practices, type of equipment used or hole conditions require an experimental approach. It has been noted in development drilling that those operators who consistently drill the "fastest" wells usually employ several types of bits.

All manufacturers use their own classification numbers for their bits. This results in mass confusion about which bit to use in what formation and whose bit is better. The International Association of Drilling Contractors (IADC) has addressed this classification problem through the development of a unified system. But whose bit is better is left to trial-and-error experimentation by the individual operator.

Rotary drilling bits are classified into the following types:

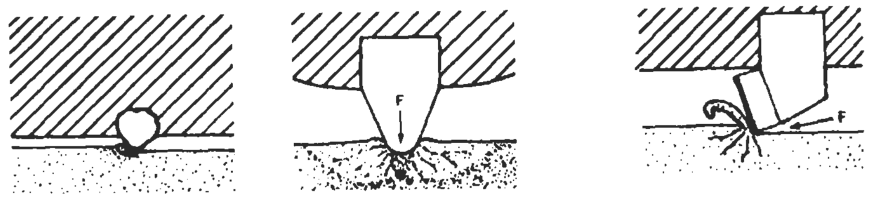
1. Roller rock bits (milled tooth bits)
2. Tungsten carbide insert roller bits
3. Diamond bits and core bits
4. Polycrystalline diamond compacts (PCD) bits

The cutting mechanics of different types of bits are shown in Figure 4-135 [43].

IADC Classification Chart and Bit Codes

In 1987, IADC developed a revised standard nomenclature for roller bits which includes a classification chart and a four-character bit code. All manufacturers must classify their bits in a prescribed manner on the IADC classification chart. The classification includes four categories: series, types, feature, and additional features. Figure 4-136 shows an IADC classification chart. A letter used in the fourth position of the four-character IADC code indicates additional design features specified in Table 4-91.

Series. Numbers 1, 2 and 3 are for milled tooth bits and designate soft, medium and hard formations, respectively. Numbers 4, 5, 6, 7 and 8 are for insert bits



Diamond Bit (Plowing/ Grinding)

Roller Cone Bit (Crushing)

Diamond Compact PCD Bit (Shearing)

Figure 4-135. Rock cutting mechanics of different bit types [43A]. *(Courtesy Hughes Christensen.)*

	SERIES	FORMATIONS	TYPES	FEATURES: STANDARD ROLLER BEARING (1)	ROLLER BRG., AIR COOLED (2)	ROLLER BRG., GAGE PROTECTED (3)	SEALED ROLLER BEARING (4)	SEALED ROLLER, BRG., GAGE PROTECTED (5)	SEALED FRICTION BEARING (6)	SEALED FRICTION, BRG., GAGE PROTECTED (7)
MILLED TOOTH BITS	1	SOFT FORMATIONS WITH LOW COMPRESSIVE STRENGTH AND HIGH DRILLABILITY	1 2 3 4							
	2	MEDIUM TO MEDIUM HARD FORMATIONS WITH HIGH COMPRESSIVE STRENGTH	1 2 3 4							
	3	HARD SEMI-ABRASIVE AND ABRASIVE FORMATIONS	1 2 3 4							
INSERT BITS	4	SOFT FORMATIONS WITH LOW COMPRESSIVE STRENGTH AND HIGH DRILLABILITY	1 2 3 4							
	5	SOFT TO MEDIUM FORMATIONS WITH LOW COMPRESSIVE STRENGTH	1 2 3 4							
	6	MEDIUM HARD FORMATIONS WITH HIGH COMPRESSIVE STRENGTH	1 2 3 4							
	7	HARD SEMI-ABRASIVE AND ABRASIVE FORMATIONS	1 2 3 4							
	8	EXTREMELY HARD AND ABRASIVE FORMATIONS	1 2 3 4							

NOTE: Bit classifications are general guidelines only.
All bit types will drill effectively in formations other than those specified. • Limited Availability

Figure 4-136. 1987 IADC roller bit classification chart.

and designate soft, soft to medium, medium, hard, and extremely hard formations, respectively.

Types. There are four grades of hardness in each series. These four grades (or types) are numerically 1, 2, 3 and 4.

Features. Seven categories of bearing design and gauge protection are defined as features. Features 8 and 9 are reserved for future use.

Table 4-91
Roller Bit Additional Design Feature [44]

Code	Feature	Code	Feature
A	Air application[1]	N	
B		O	
C	Center jet	P	
D	Deviation control	Q	
E	Extended jets	R	Reinforced welds[2]
F		S	Standard steel tooth model[3]
G	Extra gauge/body protection	T	
H		U	
I		V	
J	Jet deflection	W	
K		X	Chisel insert
L		Y	Conical insert
M		Z	Other insert shape

[1]Journal bearing bits with air circulation nozzles
[2]For percussion applications
[3]Milled tooth bits with none of the extra features listed in this table
Courtesy SPE

Additional Features. Additional features are important since they can affect bit cost, applications and performance. The fourth character of the IADC code is used to indicate additional features. Eleven such alphabetic characters are presently defined as shown in Table 4-91 [44]. Additional alphabetic characters may be utilized as required by future roller bit designs. Although the fourth character does not appear on the IADC bit comparison chart, it appears everywhere else that the IADC code is recorded such as on the shipping container and bit record.

The IADC code should be interpreted as shown in the following examples: (1) 124E–a soft formation, sealed roller bearing milled tooth bit with extended jets, (2) 437X–a soft formation, sealed friction bearing insert bit, with gauge protection and chisel-shaped teeth.

Some bit designs may have a combination of additional features. In such cases the manufacturer selects the most significant feature for the fourth character of the classification code.

IADC publishes the current bit classification charts for nearly all of the major roller bit manufacturers. In addition, IADC publishes reference charts for "obsolete" bits that are no longer available. These are useful when reviewing older bit records in order to plan a well.

The current IADC classification charts for seven roller bit manufacturers are shown in Ref. [44].

Bit classification is general and is to be used simply as a guide. All bit types will drill effectively in formations other than those specified. It is the responsibility of the manufacturer to classify his bits at his or her own discretion.

Roller Rock Bit Design

The elements of the roller rock bit are shown in Figure 4-137 [45]. Roller rock bits have three major components: the cone cutter, the bearings and the bit body. The cutting elements are circumferential rows of teeth extending from

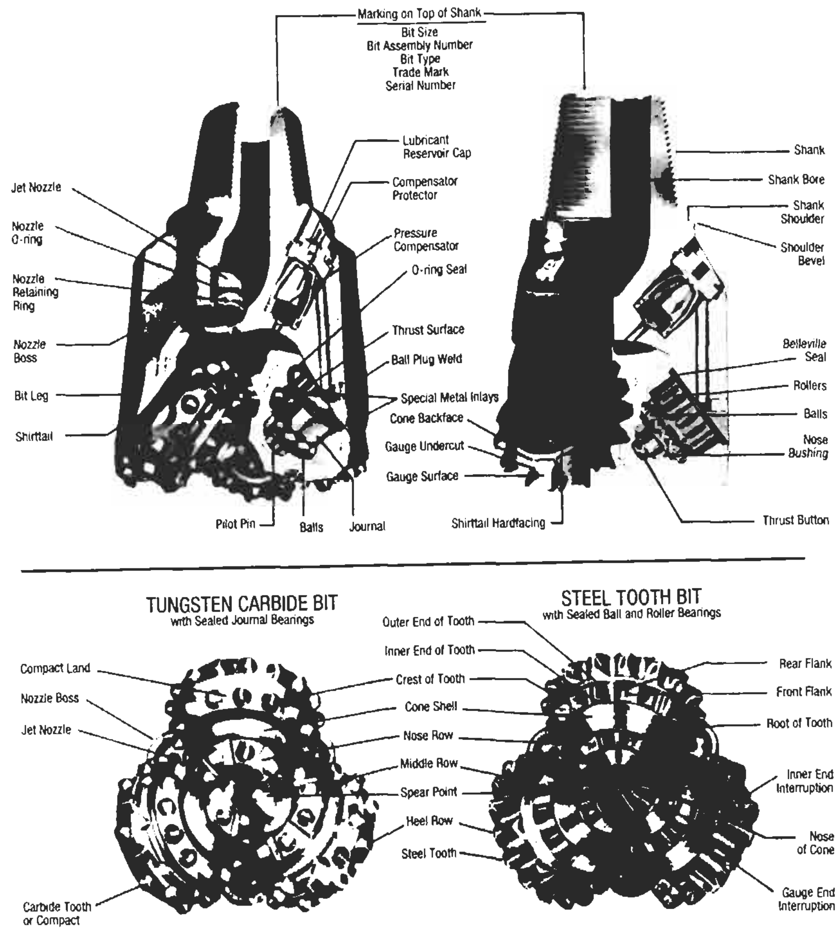


Figure 4-137. Roller (rock) bit elements [45]. *(Courtesy Canadian Association of Oilwell Drilling Contractors.)*

each cone and interfitting between rows of teeth on the adjacent cones. The teeth are either steel and machined as part of the cone, or tungsten carbide compacts pressed into holes machined in the cone surfaces. The cutters are mounted on bearings and bearing pins that are an integral part of the bit body.

The size or thickness of the various bit components depends on the type of formation to be drilled. For instance, soft formation bits generally require light weights and have smaller bearings, thinner cone shells and thinner bit leg sections than hard formation bits. This allows more space for long, slender cutting elements. Hard formation bits, which must be run under heavy weights, have stubbier cutting elements, larger bearings and sturdier bodies. Shown in Figure 4-138 are the changes of various bit design factors across the IADC classification chart.

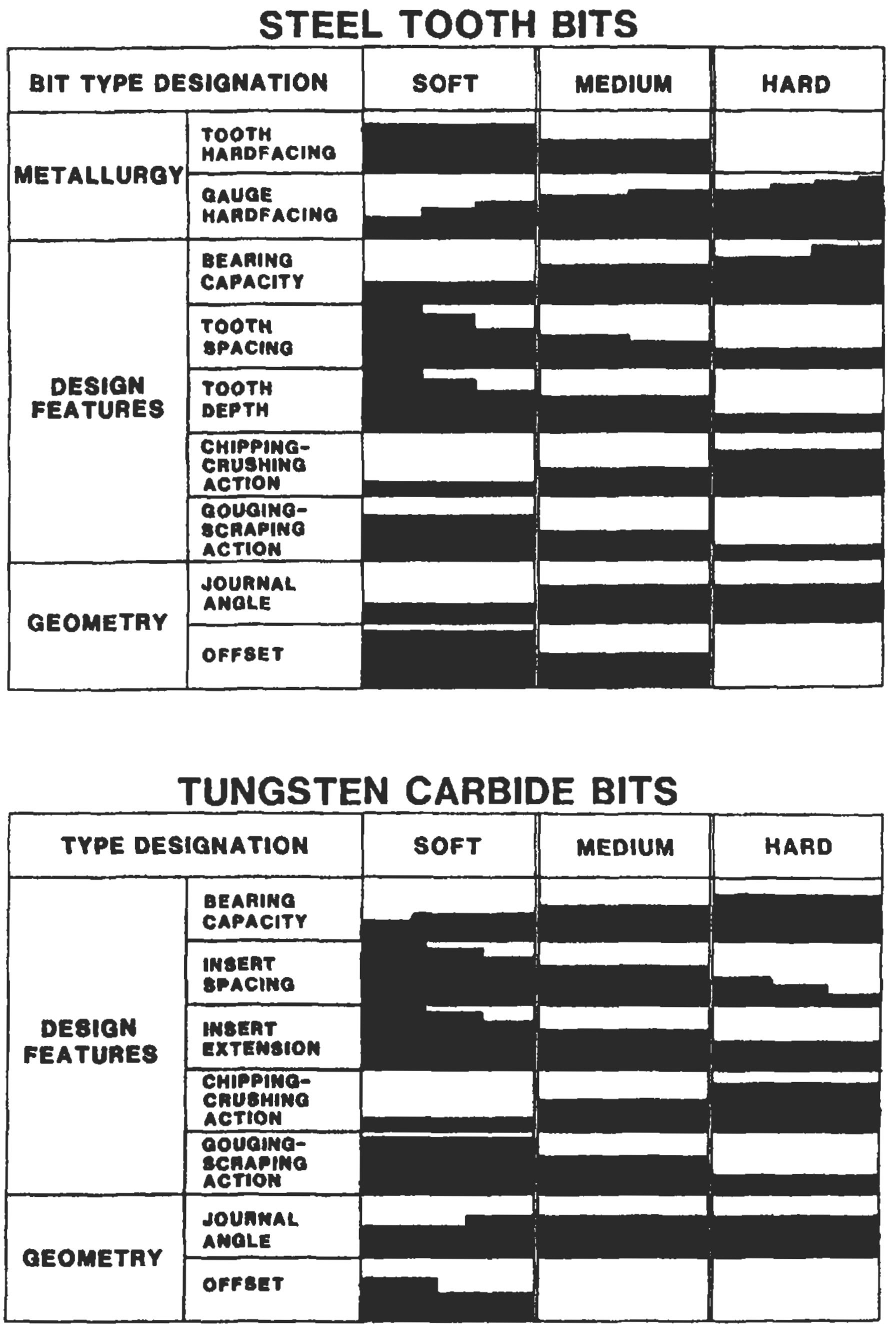


Figure 4-138. Roller cone bit design trends [44]. *(Courtesy SPE.)*

Cone Cutter Design

To understand how cone geometry can effect the way rock bit teeth cut rock, consider the moderate soft formation cone shown schematically in Figure 4-139 [45]. Such cones are designed to depart substantially from true rolling action on the bottom of the borehole. They have two or more basic cone angles, none of which has its apex at the center of bit rotation. The conical heel surface tends to rotate about its theoretical apex and the inner row surface about the center of its own apex. Since the cones are forced to rotate about the bit centerline, they slip as they rotate and produce a tearing, gouging action. This action is obtained by moving the cone centerline away from the center of bit rotation, as shown in Figure 4-139. Bits for hard formation have cones that are more nearly true rolling and use little or no cone offset. As a result, they break rock primarily by crushing.

The bearing journal angle specified in Figure 4-139 (relative to horizontal) is reduced for softer bits and increased for harder bits. This alters the cone profile which in turn affects tooth action on the bottomhole and gauge cutter action on the wall of the hole. No roller cone bit has truly conical-shaped cones, but softer bits have more highly profiled, i.e., less-conical cones than harder bits. This increases the scraping action of both bottomhole cutters and gauge surfaces. The scraping action is beneficial for drilling soft formations but it will result in accelerated tooth and gauge wear if the formation is relatively abrasive. Scraping action is minimized on hard formation bits where strength and abrasion resistance are emphasized in the design.

Bearing Design

The major bearing design used in present rock bits are shown in Fig. 4-140 [44]. Three styles of bearing designs are generally available: non-sealed roller bearings, sealed roller bearings, and sealed friction bearings. Another name for friction bearings is journal bearings. A fourth style features air-cooled non-sealed roller bearings intended for air drilling applications.

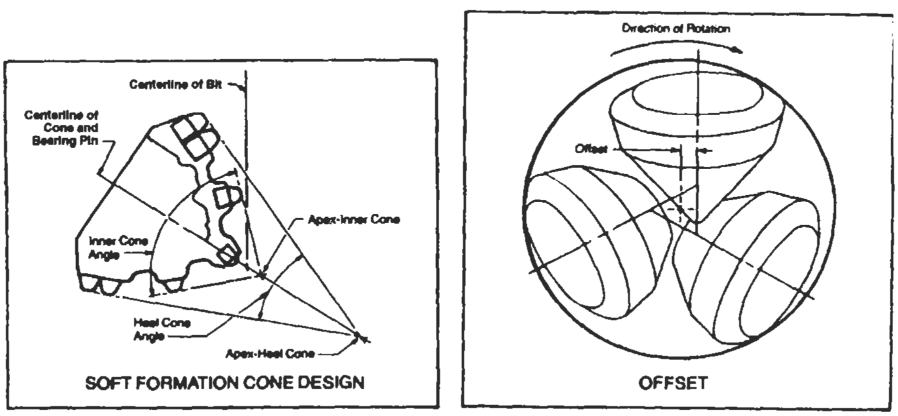


Figure 4-139. Roller bit cone design features. *(Courtesy Canadian Association of Oilwell Drilling Contractors.)*

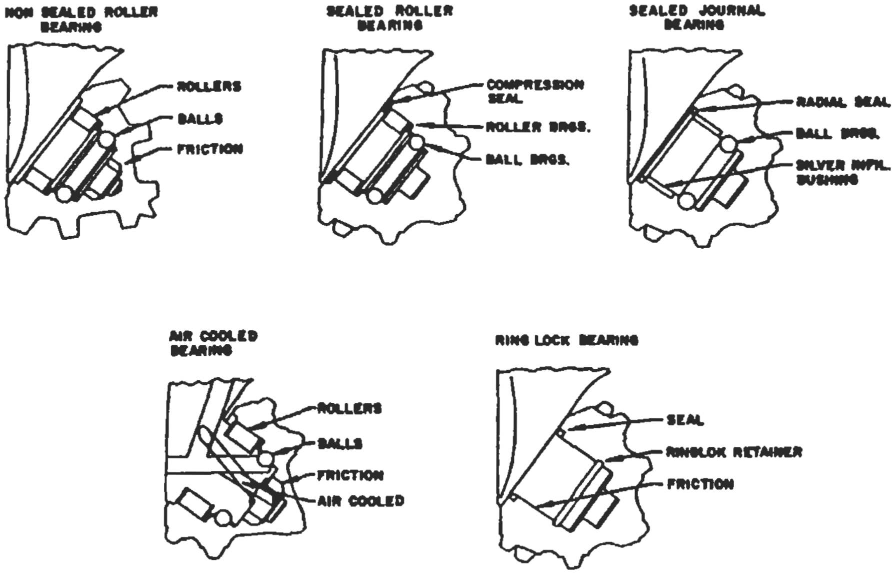


Figure 4-140. Roller cone bit bearings design [44]. *(Courtesy SPE.)*

Sealed Friction Bearing (Journal Bearing). The journal bearing, developed to match the life of carbide cutting structures, does not contain rollers; but contains only a solid journal pin mated to the inside surface of the cone. This journal becomes the primary load carrying element for the cone loads.

Advances in product design, metallurgy and manufacturing processes have produced a journal-bearing featuring precisely controlled journal, pilot pin and thrust-bearing surfaces. The bearing is designed and manufactured to ensure that all bearing elements are uniformly loaded. Substantially higher weights and rotary speeds can be run without decreasing bearing life. Sealed journal bearings provide the best wear resistance at normal rotary speeds through a combination of better load distribution and precision-machined surfaces.

Sealed Ball and Roller Bearing (Self-Lubricating). The sealed ball and roller bearing was introduced in carbide tooth bits, but is now primarily in steel tooth bits and generally lasts as long as the cutting structure. Some carbide tooth bits of 12 $\frac{1}{4}$-in. and larger sizes also are available with this type bearing. Sealed roller bearings are lubricated by clean grease rather than drilling mud and thus tend to last longer than standard roller bearings.

Nonsealed Ball and Roller Bearings. The nonsealed ball and roller bearings were introduced to replace the primitive friction journal bearing at a time when only steel tooth bits were available. They operated well in mud, and in many cases were adequate to last as long as or longer than the cutting structures they served. Today, the nonlubricating bearings are used in steel tooth bits to drill the top section of the hole where trip time is low and rotary speed are often high.

The major portion of the radial load on the cone cutter is absorbed by the roller race, with the nose bearing absorbing a lesser amount. The thrust surface is perpendicular to the pilot pen and the thrust button is designed to take outward thrust. The ball bearings allow the cutter to take inward thrust. When other bearing parts are worn out, the balls will also take some radial and outward loading.

Air Circulating Ball and Roller Bearings. When air, gas or mist are used as a drilling fluid, nonsealed ball land roller bearing bits are used. The design allows a portion of the drilling fluid to be diverted through the bearing for cooling, cleaning and lubrication. Since free water in contact with loaded bearing surfaces will reduce their life, bits are equipped with a water separator to prevent this action in cases where water is injected into the air or gas.

Also available for the prevention of bit plugging are backflow valves that prevent cuttings suspended in water from backing up through the bit into the drill pipe when the flow of air or gas is interrupted.

The *"ring lock" bearing* is a newer friction bearing design which is also classified under Columns 6 or 7 on the IADC chart. Instead of ball bearings, a *snap-ring retainer* holds the cone shell in place. This provides greater load-bearing area and cone shell thickness in the region where the ball bearing race has been eliminated. A compressed O-ring seal prevents drilling mud from contaminating the bearing grease.

Steel Tooth Cutting Structure Design

The designs of steel tooth bits cutting structure are shown in Figure 4-141 [44]. Steel tooth bits are employed in soft formations where high rotary speeds can be used. All steel tooth cones have tungsten carbide hardfacing material applied to the gage surface of the bit body and to the teeth as dictated by the intended use of a specific roller cone design. Tooth hardfacing improves wear resistance but reduces resistance to chipping and breaking. For this reason, hard formation steel tooth cones usually have gage hardfacing only, while soft formation steel tooth cones usually have hardfacing on tooth surfaces as well as the gauge surface.

Soft Formation Bits. Bits for drilling soft formations are designed with long, widely spaced teeth to permit maximum penetration into the formation and removal of large chips.

Medium Formation Bits. Medium and medium-hard formation bits are designed with more closely spaced teeth, since the bit cannot remove large pieces of the harder rock from the bottom of the borehole. The teeth also have slightly larger angles to withstand loads needed to exceed formation strength and produce chips.

Hard Formation Bits. The heel or outermost row on each cone is the driving row, that is, this row generates a rock gear pattern on the bottom of the borehole that, in the case of these strong rocks, is not easily broken away from the wall of the borehole. The numbers of heel row teeth used on each of the three cones are selected to prevent the heel teeth from "tracking," or exactly following in the path of the preceding cone, which would cause abnormally deep rock tooth holes on the borehole bottom.

Insert Bit Tooth Design

The companion of insert bits cutting structure is shown in Figure 4-142 [44]. Initially, the tungsten carbide tooth bit was developed to drill extremely hard, abrasive cherts and quartzites that had been very costly to drill because of the

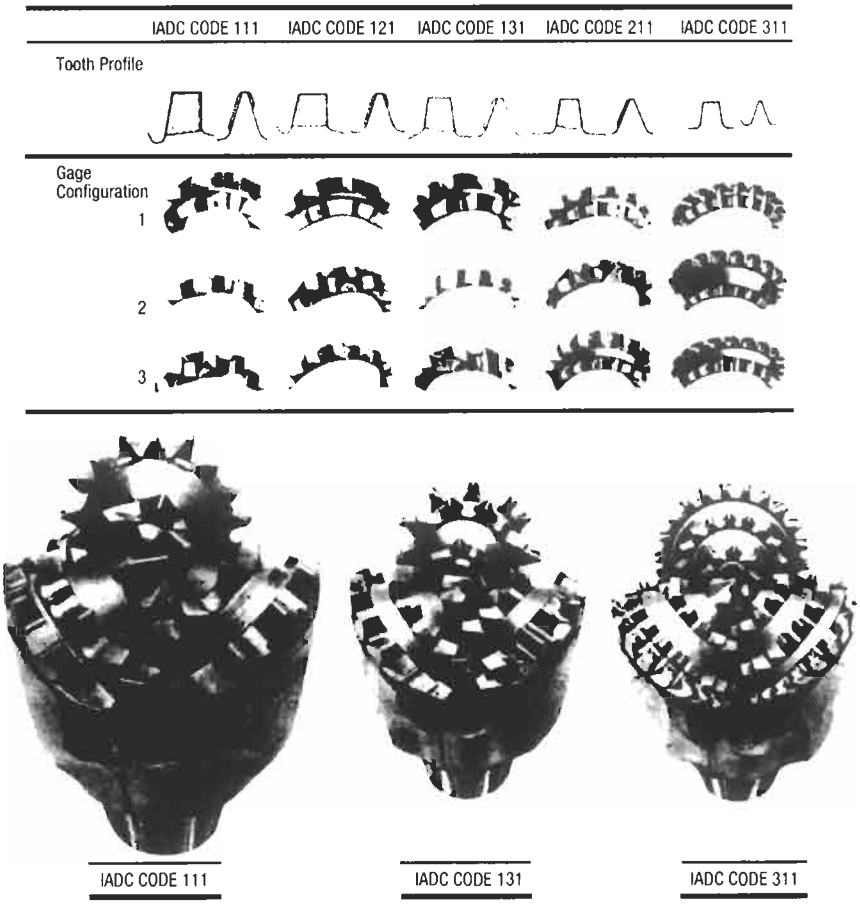


Figure 4-141. Steel tooth bit cutting structure design [44]. *(Courtesy SPE.)*

relatively short life of steel tooth bits in such formations. In this type of bit, tungsten carbide and forged alloy steel are combined to produce a cutting structure having a high resistance to abrasive wear and extremely high resistance to compressive loads. Compacts of cylindrical tungsten carbide with various shaped ends are pressed into precisely machined holes in case-hardened alloy steel cones to form the teeth. The grain size and cobalt content of tungsten carbide inserts is varied to alter the impact toughness and abrasion resistance of the cutter. Softer formation inserts, which are usually run in less abrasive rocks at higher rotary speeds, require increased toughness to resist breakage of the relatively long cutters. A cobalt content of 16% and average grain size of 6 μm is typical for such inserts. Hard formation inserts are generally run in more abrasive rocks at higher WOB levels. Hard formation inserts have a more breakage-resistant geometry so abrasion resistance becomes the most important factor. Thus the cobalt content is reduced to about 10% and the average grain size is approximately 4 μm.

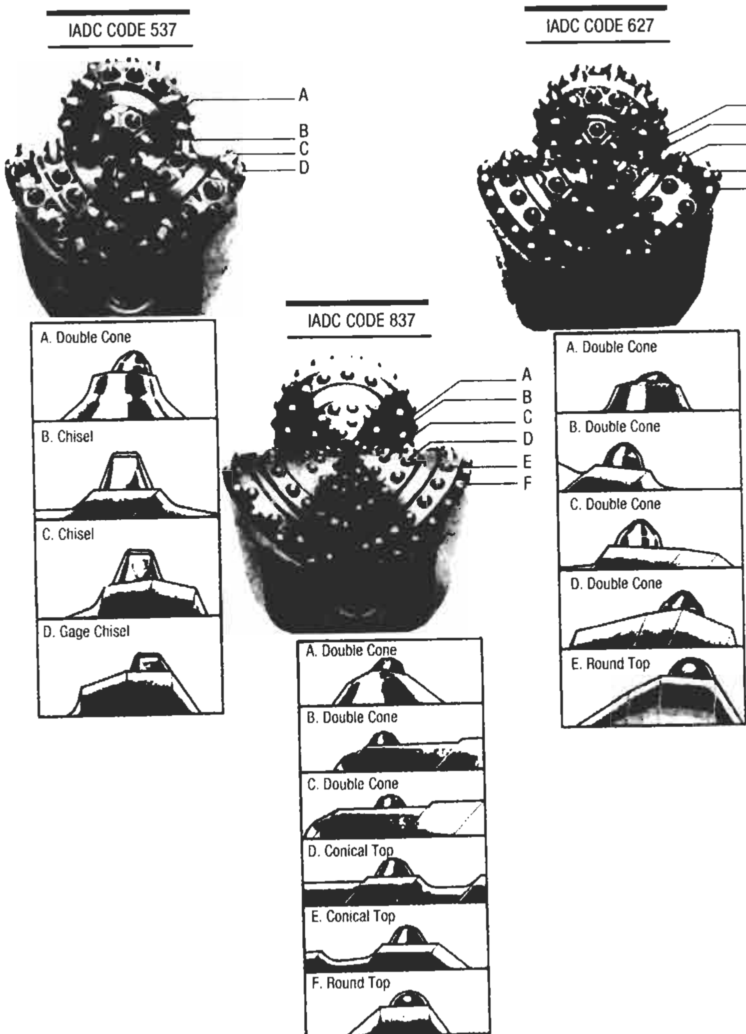


Figure 4-142. Cutting structures of insert bits [44]. *(Courtesy SPE.)*

Dull Grading for Roller Cone Bits

Grading a dull bit and evaluating the findings can increase drilling efficiency while lowering drilling cost. Also, the examination of the dull bit can often furnish information that will assist the selection of bit types and also help determine the advisability of changing operating practices. The bit life need not be totally used before it is graded, since the grading is to determine what happened to the bit during a specific drilling run. The condition of each bit should be reported in the "Bit Record" section of the IADC Daily Drilling Report form.

Tooth Wear. Tooth wear is estimated in eighths ($\frac{1}{8}$) of the initial tooth height. Since tooth wear is likely not uniform on any row of teeth of a given cone, it is advisable to take several readings and report an average figure. The following is the terminology used to report tooth wear:

Tooth Dullness	Milled Tooth	Insert Bits
T1	Tooth height $\frac{1}{8}$ gone	$\frac{1}{8}$ of inserts lost or broken
T2	Tooth height $\frac{1}{4}$ gone	$\frac{1}{4}$ of inserts lost or broken
T3	Tooth height $\frac{3}{8}$ gone	$\frac{3}{8}$ of inserts lost or broken
T4	Tooth height $\frac{1}{2}$ gone	$\frac{1}{2}$ of inserts lost or broken
T5	Tooth height $\frac{5}{8}$ gone	$\frac{5}{8}$ of inserts lost or broken
T6	Tooth height $\frac{3}{4}$ gone	$\frac{3}{4}$ of inserts lost or broken
T7	Tooth height $\frac{7}{8}$ gone	$\frac{7}{8}$ of inserts lost or broken
T8	Tooth height all gone	All of inserts lost or broken

Bearing Condition. The measurement of the bearing wear is very subjective. It is recommended to estimate it in eighths of the life of the bearing.

Since mechanical aids are not available, it is necessary to eyeball the bearing wear and estimate rotating hours left. Knowing the rotating hours of the bit at the bottom of the well, it is possible to calculate the ratio. An estimation of the total bearing life is expressed by a ratio of eighths of the bearing life as follows:

Bearing Condition		
B1	Bearing life used:	$\frac{1}{8}$
B2	Bearing life used:	$\frac{1}{4}$ (tight)
B3	Bearing life used:	$\frac{3}{8}$
B4	Bearing life used:	$\frac{1}{2}$ (medium)
B5	Bearing life used:	$\frac{5}{8}$
B6	Bearing life used:	$\frac{3}{4}$ (loose)
B7	Bearing life used:	$\frac{7}{8}$
B8	Bearing life all gone:	(Locked or lost)

Example

A roller rock bit is pulled out of the hole after 12 hr of rotation at the bottom. The driller estimates that the worst cone could rotate 4 hr more before being completely worn out; thus total bearing life estimated is 16 hr.

Therefore: $\frac{12}{16} = \frac{3}{4} = \frac{6}{8}$, i.e. B6 is reported.

Gauge Wear. When the bit pulled out of the hole is in gage, this is reported by the letter "I." When the bit pulled out of the hole is out of gage, this is reported by the amount of gage wear in $\frac{1}{16}$ of an inch.

To measure the amount of gage wear on a used bit, set the ring gauge on two cones and measure the distance between the ring gauge and the third cone in fractions of an inch, or in millimeters.

Dull Grading of Roller Cone Bits. The grading is accomplished by using an eight-column dull code as follows [46].

Cutting Structure				B	G	Remarks	
Inner Rows (I)	Outer rows (O)	Dull char. (D)	Location (L)	Bring seal (B)	Gage $\frac{1}{16}$ (G)	Other dull (O)	Reason pulled (R)

1. Column 1 (I) is used to report the condition of the cutting structure on the inner two-thirds of the bit.
2. Column 2 (O) is used to report the condition of the cutting structure on the outer one-third of the bit.

   In columns 1 and 2 a linear scale from 0 to 8 is used to describe the condition of the cutting structure as explained above.

   For example: a bit missing half of the inserts on the inner two-thirds of the bit due to loss or breakage with the remaining teeth on the inner two-thirds having a 50% reduction in height due to wear, should be graded a 6 in column 1. If the inserts on the outer one-third of the bit were all intact but were reduced by wear to half of their original height, the proper grade for column 2 would be 4.
3. Column 3 (D) uses a two-letter code to indicate the major dull characteristic of the cutting structure. Table 4-92 lists the two-letter codes for the dull characteristics to be used in this column.

Table 4-92
Major/Other Dull Characteristics [46]

* BC — Broken cone	LN — Lost nozzle
BT — Broken teeth/cutters	LT — Lost teeth/cutters
BU — Balled up	OC — Off center wear
* CC — Cracked cone	PB — Pinched bit
* CD — Cone dragged	PN — Plugged nozzle
CI — Cone interference	RG — Rounded gauge
CR — Core	RO — Ring out
CT — Chipped teeth	SD — Shirttail damage
ER — Erosion	SS — Self sharpening wear
FC — Flat crested wear	TR — Tracking
HC — Heat checking	WO — Wash out on bit
JD — Junk damage	WT — Worn teeth/cutters
* LC — Lost cone	NO — No other major/other dull characteristic

*Show cone number(s) under LOCATION (L) column 4 of the IADC dull code.
Courtesy SPE

4. Column 4 (L) uses a letter or number code to indicate the location on the face of the bit where the major cutting structure dulling characteristic occurs. Table 4-93 lists the codes to be used for describing locations on roller cone bits.
5. Column 5 (B) uses a letter or a number code, depending on bearing type, to indicate bearing condition on roller cone bits. For nonsealed bearing roller cone bits a linear scale from 0 to -8 is used to indicate the amount of bearing life that has been used. A 0 indicates that no bearing life has been used (a new bearing), and an 8 indicates that all of the bearing life has been used (locked or lost). For sealed bearing (journal or roller) bits a letter code is used to indicate the condition of the seal. An "E" indicates an effective seal, and an "F" indicates a failed seal(s).
6. Column 6 (G) is used to report on the gage of the bit. The letter "I" indicates no gage reduction. If the bit does have a reduction in gauge it is to be recorded in $\frac{1}{16}$ of an inch. The "two-thirds rule" is correct for three-cone bits. The two-thirds rule, as used for three-cone bits, requires that the gauge ring be pulled so that it contacts two of the cones at their outermost points. Then the distance between the outermost point of the third cone and the gage ring is multiplied by two-thirds and rounded to the nearest $\frac{1}{16}$ of an inch to give the correct diameter reduction.
7. Column 7 (O) is used to report any dulling characteristic of the bit, in addition to the major cutting structure dulling characteristic listed in column 3 (D). Note that this column is not restricted to only cutting structure dulling characteristics. Table 1 lists the two-letter codes to be used in this column.
8. Column 8 (R) is used to report the reason for pulling the bit out of the hole. Table 4-94 lists the two-letter or three-letter codes to be used in this column.

Table 4-93
Location (Roller Cone Bits) [46]

N — Nose rows	Cone # or #'s
M — Middle rows	1
H — Heel rows	2
A — All rows	3

Courtesy SPE

Table 4-94
Reason Pulled [46]

BHA — Change bottom hole assembly	HP — Hole problems
DMF — Down hole motor failure	HR — Hours on bit
DSF — Drill string failure	PP — Pump pressure
DST — Drill stem test	PR — Penetration rate
DTF — Down hole tool failure	RIG — Rig repairs
LOG — Run logs	TD — Total depth/casing depth
CM — Condition mud	TQ — Torque
CP — Core point	TW — Twist off
DP — Drill plug	WC — Weather conditions
FM — Formation change	

Courtesy SPE

Example [46]

We will grade three dulled roller cone bits, and discuss some possible interpretations of the wear as it relates to bit selection and application. It should be noted that there may be more than one "correct" dull grading for each bit. This can happen if two persons should disagree on the primary cutting structure dulling characteristic or on what the other dulling characteristic should be. Regardless, the IADC dull grading system provides the man on the rig with ample opportunity to report what he sees when examining a dull.

The first dull bit is a $7\frac{7}{8}$" IADC 5-1-7-X bit and has been graded as a 6, 2, BT, M, E, I, NO, PR (see Table 4-95). The bit looks to have been dulled by encountering a harder formation than the bit was designed for. This is indicated by the heavy tooth breakage on the inner teeth, and by the bit having been pulled for penetration rate (the reduced penetration rate having been caused by the tooth breakage occurring when the bit encountered the hard formation). Excessive weight on the bit could also cause the dull to have this appearance. If the run was of reasonable duration, then the bit application was proper as evidenced by the lack of "other" dulling features, the effective seals, and the fact that the bit is still in gage. However if the bit had a shorter than expected run, it is probable that the application was improper. The bit may have been too "soft" for the formation, or it may have been run with excessive weight on the bit.

The second dull bit is a $7\frac{7}{8}$-in. IADC 8-3-2-A bit that was graded 5,8,WT, A,3,2,FC,HR (see Table 4-95). This dull grade indicates proper bit selection and application. The tooth wear (WT is normal in the harder tungsten carbide insert bits as opposed to chipped or broken teeth which could indicate excessive WOB or RPM) is not a great deal more on the outer cutters than on the inner cutters, indicating proper RPM and WOB. The bit was still drilling well when pulled as indicated by listing HRS as the reason pulled. However the bit was slightly under gage ($\frac{2}{16}$ in.) at this point and may well have lost more gage rapidly if left in

Table 4-95
Example of Dull Bit Gradings [46]

BIT RECORD												
BIT NO.	1				2				3			
SIZE	7 7/8"				7 7/8"				12 1/4"			
IADC CODE	5	1	7	X	8	3	2	A	5	1	7	X
DULL CUT. STRUC. I O D L	6	2	BT	M	5	8	WT	A	0	0	NO	A
B G O R	E	I	NO	PR	3	2/16	FC	HR	E	I	LN	PP

Courtesy SPE

the hole. This supports the decision to pull the bit based on the hours. A bearing condition of 3 on the air bearings indicates good bearing life still remaining. Since there are no harder bits available, and the dull grade indicates that a softer bit would not be appropriate, this seems to have been a proper bit application.

The third dull bit is a $12\frac{1}{4}$-in. IADC 5-1-7-X bit and was graded 0,0,NO, A,E,I,LN,PP (see Table 4-95). Since there is no evidence of any cutting structure dulling, the 0,0,NO,A is used to describe the cutting structure. If this bit had been run for a long time before losing the nozzle, this dull grading would indicate that a softer bit (possibly a milled tooth bit) might be better suited to drill this interval. If the run was very short, then the indication is that the nozzle was not the proper one, or that it was improperly installed. If this was the case, then no other information concerning the proper or improper bit application can be determined.

Steel Tooth Bit Selection

The decision to run a specific bit can only be based on experience and judgment. Usually, a bit manufacturer provides qualitative recommendations on selection of his bits.

General considerations are:

1. Select a bit that provides the fastest penetration rate when drilling at shallow depths.
2. Select a bit that provides maximum footage rather than maximum penetration rate when drilling at greater depths where trip time is costly.
3. Select a bit with the proper tooth depths, as maximum tooth depth is sometimes overemphasized. When drilling at 200 rpm at a rate of 125 ft/hr, only $\frac{1}{8}$ of the hole is cut per revolution of the bit. Bits are designed with long teeth and tooth deletions for tooth cleaning.
4. Select a bit with enough teeth to efficiently remove the formations, as that often can be more important than using a bit with maximum tooth depth.
5. Select a bit with enough gage tooth structure so that the gage structure will not round off before the inner-tooth structure is gone.
6. Select a bit with tungsten carbide inserts on gage if sand streaks are expected in the formation. Do not depend on gage hardfacing alone to hold the hole to gauge.

Crooked hole considerations are:

1. Select a bit with less offset.
2. Select a bit with open gage teeth to straighten hole.
3. Selecting a bit with more teeth and with shorter crested teeth results in smoother running and reduced rate of tooth wear.
4. Selecting a bit with "T"-shaped gage teeth reduces the tendency for the bit walk.

Pinching considerations are:

1. Select a bit with less offset and harder formation type (more vertical gage angle).
2. Do not select a bit with reinforced gage teeth unless excessive gage tooth rounding is the reason for pinching.

Reaming considerations are:

1. Select a bit with minimum offset.
2. Select a bit with "L" or "T"-shaped gage structure.

Insert Bit Selection

The decision to run a specific insert bit can only be based on experience and judgment.

General considerations are:

1. Select a tungsten carbide bit with chisel crest inserts when drilling a formation that is predominantly shale. Use bit type 4-2, 5-2, 6-1 or 6-2.
2. Select a tungsten carbide bit with high offset and chisel inserts if the shale content of the formation increases and/or the mud density is high. Use bit type 5-2 or 5-3.
3. Select a tungsten carbide bit with shorter chisel inserts and less offset if the formations become more abrasive and unconsolidated. Use bit type 6-3 or 6-4.
4. Select a tungsten carbide bit with projectile or conical inserts when drilling a formation that is predominantly limestone. Use bit type 6-3 or 6-4.
5. Select a tungsten carbide bit with projectile or conical inserts if the sand content and abrasiveness of the formation increases. Use bits type 7-1 to 8-3.

Specific considerations are:

1. Select a tungsten carbide insert with the greatest amount of offset and the longest chisel crested inserts when drilling shale and soft limestone.
2. Select a tungsten carbide insert bit with a medium offset and long chisel crested inserts when drilling sandy shale with limestone and dolomite. Use bits type 4-1 to 5-3.
3. Select a tungsten carbide insert bit with a minimum offset and projectile or conical inserts when drilling limestone, brittle shale, nonporous dolomite and broken formations. Use bit type 6-3 to 7-3.
4. Select a tungsten carbide bit with medium or no offset and chisel crested inserts when drilling sandy shales, limestones and dolomites. Use bit type 5-3 or 6-4.
5. Select a tungsten carbide insert bit with no offset and conical or double cone inserts when drilling hard and abrasive limestone, hard dolomite, chert, pyrite, quartz, basalt, etc. Use bit type 7-4 to 8-3.

Quantitative Method of Bit Selection

This method is based on cost comparison between bit records and the current bit run.

The following example illustrates the application of cost-per-foot data in evaluating the economics of insert bits [34].

Example

Determine the economics for insert bits using the data below.

Applicable costs are:

Mill tooth bits, each	$ 260.00
Insert bits	1,250.00
Mud, per day	500.00
Water, per day	200.00
Desilter, per day	150.00
Supervision, per day	250.00
Total daily cost	$2,610.00
Hourly rig cost	$ 108.00

Trip time is 0.7 per hour per 1,000 ft.

The cost equation is

$$C = B/F + C_d T_d/F + C_t T_t/F \tag{4-75}$$

where C = drilling cost per foot in \$/ft
B = bit cost in \$
F = footage drilled in ft
C_d = rig cost for drilling in \$/hr
T_d = drilling time in hr
T_t = trip time in hr
C_t = rig cost for trip in \$/hr

Assumptions are (1) comparable lithology, (2) $C_d = C_t$ = \$108/hr, and (3) the well in question is to be deepened from 6000 to 7650 ft.

The bit record from the offset control well is presented in Table 4-96.

The cost per foot for each bit run is calculated as follows:

Bit No. 1
Drilling hours = 10.5
Trip hours = (0.7 hr/1000 ft)(5.958 × 1000 ft) = 4.1 hr
Total hours = 14.6 hr
Total footage = 160 ft

Therefore,

$$C = 1/F\ [B + C_d(T_d + T_t)] = 1/160\ \text{ft}[\$260 + \$108/\text{hr}\ (10.5 + 4.1)\text{hr}]$$

$$= \$11.51/\text{ft}$$

Table 4-96
Cost of Steel Tooth Bits [34]

Bit No	Depth out, ft	Footage, ft	Bottom time, hr
1	6008	174	19.0
2	6268	260	19.5
3	6518	250	25.0
4	7444	926	99.75
5	7650	206	25.5

Similar calculations are made for each bit run and recorded on the bit record. The bit record is presented in Table 4-97. Inserts were run below 6500 ft. Cost-per-foot data was calculated for each bit and is presented.

From Table 4-98 insert bits drilled 1132 ft at a cost of $17,205.00.

Cost of conventional bits from the offset well were $27,803.00.

Savings with inserts: $10,597.00.

Roller Rock Bit Hydraulics

Roller rock bit nozzle sizes are given in Table 4-98. The total pressure drop across a roller rock bit, P_b(psi), is [47]

$$P_b = \frac{\bar{\gamma}_m q^2}{7430C^2(d_1^2 + d_2^2 + d_3^2)^2} \tag{4-76}$$

where $\bar{\gamma}$ = specific weight of drilling mud in lb/gal
q = volumetric rate of flow of drilling mud through the bit in gal/min
d_1, d_2, d_3 = the diameters of the three bit nozzles, respectively in in.
C = the nozzle coefficient (usually taken to be about 0.98 or else)

Table 4-97
Cost of Insert Bits [34]

Bit No	Depth out, ft	Footage, ft	Bottom time, hr
1	5958	160	10.5
2	9260	302	19.0
3	6329	69	3.5
4	6469	140	15.5
5	6565	96	9.5
6	6692	127	9.0
7	6873	181	15.5
8	7031	158	16.0
9	7180	149	18.5
10	7243	63	11.0
11	7295	52	11.0
12	7358	53	12.5
13	7425	67	12.5
14	7460	35	10.0
15	7527	67	10.5

Table 4-98
Jet Nozzles Sizes

Inches (32's)	8	9	10	11	12	13	14	15	16	18	20	22	24	26	28
Millimeters (mm)	6.4	7.1	7.9	8.7	9.5	10.3	11.1	11.9	12.7	14.3	15.9	17.5	19.0	20.6	22.2

The bit hydraulic horsepower HP_h is

$$HP_h = \frac{q\Delta P_b}{1714} \tag{4-77}$$

Jet nozzle impact force F_b(lb) is

$$F_b = 0.01823\ Cq(\gamma\Delta P_b)^{1/2} \tag{4-78}$$

The velocity of flow from the nozzles v_n(ft/s) is given by

$$\bar{v}_n = \frac{q_1}{A_1} = \frac{q_2}{A_2} = \frac{q_3}{A_3} \tag{4-79}$$

where q_1, q_2, q_3 = the volumetric flow rate from each nozzle in, $ft^3 \bullet s$
A_1, A_2, A_3 = the cross-sectional area of each nozzle (i.e., $Ai = (\pi/4)di^2$) in ft^2

The total volumetric flow rate q (ft^3/s) can also be expressed as

$$q = \bar{v}_n(A_1 + A_2 + A_3) = \bar{v}_n A_t \tag{4-80}$$

where

$$A_t = A_1 + A_2 + A_3$$

The nozzle velocity v_n(ft/s) is

$$\bar{v}_n = \frac{Q}{3.117a_t} \tag{4-81}$$

where a_t = the total nozzle area, $in.^2$

The maximum cross flow velocity under the bit, v_c(ft/s), is [48]

$$v_c = \frac{5.9}{d_n}\frac{Q\bar{v}_n}{n} \tag{4-82}$$

where d_n = the borehole diameter in in.
n = the number of open nozzles

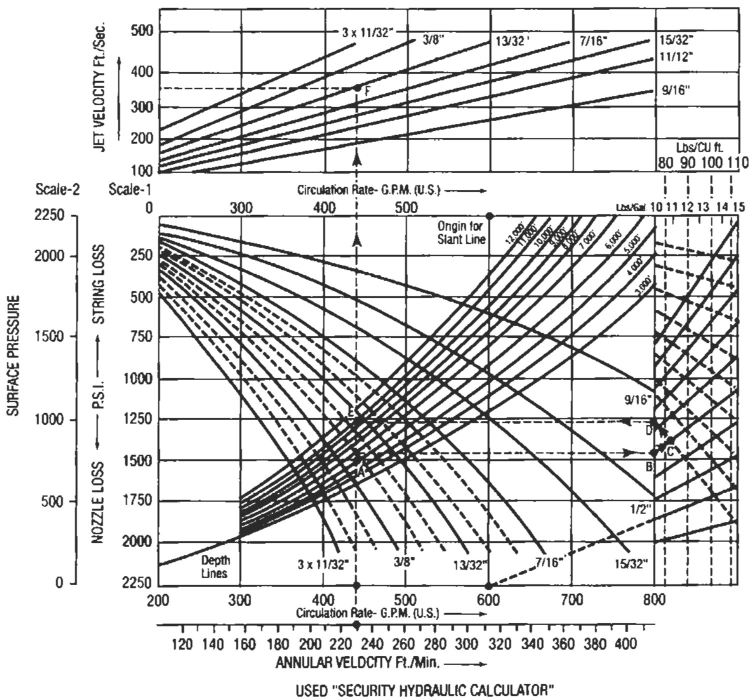
Figure 4-143 gives convenient graphs for nozzle selection for roller rock bits [47].

Example

Determine the pressure drop across the bit and the velocity of the nozzle flow where the total rate of flow through the drill string (and bit) is 300 gal/min, the specific weight of the mud is 12.0 lb/gal, and the three nozzle openings are to be $\frac{15}{32}$ in. in diameter. Use c = 0.95.

The pressure drop across the bit is found by equation 4-76.

Nozzle Selection Chart for Given Drilling String, Hole Size & Working Pressure With Varying Depth & Mud Density



DRILLING STRINGS:
400' of 6 1/4" O.D. Drill Collars
+5" Extra Hole Drill Pipes
Hole size: 8 1/2"
Working Pressure= 2250 P.S.I.

EXAMPLE:
Assume Annular Velocity= 230 Ft/min
Mud Density= 11 lbs/gal
Depth= 7000'

Start at 230 F.P.M. Reach 'A' (7000' Line)
Turn for 'B' (on 10 P.P.G. Line)
Slant to 'C' (on 11 P.P.G. Line)
Slant back to 'D' (on 10 P.P.G. Line)
Reach for 'E' (on 230 F.P.M. Line)

(Nearest higher nozzle line to be taken)
Nozzle set 3 x 13/32"
Follow 230 F.P.M. Line to 'F'
Find Jet Velocity (360 F.P.S.)

A typical nozzle program

Ann. Vel	G.P.M.	Depth	Mud Density Lb/cft	Nozzle
205	390	3000	80	2 x 11/13", 1 x 3/8"
205	390	5000	78	1 x 11/32", 2 x 3/8"
205	390	8000	78	3 x 3/8"
205	390	12,000	78	2 x 3/8", 13/32"

Curves drawn for 10 lbs/gal. for 10 P.P.G.
A&E will be the same point

Figure 4-143. Bit nozzle selection nomogram [47]. (*Courtesy* Harcourt Brace & Co.)

This becomes

$$\Delta P_b = \frac{12.0(300)^2}{7430(0.95)^2[3(0.4688)^2]^2} = 370.7 \text{ psi}$$

The velocity of the nozzle flow is found by equation 4-81.
The total area of the nozzle openings is

$$a_t = 3\frac{\pi}{4}(0.04688)^2 = 0.5178 \text{ in}^2$$

Equation 4-81 becomes

$$\bar{v}_n = \frac{300}{3.117(0.5178)} = 185.9 \text{ ft/s}$$

Diamond Bits

Diamond bits are being employed to a greater extent because of the advancements in mud motors. High rpm can destroy roller rock bit very quickly. On the other hand, diamond bits rotating at high rpm usually have longer life since there are no moving parts.

Diamond Selection. Diamonds used as the cutting elements in the bit metal matrix has the following advantages:

1. Diamonds are the hardest material.
2. Diamonds are the most abrasive resistant material.
3. Diamonds have the highest compressive strength.
4. Diamonds have a high thermal conductivity.

Diamonds also have some disadvantages as cutting elements such as: they are very weak in shear strength, have a very low shock impact resistance, and can damage or crack under extremely high temperatures.

When choosing diamonds for a particular drilling situation, there are basically three things to know. First, the quality of the diamond chosen should depend on the formations being drilled. Second, the size of the diamond and its shape will be determined by the formation and anticipated penetration rate. Third, the number of diamonds used also is determined by formation and the anticipated penetration rate.

There are two types of diamonds, synthetic and natural. Synthetic diamonds are man made and are used in PDC STRATAPAX type bit designs. STRATAPAX PDC bits are best suited for extremely soft formations. The cutting edge of synthetic diamonds are round, half-moon shaped or pointed.

Natural diamonds are divided into three categories. First are the carbonate or black diamonds. These are the hardest and most expensive diamonds. They are used primarily as gage reinforcement at the shockpoint. Second are the West African diamonds. These are used in abrasive formations and usually are of gemstone quality. About 80% of the West African diamonds are pointed in shape and, therefore, 20% are the desirable spherical shape. Third are the Congo or coated diamonds. These are the most common category. Over 98% of these diamonds are spherical by nature. They are extremely effective in soft formations. The other 2% are usually cubed shaped, which is the weakest of the shapes available.

By studying specific formations, diamonds application can be generalized as follows:

Soft, gummy formations–Congo, cubed shaped
Soft formation–large Congo, spherically shaped
Abrasive formation–premium West Africa
Hard and abrasive–Special premium West Africa

Diamond Bit Design. Diamond drill bit geometry and descriptions are given in Figure 4-144 [49]. Diamond core bit geometry and descriptions are given in Figure 4-145 [50].

There are two main design variables of diamond bits, the crown profile and face layout (fluid course configuration).

The crown profile dictates the type of formation for which the bit is best suited. They include the round, parabolic, tapered and flat crown used in hard to extremely hard formations, medium to hard formations, soft formations and for fracturing formations or sidetracks and for kick-offs, respectively.

Cone angles and throat depth dictate the bit best suited for stabilization. Cone angles are steep (60° to 70°), medium (80° to 90°), flat (100° to 120°), best suited for highly stable, stable and for fracturing formation, respectively.

Diamond drill bits with special designs and features include:

1. Long gage bits, used on downhole motors for drilling ahead in vertical boreholes.
2. Flat-bottom, shallow-cone bit designs, used on sidetracking jobs or in sidetracking jobs with downhole motors.
3. Deep cones having a 70° apex angle are normally used in drill bits to give built-in stability and to obtain greater diamond concentration in the bit-cone apex.

Diamond Bit Hydraulics. The hydraulics for diamond bits should accomplish rapid removal of the cuttings, and cooling and lubrication of the diamonds in the bit metal matrix.

Bit Hydraulic Horsepower. The effective level of hydraulic energy (hydraulic horsepower per square inch) is the key to optimum bit performance. The rule-of-thumb estimate of diamond bit hydraulic horsepower HP_h and penetration rates is shown in Table 4-99. The bit hydraulic horsepower is dependent upon the pressure drop across the bit and the flowrate.

Bit Pressure Drop. The pressure drop across the bit is determined on the rig as the difference in standpipe pressure when the bit is on bottom, and when the bit is off bottom, while maintaining constant flowrate.

Maximum Drilling Rate. In fast drilling operations (soft formations), the maximum penetration rate is limited by the maximum pressure available at the bit. This is the maximum allowable standpipe pressure minus the total losses in the circulating system.

Optimum Pump Output. In harder formations where drilling rates are limited by maximum available bit weight and rotary speed, the optimum value of

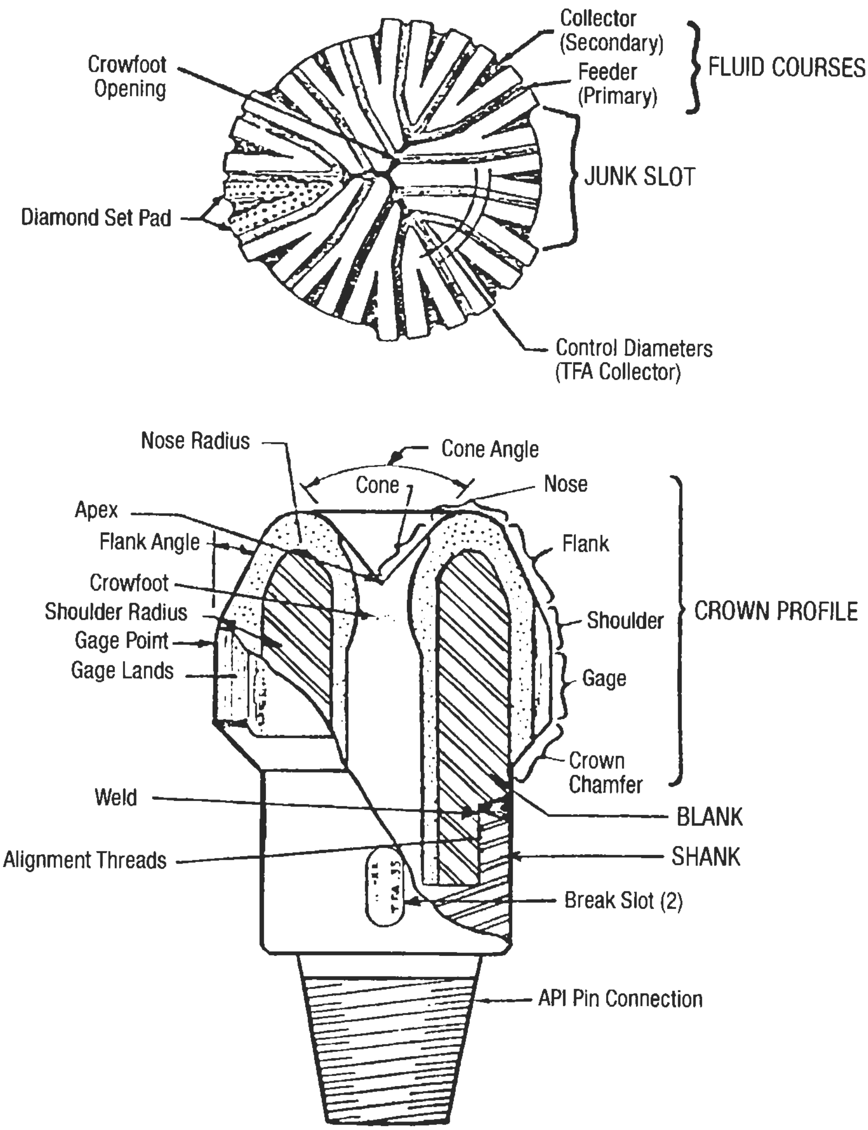


Figure 4-144. Diamond drill bit nomenclature [49]. *(Courtesy Hughes Christensen.)*

flowrate should be adjusted to achieve the bit hydraulic horsepower required. The minimum pump discharge required to maintain annular velocity and bit cooling is shown in Figure 4-146.

Hydraulic Pumpoff. The bit pressure drop acts over the bit face area between the cutting face of the bit and the formation and tends to lift the bit off the

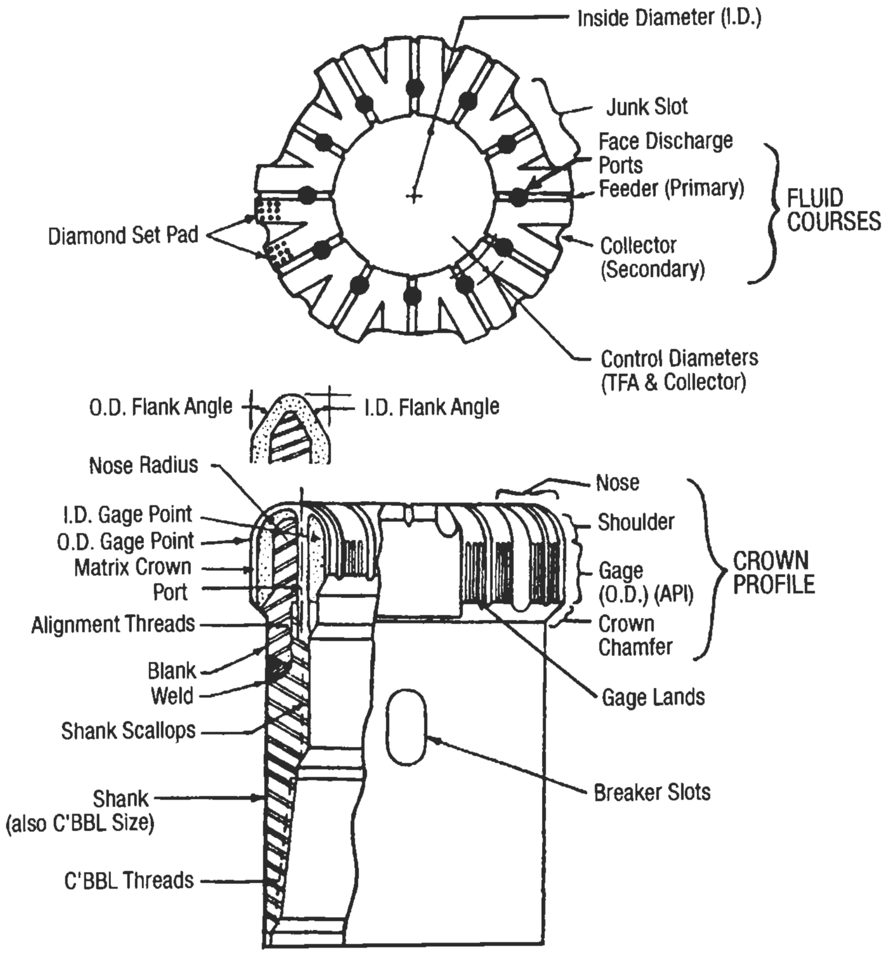


Figure 4-145. Diamond core bit nomenclature [50]. *(Courtesy Hughes Christensen.)*

Table 4-99
Bottomhole Hydraulic Horsepower Required for Diamond Drilling [49]

Penetration Rate, ft/hr	1–2	2–4	4–6	6–10	over 10
Hydraulic Horsepower Required, HP_h;/sq. inch	1–1.5	1.5–2	2–2.5	2.5–3	3–3.5

bottom of the hole. This force is large at the higher bit hydraulic horsepower being utilized today and in some cases may require additional bit weight to compensate. For example, the pumpoff force on an $8\frac{7}{16}$-in. diamond bit having a pressure drop across the bit of 900 psi would be about 6,000 lb.

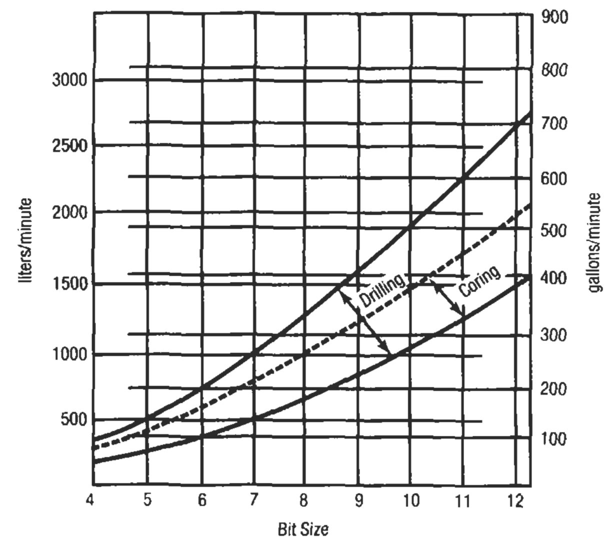


Figure 4-146. Pump discharge for diamond bits [50]. *(Courtesy Hughes Christensen.)*

The hydraulic pumpoff force F_{po}(lb) can be approximated by [50]

$$F_{po} = 1.29(\Delta P_b)(d_h - 1) \tag{4-83}$$

for the radial flow watercourse design bits, and

$$F_{po} = 0.32(\Delta P_b)(d_h - 1) \tag{4-84}$$

for cross flow watercourse system (refer to IADC classification of fixed-cutter bits).

Diamond Bit Weight on Bit and Rotary Speed

Weight on Bit. Drilling weight should be increased in increments of 2,000 lb as the penetration rate increases. As long as no problems are encountered with the hydraulics and torque, weight can be added. However, when additional weight is added and the penetration rate does not increase, the bit may be balling up, and the weight on the bit should be decreased.

Rotary Speed. Diamond bits can usually be rotated at up to 150 rpm without any problem when hole conditions and drill string design permit. Rotary speeds of 200 and 300 rpm can be used with stabilized drill strings in selected areas. Diamond bits have also operated very successfully with downhole motors at 600 to 900 rpm. The actual rotary speed limits are usually imposed by safety.

Core Bits

Most core barrels utilize diamonds as the rock cutting tool. There are three types of core barrels.

Wireline Core Barrel Systems. The wireline system can be used for continuous drilling or coring operations. The inner barrel or the drill plug center of the core bit can be dropped from the surface and retrieved without pulling the entire drill string.

Marine Core Barrels. Marine barrels were developed for offshore coring where a stronger core barrel is required. They are similar to the conventional core barrels except that they have heavier outer tube walls.

Rubber Sleeve Core Barrels. Rubber sleeve core barrels are special application tools designed to recover undisturbed core in soft, unconsolidated formations. As the core is cut, it is encased in the rubber sleeve that contains and supports it. Using face discharge ports in the bit, the contamination of the core by circulating fluid is reduced. The rubber sleeve core barrel has proven to be a very effective tool, in spite of the fact that the rubber sleeve becomes weak with a tendency to split as the temperature increases about 175°F.

Core Barrel Specifications. Core barrel sizes, recommended make-up torques, maximum recommended pulls and recommended fluid capacities are shown in Tables 4-100 and 4-101 [50].

Table 4-100
Core Barrels: Recommended Makeup Values [50]

Core Barrel Size		3.5 x 1.75	4.12 x 2.12	4.50 x 2.12	4.75 x 2.62	5.75 x 3.50	6.25 x 3	6.25 x 4	6.75 x 4	8.0 x 5.25
Recommended Make up Torque foot-pounds		1,700 to 2,050	3,000 to 3,600	5,000 to 6,000	4,050 to 4,850	7,400 to 8,800	14,900 to 17,800	8,150 to 9,800	9,900 to 12,000	19,000 to 22,700
Pounds Line Pull for Different Length Tong Levers	60"	375	660	1,100	890	1,620	3,270	1,795	2,190	4,170
	55"	410	720	1,200	970	1,770	3,570	1,960	2,390	4,550
	53"	425	750	1,250	1,010	1,830	3,700	2,030	2,480	4,720
	47"	480	840	1,400	1,140	2,070	4,170	2,290	2,800	5,320
	44"	510	900	1,500	1,210	2,200	4,460	2,450	2,990	5,690
	42"	540	940	1,570	1,270	2,310	4,670	2,560	3,130	5,960
	36"	625	1,100	1,830	1,480	2,700	5,450	2,990	3,650	6,950

Courtesy Hughes Christensen

Table 4-101
Core Barrels Characteristics [50]

Size	(ft) Standard Length	# of Turns In Safety Joint	(GPM) Fluid Capacity	Recommended Maximum Pull
3-1/2 x 1-3/4	30	7	118	74,000
4-1/8 x 2-1/8	60	7	141	101,400
4-1/2 x 2-1/8	60	7	141	194,700
4-3/4 x 2-5/8	60	13	164	137,400
5-3/4 x 3-1/2	60	6	204	200,000
6-1/4 x 3	60	7	245	290,000
6-1/4 x 4	60	6	227	193,500
6-3/4 x 4	60	6	387	275,000
8 x 5-1/4	60	7	295	310,000

The Maximum Pull is based upon the ultimate tensile strength in the pin thread area with a safety factor of three.

Courtesy Hughes Christensen

Weight on Bit and Rotary Speed for Core Bits

Weight on Bit. Figure 4-147 shows the drilling weights for diamond core bits in various formations. These are average values determined in field tests [50]. The proper weight on the bit for each core run can be determined by increasing the bit weight in steps of 1,000 to 2,000 lb, with an average speed of 100 rpm. Coring should be continued at each interval while carefully observing the penetration rate. Optimum weight on the bit has been reached when additional weight does not provide any further increase in penetration rate or require excessive torque to rotate the bit. Using too much weight can cause the diamonds to penetrate too deeply into a soft formation with an insufficient amount of mud flow able to pass between the diamonds and the formation, resulting in poor removal of the cuttings. The core bit could clog or even burn, and penetration rate and bit life will be reduced. In harder formations, excessive weight will cause burning on the tips of the diamonds or shearing with a resulting loss in salvage.

Rotary Speed. The best rotational speed for coring is usually established by the limitations of the borehole and drill string. The size and number of drill collars in the string and the formation being cored must be considered when establishing the rotational speed. Figure 4-148 shows the recommended rotating speed range for optimal core recovery in different formations [49]. Concern should also be given to the harmonic vibrations of the drill string. Figure 4-149 gives critical rotary speeds [51] which generate harmonic vibrations.

Polycrystalline Diamond Compacts (PDC) Bits

PDC bits get their name from the polycrystalline diamond compacts used for their cutting structure. The technology that led to the production of STRATAPAX drill blanks grew from the General Electric Co. work with polycrystalline manufactured diamond materials for abrasives and metal working tools. General Electric Co. researched and developed the STRATAPAX (trade

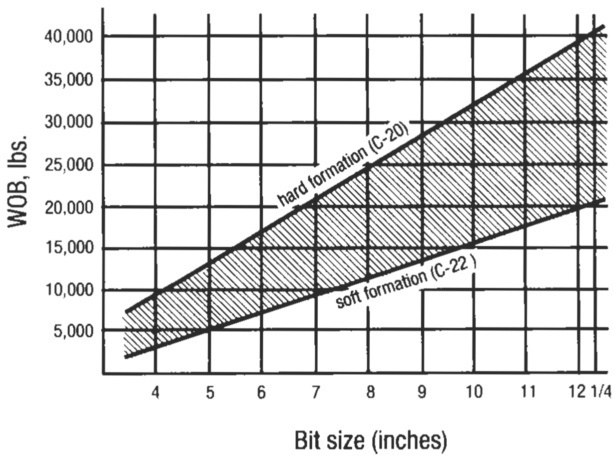


Figure 4-147. Bit weight for core bits [50]. *(Courtesy Hughes Christensen.)*

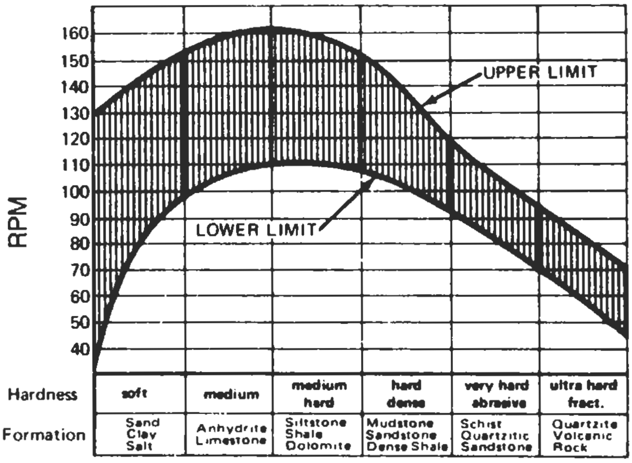


Figure 4-148. Recommended rotary speed for core bits [49]. *(Courtesy Hughes Christensen.)*

name) drill blank in 1973 and Christensen, Inc. used these in PDC bit field tests. The bits were successfully applied in offshore drilling in the North Sea area in the late 1970s and in on-shore areas in the United States in the early 1980s. In some areas, the PDC bits have out-drilling roller rock bits, reducing overall cost per foot by 30 to 50% and achieving four times the footage per bit at higher penetration rates [52,53].

Figure 4-150 shows the major components and design of the PDC bit. The polycrystalline diamond compacts, shown in Figure 4-151. The polycrystalline diamond compacts (of which General Electric's) consist of a thin layer of synthetic diamonds on a tungsten carbide disk. These compacts are produced as an integral blank by a high-pressure, high-temperature process. The diamond layer consists of many tiny crystals grown together at random orientations for maximum strength and wear resistance.

The tungsten carbide backing provides mechanical strength and further reenforces the diamond compact wear-resistant properties. During drilling, the polycrystalline diamond cutter wears down slowly with a self-sharpening effect. This helps maintain sharp cutters for high penetration-rate drilling throughout the life of the bit.

PDC Bit Design. Figures 4-152 and 4-153 show typical PDC bits. Figure 4-152 is for soft formation. Figure 4-153 is for hard and abrasive formation [43A].

Bit Body Material (Matrix). There are two common body materials for PDC bits, steel and tungsten carbide. Heat-treated steel body bits are normally a "stud" bit design, incorporating diamond compacts on tungsten carbide posts. These stud cutters are typically secured to the bit body by interference fitting and shrink fitting. Steel body bits also generally incorporate three or more carbide nozzles (often interchangeable) and carbide buttons on gauge. Steel body bits have limitations of erosion of the bit face by the drilling mud and wear of the gauge section. Some steel body bits are offered with wear-resistant coatings applied to the bit face to limit mud erosion.

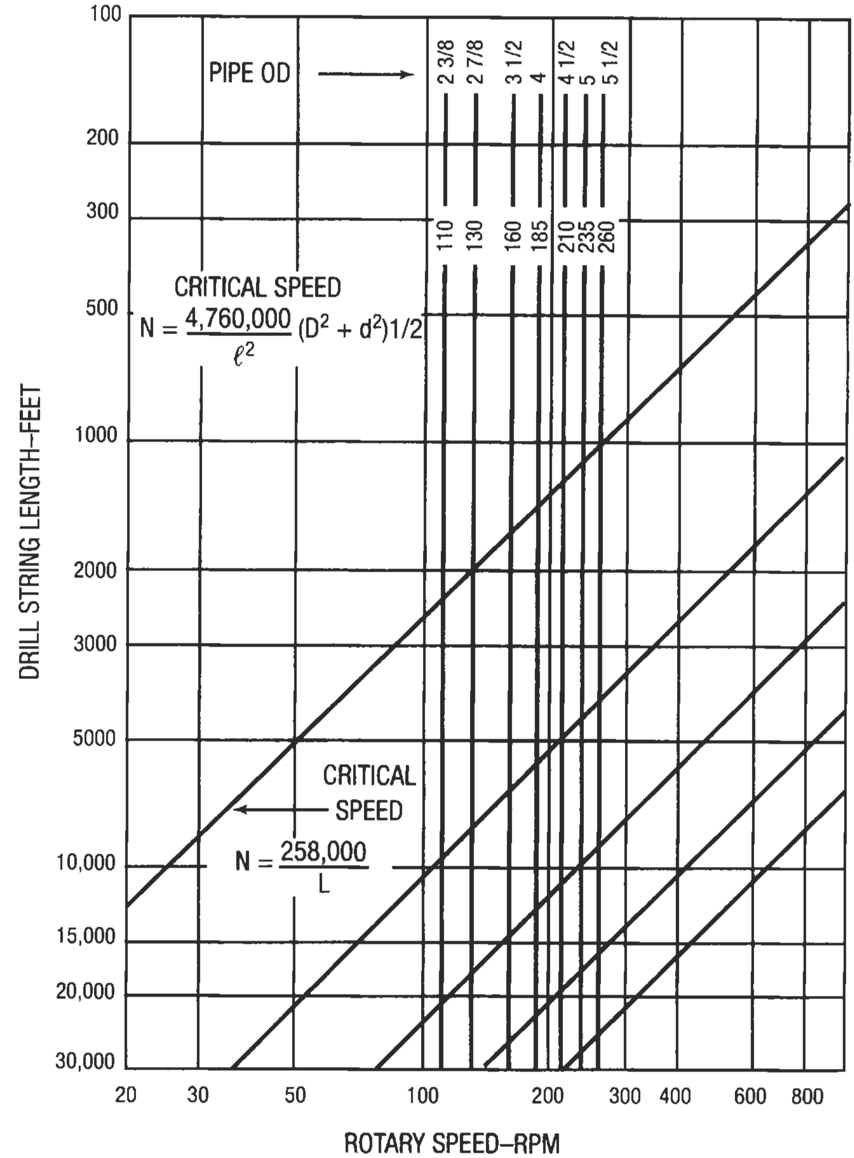


Figure 4-149. Critical rotary speed for core bits [51]. *(Courtesy API.)*

Greater bit design freedom is generally available with matrix body bits because they are "cast" in a moldlike natural diamond bits. Thus, matrix body bits typically have more complex profiles and incorporate cast nozzles and waterways. In addition to the advantages of bit face configuration and erosion resistance with matrix body bits, diamond compact matrix bits often utilize natural

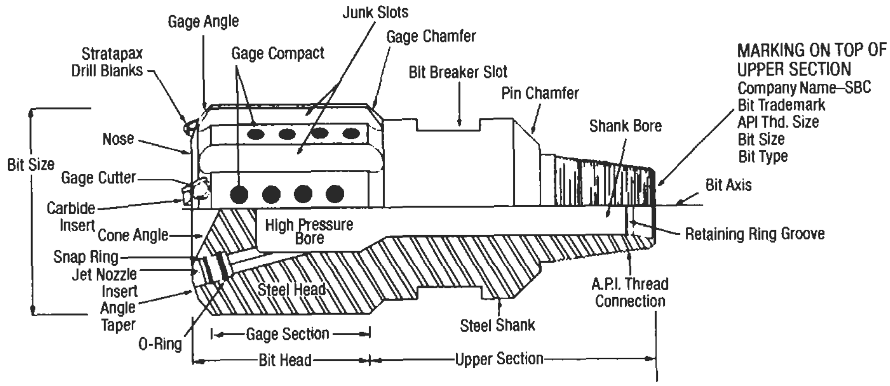


Figure 4-150. PDC bit nomenclature. *(Courtesy Strata Bit Corp.)*
Source: Strata Bit Corporation, 600 Kenrick, Suite A-1, Houston, TX 77060 ph (713) 999-4530. unknown booklet of the company.

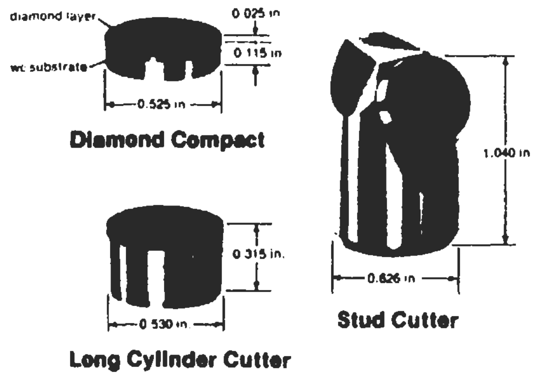


Figure 4-151. Polycrystalline diamond compacts [43A]. *(Courtesy Hughes Christensen.)*

diamonds to maintain full gage hole. Matrix body bits generally utilize long cylinder-shaped cutters secured to the bit by brazing.

Bit Profile. Bit profile can significantly affect bit performance based upon the influence it has on bit cleaning, stability and hole deviation control. The "double-cone" profile will help maintain a straight hole even in crooked hole country. The sharp nose will attack and drill the formation aggressively while the apex and reaming flank stabilize the bit. This sharp profile may be more vulnerable to damage when a hard stringer is encountered as only the cutters on the sharp nose will support the impact loading. A shallow cone profile appears to be the easiest to clean due to the concentration of hydraulics on the reduced surface area of the bit face. This profile relies heavily upon the gage section for directional stability. The shallow cone profile will hold direction and angle with sufficient gauge length and proper stabilization of the bit.

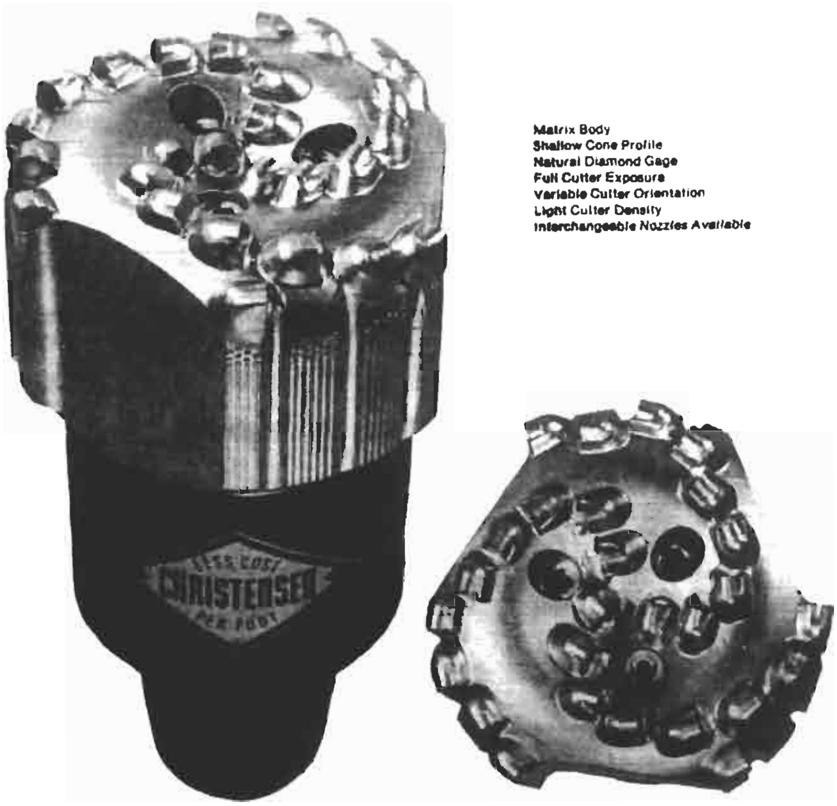


Figure 4-152. PDC bit designed for soft formations [43A]. *(Courtesy Hughes Christensen.)*

Cutter Exposure. Figure 4-154 shows the types of cutter exposure on PDC bits [43]. Cutter exposure is the distance between the cutting edge and the bit face. Stud bits typically have full exposure that proves very aggressive in soft formation. In harder formations, less than full exposure may be preferred for added cutter durability and enhanced cleaning. Matrix body bits are designed with full or partial exposure depending on formation and operating parameters.

Cutter Orientation. Figure 4-155 shows the cutter orientation for typical PDC bits. The displacement of cuttings can be affected by side and back rake orientation of the cutters. Back rake angle typically varies from 0 to –25°. The greater the degree of back rake, generally the lower the rate of penetration, but the greater the resistance to cutting edge damage when encountering a hard section. Side rake has been found to be effective in assisting bit cleaning in some formations by mechanically directing cuttings toward the annulus. Matrix body bits allow greater flexibility in adjusting cutter orientation for best drilling performance in each formation.

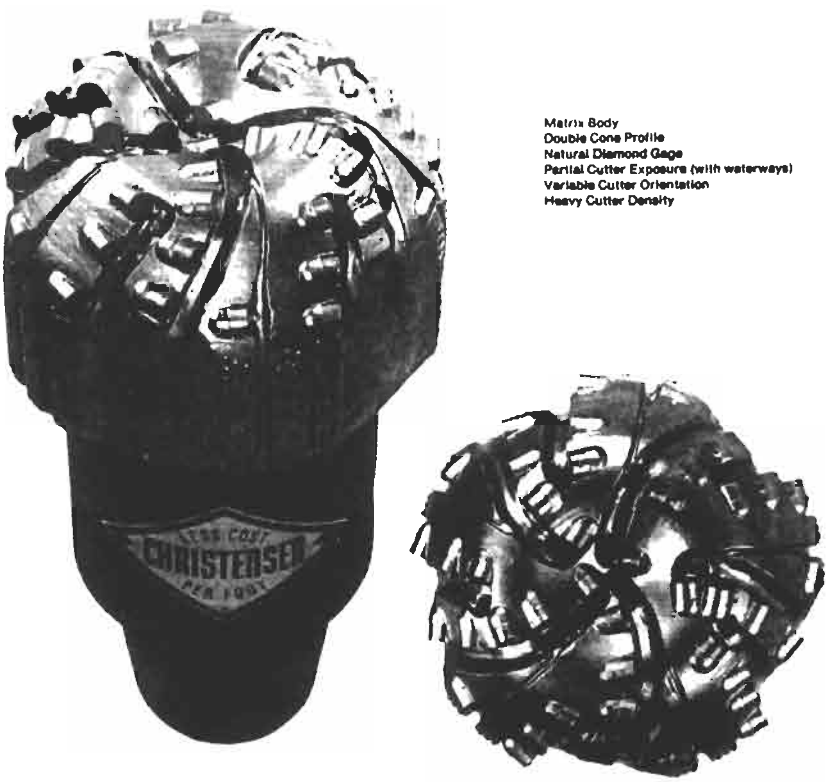


Figure 4-153. PDC bit designed for hard and abrasive formations [43A]. *(Courtesy Hughes Christensen.)*

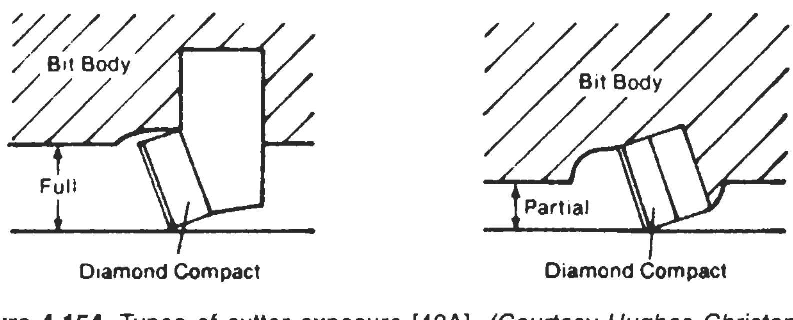


Figure 4-154. Types of cutter exposure [43A]. *(Courtesy Hughes Christensen.)*

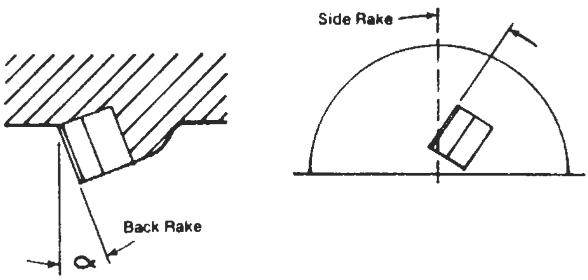


Figure 4-155. Cutter orientation [43A]. (*Courtesy Hughes Christensen.*)

IADC Fixed Cutter Bit Classification System

The term fixed cutter is used as the most correct description for the broad category of nonroller cone rock bits. The cutting elements may be comprised of any suitable material. To date, several types of diamond materials are used almost exclusively for fixed cutter petroleum drilling applications. This leads to the widespread use of the term "diamond" bits and PDC bits in reference to fixed cutter designs.

The IADC Drill Bits Subcommittee began work on a new classification method in 1985. It was determined from the outset that (1) a completely new approach was required, (2) the method must be simple enough to gain widespread acceptance and uniform application, yet provide sufficient detail to be useful, (3) emphasis should be placed on describing the form of the bit, i.e., "paint a mental picture of the design", (4) no attempt should be made to describe the function of the bit, i.e., do not link the bit to a particular formation type or drilling technique since relatively little is certain yet about such factors for fixed cutter bits, (5) every bit should have a unique IADC code, and (6) the classification system should be so versatile that it will not be readily obsolete.

The resultant four-character diamond bit classification code was formally presented to the IADC Drilling Technology Committee at the 1986 SPE/IADC Drilling Conference. It was subsequently approved by the IADC Board of Directors and designated to take effect concurrent with the 1987 SPE/IADC Drilling Conference. A description of the 1987 IADC Fixed Cutter Bit Classification Standard follows [54].

Four characters are utilized in a prescribed order (Figure 4-156) to indicate seven fixed cutter bit design features: cutter type, body material, bit profile, fluid discharge, flow distribution, cutter size, and cutter density. These design traits were selected as being most descriptive of fixed cutter bit appearance.

The four-character bit code is entered on an IADC-API Daily Drilling Report Form as shown in Figure 4-157. The space requirements are consistent with the four-character IADC roller bit classification code. The two codes are readily distinguished from one another by the convention that diamond bit codes begin with a letter, while roller bit codes begin with a number.

Each of the four characters in the IADC fixed cutter bit classification code are further described as follows:

Cutter Type and Body Material. The first character of the fixed cutter classification code describes the primary cutter type and body material (Figure 4-156).

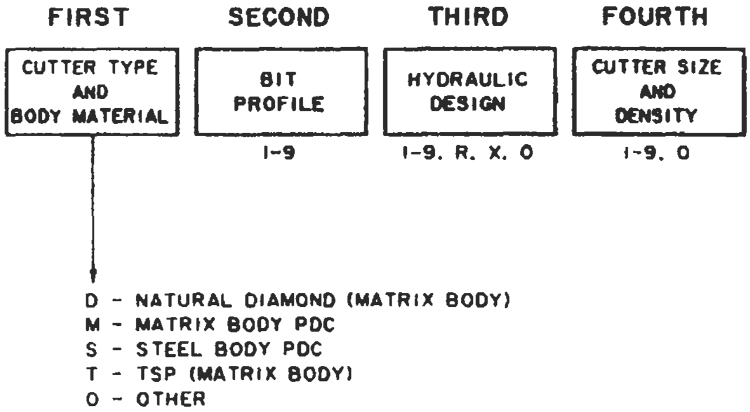
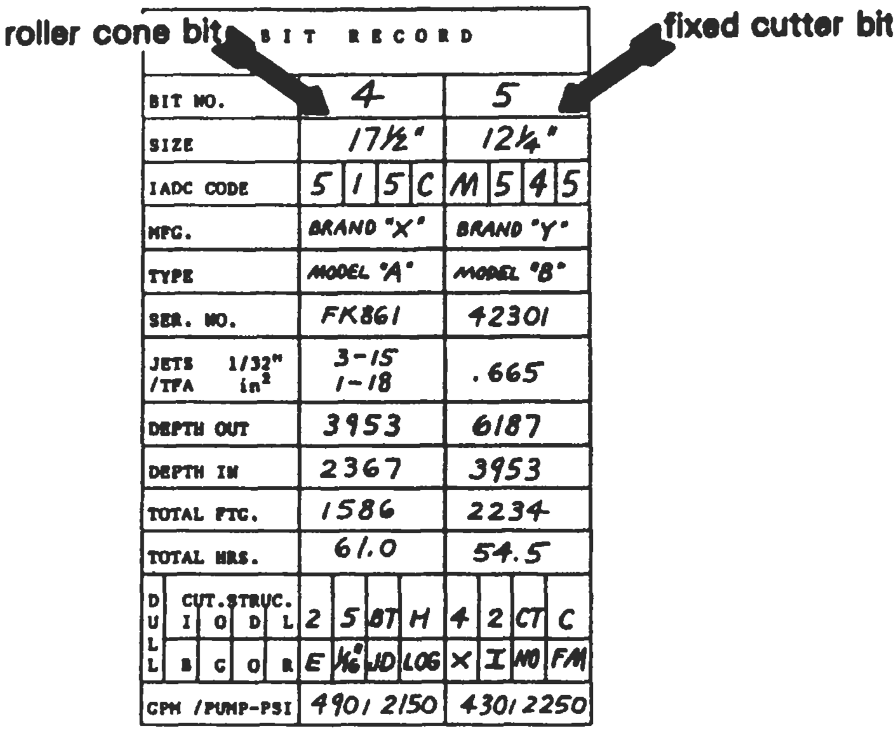


Figure 4-156. Four-character classification code for fixed-cutter bits [54]. *(Courtesy SPE.)*



BIT RECORD		
BIT NO.	4	5
SIZE	17½"	12¼"
IADC CODE	5 1 5 C	M 5 4 5
MFG.	BRAND "X"	BRAND "Y"
TYPE	MODEL "A"	MODEL "B"
SER. NO.	FK861	42301
JETS 1/32" /TFA in²	3-15 1-18	.665
DEPTH OUT	3953	6187
DEPTH IN	2367	3953
TOTAL FTG.	1586	2234
TOTAL HRS.	61.0	54.5
DULL CUT. STRUC. I O D L	2 5 BT H	4 2 CT C
DULL B G O R	E 1/16 JD LOG	X I NO FM
CPM /PUMP-PSI	490/2150	430/2250

Figure 4-157. Fixed-cutter bit code entry in IADC-API Daily Report [54]. *(Courtesy SPE.)*

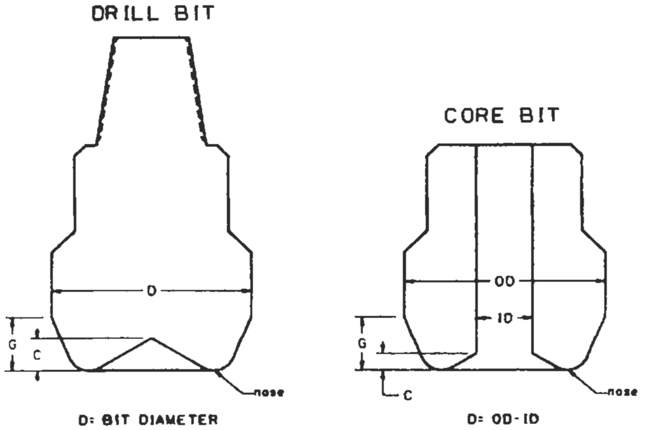
Five letters are presently defined: D—natural diamond/matrix body, M—PDC/matrix body, S—PDC/steel body, T—TSP/matrix body, O—other.

The term PDC is defined as "polycrystalline diamond compact." The term TSP is defined as "thermally stable polycrystalline" diamond. TSP materials are composed of manufactured polycrystalline diamond which has the thermal stability of natural diamond. This is accomplished through the removal of trace impurities and in some cases the filling of lattice structure pore spaces with a material of compatible thermal expansion coefficient.

The distinction of *primary* cutter types is made because fixed cutter bits often contain a variety of diamond materials. Typically one type of diamond is used as the primary cutting element while another type is used as backup material.

Profile. The numbers 1 through 9 in the second character of the fixed cutter classification code refer to the bit's cross-sectional profile (Figure 4-158). The

BIT PROFILE CODES



G-GAGE HEIGHT	C-CONE HEIGHT		
	HIGH C > 1/4D	MEDIUM 1/8D ≤ C ≤ 1/4D	LOW C < 1/8D
HIGH G > 3/8D	1	2	3
MED 1/8D ≤ G ≤ 3/8D	4	5	6
LOW G < 1/8D	7	8	9

Figure 4-158. Bit profile codes for fixed cutter bits [54]. *(Courtesy SPE.)*

term profile is used here to describe the cross-section of the cutter/bottomhole pattern. This distinction is made because the cutter/bottomhole profile is not necessarily identical to the bit body profile.

Nine basic bit profiles are defined by arranging two profile parameters—outer taper (gage height) and inner concavity (cone height)—in a 3 × 3 matrix (Figure 4-159). The rows and columns of the matrix are assigned high, medium and low values for each parameter. Gage height systematically decreases from top to bottom. Cone height systematically decreases from left to right. Each profile is assigned a number.

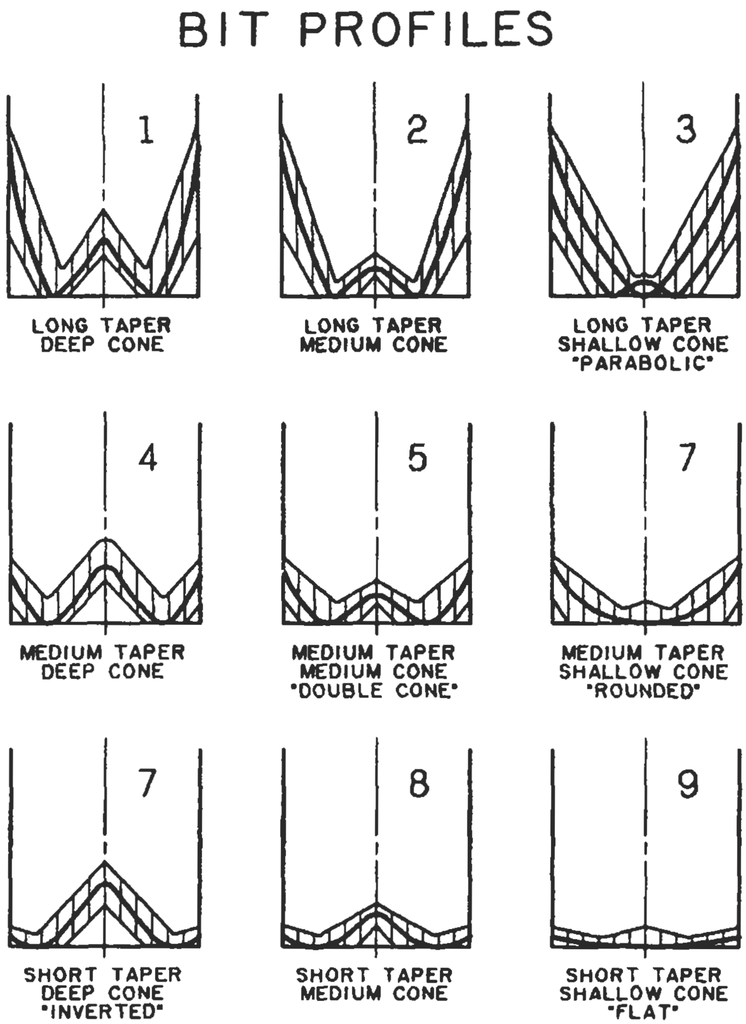


Figure 4-159. Nine basic profiles of fixed-cutter bits [54]. *(Courtesy SPE.)*

Two versions of the profile matrix are presented. One version (Figure 4-158) is primarily for the use of manufacturers in classifying their bit profiles. Precise ranges of high, medium and low values are given. In Figure 4-158 gage height and cone height dimensions are normalized to a reference dimension which is taken to be the bit diameter for drill bits and the (OD-ID) for core bits. Figure 4-159 provides a visual reference which is better suited for use by field personnel. Bold lines are drawn as examples of typical bit profiles in each category. Cross-hatched areas represent the range of variation for each category. Each of the nine profiles is given a name. For example, "double cone" is the term used to describe the profile in the center of the matrix (code 5). The double-cone profile is typical of many natural diamond and TSP bits.

The number 0 is used for unusual bit profiles which cannot be described by the 3 × 3 matrix of Figure 4-158. For example, a "bi-center" bit which has an asymmetrical profile with respect to the bit pin centerline should be classified with the numeral 0.

Hydraulic Design. The numbers 1 through 9 in the third character of the fixed cutter classification code refer to the hydraulic design of the bit (Figure 4-160).

HYDRAULIC DESIGN

	CHANGEABLE JETS	FIXED PORTS	OPEN THROAT
BLADED	1	2	3
RIBBED	4	5	6
OPEN FACED	7	8	9

ALTERNATE CODES

R - RADIAL FLOW

X - CROSS FLOW

O - OTHER

Figure 4-160. Hydraulic design code for fixed-cutter bits [54]. *(Courtesy SPE.)*

The hydraulic design is described by two components: the type of fluid outlet and the flow distribution. A 3 × 3 matrix of orifice types and flow distributions defines 9 numeric hydraulic design codes. The orifice type varies from changeable jets to fixed ports to open throat from left to right in the matrix. The flow distribution varies from bladed to ribbed to open face from top to bottom. There is usually a close correlation between the flow distribution and the cutter arrangement.

The term *bladed* refers to raised, continuous flow restrictors with a standoff distance from the bit body of more than 1.0 in. In most cases cutters are affixed to the blades so that the cutter arrangement may also be described as bladed. The term *ribbed* refers to raised continuous flow restrictors with a standoff distance from the bit body of 1.0 in. or less. Cutters are usually affixed to most of the ribs so that the cutter arrangement may also be described as ribbed. The term *open face* refers to nonrestricted flow arrangements. Open face flow designs generally have a more even distribution of cutters over the bit face than with bladed or ribbed designs.

A special case is defined: the numbers 6 and 9 describe the crowfoot/water course design of most natural diamond and many TSP bits. Such designs are further described as having either radial flow, crossflow (feeder/collector), or other hydraulics. Thus, the letters R (radial flow), X (crossflow), or O (other) are used as the hydraulic design code for such bits.

Cutter Size and Placement Density. The numbers 1 through 9 and 0 in the fourth character of the fixed cutter classification code refer to the cutter size and placement density on the bit (Figure 4-161). A 3 × 3 matrix of cutter sizes and placement densities defines 9 numeric codes. The placement density varies from light to medium to heavy from left to right in the matrix. The cutter size varies from large to medium to small from top to bottom. The ultimate combination of small cutters set in a high density pattern is the impregnated bit, designated by the number 0.

Cutter size ranges are defined for natural diamonds based on the number of stones per carat. PDC and TSP cutter sizes are defined based on the amount of usable cutter height. Usable cutter height rather than total cutter height is the functional measure since various anchoring and attachment methods affect the "exposure" of the cutting structure. The most common type of PDC cutters, which have a diameter that is slightly more than $\frac{1}{2}$ in., were taken as the basis for defining medium size synthetic diamond cutters.

Cutter density ranges are not explicitly defined. The appropriate designation is left to the judgment of the manufacturer. In many cases manufacturers build "light-set" and "heavy-set" versions of a standard product. These can be distinguished by use of the light, medium, or heavy designation which is encoded in the fourth character of the IADC fixed cutter bit code. As a general guide, bits with minimal cutter redundancy are classified as having light placement density and those with high cutter redundancy are classified as having heavy placement density.

Examples of Fixed-Cutter Bits Classification

Figure 4-162 shows a natural diamond drill bit which has a long outer taper and medium inner cone, radial flow fluid courses, and five to six stones per carat (spc) diamonds set with a medium placement density. Using the definitions in Figures 4-156, 4-158, 4-159, and 4-160, the characteristics of this bit are coded D 2 R 5 as follows:

CUTTER SIZE AND DENSITY

SIZE \ DENSITY	light	medium	heavy
large	1	2	3
medium	4	5	6
small	7	8	9

0 – impregnated

CUTTER SIZE RANGES	NATURAL DIAMONDS stones per carat	SYNTHETIC DIAMONDS usable cutter height
large	< 3	> 5/8"
medium	3–7	3/8–5/8"
small	> 7	< 3/8"

CUTTER DENSITY IS DETERMINED BY MANUFACTURER

Figure 4-161. Code of cutter size and placement [54]. *(Courtesy SPE.)*

Cutter/Body Type	D—natural diamond, matrix body
Bit Profile	2—long taper, medium cone
Hydraulic Design	R—open throat/open face radial flow
Cutter Size/Density	5—med. cutter size, med. placement density

Figure 4-163 shows a steel body PDC bit with standard-size cutters lightly set on a deep inner cone profile. This bit has changeable nozzles and is best described as having a ribbed flow pattern although there are open face characteristics near the center and bladed characteristics near the gage. The IADC classification code in this case is S 7 4 4.

Figure 4-164 shows a steel body core bit with a long-taper, stepped profile fitted with impregnated natural diamond blocks as the primary cutting elements. The bit has no inner cone. Since there is no specific code for the natural diamond/steel body combination, the letter O (other) is used as the cutter type/body material code. The profile code 3 is used to describe the long outer taper with little or no inner cone depth. The hydraulic design code 5 indicates a fixed

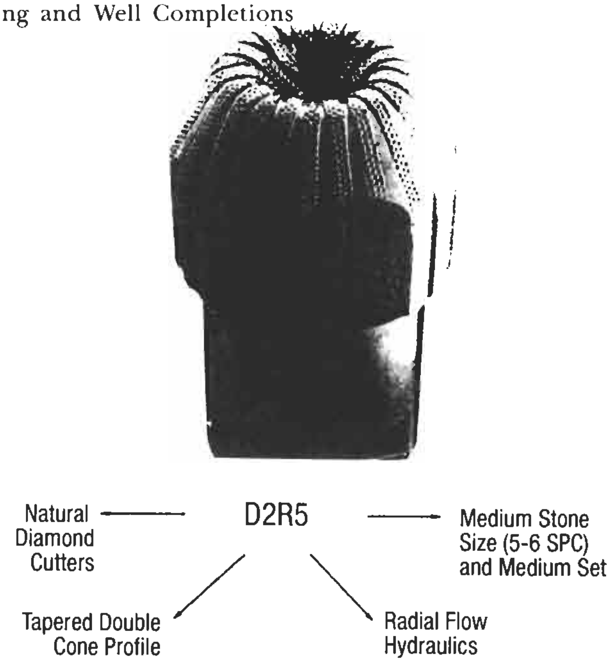


Figure 4-162. Example of natural diamond bit with radial flow hydraulic design [54]. *(Courtesy SPE.)*

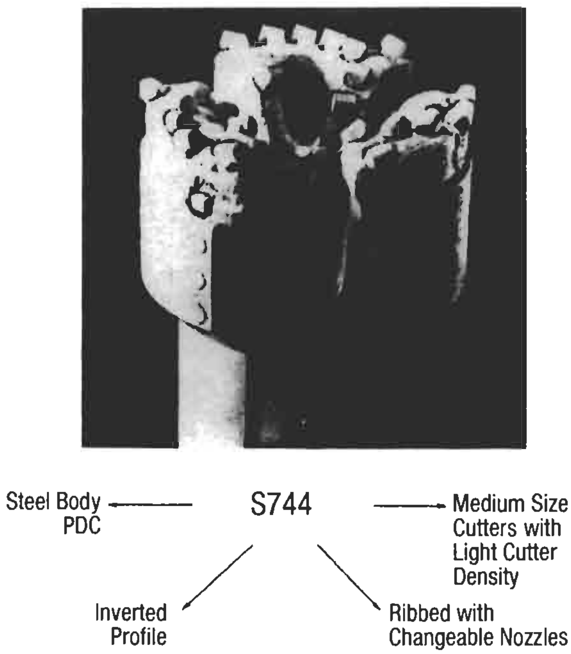


Figure 4-163. Example of steel body PDC bit with inverted profile [54]. *(Courtesy SPE.)*

Figure 4-164. Example of steel body impregnated core bit with face discharge flow [54]. *(Courtesy SPE.)*

port, ribbed design. Finally, the number 0 is used for impregnated natural diamond bits. Therefore the complete IADC classification code for this fixed cutter bit is 0 3 5 0. Although the classification code for this bit does not explicitly indicate the cutter type and body material, it can be inferred from the rest of the code that this is an impregnated natural diamond, nonmatrix body bit, in which case steel is the most likely body material.

Dull Grading for Fixed Cutter Bits

The section describes the first IADC standardized system for dull grading natural diamond, PDC, and TSP (thermally stable polycrystalline diamond) bits, otherwise known as fixed cutter bits [55]. The new system is consistent with the recently revised dull grading system for roller bits. It describes the condition of the cutting structure, the primary (with location) and secondary dull characteristics, the gage condition, and the reason the bit was pulled.

The format of the dull grading system is shown in Figure 4-165. For completeness, Figure 4-165 contains all of the codes needed to dull grade fixed cutter bits and roller bits. Those codes which apply to fixed cutter bits are in boldface.

Eight factors about a worn fixed cutter bit can be recorded. The first four spaces are used to describe the cutting structure. In the first two spaces, the amount of cutting structure wear is recorded using the linear scale 0 to 8, based on the initial useable cutter height. This is consistent with grading tooth wear

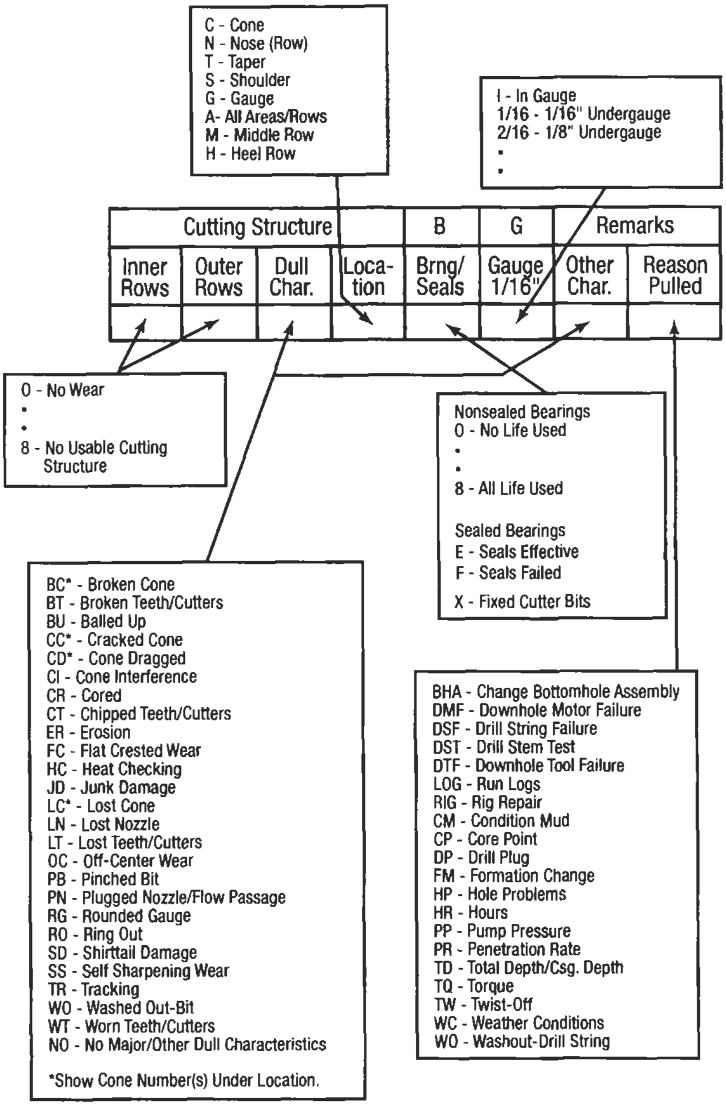


Figure 4-165. IADC bit dull grading codes—bold characters for fixed-cutter bits [55]. *(Courtesy SPE.)*

on roller bits. The amount of cutter wear represented by 0 through 7 is shown schematically in Figure 4-166. An 8 means there is no cutter left. This same scale is to be used for TSP and natural diamond bits, with 0 meaning no wear, 4 meaning 50% wear, and so forth.

The first two spaces of the dull grading format are used for the inner two-thirds of the bit radius and the outer one-third of the bit radius, as shown

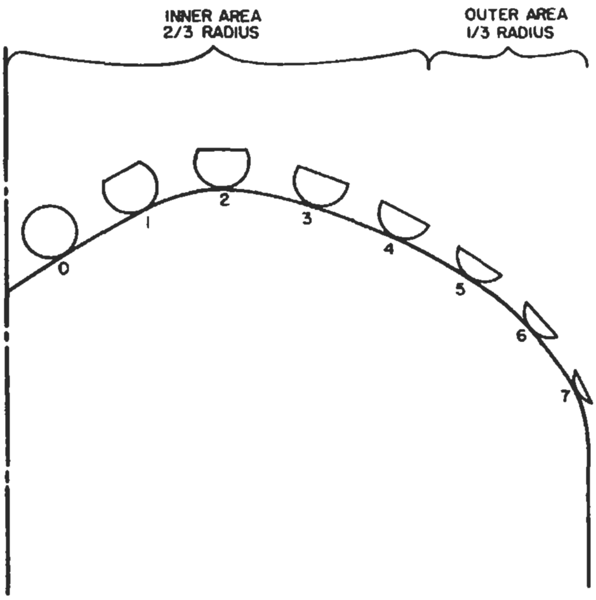


Figure 4-166. Schematics of cutters wear [55]. *(Courtesy SPE.)*

schematically in Figure 4-166. When grading a dull bit, the average amount of wear in each area should be recorded. For example, in Figure 4-166 the five cutters in the inner area would be graded a 2. This is calculated by averaging the grades of the individual cutters in the inner area as follows: (4+3+2+1+0)/5=2. Similarly, the grade of the outer area would be a 6. On an actual bit the same procedure would be used. Note that for a core bit, the centerline in Figure 4-166 would be the core bit ID.

The third space is used to describe the primary dull characteristic of the worn bit, i.e., the obvious physical change from its new condition. The dull characteristics which apply to fixed cutter bits are listed in Figure 4-165.

The location of the primary dull characteristic is described in the fourth space. There are six choices: cone, nose, taper, shoulder, gauge, and all areas. Figure 4-167 shows four possible fixed cutter bit profiles with the different areas labeled. It is recognized that there are profiles for which the exact boundaries between areas are debatable and for which certain areas may not even exist. Notice that in the bottom profile there is no taper area shown. However, using Figure 4-167 as a guide, it should be possible to clearly define the different areas on most profiles.

The fifth space will always be an "X" for fixed cutter bits, since there are no bearings. This space can be used to distinguish dull grades for fixed cutter bits from dull grades for roller bits.

The measure of the bit gauge is recorded in the sixth space. If the bit is still in gauge, an "I" is used. Otherwise, the amount the bit is undergauge is noted to the nearest $\frac{1}{16}$ of an inch.

The seventh space is for the secondary dull characteristic of the bit, using the same list of codes as was used for the primary dull characteristic. The reason

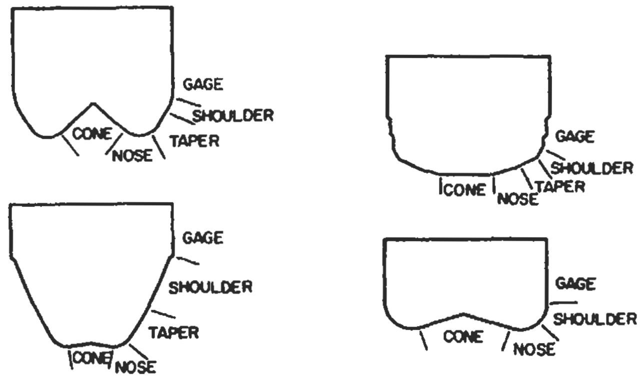


Figure 4-167. Locations of wear on fixed-cutter bit [55]. *(Courtesy SPE.)*

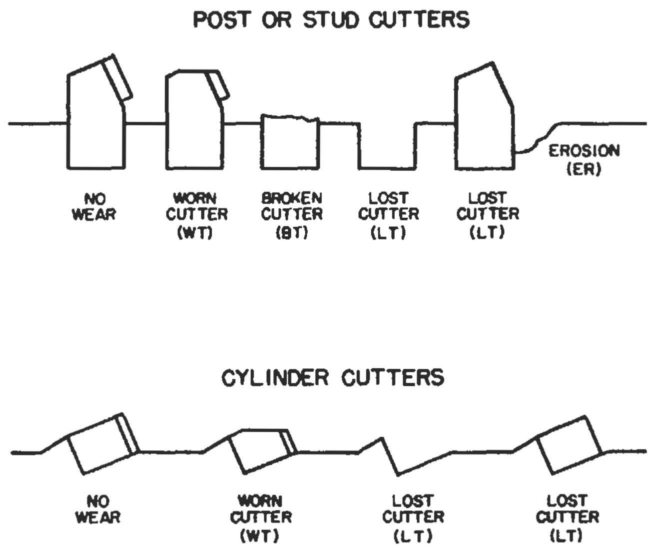


Figure 4-168. Schematic of common dull characteristics [55]. *(Courtesy SPE.)*

the bit was pulled is shown in the eighth space using the list of codes shown in Figure 4-165.

Downhole Tools

Downhole drilling tools are the components of the lower part of a drill string used in normal drilling operations such as the drill bits, drill collars, stabilizers, shock absorbers, hole openers, underreamers, drilling jars as well as a variety of drill stem subs.

As drill bits, drill collars and drill stem subs are discussed elsewhere this section regards shock absorbers, jars, underreamers, and stabilizers.

Shock Absorbers

Extreme vertical vibration throughout the drill string are caused by hard, broken or changing formations, and the drilling bit chafing against the bottom formation as it rotates.

In shallow wells, the drill string transmits the vibration oscillations all the way to the crown block of the drilling rig. The affect can be devastating as welds fail, seams split and drill string connections break down under the accelerated fatigue caused by the vibrations. In deep holes, these vibrations are rarely noticed due to elasticity and self-dampening effect of the long drill string. Unfortunately, the danger of fatigue still goes on and has resulted in many fishing operations.

The drill string vibration dampeners are used to absorb and transfer the shock of drilling to the drill collars where it can be borne without damaging or destroying other drill string equipment. Their construction and design vary with each manufacturer. To effectively absorb the vibrations induced by the drill bit, an element with a soft spring action and good dampening characteristics is required. There are six basic spring elements used: (1) vulcanized elastometer, (2) elastomeric element, (3) steel wool, (4) spring steel, (5) Belleville steel springs, (6) gas compression.

Types of Shock Absorbers. There are eight commonly used commercial shock absorbers.

Drilco Rubber Type. See Figure 4-169 and Table 4-102 [56]. Shock is absorbed by an elastometer situated between the inner and outer barrels. This shock absorbing element is vulcanized to the barrels. The torque has to be transmitted from the outer into the inner barrel. This tool is able to absorb shocks in axial or in radial directions. There is no need to absorb shocks in the torque because the drill string itself acts like a very good shock absorber so the critical shocks are in axial directions. These tools cannot be used at temperatures above 200°F. Though they produce a small stroke the dampening effect is good [56].

Christensen Shock-Eze. See Figure 4-170 [57]. A double-action vibration and shock absorber employing Belleville spring elements are immersed in oil.

The tool features a spline assembly that transmits high torque loads to the bit through its outer tube, while the inner assembly absorbs vibration through a series of steel-disc springs. The spring system works in both suspension and compression.

The high shock-absorbing capabilities of this tool are attained by compression of the stack of springs within a stroke of five inches. The alternating action of the patented spring arrangement provides a wide working range, under all possible conditions of thrust and mud pressure drop.

Placement of Shock Absorber in Drill String. Many operators have their own way of placing shock absorbers in the drill string (see Figure 4-171) [57]. In general, the optimum shock absorbing effect is obtained by running the tool as near to the bit as possible. With no deviation expected, the tool should be installed immediately above the bit stabilizer as shown in Figure 4-171C.

In holes with slight deviation problems, the shock absorber could be run on top of the first or second string stabilizer. For situations where there are severe

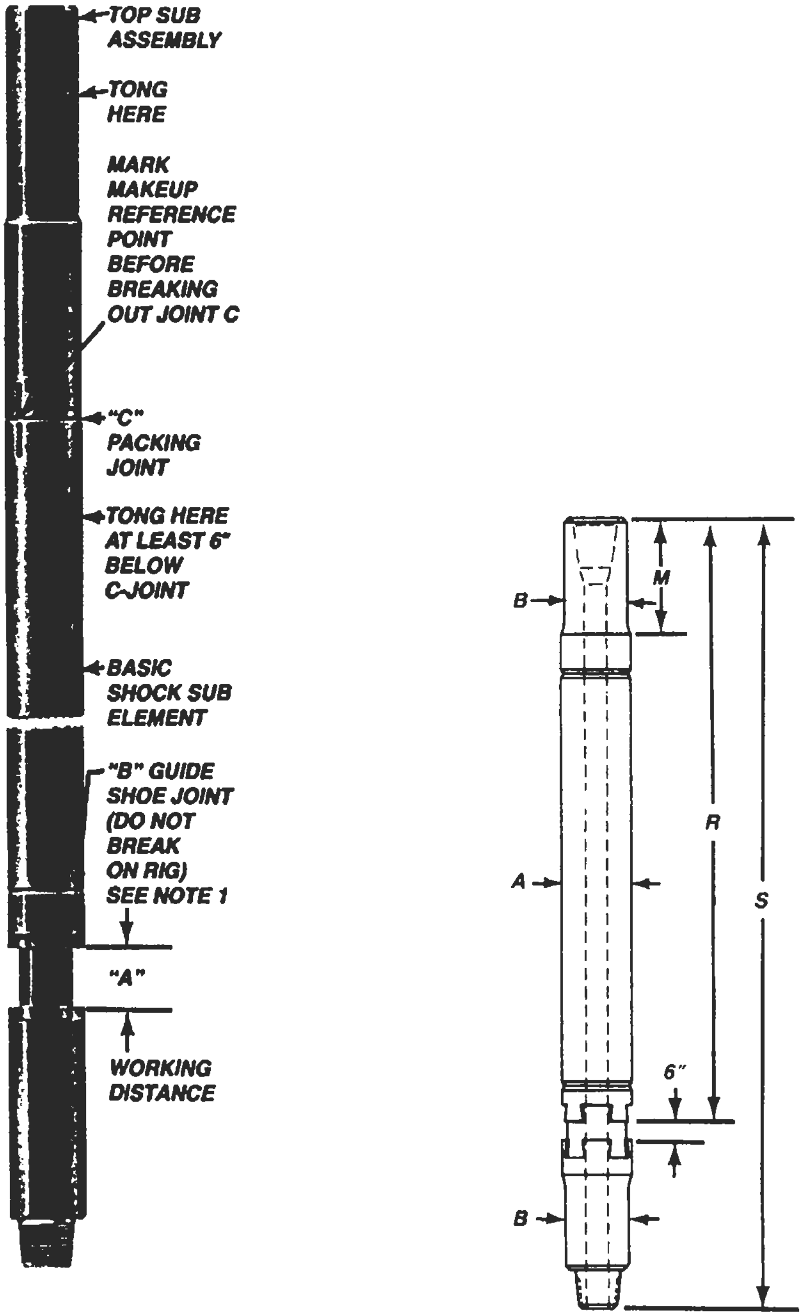


Figure 4-169. Drilco rubber-spring shock dampener. *(Courtesy Smith International, Inc.)*

Table 4-102
Drilco Rubber Spring Shock Dampener Model "E"*

Nominal Size Tool (A)	*Suggested Hole Size Recommended for Best Performance	Body Specifications				Misc. Lengths			Approx. Complete Weight (lb)	C-Joint Center Packing Joint Makeup Torque (ft-lb)
		Top Sub Bore	Main Body Bore	End Conn.	OD (B)	S	M	R		
12	17½ thru 30	2 13/16	2 13/16	Specify	To Be	160	38	126	3070	70,000
10	12¼ thru 15	2 13/16	2¼		Specified	148	23	116	2050	55,000
9	9⅝ thru 12¼	2 13/16	2¼	Size and	Same OD	149	23	116	1635	41,000
8	8½ thru 11	2 13/16	2¼		As Drill	145	23	112	1400	35,000
7	7⅞ thru 9	2¼	2	Type	Collar	143	23	111	1100	27,000
6¼	6¾ thru 9	2¼	1½			144	23	111	890	20,000

Note: 1. All dimensions are given in inches, unless otherwise stated.
*2 Recommended for optimum tool life.

*Courtesy Smith International, Inc.

deviation problems, the shock absorber should be place as shown in Figures 4-171B and 4-171D.

For turbine drilling, it is recommended that the shock absorber be placed on top of the first stabilizer above the turbine as in Figure 4-171A.

Jars

Jars provide an upward or downward shock (or jar) to the entire drill string. Early attempts to recover stuck drill pipe motivated the development of jars.

Types of Jars. There are two general classes of jars: fishing jars and drilling jars.

A fishing jar is used to free stuck drill string, and is added to the drill string only when the string becomes stuck.

The drilling jar is used as a part of the drill string to work any time it is needed. With modern drilling requiring more safety and less cost per foot, it has become more economical to use drilling jars. In areas where possible sticking conditions exist, the drilling jar is ready to free a stuck pipe through calculated string over-pull or slack-off. The jars are used immediately when the string becomes stuck, which prevents excessive downtime and costly tripping. Unlike the fishing jar, the drilling jar has the additional function of transmitting the high amount of drilling torque to the bit.

Drilling Jar Design. The jarring and bumping characteristic of drilling jars is determined by their specific type of release elements and stroke. There are three basic types of release elements: (1) hydraulic, (2) mechanical, and (3) a combination of hydraulic and mechanical. Hydraulic mechanism employs a sleeve, or valve that is pulled through a restricted area that allows only a small amount of hydraulic oil to pass through. Once the sleeve, or valve passes through the restricted area and enters the larger chamber, it is free to travel upwards until reaching the anvil that creates a sudden stop and sends a shock throughout the string.

Mechanical mechanisms involve the following types of dynamic action:

1. Adjustable spring pressure against locking system.
2. T-slots system. A combination of upward overpull or string weight, and right-hand torque is required. When torque is released, shots disengage for jarring.

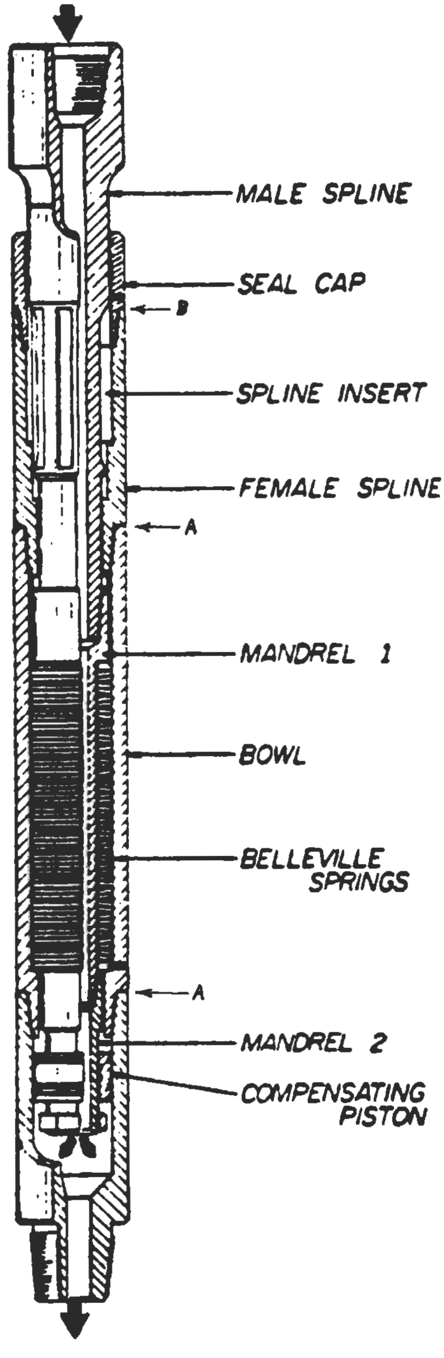


Figure 4-170. Christensen's Shock Eze. *(Courtesy Hughes Christensen.)*

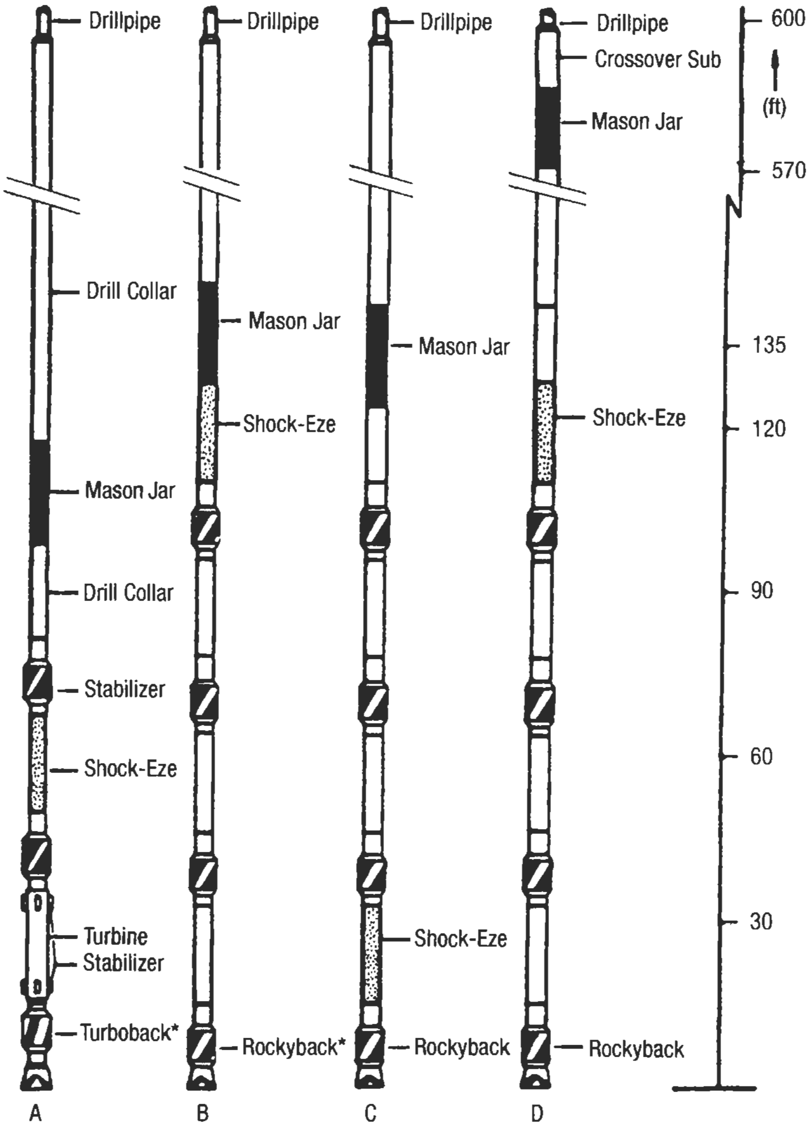


Figure 4-171. Recommended placement of the Shock Eze. *(Courtesy Hughes Christensen.)*

3. J-slots system. The slots will roll out with enough overpull or string weight. Sets of springs apply lateral pressure holding mandrel in place.
4. Firing racks system. A lateral pressure is applied by adjusting racks. Angle on firing racks give it a 2:1 release factor when overpull or string weight is applied.

Types of Drilling Jars. There are two types commonly used commercial drilling jars, combination of hydraulic (upward) and mechanical (downward) motion, and purely mechanical action. The examples of both types follow.

Christensen-Mason Jar. (See Figure 4-172 and Table 4-103 [57].) This is a combination tool, offering possibility to jar upwards hydraulically and to bump downward mechanically. The jar is equipped with a special releasing (locking) mechanism, so that the jar cannot be fired upwards until the locking system has been released. It has a 6-in. jarring stroke upwards and 30-in. for downward bumping [57].

Hevi-Hitter™ Christensen Jar. (See Figure 4-173 [57A].) This is a mechanical drilling jar with firing racks system applied. Its jar force is constant regardless of torque applied.

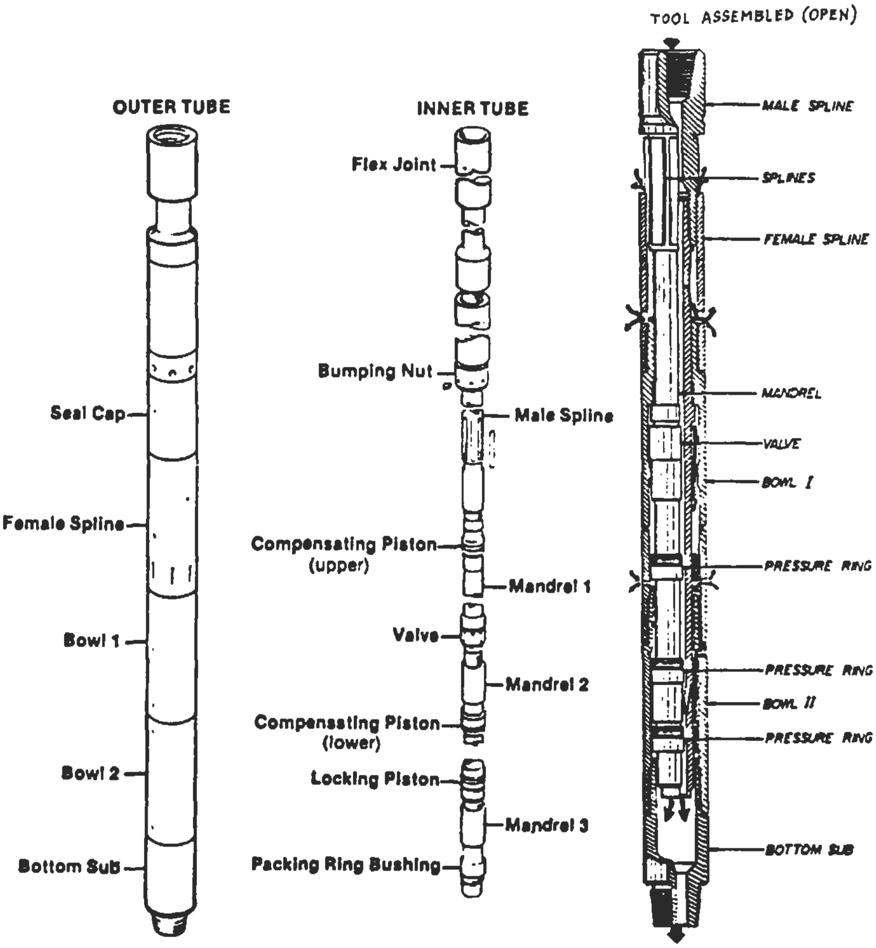


Figure 4-172. Christensen's mason drilling jar. *(Courtesy Hughes Christensen.)*

Table 4-103
Christensen's Mason Drilling Jar

O. D.	I. D.	Max Jarring load	Tool Weight
Inch	Inch	lbs	lbs
$4^3/_4$	2	80,000	1,100
$6^1/_4$	$2^1/_4$	165,000	1,800
$7^1/_4$	$2^1/_2$	165,000	3,300
$8^1/_4$	$2^1/_2$	190,000	3,900
$9^1/_2$	$2^{13}/_{16}$	190,000	5,000
$11^1/_4$	$2^{13}/_{16}$	250,000	7,000

Courtesy Hughes Christensen.

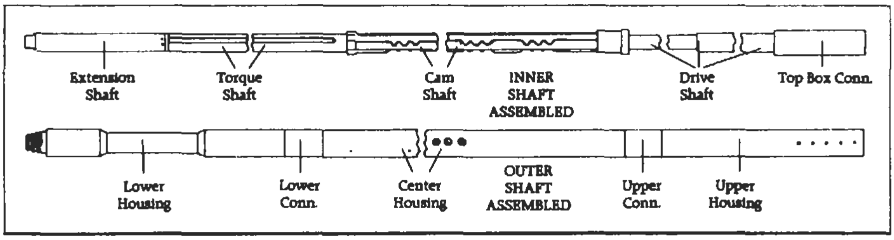


Figure 4-173. Hevi-Hitter™ mechanical drilling jar. *(Courtesy Hughes Christensen.)*

Included in the drill string, the Hevi-Hitter jar can fire either upward or downward. The jarring force, which can exceed 700,000 pounds on the larger sizes, can be controlled from the surface by applying and holding right-hand torque to increase impact, and by applying and holding left-hand torque to decrease impact. Because the Hevi-Hitter jar recocks automatically, jarring operations can proceed swiftly until the stuck pipe is free. Various impact forces can be generated dependant upon weight of drill collars (or heavy weight drill pipe) above the jars and the value of surface pull, as shown in Table 4-104 [57A].

Underreamers

The term "underreaming" has been used interchangeably with "hole opening." Underreaming is the process of enlarging the hole bore beginning at some point below the surface using a tool with expanding cutters. This permits lowering the tool through the original hole to the point where enlargement of the hole is to begin.

Table 4-104
Hevi-Hitter™ Jar Impact Forces (1,000 lb/ft) [57A]

Pull Above Pipe Weight (lbs. x 10,000)	Heavy-Weight Drill Pipe or Drill Collar Weight Above Jars (lbs. × 1,000)								
	4	6	8	10	12	14	16	18	20
3	45	68	90	113	135	157	180	202	225
4	60	90	120	150	180	210	240	270	300
5	75	113	150	187	225	262	300	337	374
6	90	135	180	225	270	315	359	404	449
7	105	158	210	262	315	367	419	472	524
8	120	180	240	300	359	419	479	539	599
9	135	202	270	337	404	472	539	606	674
10	150	225	300	374	449	524	599	674	748
11	165	247	330	412	494	576	659	741	823
12	180	270	359	449	539	629	719	808	898
13	195	229	389	487	584	681	778	876	973
14	210	315	419	524	629	733	838	943	1048
15	225	337	449	562	674	786	898	1010	1122
16	240	359	479	599	719	838	958	1078	1197

Courtesy Hughes Christensen.

Hole opening is considered as opening or enlarging the hole from the surface (or casing shoe) downward using a tool with cutter arms at a fixed diameter.

Thus, the proper name for the tools with expandable cutting arms is underreamers. The cutting arms are collapsed in the tool body while running the tool in the hole. Once the required depth is reached, mud circulation pressure moves the cutters opening for drilling operation. Additional pressure drop across the underreamer orifice gives the operator positive indication that the cutter arms are extended fully and the tool is underreaming at full gauge.

Underreamer Design. There are two basic types of underreamers: (1) roller cone rock-type underreamers and (2) drag-type underreamers. The roller cone rock-type underreamers are designed for all types of forma-tions depending upon the type of roller cones installed. The drag-type under-reamers are used in soft to medium formations. Both types can be equipped with a bit to drill and underream simultaneously. This allows for four different combinations of underreamers as shown in Figure 4-174 [58]. Nomenclature of various underreamer designs are shown in Figures 4-175, 4-176, and 4-177 [58].

Underreamer Specifications. Table 4-105 [58] shows example specifications for nine models of Servco roller cone rock-type underreamers. Table 4-106 contains example data for four models of Servco drag-type underreamers [58].

Underreamer Hydraulics. Pressure losses across the underreamer nozzles (orifice) are shown in Figures 4-178 and 4-179 [58]. The shaded area represents the recommended pressure drop required for cutters to fully open. These pressure drop graphs can be used for pressure losses calculations (given pump output and nozzles) or for orifice (nozzle) selection (given pump output and pressure loss required).

Figure 4-174. Types of underreamers. *(Courtesy Smith International, Inc.)*

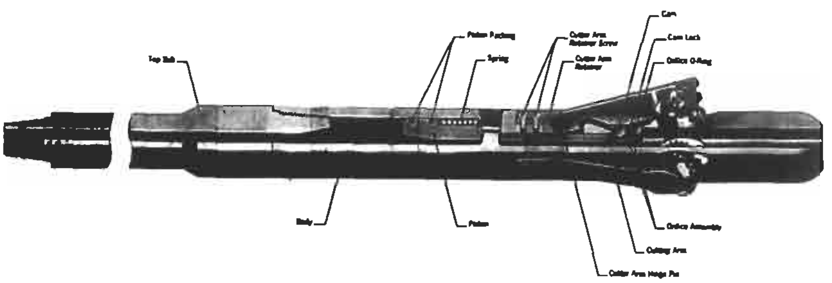


Figure 4-175. Rock-type underreamer nomenclature. *(Courtesy Smith International, Inc.)*

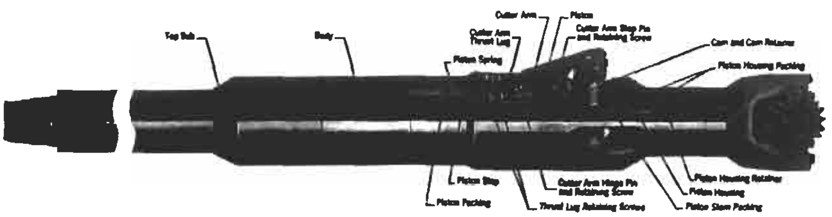


Figure 4-176. Rock-drilling underreamer nomenclature. *(Courtesy Smith International, Inc.)*

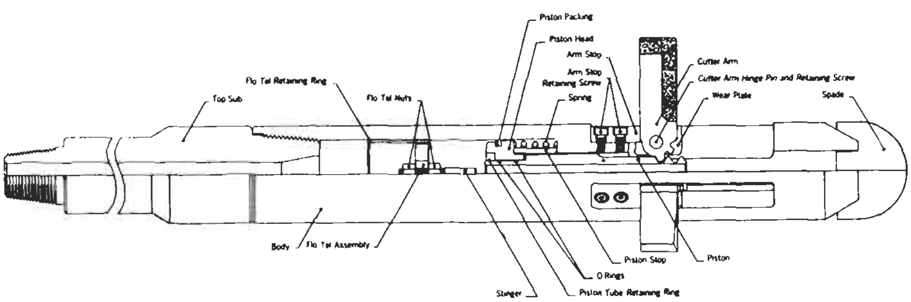


Figure 4-177. Drag-type underreamer nomenclature (open arms). *(Courtesy Smith International, Inc.)*

Table 4-105
Rock-type Underreamers (Servco) [58]

Through Casing, Inches	Underreamer Body Dia., Inches	Top Connections, API Reg. Pin	Hole Opening Dimensions Inches
4-1/2	3-5/8	2-3/8	4-3/4 through 6-1/2
5-1/2	4-1/2	2-7/8	6 through 9
7	5-3/4	3-1/2	8 through 11
7-5/8	6	3-1/2	8 through 13
8-5/8	7-1/4	4-1/2	9 through 14
9-5/8	8-1/4	4-1/2	10 through 15
10-3/4	9-1/2	6-5/8	13 through 18
13-3/8	11-3/4	6-5/8	15 through 22
18-5/8	14-3/4	6-5/8	22 through 28

Courtesy Smith International, Inc.

Table 4-106
Drag-type Underreamers (Servco) [58]

	Model Number			
Specifications	57DP	72 DP	95DP	110DP
Body Dia.	5-3/4"	7-1/4"	9-1/2"	11"
Top Conn.	3-1/2" Reg. Pin	4-1/2" Reg. Pin	6-5/8" Reg. Pin	6-5/8" Reg. Pin
Length (shoulder)	66"	71"	57"	57"
Expanded Dia. (max.)	16"	22"	28"	30"
Standard Orifice	Flo-Tel	Flo-Tel	3/4"	3/4"

Courtesy Smith International, Inc.

Stabilizers

Drill collar stabilizers are installed within the column of drill collars. Stabilizers guide the bit straight in vertical-hole drilling or help building, dropping, or maintaining hole angle in directional drilling. The stabilizers are used to

1. provide equalized loading on the bit
2. prevent wobbling of the lower drill collar assembly
3. minimize bit walk
4. minimize bending and vibrations that cause tool joint wear
5. prevent collar contact with the sidewall of the hole
6. minimize keyseating and differential pressure.

The condition called "wobble" exists if the bit centerline does not rotate exactly parallel to and on the hole centerline so the bit is tilted.

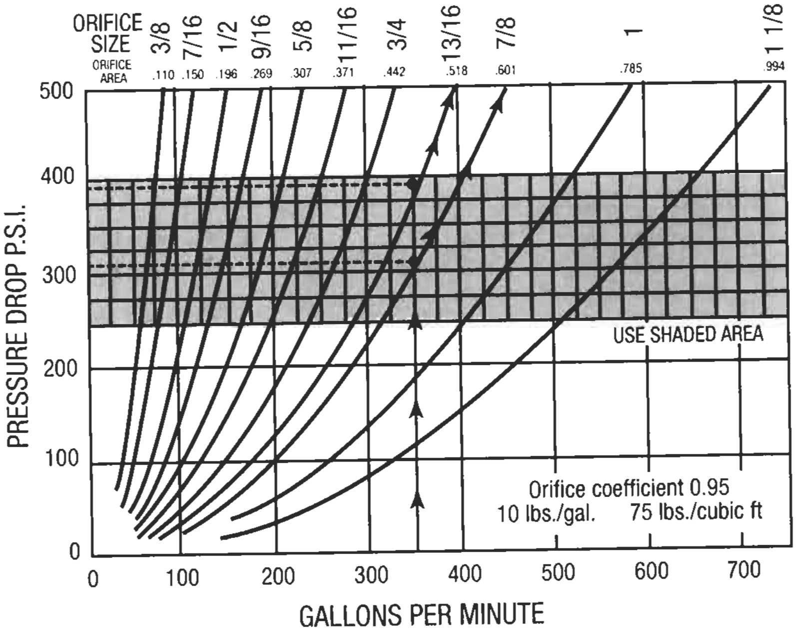


Figure 4-178. Pressure drop across underreamer (rock-type or drag-type underreamer with one nozzle). *(Courtesy Smith International, Inc.)*

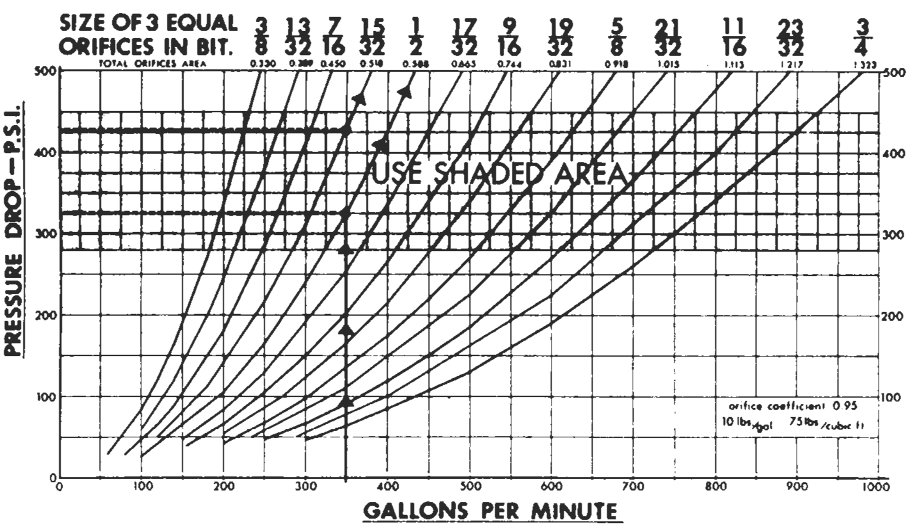


Figure 4-179. Pressure drop across underreamer (rock drilling underreamer with three nozzles). *(Courtesy Smith International, Inc.)*

Stabilizer Design. There are four commonly used stabilizer designs.

Solid-Type Stabilizers. (See Figure 4-180.) These stabilizers have no moving or replaceable parts, and consist of mandrel and blades that can be one piece alloy steel (integral blade stabilizer) or blades welded on the mandrel (weld-on blade stabilizer). The blades can be straight, or spiral, and their working surface is either hardfaced with tungsten carbide inserts or diamonds [57,58].

Replaceable-Blades Stabilizers. (See Figure 4-181 [58A].) These stabilizers can maintain full gauge stabilization. Their blades can be changed at the rig with hand tools; no machining or welding is required.

Sleeve-Type Stabilizers. (See Figure 4-182.) These stabilizers have replaceable sleeve that can be changed in the field. There are two types of sleeve-type stabilizers: the rotating sleeve-type stabilizer (Figure 4-182A [58]) and the nonrotating sleeve-type stabilizer (Figure 4-182B [59]). Rotating sleeve-type stabilizers have no moving parts and work in the same way as solid-type stabilizers. Nonrotating sleeve-type stabilizers have a nonrotating rubber sleeve supported by the wall of the borehole. The rubber sleeve stiffens the drill collar string in packed hole operations just like a bushing.

Reamers. (See Figure 4-183 [59].) Reamers are stabilizers with cutting elements embedded in their fins, and are used to maintain hole gage and drill out doglegs and keyseats in hard formations. Because of the cutting ability of the reamer, the bit performs less work on maintaining hole gauge and more work on drilling

A	B	C	D
Integral Blade Stabilizer. Hardfacing with tungsten carbide compacts. (Servco)	Weld-On Blade Stabilizer. Alloy steel hardfacing. (Servco)	Big Bear™ Near-Bit Stabilizer. Granular tungsten carbide hardfacing. (Servco)	Diamond Near-Bit Stabilizer (Christensen)

Figure 4-180. Solid-type stabilizers. *(Courtesy Smith International and Baken Hughes INTEQ)*

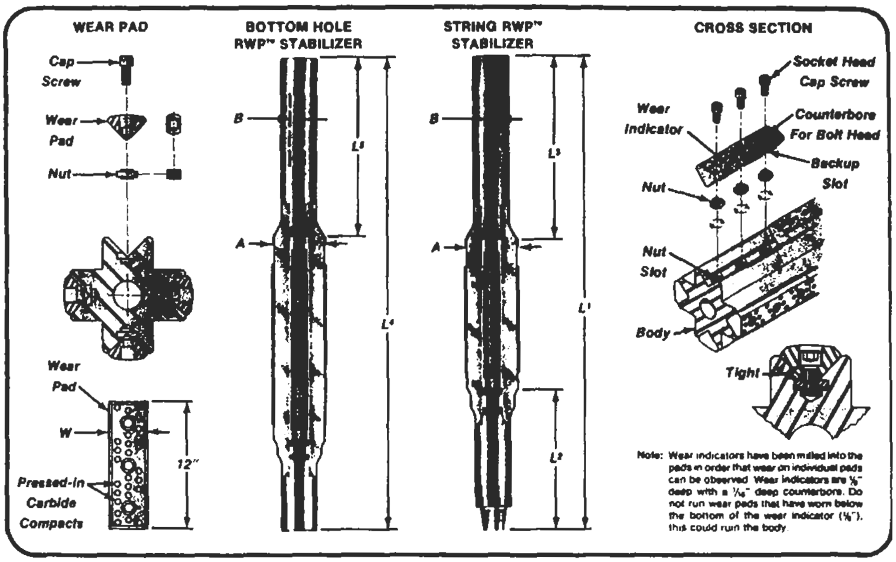


Figure 4-181. Replaceable-blades stabilizers. *(Courtesy Smith International, Inc.)*

ahead. Reamers can be used as near-bit stabilizers in the bottomhole assembly or higher up in the string. There are basically three types of reamer body:

- Three-point bottom hole reamer. This type of reamer is usually run between the drill collars and the bit to ensure less reaming back to bottom with a new bit.
- Three-point string reamer. The reamer is run in the drill collar string. This reamer provides stabilization of the drill collars to drill a straighter hole in crooked hole country. When run in the string, the reamer is effective in reaming out dog-legs, keyseats and ledges in the hole.
- Six-point bottom hole reamer. This type of reamer is run between the drill collars and the bit when more stabilization or greater reaming capacity is required. Drilling in crooked hole areas with a six-point reamer has proven to be very successful in preventing sharp changes in hole angles.

Application of Stabilizers. Figure 4-184 [16] illustrates three applications of stabilizers, pendulum, fulcrum, and lock-in (stiff) hook-up.

The stiff hookup consists of three or more stabilizers placed in the bottom 50 to 60 ft of drill collar string. In mild crooked hole conditions, the stiff hookup will hold the deviation to a minimum. In most cases, deviation will be held below the maximum acceptable angle. In severe conditions, this hookup will slow the rate of angle buildup, allowing more weight to be run for a longer time. This method prevents sudden increases or decreases of deviation, making dog-legs less severe and decreasing the probability of subsequent keyseats and other undesirable hole conditions. The stiff hookup is beneficial only until the maximum acceptable angle is reached. The pendulum principle should then be used.

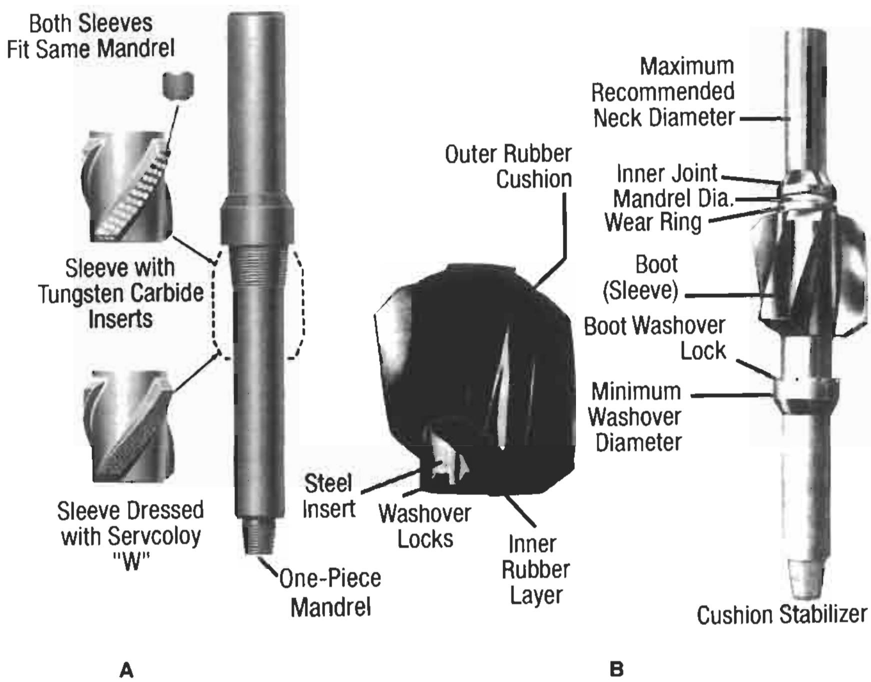


Figure 4-182. Sleeve-type stabilizers. (A) Rotating sleave-type stabilizer (Servco). (B) Grant cushion stabilizers (nonrotating sleave-type stabilizer). *(Courtesy Smith International and Masco Tech Inc.)*

To employ pendulum effect in directional drilling, usually one stabilizer is placed in the optimum position in the drill collar string. The position is determined by the hole size, drill collar size, angle of deviation and the weight on the bit. A properly placed stabilizer extends the suspended portion of the drilling string (that portion between the bit and the point of contact with the low side of the hole). The force of gravity working on this extended portion results in a stronger force directing the bit toward vertical so the well trajectory returns to vertical.

To employ fulcrum effect one stabilizer is placed just above the bit and additional weight is applied to the bit. The configuration acts as a fulcrum forcing the bit to the high side of the hole. The angle of hole deviation increases (buildup) as more weight is applied.

To employ a restricted fulcrum effect one stabilizer is placed just above the bit while second stabilizer is placed above the nonmagnetic drill collar. The hookup allows a gradual buildup of inclination with no abrupt changes.

To prevent key-seating one stabilizer is placed directly above the top of drill collars. The configuration prevents drill collars wedging into a key seat during tripping out of the hole.

To prevent differential sticking across depleted sands stabilizers are placed throughout the drill collar string. The area of contact between drill collars and hole is reduced, thus reducing the sticking force.

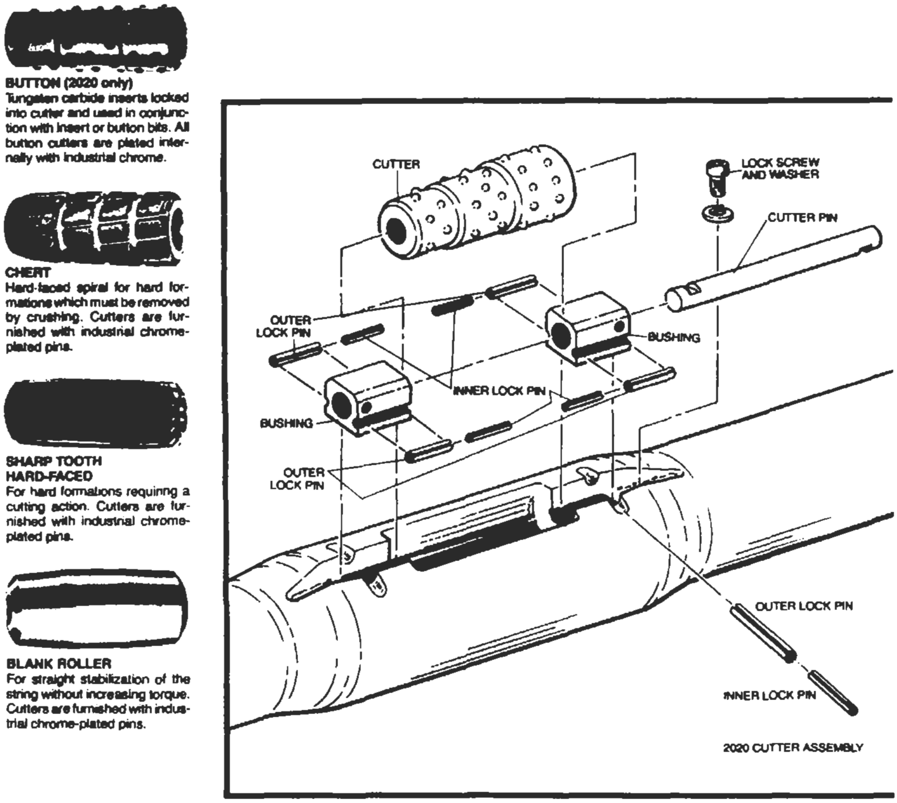


Figure 4-183. Stabilizer/reamers; various cutters and schematics of cutter assembly. *(Courtesy Masco Tech Inc.)*

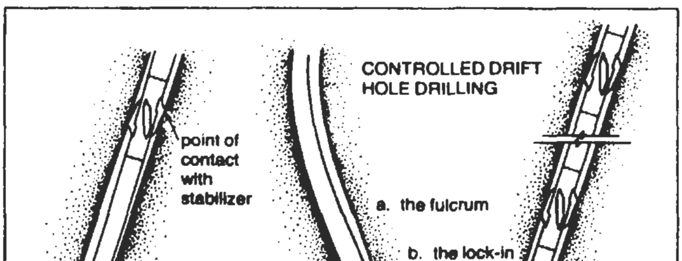


Figure 4-184. Applications of stabilizers in directional drilling [16].

DRILLING MUD HYDRAULICS

Rheological Classification of Drilling Fluids

Experiments performed on various drilling muds have shown that the shear stress-shear rate characteristic can be represented by one of the functions schematically depicted in Figure 4-185. If the shear stress-shear rate diagram is a straight line passing through the origin of the coordinates, the drilling fluid is classified as a Newtonian fluid, otherwise it is considered to be non-Newtonian.

The following equations can be used to describe the shear stress-shear rate relationship:

Newtonian fluid

$$\tau = \mu\left(-\frac{dv}{dr}\right) \tag{4-85}$$

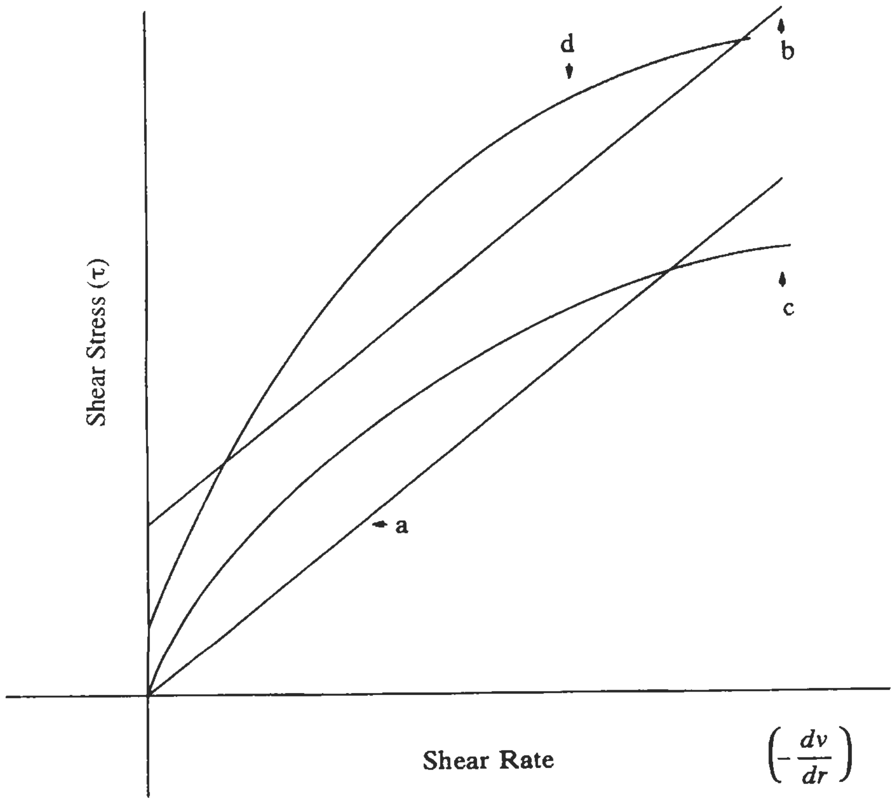


Figure 4-185. Shear stress-shear rate diagram. (a) Newtonian fluid. (b) Bingham plastic fluid. (c) Power law fluid. (d) Herschel-Buckley fluid.

Bingham plastic fluid

$$\tau = \tau_y + \mu_p\left(-\frac{dv}{dr}\right)^n \tag{4-86}$$

Power law fluid

$$\tau = K\left(-\frac{dv}{dr}\right)^n \tag{4-87}$$

Herschel and Buckley fluid

$$\tau = \tau_y + K\left(-\frac{dv}{dr}\right)^n \tag{4-88}$$

where τ = shear stress
v = the velocity of flow
dv/dr = shear rate (velocity gradient in the direction perpendicular to the flow direction)
μ = dynamic viscosity
τ_y = yield point stress
μ_p = plastic viscosity
K = consistency index
n = flow behavior index

The μ, τ_y, μ_p, K and n are usually determined with the Fann rotational viscosimeter. The Herschel and Buckley model is not considered in this manual.

Flow Regimes

The flow regime, i.e., whether laminar or turbulent, can be determined using the concept of the Reynolds number. The Reynolds number, Re, is calculated in consistent units from

$$Re = \frac{dv\rho}{\mu} \tag{4-89}$$

where d = diameter of the fluid conduit
v = velocity of the fluid
ρ = density of fluid
μ = viscosity

In oilfield engineering units

$$Re = 928\frac{d_e v\bar{\gamma}}{\mu} \tag{4-90}$$

where d_e = equivalent diameter of a flow channel in in.
v = average flow velocity in ft/s
$\bar{\gamma}$ = drilling fluid specific weight in lb/gal
μ = drilling fluid dynamic viscosity in cp

The equivalent diameter of the flow channel is defined as

$$d_e = \frac{4(\text{flow cross-sectional area})}{\text{wetted perimeter}} \tag{4-91}$$

The flow changes from laminar to turbulent in the range of Reynolds numbers from 2,100 to 4,000 [60]. In laminar flow, the friction pressure losses are proportional to the average flow velocity. In turbulent flow, the losses are proportional to the velocity to a power ranging from 1.7 to 2.0.

The average flow velocity is given by the following equations:

- Flow in circular pipe

$$v = \frac{q}{2.45d^2} \tag{4-92}$$

- Flow in an annular space between two circular pipes

$$v = \frac{q}{2.45(d_1^2 - d_2^2)} \tag{4-93}$$

where q = mud flow rate in gpm
d = inside diameter of the pipe in in.
d_1 = larger diameter of the annulus in in.
d_2 = smaller diameter of the annulus in in.

For non-Newtonian drilling fluids, the concept of an effective viscosity* can be used to replace the dynamic viscosity in Equation 4-89.

For a Bingham plastic fluid flow in a circular pipe and annular space, the effective viscosities are given as [61].

- Pipe flow

$$\mu_e = \mu_p + 6.65\frac{\tau_y d}{v} \tag{4-94}$$

- Annular flow

$$\mu_e = \mu_p = 4.99\frac{\tau_y(d_1 - d_2)}{v} \tag{4-95}$$

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*Also called equivalent or apparent viscosity in some published works.

For a Power law fluid flow, the following formulas can be used:

- Pipe flow

$$\mu_e = \left(\frac{1.6v}{d}\frac{3n+1}{4n}\right)^n\left(\frac{300Kd}{v}\right) \tag{4-96}$$

- Annular flow

$$\mu_e = \left(\frac{2.4v}{d_1 - d_2}\frac{2n+1}{3n}\right)^n \frac{200K(d_1 - d_2)}{v} \tag{4-97}$$

The mud rheological properties μ_p, τ_y, n and K are typically calculated based upon the data from two (or more)-speed rotational viscometer experiments. For these experiments, the following equations are applicable:

$$\mu_p = \theta_{600} - \theta_{300}(c_p) \tag{4-98}$$

$$\tau_y = \theta_{300} - \mu_p(\text{lb}/100\ \text{ft}^2) \tag{4-99}$$

$$n = 3.32 \log \frac{\theta_{600}}{\theta_{300}} \tag{4-100}$$

$$K = \frac{\theta_{300}}{(511)^n}(\text{lb}/100\text{ft}^2\text{s}^{-n}) \tag{4-101}$$

where θ_{600} = viscometer reading at 600 rpm
θ_{300} = viscometer reading at 300 rpm

Example

Consider a well with the following geometric and operational data:

Casing $9\frac{5}{8}$ in., unit weight = 40 lb/ft, ID = 8.835 in. Drill pipe: $4\frac{1}{2}$ in., unit weight = 16.6 lb/ft, ID = 3.826 in. Drill collars: $6\frac{3}{4}$ in., unit weight = 108 lb/ft, ID = $2\frac{1}{4}$ in. Hole size: $8\frac{1}{2}$ in. Drilling fluid properties: θ_{600} = 68, θ_{300} = 41, density = 10 lb/gal, circulating rate = 280 gpm.

Calculate Reynolds number for the fluid (1) inside drill pipe, (2) inside drill collars, (3) in drill collar annulus, and (4) in drill pipe annulus.

To perform calculation, a Power law fluid is assumed.

Flow behavior index (use Equation 4-100) is

$$n = 3.32 \log \frac{68}{41} = 0.729$$

Consistency index (use Equation 4-101) is

$$K = \frac{41}{(511)^{0.729}} = 0.433\text{lb}/100\text{ft}^2\text{s}^{-0.729}$$

The average flow velocities are

- Inside drill pipe (use Equation 4-92)

$$v = \frac{280}{(2.45)(3.826)^2} = 7.807\,\text{ft/s}$$

- Inside drill collars

$$v = \frac{280}{(2.45)(2.25)^2} = 22.575\,\text{ft/s}$$

- In drill collar annulus, an open hole (use Equation 4-93)

$$v = \frac{280}{(2.45)(8.5^2 - 6.75^2)} = 4.282\,\text{ft/s}$$

- In drill pipe annulus (in the cased hole)

$$v = \frac{280}{(2.45)(8.835^2 - 4.5^2)} = 1.977\,\text{ft/s}$$

The effective viscosities are

- Inside drill pipe (use Equation 4-96)

$$\mu_e = \left(\frac{(1.6)(7.807)}{3.826}\,\frac{(3)(0.729)+1}{(4)(0.729)}\right)^{0.729}\left(\frac{(300)(0.433)(3.826)}{7.807}\right) = 160.9\,\text{cp}$$

- Inside drill collars

$$\mu_e = \left(\frac{(1.6)(22.575)}{2.25}\,\frac{(3)(0.729)+1}{(4)(0.729)}\right)^{0.729}\left(\frac{(300)(0.433)(2.25)}{22.575}\right) = 104.5\,\text{cp}$$

- In drill collar annulus (use Equation 4-97)

$$\mu_e = \left(\frac{(2.4)(4.282)}{8.5-6.75}\,\frac{(2)(0.729)+1}{(3)(0.729)}\right)^{0.729}\left(\frac{(200)(0.433)(8.5-6.75)}{4.282}\right) = 140.1\,\text{cp}$$

- In drill pipe annulus

$$\mu_e = \left(\frac{(2.4)(1.977)}{8.835-4.5}\,\frac{(2)(0.729)+1}{(3)(0.729)}\right)^{0.729}\left(\frac{(200)(0.433)(8.835-4.5)}{1.977}\right) = 233.6\,\text{cp}$$

Reynolds number (use Equation 4-90)

- Inside drill pipe

$$Re = 928\frac{(3.826)(7.807)(10)}{(160.9)} = 1723$$

- Inside drill collars

$$Re = 928\frac{(2.25)(22.575)(10)}{(104.5)} = 4511$$

- In drill collar annulus

$$Re = 928\frac{(8.5 - 6.75)(4.282)(10)}{(140.1)} = 496$$

- In drill pipe annulus

$$Re = 928\frac{(8.835 - 4.5)(1.977)(10)}{(233.6)} = 340$$

Principle of Additive Pressures

Applying the conservation of momentum to the control volume for a one-dimensional flow conduit, it is found that [62]

$$\rho A\frac{dv}{dt} = -A\frac{dP}{dl} - P_w\tau_w - A\rho g\cos\alpha \tag{4-102}$$

where ρ = fluid density
A = flow area
dv/dt = acceleration (total derivative)
v = flow velocity
τ_w = average wall shear stress
P_w = wetted perimeter
g = gravity acceleration
α = inclination of a flow conduit to the vertical
dP/dl = pressure gradient
l = length of flow conduit

For a steady-state flow, Equation 4-102 is often written as an explicit equation for the pressure gradient. This is

$$\frac{dP}{dl} = -\frac{P_w}{A}\tau_w - \rho v\frac{dv}{dl} - \rho g\cos\alpha \tag{4-103}$$

The three terms on the right side are known as frictional, accelerational (local accelaration) and gravitational components of the pressure gradient. Or, in other words, the total pressure drop between two points of a flow conduit is the sum of the components mentioned above. Thus,

$$\Delta P = \Delta P_F + \Delta P_A + \Delta P_G \tag{4-104}$$

where ΔP_F = frictional pressure drop
ΔP_A = accelerational pressure drop
ΔP_G = gravitational pressure drop (hydrostatic head)

Equation 4-104 expresses the principle of additive pressures. In addition to Equation 4-104, there is the equation of state for the drilling fluid.

Typically, water based muds are considered to be incompressible or slightly compressible. For the flow in drill pipe or drill collars, the acceleration component (ΔP_A) of the total pressure drop is negligible, and Equation 4-104 can be reduced to

$$\Delta P = \Delta P_F + \Delta P_G \tag{4-105}$$

Equations 4-102 through 4-105 are valid in any consistent system of units.

Example

The following data are given:

- Pressure drop inside the drill string = 600 psi
- Pressure drop in annular space = 200 psi
- Pressure drop through the bit nozzle = 600 psi
- Hole depth = 10,000 ft
- Mud density = 10 lb/gal

Calculate

- bottomhole pressure
- pressure inside the string at the bit level (above the nozzles)
- drill pipe pressure

Because the fluid flow in annular space is upward, the total bottom hole pressure is equal to the hydrostatic head plus the pressure loss in the annulus.

Bottom hole pressure P_{bottom} (psi),

$$P_{bottom} = (0.052)(10)10{,}000 + 200 = 5{,}200 \text{ psi}$$

Note that when the circulation is stopped, the friction pressure loss in annular space diminishes to zero and the bottom hole pressure is reduced to 5,000 psi.

Pressure inside the string above the nozzle, p_{is} (psi),

$$p_{is} = 5200 + 1600 = 6800 \text{ psi}$$

Drill pipe pressure, p_{dp} (psi),

$$p_{dp} = 6800 + 600 - (0.052)(10)(10{,}000) = 2{,}200 \text{ psi}$$

Note that the drill pipe pressure is a sum of all the pressure losses in the circulating system.

Friction Pressure Loss Calculations

Laminar Flow

For pipe flow of Bingham plastic type drilling fluid, the following can be used:

$$\Delta p = \frac{\mu_p L v}{1500 d^2} + \frac{\tau_y L}{225 d} \tag{4-106}$$

Corresponding equation for a Power law type drilling fluid is

$$\Delta p = \left[\left(\frac{1.6 v}{d}\right)\left(\frac{3n+1}{4n}\right)\right]^n \frac{KL}{300d} \tag{4-107}$$

For annular flow of Bingham plastic and Power law fluids, respectively,

$$\Delta p = \frac{\mu_p L v}{1000(d_1 - d_2)^2} + \frac{\tau_y L}{200(d_1 - d_2)} \tag{4-108}$$

and

$$\Delta p = \left[\left(\frac{2.4 v}{(d_1 - d_2)}\right)\left(\frac{2n+1}{3n}\right)\right]^n \frac{KL}{300(d_1 - d_2)} \tag{4-109}$$

Turbulent Flow

Turbulent flow occurs if the Reynolds number as calculated above exceeds a certain critical value. Instead of calculating the Reynolds number, a critical flow velocity may be calculated and compared to the actual average flow velocity [60].

The critical velocities for the Bingham plastic and Power law fluids can be calculated as follows:

- Bingham plastic fluid

$$v_c = \frac{1.08\mu_p + 1.08\sqrt{\mu_p^2 + 9.256(d_1 - d_2)^2 \tau_y \rho}}{(d_1 - d_2)} \tag{4-110}$$

- Power law fluid

$$v_c = \left[\frac{3.878 \times 10^4 K}{\rho}\right]^{1/(2-n)} \left[\left(\frac{2.4}{d_2 - d_2}\right)\left(\frac{2n+1}{3n}\right)\right]^{n/(2-n)} \tag{4-111}$$

In the case of pipe flow, for practical purposes, the corresponding critical velocities may be calculated using Equation 4-110 and 4-111, but letting $d_2 = 0$.

In the above equation, the critical flow velocity is in ft/min and all other quantities are specified above.

In turbulent flow the pressure losses, Δp (psi), can be calculated from the Fanning equation [60].

$$\Delta p = \frac{f\gamma L v^2}{25.8d} \tag{4-112}$$

where f = Fanning factor
L = length of pipe, ft

The friction factor depends on the Reynolds number and the surface conditions of the pipe. There are numerous charts and equations for determining the relationship between the friction factor and Reynolds number. The friction factor can be calculated by [63]

$$f = 0.046 \text{ Re}^{-0.2} \tag{4-113}$$

Substituting Equation 4-91 (4-92), and 4-113 into Equation 4-112 yields [63]

- Pipe flow

$$\Delta p = \frac{7.7 \times 10^{-5} \bar{\gamma}^{0.8} q^{1.8} \mu_p^{0.2} L}{d^{4.8}} \tag{4-114}$$

- Annular flow

$$\Delta p = \frac{7.7 \times 10^{-5} \bar{\gamma}^{0.8} \mu_p^{0.2} q^{1.8} L}{(d_1 - d_2)^3 (d_1 + d_2)^{1.8}} \tag{4-115}$$

Example

The wellbore, drill string and drilling fluid data from the previous example are used. Casing depth is 4,000 ft. Assuming a drill pipe length of 5,000 ft and a drill collar length of 500 ft, find the friction pressure losses.

- Flow inside the drill pipe

The critical flow velocity is

$$v_c = \left[\frac{(3.878 \times 10^4)(0.433)}{10}\right]^{1/(2-0.729)} \left[\frac{2.4(2)(0.7.29)+1}{}\right]^{0.729/(2-0.729)}$$

$$= 343.54 \text{ ft/min} = 5.73 \text{ ft/s}$$

Since $v < v_c$, the flow is laminar, and Equation 4-107 is chosen to calculate the pressure loss p_1.

$$\Delta p_1 = \left[\left(\frac{(1.6)(468.42)}{3.826}\right)\left(\frac{(3)(0.729)+1}{(4)(0.729)}\right)\right]^{0.729}\frac{(0.43)(5000)}{(300)(3.826)}$$

$$= 94.32\text{ psi}$$

- Flow inside the drill collars

It is easy to check that the flow is turbulent and thus Equation 4-114 is chosen to calculate the pressure loss Δp_2.

$$\Delta p_2 = \frac{(7.7\times 10^{-5})(10^{0.8})(27)^{0.2}(280)^{1.8}(500)}{(2.25)^{4.8}} = 243.23\text{ psi}$$

- Annulus flow around the drill collars

To calculate the critical velocity, Equation 4-111 was used

$$v_c = \left[\frac{(3.878)(10^4)(0.433)}{10}\right]^{1/(2-0.729)}\left[\frac{2.4}{8.5-6.75}\frac{(2)(0.729)+1}{(3)(0.729)}\right]^{0.729/(2-0.729)}$$

$$= 441.79\text{ ft/min} = 7.36\text{ ft/s}$$

Since $v < v_c$, the flow is laminar and Equation 4-109 is chosen to calculate the pressure loss Δp_3.

$$\Delta p_3 = \left[\frac{(2.4)(25.92)}{(8.5-6.75)}\frac{(2)(0.729)+1}{(3)(0.729)}\right]^{0.729}\frac{(0.433)500}{(300)(8.5-6.75)}$$

$$= 32.27\text{ psi}$$

- Annulus flow around the drill pipe in the open hole section

It is found from Equation 4-111 that the flow is laminar, thus Equation 4-109 can be used

$$\Delta p_4 = \left[\frac{(2.4)(131.87)}{(8.5-4.5)}\frac{(2)(0.729)+1}{(3)(0.729)}\right]^{0.729}\frac{(0.433)1000}{(300)(8.5-4.5)}$$

$$= 9.51\text{ psi}$$

- Annulus flow around the drill pipe in a cased section

It is found from Equation 4-111 that the flow is laminar, thus Equation 4-109 can be used.

$$\Delta p_5 = \left[\frac{(2.4)(118.62)}{(8.835-4.5)}\frac{(2)(0.729)+1}{(3)(0.729)}\right]^{0.729}\frac{(0.433)(4000)}{(300)(8.835-4.5)}$$

$$= 32.0 \text{ psi}$$

The total frictional pressure loss is

$$\Delta p_f = 94.32 + 243.23 + 32.27 + 9.51 + 32.0 = 411.33 \text{ psi}$$

Pressure Loss through Bit Nozzles

Assuming steady-state, frictionless (due to the short length of the nozzles) drilling fluid flow, Equation 4-102 is written

$$\rho v \frac{\partial v}{\partial l} = \frac{\partial p}{\partial l} \tag{4-116}$$

Integrating Equation 4-116 assuming incompressible drilling fluid flow (ρ is constant) and after simple rearrangements yields the pressure loss across the bit Δp_b(psi) which is

$$\Delta p_b = \frac{\rho v^2}{2} \tag{4-117}$$

Introducing the nozzle flow coefficient of 0.95 and using field system of units, Equation 4-117 becomes

$$\Delta p_b = \frac{\bar{\gamma} v^2}{1,120} \tag{4-118}$$

or

$$\Delta p_b = \frac{\bar{\gamma} q^2}{10,858 A^2} \tag{4-119}$$

where v = nozzle velocity in ft/s
q = flow rate in gpm
$\bar{\gamma}$ = drilling fluid density in lb/gal
A = nozzle flow area in in.^{2}

If the bit is furnished with more than one nozzle, then

$$A = A_1 + A_2 + A_3 + \ldots A_n \tag{4-120}$$

where n is the number of nozzles, and

$$d_{en} = \sqrt{d_1^2 + d_2^2 + \ldots d_n^2} \tag{4-121}$$

where d_{en} is the equivalent nozzle diameter.

Example

A tricone roller rock bit is furnished with three nozzles with the diameters of $\frac{9}{32}$, $\frac{10}{32}$ and $\frac{12}{32}$ in. Calculate the bit pressure drop if the mud weight is 10 lb/gal and flowrate is 300 gpm.

Nozzle equivalent diameter is

$$d_{en} = \sqrt{\left(\frac{9}{32}\right)^2 + \left(\frac{10}{32}\right)^2 + \left(\frac{12}{32}\right)^2} = 0.5643 \text{ in.}$$

and the corresponding flow area is

$$A = \frac{\pi}{4}\left(\frac{18.0277}{32}\right)^2 = 0.2493 \text{ in.}^2$$

The pressure loss through bit nozzles is

$$\Delta p_b = \frac{(10)(300)^2}{10858(0.2493)^2} = 1334 \text{ psi}$$

AIR AND GAS DRILLING

Types of Operations

Air and natural gas have been used as drilling fluids to drill oil and gas wells since 1953. There are basically four distinct types of drilling using these fluids: air and gas drilling with no additives (often called dusting), unstable foam drilling (also called misting), stable foam drilling and aerated mud drilling [64].

Air and natural gas have also been used as drilling fluids in slim-hole-drilling mining operations, special large-diameter boreholes for nuclear weapons tests, and, more recently, in geothermal drilling operations.

Air and natural gas drilling techniques are used principally because of their ability to drill in loss-of-circulation areas where mud drilling operations are difficult or impossible. These drilling fluids have other specific advantages over mud drilling fluids when applied to oil and gas well drilling operations, which will be discussed later in this section. In general, air and gas drilling techniques are restricted to mature sedimentary basins where the rock formations are well cemented and exhibit little plastic flow characteristics. Also, to varying degrees, air and gas drilling techniques are restricted to drilling in rock formations that have limited formation water or other fluids present.

In the United States, air and gas drilling techniques are used extensively in parts of the southwest in and around the San Juan Basin, in parts of the Permian Basin, in Arkansas and eastern Oklahoma, in Maryland, Virginia and parts of Tennessee. Internationally, oil and gas drilling operations are carried out with air and gas drilling techniques in parts of the Middle East, North Africa and in the Western Pacific.

Air and Gas

Air and natural gas are often used as a drilling fluid with no additives placed in the injected stream of compressed fluid. This type of drilling is also often referred to as "dusting" because great dust clouds are created around the drill rig when no formation water was present. However, modern air and gas drilling operations utilize a spray at the end of the blooey line to control the dust ejected from the well. Figure 4-185 shows a typical site plan for air drilling operations.

In air drilling operation, large compressors and usually a booster compressor are used to compress atmospheric air and supply the required volumetric flowrate to the standpipe in much the same way that mud pumps supply mud for drilling. The volumetric flowrate of compressed air needed (which is usually stated in SCFM of air) depends upon the drilling rate, the geometry of the borehole to be drilled and the geometry of the drill string to be used to drill the hole [64,65].

Natural gas drilling is carried out in regions where there is significant natural gas production, and it is extremely useful in drilling into potential fire or explosive zones such as coal seams, or oil and gas production formations. Instead of utilizing atmospheric air and compressing the air for supply to the standpipe, natural gas is taken from the nearby gas collection pipeline and supplied to the standpipe, often at pipeline pressure. If high standpipe pressures are needed, a booster is used to raise the pressure. The pressure needed at the standpipe, and thus the need for a booster, depends upon the volumetric flow rate required, the geometry of the borehole to be drilled, and the geometry of the drill string to be used to drill the hole [65,66].

The volumetric flowrates required for air and natural gas drilling are basically determined by the penetration rate and the geometry of the borehole and drill string. There must be sufficient compressed air (or gas) circulating through the drill bit to carry the rock cuttings from the bottom of the borehole.

The actual engineering calculations to determine the required volumetric flowrate and various pressure calculations will be discussed later in this section.

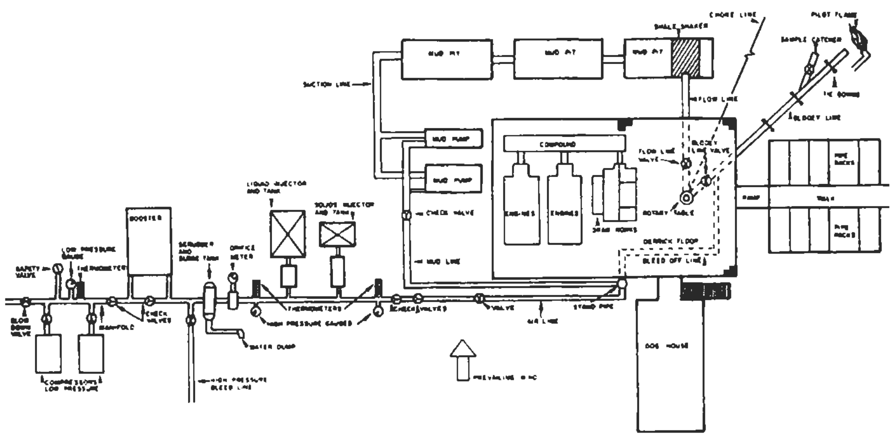


Figure 4-186. Air drilling surface equipment site plan.

Unstable Foam (Mist)

In order to increase the formation water-carrying capacity of the air and natural gas drilling fluids, water is often injected at the surface just after the air has been compressed and prior to the standpipe (water injector is shown in Figure 4-186). An amount of water is injected that will saturate the compressed air when it reaches the bottom of the hole. Thus, if the water-saturated returning airflow encounters formation water, internal energy in the airflow will not be required to change the formation water to vapor. The formation water will be carried to the surface as water particles much like the rock cuttings. If only water is injected at the surface, then the drilling fluid is called "mist." Usually a surfactant is injected with the injected water. This surfactant will cause the air and water to foam. This foam, however, is not continuous (i.e., it will have large voids in the annulus section because of the high velocity of the returning airflow). This is the reason why this type of drilling operation is also denoted as unstable foam.

Stable Foam

Stable foam drilling operations are used when even more formation water-carrying capability is needed (relative to air and gas and unstable foam). Also, stable foam provides significant bottomhole pressure that can counter formation pore pressures and thus provide some well control capabilities. Stable foam drilling operations provide a continuous column of foam in the annulus from the bottom of the borehole annulus to the back pressure valve at the end of the blooey line. Air and natural gas and unstable foam require large compressors to produce a fixed volumetric flowrate of air. Stable foam drilling requires far less compressed air, and the compressed air is provided by a flexible system. The air compressors used in stable foam drilling should be capable of supplying air at various pressures and volumetric flowrates. In general, the back pressure valve at the end of the blooey line is adjusted to ensure that a continuous foam column exists in the annulus. However, if the back pressure is too high, the foam at the bottom of the borehole (in the annulus) will break down into the individual phases of liquid and gas. Foam quality at the bottom hole in the annulus should not drop below about 60% [67–69]. Engineering calculations for determining the appropriate parameters for stable foam drilling operations are quite complicated. There are a few stable foam simulation programs available for well planning [70]. Those interested in stable foam engineering calculations are advised to consult service companies specializing in stable foam drilling operations.

Aerated Mud

Aerated mud drilling operations are used throughout the drilling industry, onshore and offshore. Aerated mud drilling is usually employed as an initial remedy to loss-of-circulation problems. To aerate water-based mud or oil-based mud, air is injected into the drilling mud flow at the surface prior to the mud entering the standpipe (primary aeration) or in the return annulus flow through an air line set with the casing string (parasite tubing aeration) [71,72]. Primary aeration is the most commonly used technique for aerating mud. But because of the high resistance to flow of aerated liquids, as aeration is needed at depth, parasite tubing aeration offers a usable alternative.

The relative advantages and disadvantages of the various types of air and gas drilling operations discussed are listed as follows:

Air and Gas (Dusting)	
Advantages	**Disadvantages**
• No loss-of-circulation problem • No formation damage • Very high penetration rate • Low bit costs • Low water requirement • No mud requirement	• No ability to counter subsurface pore pressure problems • Little ability to carry formation water from hole • Hole erosion problems are possible if formations are soft • Possible drill string erosion problems • Downhole fires are possible if hydrocarbons are encountered (gas only) • Specialized equipment necessary

Unstable Foam (Misting)	
Advantages	**Disadvantages**
• No loss-of-circulation problem • Ability to handle some formation water • No formation damage • Very high penetration rate • Low bit costs • Low water requirement • No mud requirement • Low chemical additive costs • Downhole fires are normally not a problem even with air	• Very little ability to counter subsurface pore pressure problems • No ability to carry a great deal of formation water from hole • Hole erosion problems are possible if formations are soft • Possible drill string erosion problems • Specialized equipment necessary

Stable Foam	
Advantages	**Disadvantages**
• No loss-of-circulation problem • Ability to handle considerable formation water • Little or no formation damage • High penetration rate • Low bit costs • Low water requirements • No mud requirements • Some ability to counter subsurface pore pressure problems	• Considerable additive (foamer) costs • Careful and continuous adjusting of proportions necessary • Specialized equipment necessary

Aerated Mud	
Advantages	**Disadvantages**
• Loss of circulation is not a big problem • Ability to handle very high volumes of formation water	• High mud pump pressure requirements • High casing/air line costs if parasite tubing is used • Some specialized equipment

Aerated Mud (continued)	
Advantages	**Disadvantages**
• Improved penetration rates (relative to mud drilling) • Ability to counter high subsurface pore pressure problems	

The listing is basically in descending order in terms of ability to counter loss-of-circulation problems (i.e., air and gas being the most useful technique) and a lack of causing formation damage (i.e., air and gas cause no formation damage). The listing is in ascending order in terms of ability to carry formation water from the hole and ability to counter subsurface pore pressure.

Equipment

Surface and subsurface specialized equipment are required for air and gas drilling operations.

Surface Equipment

Figure 4-186 shows the layout of surface equipment for a typical air drilling operation. Described below are specialized surface components unique to air drilling operations.

Blooey Line. This special pipeline carries exhaust air and cuttings from the annulus to the flare pit. The length of the blooey line should be sufficient to keep dust exhaust from interfering with rig operations. The blooey line should have no constrictions or curved joints.

Bleed-Off Line. This line bleeds off pressure within the standpipe, rotary base, kelly and the drill pipe to the depth of the top float valve. The bleed-off line allows air (or gas) under pressure to be fed directly to the blooey line.

Air (or Gas) Jets. The jets are often used when there is the possibility that relatively large amounts of natural gas may enter the annulus from a producing formation as the drilling operation progresses. The air (or gas) jets pull a vacuum on the blooey line and therefore on the annulus, thereby keeping gases in the annulus moving out of the blooey line.

Compressors and Boosters. In a typical air drilling operation the compressors supply compressed air from the atmosphere for discharge to the standpipe or for the boosters. For air drilling operations, these primary compressors are usually multistage machines that compress atmospheric air to about 200–300 psig. Air drilling operations require a fixed volumetric rate of flow, thus compressors are usually rated by the capacity at sea level conditions, or the actual cubic feet per minute of higher altitude atmospheric air they will operate with. In addition, these primary compressors are also rated by the maximum output air pressure. This is somewhat confusing since some of the primary compressors used in the field are fixed ratio screw-type compressors. These primary compressors produce only their maximum or fixed output pressure.

The booster, which can compress air coming from the primary compressors to higher levels (i.e., on the order of 1,000 psig or higher), is always a piston-type compressor capable of variable volumetric flow and variable pressure output.

The volumetric rate of flow requirements for air drilling operations and unstable foam drilling operations are quite large, on the order of 1,000 actual cfm to possibly as high as 4,000 actual cfm.

For stable foam drilling operations, much less volumetric rate of air flow is needed (i.e., usually less than 500 actual cfm). Also, the compressor should be capable of variable volumetric rate of flow and variable output pressure. The back pressure must be continuously adjusted to maintain a continuous column of stable foam in the annulus. This continuous adjustment of back pressure requires, therefore, continuous adjustment of input volumetric rate of airflow and output pressure (also, water and surfactant must be adjusted).

For aerated mud drilling operations, the compressor should also be capable of variable volumetric rate of airflow and variable output pressure. Again, as drilling progresses, the input volume of compressed air and the output pressure are continuously adjusted.

Chemical (and Water) Tank and Pump. The pump injects water, liquid foamers and chemical corrosion inhibitors into the high-pressure air (or gas) line after compression of the air and prior to the standpipe.

Solids Injector. This is used to inject hole-drying powder into the wellbore to dry water seeping into the borehole from water-bearing formations.

Rotating Kelly Packer (Rotating Head). Figure 4-187 shows the details of a rotating head. This surface equipment is critical and is also a unique piece of

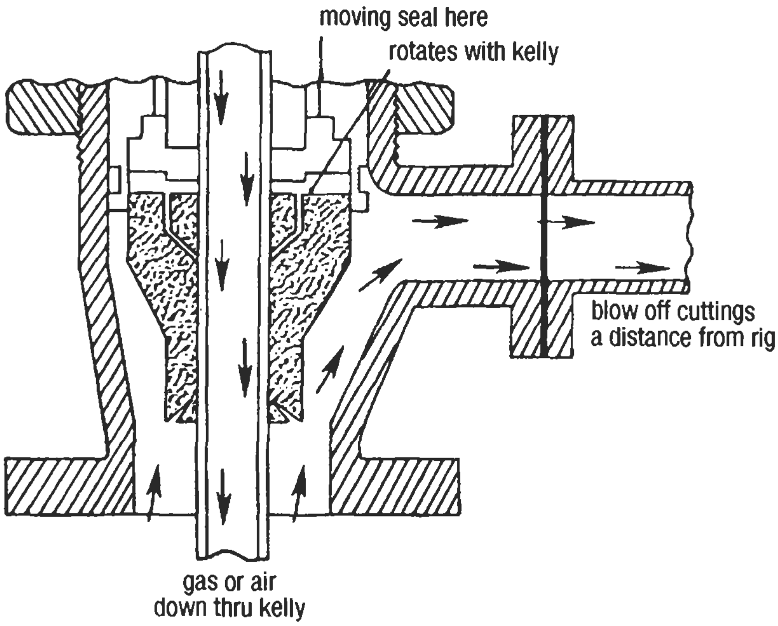


Figure 4-187. Rotating head.

equipment to air drilling operations. The rotating head packs off the annulus return flow from the rig floor (i.e., seals against the rotating kelly) and diverts the upward flowing air (or gas) and cuttings to the blooey line. Little pressure (a few psig) exists in the annulus flow at the rotating head.

Kelly. Because of its greater seal effectiveness within the rotating head, a hexagonal rather than a square kelly should be used in air (or gas) drilling operations.

Scrubber. This removes excess water from the injected air (or gas) stream to ensure that a minimum of moisture is circulated (if dry air for drilling is required) and to protect the booster.

Sample Catchers. A small-diameter pipe (about 2 in.) is fixed to the bottom of the blooey line to facilitate the catching and retaining of downhole cutting samples for geologic examination.

De-Duster. This provides a spray of water at the end of the blooey line to wet down the dust particles exiting the blooey line.

Gas Sniffer. This instrument can be hooked into the blooey line to detect very small amounts of natural gas entering the return flow from the annulus.

Pilot Light. This is a small continuously operated flame at the end of the blooey line, and it will ignite any natural gas encountered while drilling.

Burn Pit. This pit is at the end of the blooey line and provides a location for the cutting returns, foam and for natural gas or oil products from the subsurface to be ignited by the pilot light and burned off. The burn pit should be located away from the standard mud drilling reserve pit.

Meter for Measuring Air (or Gas) Volume. A standard orifice meter is generally used to measure air (or gas) injection volume rates.

Downhole Equipment

Special pieces of downhole equipment and special concerns must be considered during downhole air (or gas) drilling operations.

Float-valve Subs. These subs are at the bottom and near the top of the drill string. The bottom float-valve sub prevents the backflow of cuttings into the drill string during connections or other air (or gas) flow shutdowns that would otherwise plug the bit. The bottom float-valve sub also aids in preventing extensive damage to the drill string in the event of a downhole fire. The top (or upper) float-valve subs aid in retaining high pressure air (or gas) within a long drill string while making connections or other shutdowns.

Bottomhole Assemblies. In general, the drill pipe, drill collars and, in particular, bottomhole assemblies for air (or gas) drilling operations are the same as those in mud drilling. However, because the penetration rate is much greater in air (or gas) drilling operations due to the lack of confining pressure on the bit cutting surface, care must be taken to control unwanted deviation of the borehole. Thus, for air (or gas) drilled boreholes, a packed-hole or stiff bottomhole assembly is recommended.

Drill Pipe Wear. Erosion can occur between the hard band and the tool joint metal when the box end is hardbanded. This erosion is due to the high-velocity flow of cuttings in the annulus section of the borehole.

Bits. Bits that offer the best gage protection should be selected for air (or gas) drilling operations. Gage reduction always occurs in air (or gas) drilled boreholes, particularly near the end of a bit run. Returning to an air-drilled borehole with the same gage bit is dangerous unless care is taken in returning to the bottom. It is nearly always necessary to ream the last third of the borehole length (i.e., the last bit run) to get to the bottom. Most manufacturers of bits make tricone bits that are specially designed for air (or gas) drilling operations. These bits have the same cutting structures as the mud bits. The differences between air bits and mud bits are in the design of the internal passages for airflow and in the cooling of the internal bearings of the bits. It is common practice in air and gas drilling operations to operate the bits with no nozzle plates in the orifice openings.

Air Hammer. This is a special downhole drilling tool for controlling severe deviation problems and for drilling very hard formations. The air hammer is an air percussion hammer system that operates from compressed air and the rotary motion of the drill string.

Air (or Gas) Downhole Motors. Some positive displacement mud motors can be operated on unstable foam. In general, these mud motors must be low-torque, high-rotational-speed motors. Such motors have found limited use in air and gas drilling operations where directional boreholes are required. Recently a downhole turbine motor has been developed specifically for air and gas drilling operations. This downhole pneumatic turbine motor is a high-torque, low-rotational-speed motor.

Well Completion

In general, well completion procedures for an air- (or gas-) drilled borehole are nearly the same as those for a mud-drilled borehole.

For mud drilling operations, the depth at which a casing is to be set is usually dictated by the pore-pressure and fracture-pressure gradients. In air (or gas) drilling operations, a casing is set to the depth at which significant formation water will occur. If at all possible, casing should be set just after a significant water zone has been penetrated so that the air drilling difficulties encountered with all water influx to the annulus, can be minimized by sealing off the water zones promptly after drilling through the zone. Thus, stand-by air compressors and expensive foaming additives are used only for a short time.

In most air and gas drilling operations, open-hole well completions are common. This type of completion is consistent with low pore pressure and the desire to avoid formation damage. It is often used for gas wells where nitrogen foam fracturing stimulation is necessary to provide production. In oil wells drilled with natural gas as the drilling fluid, the well is often an open hole completed with a screen set on a liner hanger to control sand influx to the well.

Liners are used a great deal in completion of wells drilled with air (or gas) drilling techniques. The low pore-pressure subsurface limitations necessary to allow air (or gas) drilling give rise to minimum casing design requirements. Thus, liners can be used nearly throughout the casing program.

In many air and gas drilling operations when casing or a liner is set, the casing or liner is lowered into the dry borehole and once the bottom has been reached, the casing or liner is landed (with little or no compression on the lower part of the casing or liner string). After landing the casing or liner, the borehole is then filled with water (with appropriate additives), and cement is pumped to the annulus around the casing or liner and the water in the borehole is displaced to the surface. The cement is followed by water and the cement is allowed to set. After the cement has set, there is water inside the casing (or liner) that must be removed before air (or gas) drilling can proceed.

The following procedure is recommended for unloading and drying a borehole prior to drilling ahead on air [73]:

1. Run the drill string complete with desired bottomhole assembly and bit to bottom.
2. Start the mud pump, running as slowly as possible, to pump fluid at a rate of 1.5 to 2.0 bbl/min. This reduces fluid friction resistance pressures to a minimum and pumps at minimum standpipe pressure for circulation. The standpipe pressure (for 1.5 to 2.0 bbl/min) can be found from standard fluid hydraulic calculations.
3. Bring one compressor and booster on line to aerate the fluid pumped downhole; approximately 100 to 150 scfm/bbl of fluid should be sufficient for aeration.

   If the air volume used is too high, standpipe pressure will exceed the pressure rating of the compressor (and/or booster). Therefore, the compressor must be slowed down until air is mixed with the fluid going downhole.

   The mist pump should inject water at a rate of about 12 bbl/hr; the foam injection pump should inject about 3 gal/hr of surfactant; this binds fluid and air together for more efficient aeration.

   As the fluid column in the annulus is aerated, standpipe pressure will drop. Additional compressors (i.e., increased air volume) can then be added to further lighten the fluid column and unload the hole.

   Compared to the slug method of unloading the hole, the aeration method is more efficient. The slug method consists of alternately pumping first air (injected up to an arbitrary maximum pressure) and then water (to reduce the pressure to an arbitrary minimum); this procedure is repeated until air can be injected continuously. The aeration method takes less time, causes no damage to pit walls from surges (as can happen with alternate slugs of air and water) and can generally be done at lower operating pressures.
5. After the hole has been unloaded, keep mist and foam injection pumps in operation to clean the hole of sloughing formations, providing a mist of 1.5 barrels of water per hour per inch of hole diameter and 0.5 to 4 gal of surfactant per hour, respectively.
6. At this point, begin air or mist drilling. Drill 20 to 100 ft to allow any sloughing hole to be cleaned up.
7. Once the hole has been stabilized (i.e., after sloughing), stop drilling and blow the hole with air mist to eliminate cuttings. Continue this procedure for 15 to 20 min or until the air mist is clean (i.e., shows a fine spray and white color).
8. Replace the kelly and set the bit on bottom. Since the hole is now full of air, surfactant and water will run to the bottom. Unless mixed with air and pumped up the annulus (which cannot be done if the drill bit is

above the surfactant-water mixture), the surfactant mixture cannot be properly swept out of the hole.

9. With the bit directly on bottom, start the air down the hole. Straight air should be pumped at normal drilling volumes until the surfactant sweep comes to the surface, appearing at the end of the blooey line and foaming like shave cream.
10. Continuously blow the hole with air for about 30 min to 1 hr.
11. Begin drilling. After 5 or 10 ft have been drilled, the hole should dust (although it is sometimes necessary to drill 60 to 90 ft before dust appears at the surface). If the hole does not dust after these steps have been carried out, pump another surfactant slug around. If dusting cannot be achieved, mist drilling may be required to complete the operation.

    Depending on the hole depth, the entire procedure requires 2 to 6 hr. Holes of over 11,000 ft have been successfully unloaded using the aeration method. A well can be dusted, mist-drilled, dried up and returned to dust drilling. To dry a hole properly, it is important that it be kept clean. Drying agents have been tried but without much success. The best drying agent available at the present time is the formation itself.
12. It should be noted that when drilling with natural gas as the drilling fluid from a pipeline source with limited pressure, nitrogen is often used to unload the hole.

It is quite desirable to place water into an open-hole section prior to running a casing or liner string and cementing. The water will provide a hydraulic head to hold back any formation gas in the open-hole section that could cause a fire hazard at the rig floor.

However, if the operator feels that the open-hole section would slough badly if water were placed in the hole, then the casing or liner string may have to be run into the dry open hole. This means that great care must be taken in running a casing or liner string into the open-hole section if the subsurface formations are making gas.

There are procedures that can be followed to allow the safe placement of casing or liner string in a dry open-hole section that is making gas. Figures 4-188 and 4-189 show the typical blowout prevention (BOP) stack arrangements used for air (or gas) drilled boreholes [74]. Figure 4-188 shows the BOP stack arrangement for rotary rigs that have rather high cellars. Such a rig would be appropriate for drilling beyond 8,000 ft of depth. Figure 4-189 shows a more typical BOP stack arrangement for rotary rigs used in air and gas drilling operations. These rigs typically drill boreholes of 8,000 ft in depth or less. These low cellar rigs cannot fit the larger BOP stack (i.e., the one shown in Figure 4-188) into the cellar. Small BOP stacks shown in Figure 4-189 give rise to safety problems when lowering casing or liners into the borehole during well completion operations. The safety problems arise when a well, which is making gas, is allowed to be opened to the surface when running a casing or liner string. If proper procedures are not used when the stripper rubber is changed to accommodate the outside diameter changes in drill pipe or casing or liner, formation gas can escape to the rig floor where it can be ignited.

An example of the typical safety problem that can occur when completing these wells is when a $6\frac{1}{4}$ in. open hole has been drilled below the last casing (or liner) shoe. The open section of the borehole is usually to be cased with a $4\frac{1}{2}$ in. liner. The liner is to be lowered into the open hole on a liner hanger that is made up to $3\frac{1}{2}$ in. drill pipe. It is assumed that prior to the liner operation, drilling had been under way; thus the stripper rubber in the rotating

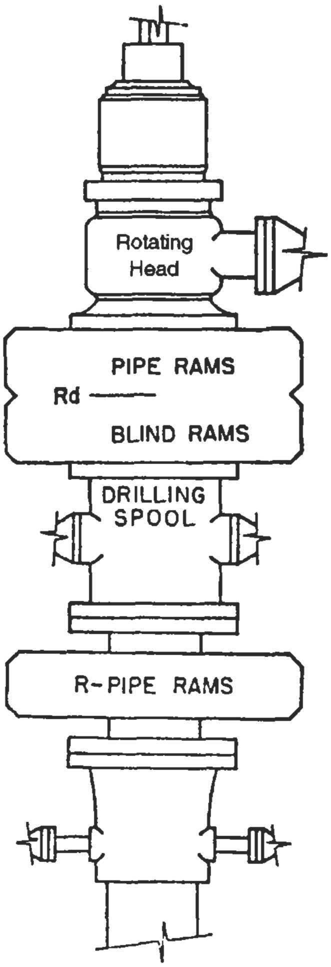


Figure 4-188. BOP stack arrangment for high cellar rigs.

head and the pipe rams are appropriate for the $3\frac{1}{2}$ in. drill pipe. Therefore, as the drill string is raised to the surface, the stripper rubber in the rotating head and the pipe ram are available to limit formation gas from coming to the rig floor. Usually the pipe rams are not employed, and protection of the rig and its crew from the formation gas is dependent upon stripper rubber in the rotating head. When drilling a $6\frac{1}{4}$ in. borehole with air, the drill collars employed are normally $4\frac{3}{4}$ in.

Beginning with the removal of the drill string, the proper procedure for placing the liner in the example borehole is:

1. Use the $3\frac{1}{2}$ in. stripper rubber in the rotating head to strip over the $4\frac{3}{4}$ in. drill collars until the drill bit is above the blind rams (but below the stripper rubber in the rotating head).

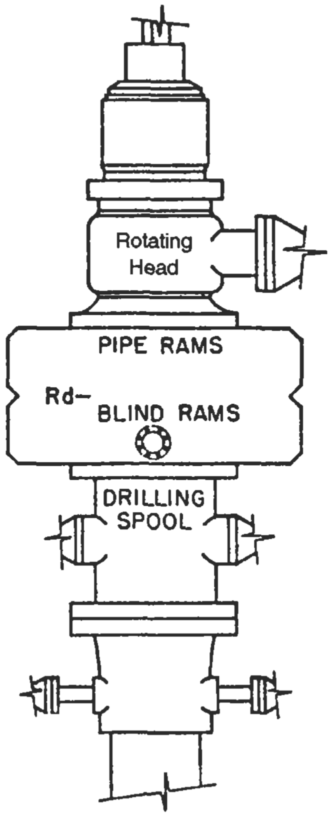


Figure 4-189. BOP stack arrangement for low cellar rig.

2. Close the blind rams, open the rotating head to release the $3\frac{1}{2}$ in. stripper rubber, pull the rotating table bushings and raise the last joint of drill collar, remove the bit, and lay this joint down (remove the $3\frac{1}{2}$ in. stripper rubber from this drill collar joint for later use).
3. Place the new $4\frac{1}{2}$ in. stripper rubber for the rotating head on the first joint of $4\frac{1}{2}$ in. liner joint. This liner joint should have the casing shoe and float valve made up to the bottom end. Lower this first joint of liner through the rotary table opening until the $4\frac{1}{2}$ in. stripper rubber can be secured in the rotating head. Once the $4\frac{1}{2}$ in. stripper rubber is in place in the rotating head, secure the stripper rubber by closing the rotating head.
4. Open the blind rams, and continue to lower the $4\frac{1}{2}$ in. liner through the rotary table opening using the casing slips.
5. Make up the liner hanger to the last liner joint while holding the liner string in the casing slips. Make up the first joint of the $3\frac{1}{2}$ in. drill pipe to the liner hanger, remove the casing slips and lower the liner string on the $3\frac{1}{2}$ in. drill pipe into the borehole until the $3\frac{1}{2}$ in. drill pipe is adjacent to the $3\frac{1}{2}$ in. pipe rams, close the $3\frac{1}{2}$ in. pipe rams onto the $3\frac{1}{2}$ in. drill pipe and replace the rotary table bushings. Place at least three joints of

$3\frac{1}{2}$ in. drill pipe onto the liner string, stripping with the pipe rams and using the rotary slips as the drill pipe goes into the hole.

6. With the $3\frac{1}{2}$ in. pipe rams closed on the $3\frac{1}{2}$ in. drill pipe, open rotating head and pull the rotary bushings. While stripping with the pipe rams, raise the liner string (with the $3\frac{1}{2}$ in. drill pipe) until the $4\frac{1}{2}$ in. stripper rubber and the drill pipe joint it is on are above the rotary table. Replace the rotary table bushings and, using the rotary slips, remove this drill pipe joint with the $4\frac{1}{2}$ in. stripper rubber (the $4\frac{1}{2}$ in. stripper rubber can be removed from this joint later).
7. Pick up a joint of $3\frac{1}{2}$ in. drill pipe with a $3\frac{1}{2}$ in. stripper rubber attached. Make up this $3\frac{1}{2}$ in. drill pipe joint to the liner string's upper drill pipe joint being held in the rotary slips.
8. Remove the rotary bushings and, while stripping with the pipe rams, lower the liner string until the $3\frac{1}{2}$ in. stripper rubber is in the rotating head. Close the rotating head to secure the $3\frac{1}{2}$ in. stripper rubber.
9. Replace the rotary bushings, open the pipe rams and continue to lower the liner string with the $3\frac{1}{2}$ in. drill pipe.

The above procedure allows for the maximum protection against formation gas escaping to the rig floor. There is basically no time at which the hole is exposed directly to the rig floor.

This rather complicated procedure has been necessary because the drilling rig can only accommodate the BOP stack that has only one set of pipe rams and a blind ram (i.e., Figure 4-188). It is obvious if the rig could accommodate the taller BOP stack with two pipe rams (namely Figure 4-187), one pipe ram could be for $3\frac{1}{2}$ in. drill pipe and the other for the $4\frac{1}{2}$ in. liner. Such an arrangement would greatly simplify the safety procedures necessary for placing the $4\frac{1}{2}$ in. liner into the borehole.

The above example is simply one of the safety problems that must be faced with the completion of wells drilled with air (or gas) and making formation gas. The principle behind the above procedure is to eliminate or limit the exposure of the rig floor to the dangerous, potentially explosive formation gas.

When a well is making (formation) gas, the rig crews must be constantly alert to safety and proper safety procedures, and rules must be followed. Nearly all drilling companies strictly forbid any smoking material, lighters or matches to be taken into the areas where air (or gas) is being used to drill a well for hydrocarbons [75].

Well Control

A blowout, which is the continuous flow of oil or gas to the surface through the annulus, is the result of a lack of sufficient bottomhole pressure from the column of circulating fluid and proper well head equipment.

Blowout and Bottomhole Pressure

Air and Gas. In the regions where air and natural gas are used as the principal drilling fluids, the potential oil and gas production zones usually have low pore pressure, or require well stimulation techniques to yield commercial production. In these production zones, air drilling (or natural gas drilling) is continued into the production zone and the initial produced formation fluids are carried to the surface by the circulating air or natural gas. This is nearly the same situation as in mud drilling, except that in air (or gas) drilling the transit time for the initial produced formation fluids to reach the surface is much shorter. In mud drilling, the entry of formation fluids to the well is considered a kick. The well is not considered to be a blowout until the formation fluids fill the annulus and are flowing uncontrolled from the well. In air (or gas) drilling, once formation fluids enter the annulus, the well is in a blowout condition. This is due to the fact that the formation fluids that enter the well can flow to the surface immediately.

Unstable Foam (Mist). The addition of water and a foaming agent into the circulating air (or gas) fluid will slightly increase the bottomhole pressure during drilling operations. However, this slight increase in bottomhole pressure does not alter the situation with regards to the potential well control capability of this circulating fluid. Again, as above, once formation fluids enter the well, the well is in a blowout condition.

Stable Foam. When a well is drilled with stable foam as the drilling fluid, there is a back pressure valve at the blooey line. The back pressure valve allows for a continuous column of foam in the annulus while drilling operations are under way. Thus, while drilling, this foam column can have significant bottomhole pressure. This bottomhole pressure can be sufficient to counter formation pore pressure and thus control potential production fluid flow into the well annulus.

Aerated Mud. In aerated mud drilling operations, the drilling mud is injected with compressed air to lighten the mud. Therefore, at the bottom of the well in the annulus, the bottomhole pressure for an aerated mud will be less than that of the mud without aeration. However, an aerated mud drilling operation will have very significant bottomhole pressure capabilities and can easily be used to control potential production fluid flow into the well annulus.

Blowout Prevention Equipment

Air and gas, unstable foam and stable foam techniques are used almost exclusively for onshore drilling operations, rarely in offshore applications. Aerated mud, however, is used for both onshore and offshore drilling operations.

The minimum requirements for well blowout prevention equipment for drilling with air and gas, unstable foam and stable foam techniques are shown in Figures 4-187 and 4-188. The BOP stack arrangement in Figure 4-187 is for a rather standard rotary rig that will accommodate at least two sets of pipe rams in addition to the rotating head and the blind rams. Figure 4-188 shows a BOP stack arrangement used for smaller rotary rigs that do not have sufficient cellar height to accommodate the second set of pipe rams.

The minimum requirements for well blowout prevention equipment for aerated mud drilling operations are basically the same as those for normal mud drilling operations.

Air, Gas and Unstable Foam Calculations

Pertinent engineering calculations can be made for air and gas drilling operations and for unstable foam drilling operations.

Volumetric Flowrate Requirements

There is a minimum air (or gas) volume rate of flow that must be maintained in order to adequately clean the bottom of the hole of cuttings and carry these cuttings to the surface. This initial calculation depends principally upon the borehole geometry and the drilling rate.

The minimum volumetric flowrate Q (actual cfm) can be obtained from [61]

$$\frac{C_1 S(CT_s + \beta h)Q^2}{(D_h^2 - D_p^2)V_{min}^2} = \left[(P_s^2 + bT_{av}^2)e^{2ah/T_{av}} - bT_{ab}^2\right]^{0.5} \tag{4-122}$$

and

$$a = \frac{SQ + C_2 K D_h^2}{53.3Q} \tag{4-123}$$

$$b = \frac{C_3 Q^2}{(D_h - D_p)^{1.333}(D_h^2 - D_p^2)^2} \tag{4-124}$$

where D_h = inside diameter of borehole in ft
D_p = outside diameter of drill pipe in ft
S = specific gravity of gas (air is 1.0)
T_s = surface atmospheric temperature in °R
T_{av} = average temperature of flow in annulus in °R
β = geothermal gradient in °F per ft
K = drilling rate in ft/hr
V_{min} = equivalent minimum velocity of standard air for removal of cuttings (V_{min} = 3000 ft/min)
h = hole depth in ft

The constants C_1, C_2, C_3 are

$$C_1 = \left(\frac{4}{\pi}\right)^2\left(\frac{T_{sl}}{P_{sl}}\right)\left(\frac{P_s}{T_s}\right)^2 \tag{4-125}$$

$$C_2 = \frac{\frac{\pi}{4}(62.2)(2.7)}{60\left(\frac{P_s}{53.3T_s}\right)} \tag{4-126}$$

$$C_3 = \frac{0.014}{2(32.2)(3600)}\left(\frac{4}{\pi}\right)^2\left(\frac{P_s}{T_s}\right)^2 \tag{4-127}$$

where T_{sl} = temperature at sea level standard atmospheric conditions (i.e., 520°R)
P_{sl} = pressure at sea level standard atmospheric conditions (i.e., 2116.8 lb/ft^2)

Thus, C_1, C_2 and C_3 are related to the actual atmospheric surface conditions where drilling is taking place. Table 4-107 gives the values of C_1, C_2 and C_3 for

Table 4-107
Constants C1, C2 and C3 for Various Surface Locations Above Sea Level

Surface Location Above Sea Level (ft)	C_1	C_2	C_3
0	6.610	28.83	1.628×10^{-6}
2,000	5.873	30.59	1.446×10^{-6}
4,000	5.207	32.48	1.282×10^{-6}
6,000	4.612	34.52	1.136×10^{-6}
8,000	4.080	36.70	1.005×10^{-6}
10,000	3.605	39.04	0.8878×10^{-6}

various surface locations above sea level. Also Table 4-109 gives atmospheric pressure, temperature and specific weight of air for various surface locations above sea level.

Example

Find the minimum Q (actual cfm) for an air drilling operation at a surface location of 6,000 ft above sea level. Drilling is to begin at 8,500 ft of depth and continue to 10,000 ft. The borehole, from top to bottom, has nearly a uniform inside diameter of slightly larger than $8\frac{3}{4}$ in. The bit to be used to drill the interval is $8\frac{3}{4}$ in. The outside diameter of drill pipe is $4\frac{1}{2}$ in. The expected drilling rate is 60 ft/hr. The geothermal gradient will be taken to be 0.01 °F/ft.

To obtain the governing minimum Q for the interval to be drilled, Equation 4-122 must be solved at 10,000 ft of depth. Since the drilling location is at a surface location of 6,000 ft above sea level, from Table 4-107, we have

$$C_1 = 4.612$$

$$C_2 = 34.52$$

$$C_3 = 1.136 \times 10^{-6}$$

Table 4-108
Atmosphere at Elevations Above Sea Level

Surface Location Above Sea Level (ft)	Pressure (psi)	Temperature (°F)	Specific Weight (lb/ft³)
0	14.696	59.00	0.0765
2,000	13.662	51.87	0.0721
4,000	12.685	44.74	0.0679
6,000	11.769	37.60	0.0639
8,000	10.911	30.47	0.0601
10,000	10.108	23.36	0.0565

Equations 4-108 and 4-109 become

$$a = \frac{(1.0)Q + 34.52(60)(0.7292)^2}{53.3Q}$$

$$b = \frac{1.136 \times 10^{-6} Q^2}{0.03835}$$

The bottomhole temperature t_{bh}(°F) is approximately (see Table 4-109)

$$t_{bh} = 37.60 + 0.01(10{,}000)$$

$$= 137.6 \text{ F}$$

Thus,

$$T_{av} = \frac{37.6 + 137.6}{2} + 460 = 547.6°R$$

Equation 4-123 becomes

$$2.003 \times 10^{-3} Q^2 = \{[(1694.7)^2 + b(547.6)^2]\ e^{2a(10{,}000)/547.6} - b(547.6)^2\}^{0.5}$$

where a and b are functions of the unknown Q and are given above.

The above equation must be solved by iteration for values of Q. The value of Q that satisfies the above equation is

Q = 2,110 actual cfm

This is the upper value of the minimum air volumetric flowrates for the interval to be drilled (i.e., 8,500 to 10,000 ft).

Surface Compressor and Booster Requirements

Once the governing minimum air volumetric flowrate has been found for an interval to be drilled, the compressors of fixed volumetric flowrate can be selected. An additional compressor is usually on site as a stand-by to have additional air available in case of unexpected downhole problems and to have a compressor available in the event one of the operational compressors needs to be shut down for maintenance.

The number of fixed volumetric flowrate compressors is selected such that the necessary minimum air volumetric flowrate is exceeded. The air volumetric flowrate that the compressors produce is shown as the real air volumetric flowrate. This real air volumetric flowrate, Q_r (actual cfm) is used to calculate the bottomhole pressure. Bottomhole pressure, P_b (lb/ft^2 abs) is determined by

$$P_b = \left[(P_s^2 + bT_{av}^2)e^{2ah/T_{av}} - bT_{av}^2\right]^{0.5} \qquad (4\text{-}128)$$

where Q_r is used to find the new values of a and b.

Knowing the bottomhole pressure, the number of bit orifice openings and the inside diameter of these openings, the pressure inside the drill pipe just above the bit and the surface injection pressure can be found.

The total area of the orifice openings $A_n(ft^2)$ is

$$A_n = n\frac{\pi}{4}\left(\frac{d_n}{12}\right)^2 \tag{4-129}$$

where n = number of orifices
d_n = diameter of the orifice openings in in.

The weight rate of airflow through the system G(lb/s) is

$$G = \frac{Q\gamma_s}{60} \tag{4-130}$$

where γ_s = specific weight of air at the surface location in lb/ft^3

The pressure above the bit P_a (lb/ft^2 abs) can be found from

$$G = A_n\left\{\frac{2gk}{k-1}P_b\gamma_b\left[\left(\frac{P_a}{P_b}\right)^{(k-1)/k} - 1\right]\right\}^{0.5} \tag{4-131}$$

where k = ratio of specific heat of air (or gas) for air k = 1.4
g = acceleration of gravity (32.2 ft/s^2)

The surface injection pressure P_i(lb/ft^2 abs) can be determined if knowing the pressure above the bit. The surface injection pressure is determined from

$$P_i = \left[\frac{P_a^2 + b'T_{av}^2(e^{2a'h/T_{av}} - 1)}{e^{2a'h/T_{av}}}\right]^{0.5} \tag{4-132}$$

and

$$a' = \frac{S}{53.3} \tag{4-133}$$

$$b'\frac{C_3Q^2}{D_{ip}^{5.333}} \tag{4-134}$$

where D_{ip} = inside diameter of the drill pipe, ft

Example 1

Using the data given below and results from the Example on p. 856, determine the real air volumetric flowrate and the expected surface injection pressure. The

$8\frac{3}{4}$ in. bit will have three open orifices and each orifice has an opening of 0.80 in. in diameter. The inside diameter of drill pipe is 3.640 in. The surface compressors and booster available at the drilling location are

Compressor (primary)
  Atlas Copco PN1200
  Positive displacement screw-type, three stages
  700 HP
  1200 actual cfm
  Fixed maximum pressure, 300 psig
  Turbocharged
  $e_m \cong 80\%$
Booster (secondary)
  Joy WB-12
  Positive displacement, piston-type, three stages
  500 HP
  Maximum pressure, 1500 psig
  Naturally aspirated
  $e_m \cong 80\%$

The minimum volumetric flowrate has been found to be Q_{min} = 2,110 actual cfm. Therefore, the real air volumetric flowrate must be provided by two of the compressors given above. Thus the Q_r will be

$$Q_r = 2(1200)$$

$$= 2400 \text{ actual cfm}$$

Using the above Q_r, Equations 4-123 and 4-124 are

$$a = 0.0274$$

$$b = 170.62$$

From Equation 4-128 the bottomhole pressure is

$$P_b = \{[(1694.7)^2 + 170.62(547.6)^2]e^{2(0.0274)(10{,}000)/547.6} - 170.62(547.6)^2\}^{0.5}$$

$$= 9789.1 \text{ lb/ft}^2 \text{ abs}$$

or

$$p_b = 68.0 \text{ psia}$$

The specific weight of the air at the bottom of the hole is

$$\gamma_b = \frac{P_b}{RT_b} = \frac{9789.1}{53.3(597.6)} = 0.3073 \text{ lb/ft}^3$$

From Equation 4-129 the total area of the orifice openings is

$$A_n = (3)\frac{\pi}{4}\left(\frac{0.8}{12}\right)^2 = 0.010472\text{ft}^2$$

From Equation 4-130 the weight rate of flow through the system is

$$G = \frac{2400(0.0639)}{60} = 2.556\,\text{lb/s}$$

The pressure above the bit can be found from Equation 4-131. Equation 4-131 is

$$2.556 = 0.010472\left[\frac{2(32.2)(1.4)}{0.4}(9789.1)(0.3073)\left\{\left(\frac{P_a}{9789.1}\right)^{0.2857} - 1\right\}\right]^{0.5}$$

Solving the above

P_a = 13,144.9 lb/ft^2 abs

or

p_a = 91.3 psia

Equations 4-133 and 4-134 are

a′ = 0.0188

b′ = 3790.7

Equation 4-132 is

$$P_i = \left[\frac{(13,144.9)^2 + 3790.7(547.6)^2(e^{2(0.0188)(10,000)/547.6} - 1)}{e^{2(0.0188)(10,000)/547.6}}\right]^{0.5}$$

$$= 25,526.4\,\text{lb/ft}^2\text{ psia}$$

or

p_i = 177.3 psia

The above injection pressure is the expected standpipe pressure when drilling at 10,000 ft of depth. The injection pressure will be somewhat less than the above when drilling the upper portion of the interval (i.e. at 8500').

Thus the primary compressors will have sufficient pressure capability to drill the interval from 8,500 to 10,000 ft. A third primary compressor should be on site and hooked up for immediate service in the event of downhole problems or the necessity to shut down one of the operating compressors. Also, the booster should be hooked up for immediate service in the event of downhole problems. For more information and engineering calculations pertaining to compressors and boosters see reference 64.

Injected Water Requirements and Formation Water

If water-bearing formations are expected while drilling with air, then it is necessary to make sure the air entering the bottom of the annulus section of the borehole is saturated with moisture. If the circulating air is saturated then the air will not lose internal energy absorbing the formation water. The loss of internal energy would affect its potential to expand and thus reduce the kinetic energy of air flow in the annulus. The loss of kinetic energy will reduce the lifting capability of the circulating air.

Once the circulating air is saturated and it enters the borehole annulus, the air will carry the formation water as droplets. Thus the formation water will be carried to the surface in much the same manner as the rock cuttings.

To saturate the circulating air with water so that the air cannot absorb formation water, the water must be injected into the compressed air at the surface prior to the standpipe (see Figure 4-185). If only water is injected into the circulating air, then the drilling operation is called mist drilling. Usually, however, a foaming agent (or surfactant) is injected with the water. This allows a foam to be created in the annulus, which aids in transportation of the cuttings to the surface. These foaming agents are pumped together with the water injected on the basis of about 0.2% of the injected and projected formation water. When water and a foaming agent are injected the drilling operation is called unstable foam drilling.

The volumetric flowrate of water to be injected into the compressed air depends upon the saturation pressure of the water vapor at the bottomhole temperature. The saturation pressure p_{sat} (psia) at the bottom of the hole depends only on the bottomhole temperature and is given by [76,77].

$$\log_{10} p_{sat} = 6.39416 - \frac{1750.286}{217.23 + 0.555 t_b} \tag{4-135}$$

where t_b = bottomhole temperature (°F)

Knowing the saturation pressure of the water vapor, the amount of injected water can be determined. The amount of injected water, q_i (gal/hr), to provide saturated air at bottomhole conditions is

$$q_i = 269.17 \left(\frac{p_s}{p_b - p_s} \right) G \tag{4-136}$$

where p_b = bottomhole pressure in psia
G = weight rate of flow of (dry) air in lb/s

Example

Using the data and results from the Examples on pp. 856 to 859, determine the approximate amount of surface injected water and foaming agent needed to saturate the air at bottomhole conditions and to provide an unstable foam in the annulus of the borehole.

The saturation pressure can be found from substitution of the bottomhole temperature of 137.6°F into Equation 4-135. This yields

$$\log_{10} p_{sat} = 0.4327$$

$$p_{sat} = 2.708 \text{ psia}$$

The volumetric flowrate of injected water is determined from Equation 4-136, which yields

$$q_i = 269.17\left(\frac{2.708}{68.0 - 2.708}\right)(2.556) = 28.5 \text{ gal/hr}$$

The approximate volumetric rate of foaming agent q_f (lb/hr) injected will be

$$q_f = 0.002(28.5)$$

$$= 0.06 \text{ gal/hr}$$

However, this foaming agent injected is only a small part of what should be injected to foam the anticipated formation water which may enter the annulus. This will be covered in the next Example.

If the circulating air has been saturated, then any formation water entering the annulus will be carried to the surface as droplets and will not reduce the temperature of the air and thereby reduce kinetic energy of the air as it expands in the annulus. The amount of formation water that can be carried from the borehole annulus by the real amount of air circulating Q_r is directly related to the additional air that is being circulated above that minimum value Q_{min} necessary to clean the hole of rock cuttings.

To calculate the amount of water that can be carried from the hole, Q_r is substituted into Equation 4-123 and the potential drilling rate such an air volumetric flowrate can support. The additional drilling rate that can be supported by Q_r is actually the weight of formation water (per hour) that can be removed from the borehole. Therefore, once the potential drilling rate is obtained for Q_r, then the formation water that can be taken in the borehole annulus per hour q_w (gal/hr) while maintaining the normal drilling rate will be

$$q_w = \frac{\pi}{4} D_h^2 (K_p - K_a) \frac{(62.4)(2.7)}{8.33} \tag{4-137}$$

where K_p = potential drilling rate for Q_r in ft/hr
K_a = actual drilling rate in ft/hr

Example

Using the data and results from the previous Examples, determine the approximate volumetric flowrate of formation water into the annulus of the well that can be removed by the actual air circulation rate of 2,400 actual cfm. Also, determine the total amount of foaming agent which should be injected into the circulating air.

Substitute into Equations 4-122, 4-123, and 4-124 the previous example values and let

$$Q = Q_r = 2{,}400 \text{ actual cfm}$$

Equation 4-122 becomes

$$2.003\times10^{-3}(2400)^2$$
$$=\left\{\left[(1694.7)^2+170.62(547.6)^2\right]e^{2a\,10{,}000/547.6}-170.62(547.6)^2\right\}^{0.5}$$

Equation 4-123 becomes (with $k = K_p$)

$$a = 0.0187617 + 0.00014349K_p$$

From the above two equations the potential drilling rate K_p for a Q_r = 2,400 actual cfm is found to be

$$K_p = 103.3 \text{ ft/hr}$$

Substitution of the above into Equation 4-137 yields

$$q_w = \frac{\pi}{4}\left(\frac{8.75}{12}\right)^2(103.3-60)\frac{(62.4)(2.7)}{8.33} = 365.7 \text{ gal/hr}$$

The total volumetric flowrate of foaming agent that should be injected into the circulating air is

$$q_w \cong 0.06 + 0.002(365.7)$$
$$\cong 0.80 \text{ gal/hr}$$

DOWNHOLE MOTORS

Background

In 1873, an American, C. G. Cross, was issued the first patent related to a downhole turbine motor for rotating the drill bit at the bottom of a drillstring with hydraulic power [78]. This drilling concept was conceived nearly 30 years before rotary drilling was introduced in oil well drilling. Thus the concept of using a downhole motor to rotate or otherwise drive a drill bit at the bottom of a fluid conveying conduit in a deep borehole is not new.

The first practical applications of the downhole motor concept came in 1924 when engineers in the United States and the Soviet Union began to design, fabricate and field test both single-stage and multistage downhole turbine motors [79]. Efforts continued in the United States, the Soviet Union and elsewhere in Europe to develop an industrially reliable downhole turbine motor that would operate on drilling mud. But during the decade to follow, all efforts proved unsuccessful.

In 1934 in the Soviet Union a renewed effort was initiated to develop a multistage downhole turbine motor [79–81]. This new effort was successful. This development effort marked the beginning of industrial use of the downhole turbine motor. The Soviet Union continued the development of the downhole turbine motor and utilized the technology to drill the majority of its oil and gas wells. By the 1950s the Soviet Union was drilling nearly 80% of their wells with the downhole turbine motors using surface pumped drilling mud or freshwater as the activating hydraulic power.

In the late 1950s, with the growing need in the United States and elsewhere in the world for directional drilling capabilities, the drilling industry in the United States and elsewhere began to reconsider the downhole turbine motor technology. There are presently three service companies that offer downhole turbine motors for drilling of oil and gas wells. These motors are now used extensively throughout the world for directional drilling operations and for some straight-hole drilling operations.

The downhole turbine motors that are hydraulically operated have some fundamental limitations. One of these is high rotary speed of the motor and drill bit. The high rotary speeds limit the use of downhole turbine motors when drilling with roller rock bits. The high speed of these direct drive motors shortens the life of the roller rock bit.

In the 1980s in the United States an effort was initiated to develop a downhole turbine motor that was activated by compressed air. This motor was provided with a gear reducer transmission. This downhole pneumatic turbine has been successfully field tested [82].

The development of positive displacement downhole motors began in the late 1950s. The initial development was the result of a United States patent filed by W. Clark in 1957. This downhole motor was based on the original work of a French engineer, René Monineau, and is classified as a helimotor. The motor is actuated by drilling mud pumped from the surface. There are two other types of positive displacement motors that have been used, or are at present in use today: the vane motor and the reciprocating motor. However, by far the most widely used positive displacement motor is the helimotor [79,83].

The initial work in the United States led to the highly successful single-lobe helimotor. From the late 1950s until the late 1980s there have been a number of other versions of the helimotor developed and fielded. In general, most of the recent development work in helimotors has centered around multilobe motors. The higher the lobe system, the lower the speed of these direct drive motors and the higher the operating torque.

There have been some efforts over the past three decades to develop positive development vane motors and reciprocating motors for operation with drilling mud as the actuating fluid. These efforts have not been successful.

In the early 1960s efforts were made in the United States to operate vane motors and reciprocating motors with compressed air. The vane motors experienced some limited test success but were not competitive in the market of that day [84]. Out of these development efforts evolved the reciprocating (compressed) air hammers that have been quite successful and are operated extensively in the mining industry and have some limited application in the oil and gas industry [85]. The air hammer is not a motor in the true sense of rotating equipment. The reciprocating action of the air hammer provides a percussion effect on the drill bit, the rotation of the bit to new rock face location is carried out by the conventional rotation of the drill string.

In this section the design and the operational characteristics and procedures of the most frequently used downhole motors will be discussed. These are the downhole turbine motor and the downhole positive displacement motor.

Turbine Motors

Figure 4-190 shows the typical rotor and stator configuration for a single stage of a multistage downhole turbine motor section. The activating drilling mud or freshwater is pumped at high velocity through the motor section, which, because of the vane angle of each rotor and stator (which is a stage), causes the rotor to rotate the shaft of the motor. The kinetic energy of the flowing drilling mud is converted through these rotor and stator stages into mechanical rotational energy.

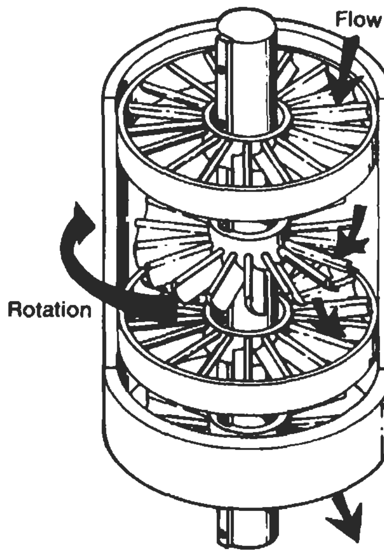


Figure 4-190. Basic turbine motor design principle. *(Courtesy Smith International, Inc.)*

Design

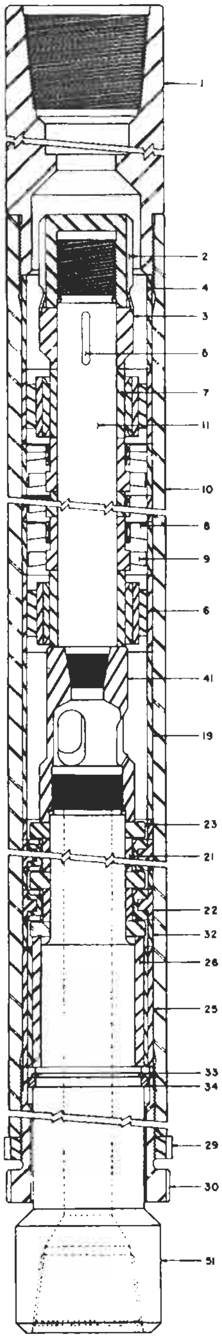
The rotational energy provided by the flowing fluid is used to rotate and provide torque to the drill bit. Figure 4-191 shows the typical complete downhole turbine motor actuated with an incompressible drilling fluid.

In general, the downhole turbine motor is composed of two sections: (1) the turbine motor section and; (2) the thrust-bearing and radial support bearing. These sections are shown in Figure 4-191. Sometimes a special section is used at the top of the motor to provide a filter to clean up the drilling mud flow before it enters the motor, or to provide a by-pass valve.

The turbine motor section has multistages of rotors and stators, from as few as 25 to as many as 300. For a basic motor geometry with a given flowrate, an increase in the number of stages in the motor will result in an increase in torque capability and an increase in the peak horsepower. This performance improvement, however, is accompanied by an increase in the differential pressure through the motor section (see Table 4-109). The turbine motor section usually has bearing groups at the upper and lower ends of the rotating shaft (on which are attached the rotors). The bearing groups only radial load capabilities.

The lower end of the rotating shaft of the turbine motor section is attached to the upper end of the main shaft. The drilling fluid after passing through the turbine motor section is channeled into the center of the shaft through large openings in the main shaft. The drill bit is attached to the lower end of the main shaft. The weight on the bit is transferred to the downhole turbine motor housing via the thrust-bearing section. This bearing section provides for rotation while transferring the weight on the bit to the downhole turbine motor housing.

In the thrust-bearing section is a radial support bearing section that provides a radial load-carrying group of bearings that ensures that the main shaft rotates



Item No.	Description
1	Top Sub
2	Shaft Cap
3	Lockwasher-Turbine Section
4	Stator Spacer
5	Shaft Key-Turbine Section
6	Intermediate Bearing Body
7	Intermediate Bearing Sleeve
8	Stator - Assembly
9	Rotor - Assembly
10	Turbine Housing
11	Turbine Shaft
19	Spacer-Bearing Section
21	Thrust Bearing Sleeve
22	Thrust Bearing Body
23	Thrust Disc
25	Lower Bearing Body
26	Lower Bearing Sleeve
29	Lower Sub Lock Ring
30	Lower Sub
31	Bearing Shaft
32	Lower Bearing Spacer
33	Retaining Ring
34	Catch Ring
40	Float Retainer Ring
41	Shaft Coupling
45	Shaft Cap Lock Screw
46	Eastco Float
	Turbodrill Complete
	[1]Optional, order by top sub too
	Repair Accessories
	Rubber Lubricant
	Assembly Compound
	Joint Compound
	Retaining Ring Pliers-External
	Retaining Ring Pliers-Internal
	¼" Lock Screw Wrench
	Eastco Float Repair Kit

Figure 4-191. Downhole turbine motor design. *(Courtesy Eastman-Christensen Co.)*

Table 4-109
Turbine Motor, $6^3/_4$-in. Outside Diameter, Circulation Rate 400 gpm, Mud Weight 10 lb/gal

Number of Stages	Torque* (ft-lbs)	Optimum Bit Speed (rpm)	Differential Pressure (psi)	Horsepower*	Thrust Load (1000 lbs)
212	1412	807	1324	217	21
318	2118	807	1924	326	30

*At optimum speed
Courtesy Eastman-Christensen

about center even when a side force on the bit is present during directional drilling operations.

There are of course variations on the downhole turbine motor design, but the basic sections discussed above will be common to all designs.

The main advantages of the downhole turbine motor are:

1. Hard to extremely hard competent rock formations can be drilled with turbine motors using diamond or the new polycrystalline diamond bits.
2. Rather high rates of penetration can be achieved since bit rotation speeds are high.
3. Will allow circulation of the borehole regardless of motor horsepower or torque being produced by the motor. Circulation can even take place when the motor is installed.

The main disadvantages of the downhole turbine motor are:

1. Motor speeds and, therefore, bit speeds are high, which limits the use of roller rock bits.
2. The required flowrate through the downhole turbine motor and the resulting pressure drop through the motor require large surface pump systems, significantly larger pump systems than are normally available for most land and for some offshore drilling operations.
3. Unless a measure while drilling instrument is used, there is no way to ascertain whether the turbine motor is operating efficiently since rotation speed and/or torque cannot be measured using normal surface data (i.e., standpipe pressure, weight on bit, etc.).
4. Because of the necessity to use many stages in the turbine motor to obtain the needed power to drill, the downhole turbine motor is often quite long. Thus the ability to use these motors for high-angle course corrections can be limited.
5. Downhole turbine motors are sensitive to fouling agents in the mud; therefore, when running a turbine motor steps must be taken to provide particle-free drilling mud.
6. Downhole turbine motors can only be operated with drilling mud.

Operations

Figure 4-192 gives the typical performance characteristics of a turbine motor. The example in this figure is a $6\frac{3}{4}$-in. outside diameter turbine motor having 212 stages and activated by a 10-lb/gal mud flowrate of 400 gal/min.

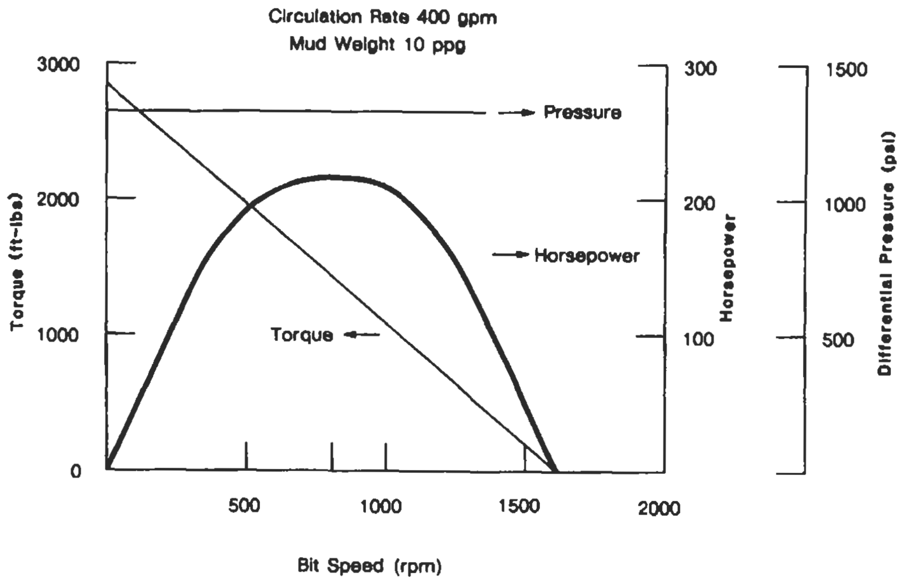


Figure 4-192. Turbine motor, 6¾-in. outside diameter, two motor sections, 212 stages, 400 gal/min, 10-lb/gal mud weight. *(Courtesy Eastman-Christensen.)*

For this example, the stall torque of the motor is 2,824 ft-lb. The runaway speed is 1,614 rpm and coincides with zero torque. The motor produces its maximum horsepower of 217 at a speed of 807 rpm. The torque at the peak horsepower is 1,412 ft-lb, or one-half of the stall torque.

A turbine device has the unique characteristic that it will allow circulation independent of what torque or horsepower the motor is producing. In the example where the turbine motor has a 10-lb/gal mud circulating at 400 gal/min, the pressure drop through the motor is about 1,324 psi. This pressure drop is approximately constant through the entire speed range of the motor.

If the turbine motor is lifted off the bottom of the borehole and circulation continues, the motor will speed up to the runaway speed of 1,614 rpm. In this situation the motor produces no drilling torque or horsepower.

As the turbine motor is lowered and weight is placed on the motor and thus the bit, the motor begins to slow its speed and produce torque and horsepower. When sufficient weight has been placed on the turbine motor, the example motor will produce its maximum possible horsepower of 217. This will be at a speed of 807 rpm. The torque produced by the motor at this speed will be 1,412 ft-lb.

If more weight is added to the turbine motor and the bit, the motor speed and horsepower output will continue to decrease. The torque, however, will continue to increase.

When sufficient weight has been placed on the turbine motor and bit, the motor will cease to rotate and the motor is described as being stalled. At this condition, the turbine motor produces its maximum possible torque. Even when the motor is stalled, the drilling mud is still circulating and the pressure drop is approximately 1,324 psi.

The stall torque M_s (ft-lb) for any turbine motor can be determined from [86]

$$M_s = 1.38386 \times 10^{-5} \frac{\eta_h \eta_m n_s \bar{\gamma}_m q^2 \tan\beta}{h} \tag{4-138}$$

where η_h = hydraulic efficiency
η_m = mechanical efficiency
n_s = number of stages
$\bar{\gamma}_m$ = specific weight of mud in lb/gal
q = circulation flowrate in gal/min
β = exit blade angle in degrees
h = radial width of the blades in in.

Figure 4-193 is the side view of a single-turbine stage and describes the geometry of the rotor and stator.

The runaway speed N_r (rpm) for any turbine motor can be determined from

$$N_r = 5.85 \frac{\eta_v q \tan\beta}{r_m^2 h} \tag{4-139}$$

where η_v = volumetric efficiency
r_m = mean blade radius in in.

The turbine motor instantaneous torque M (ft-lb) for any speed N (rpm) is

$$M = M_s\left(1 - \frac{N}{N_r}\right) \tag{4-140}$$

The turbine motor horsepower HP (hp) for any speed is

$$HP = \frac{2\pi M_s N}{33{,}000}\left(1 - \frac{N}{N_r}\right) \tag{4-141}$$

The maximum turbine motor horsepower is at the optimum speed, N_0, which is one-half of the runaway speed. This is

$$N_0 = \frac{N_r}{2} \tag{4-142}$$

Thus the maximum horsepower HP_{max} is

$$HP_{max} = \frac{\pi M_s N_r}{2(33{,}000)} \tag{4-143}$$

The torque at the optimum speed M_0 is one-half the stall torque. Thus

$$M_0 = \frac{M_s}{2} \tag{4-144}$$

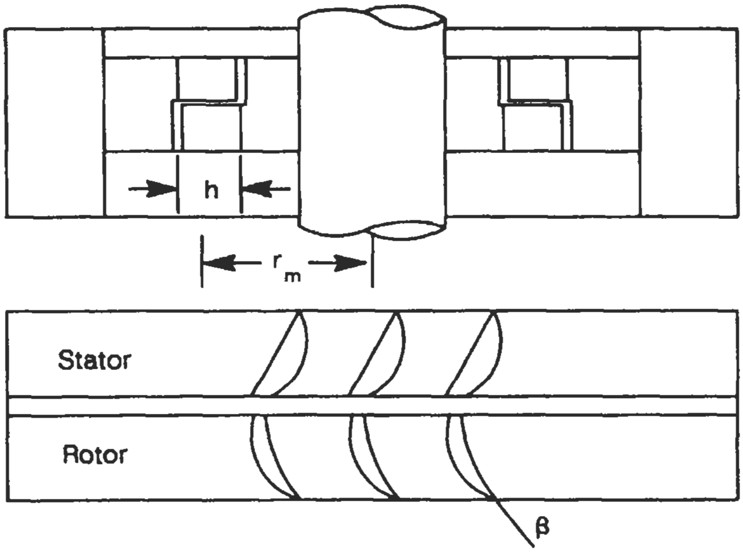


Figure 4-193. Turbine rotor and stator geometry of a single stage. *(Courtesy Smith International, Inc.)*

The pressure drop Δp (psi) through a given turbine motor design is usually obtained empirically. Once this value is known for a circulation flowrate and mud weight, the pressure drop for other circulation flowrates and mud weights can be estimated.

If the above performance parameters for a turbine motor design are known for a given circulation flowrate and mud weight (denoted as 1), the performance parameters for the new circulation flowrate and mud weight (denoted as 2) can be found by the following relationships:

Torque

$$M_2 = \left(\frac{q_2}{q_1}\right)^2 M_1 \tag{4-145}$$

$$M_2 = \left(\frac{\bar{\gamma}_2}{\bar{\gamma}_1}\right) M_1 \tag{4-146}$$

Speed

$$N_2 = \left(\frac{q_2}{q_1}\right) N_1 \tag{4-147}$$

Power

$$HP_2 = \left(\frac{q_2}{q_1}\right)^3 HP_1 \tag{4-148}$$

$$HP_2 = \left(\frac{\bar{\gamma}_2}{\bar{\gamma}_1}\right) HP_1 \tag{4-149}$$

Pressure drop

$$\Delta p_2 = \left(\frac{q_2}{q_1}\right)^2 \Delta p_1 \tag{4-150}$$

$$\Delta p_2 = \left(\frac{\bar{\gamma}_2}{\bar{\gamma}_1}\right) \Delta p_1 \tag{4-151}$$

Table 4-110 gives the performance characteristics for various circulation flowrates for the 212-stage, $6\frac{3}{4}$-in. outside diameter turbine motor described briefly in Table 4-109 and shown graphically in Figure 4-192.

Table 4-111 gives the performance characteristics for various circulation flowrates for the 318-stage, $6\frac{3}{4}$-in. outside diameter turbine motor described briefly in Table 4-109. Figure 4-194 shows the performance of the 318-stage turbine motor at a circulation flowrate of 400 gal/min and mud weight of 10 lb/gal.

The turbine motor whose performance characteristics are given in Table 4-110 is made up of two motor sections with 106 stages in each section. The turbine motor whose performance characteristics are given in Table 4-111 is made up of three motor sections.

Table 4-110
Turbine Motor, 6¾-in. Outside Diameter, Two Motor Sections, 212 Stages, Mud Weight 10 lb/gal

Circulation Rate (gpm)	Torque* (ft-lbs)	Optimum Bit Speed (rpm)	Differential Pressure (psi)	Maximum Horsepower*	Thrust Load (1000 lbs)
200	353	403	331	27	5
250	552	504	517	53	8
300	794	605	745	92	12
350	1081	706	1014	145	16
400	1421	807	1324	217	21
450	1787	908	1676	309	26
500	2206	1009	2069	424	32

*At optimum speed
Courtesy Eastman-Christensen

Table 4-111
Turbine Motor, 6¾-in. Outside Diameter, Three Motor Sections, 318 Stages, Mud Weight 10 lb/gal

Circulation Rate (gpm)	Torque* (ft-lbs)	Optimum Bit Speed (rpm)	Differential Pressure (psi)	Maximum Horsepower*	Thrust Load (1000 lbs)
200	529	403	485	40	8
250	827	504	758	79	12
300	1191	605	1092	137	17
350	1622	706	1486	218	23
400	2118	807	1941	326	30
450	2681	908	2457	464	38

*At optimum power
Courtesy Eastman-Christensen

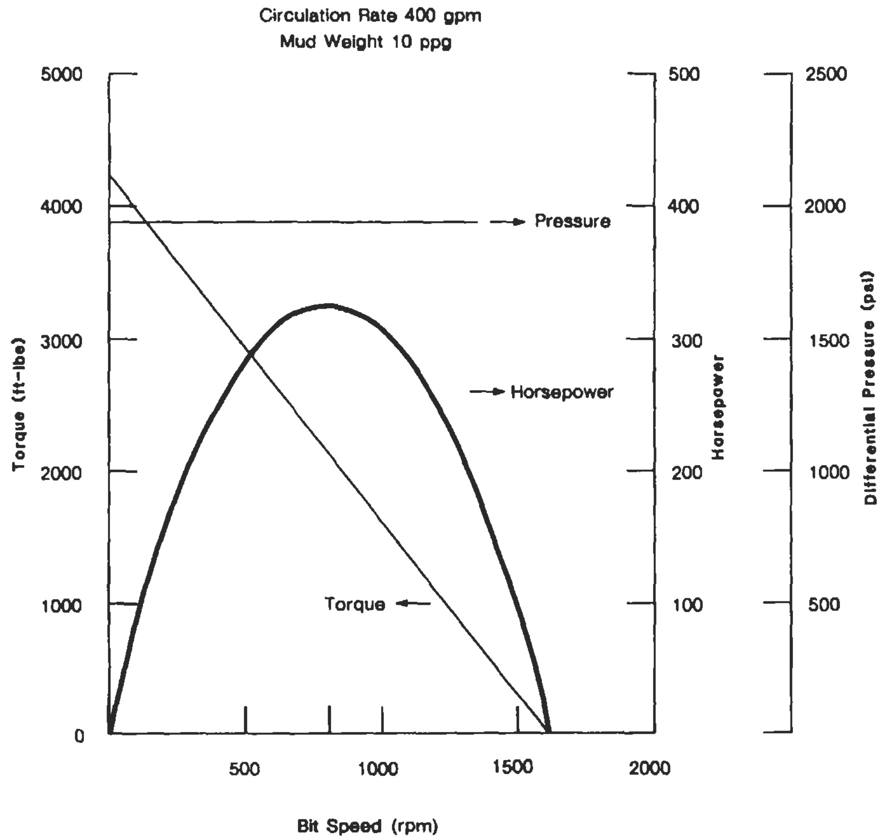


Figure 4-194. Turbine motor, 6¾-in. outside diameter, three motor sections, 318 stages, 400 gal/min, 10-lb/gal mud weight. *(Courtesy Eastman-Christensen.)*

The major reason most turbine motors are designed with various add-on motor sections is to allow flexibility when applying turbine motors to operational situations.

For straight hole drilling the turbine motor with the highest possible torque and the lowest possible speed is of most use. Thus the turbine motor is selected such that the motor produces the maximum amount of power for the lowest possible circulation flowrate (i.e., lowest speed). The high power increases rate of penetration and the lower speed increases bit life particularly if roller rock bits are used.

For deviation control drilling the turbine motor with a lower torque and the shortest overall length is needed.

Example 1

Using the basic performance data given in Table 4-110 for the $6\frac{3}{4}$-in. outside diameter turbine motor with 212 stages, determine the stall torque, maximum horsepower and pressure drop for this motor if only one motor section with 106 stages were to be used for a deviation control operation. Assume the same circulation flow rate of 400 gal/min, but a mud weight of 14 lb/gal is to be used.

Stall Torque. From Table 4-110 the stall torque for the turbine motor with 212 stages will be twice the torque value at optimum speed. Thus the stall torque for 10 lb/gal mud weight flow is

$$M_s = 2(1421)$$

$$= 2{,}842 \text{ ft-lb}$$

From Equation 4-138 it is seen that stall torque is proportional to the number of stages used. Thus the stall torque for a turbine motor with 106 stages will be (for the circulation flowrate of 400 gal/min and mud weight of 10 lb/gal)

$$M_s = 2{,}842\left(\frac{106}{212}\right)$$

$$= 1{,}421 \text{ ft-lb}$$

and from Equation 4-146 for the 14-lb/gal mud weight

$$M_s = 1{,}421\left(\frac{14}{10}\right)$$

$$= 1{,}989 \text{ ft-lb}$$

Maximum Horsepower. From Table 4-110 the maximum horsepower for the turbine motor with 212 stages is 217. From Equation 4-143 it can be seen that the maximum power is proportional to the stall torque and the runaway speed. Since the circulation flowrate is the same, the runaway speed is the same for this case. Thus, the maximum horsepower will be proportional to the stall torque. The maximum power will be (for the circulation flowrate of 400 gal/min and mud weight of 10 lb/gal)

$$HP_{max} = 217\left(\frac{1421}{2842}\right)$$

$$= 108.5$$

and from Equation 4-149 for the 14-lb/gal mud weight

$$HP_{max} = 108.5\left(\frac{14}{10}\right)$$

$$= 152$$

Pressure Drop. Table 4-110 shows that the 212-stage turbine motor has a pressure drop of 1,324 psi for the circulation flowrate of 400 gal/min and a mud weight of 10 lb/gal. The pressure drop for the 106 stage turbine motor should be roughly proportional to the length of the motor section (assuming the motor sections are nearly the same in design). Thus the pressure drop in the 106-stage turbine motor should be proportional to the number of stages. Therefore, the pressure drop should be

$$\Delta p = 1,324\left(\frac{106}{212}\right)$$

$$= 662 \text{ psi}$$

and from Equation 4-151 for the 14-lb/gal mud weight

$$\Delta p = 662\left(\frac{14}{10}\right)$$

$$= 927 \text{ psi}$$

The last column in Tables 4-110 and 4-111 show the thrust load associated with each circulation flowrate (i.e., pressure drop). This thrust load is the result of the pressure drop across the turbine motor rotor and stator blades. The magnitude of this pressure drop depends on the individual internal design details of the turbine motor (i.e., blade angle, number of stages, axial height of blades and the radial width of the blades) and the operating conditions. The additional pressure drop results in thrust, T (lb), which is

$$T = \pi r_m^2 \Delta p \qquad (4\text{-}152)$$

Example 2

A $6\frac{3}{4}$-in. outside diameter turbine motor (whose performance data are given in Tables 4-110 and 4-111) is to be used for a deviation control direction drilling operation. The motor will use a new $8\frac{1}{2}$-in. diameter diamond bit for the drilling operation. The directional run is to take place at a depth of 17,552 ft (measured depth). The rock formation to be drilled is classified as extremely hard, and it is anticipated that 10 ft/hr will be the maximum possible drilling rate. The mud weight is to be 16.2 lb/gal. The drilling rig has a National Supply Company, triplex mud pump Model 10-P-130 available. The details of this pump are given in Table 4-112 (also see the section titled "Mud Pumps" for more details). Because this is a deviation control run, the shorter two motor section turbine motor will be used.

Determine the appropriate circulation flowrate to be used for the diamond bit, turbine motor combination and the appropriate liner size to be used in the triplex pump. Also, prepare the turbine motor performance graph for the chosen circulation flowrate. Determine the total flow area for the diamond bit.

Bit Pressure Loss. To obtain the optimum circulation flowrate for the diamond bit, turbine motor combination, it will be necessary to consider the bit and the turbine motor performance at various circulation flowrates: 200, 300, 400 and 500 gal/min.

Since the rock formation to be drilled is classified as extremely hard, 1.5 hydraulic horsepower per square inch of bit area will be used as bit cleaning and cooling requirement [87]. The projected bottomhole area of the bit A_b (in.^{2}) is

$$A_b = \frac{\pi}{4}(8.5)^2$$

$$= 56.7 \text{ in.}^2$$

For a circulation flowrate of 200 gal/min, the hydraulic horsepower for the bit HP_b (hp) is

$$HP_b = 1.5\ (56.7)$$

$$= 85.05$$

The pressure drop across the bit Δp_b (psi) to produce this hydraulic horsepower at a circulation flowrate of 200 gal/min is

$$\Delta p_b = \frac{85.05(1{,}714)}{200}$$

$$= 729 \text{ psi}$$

Table 4-112
Triplex Mud Pump, Model 10-P-13 National Supply Company, Example 2

Input Horsepower 1300, Maximum Strokes per Minute, 140 Length of Stroke, 10 Inches

	Liner Size (inches)				
	5¼	5½	5¾	6	6¼
Output per Stroke (gals)	2.81	3.08	3.37	3.67	3.98
Maximum Pressure (psi)	5095	4645	4250	3900	3595

Similarly, the pressure drop across the bit to produce the above hydraulic horsepower at a circulation flowrate of 300 gal/min is

$$\Delta p_b = \frac{85.05(1,714)}{300}$$

$$= 486 \text{ psi}$$

The pressure drop across the bit at a circulation flowrate of 400 lb/gal is

$$\Delta p_b = 364 \text{ psi}$$

The pressure drop across the bit at a circulation flowrate of 500 gal/min

$$\Delta p_b = 292 \text{ psi}$$

Total Pressure Loss. Using Table 4-110 and Equations 4-150 and 4-151, the pressure loss across the turbine motor can be determined for the various circulation flowrates and the mud weight of 16.2 lb/gal. These data together with the above bit pressure loss data are presented in Table 4-113. Also presented in Table 4-113 are the component pressure losses of the system for the various circulation flowrates considered. The total pressure loss tabulated in the lower row represents the surface standpipe pressure when operating at the various circulation flowrates.

Pump Limitations. Table 4-112 shows there are five possible liner sizes that can be used on the Model 10-P-130 mud pump. Each liner size must be considered to obtain the optimum circulation flowrate and appropriate liner size. The maximum pressure available for each liner size will be reduced by a safety factor of 0.90.* The maximum volumetric flowrate available for each liner size will also be reduced by a volumetric efficiency factor of 0.80 and an additional safety factor of 0.90.** Thus, from Table 4-112, the allowable maximum pressure and allowable maximum volume, metric flowrates will be those shown in Figures 4-195 through 4-199, which are the liner sizes $5\frac{1}{4}$, $5\frac{1}{2}$, $5\frac{3}{4}$, 6 and $6\frac{1}{4}$ in., respectively. Plotted on each of these figures are the total pressure losses for the various circulation flowrates considered. The horizontal straight line on each figure is the allowable maximum pressure for the particular liner size. The vertical straight line is the allowable maximum volumetric flowrate for the particular liner size. Only circulation flowrates that are in the lower left quadrant of the figures are practical. The highest circulation flowrate (which produces the highest turbine motor horsepower) is found in Figure 4-197, the $5\frac{3}{4}$-in. liner. This optimal circulation flowrate is 340 gal/min.

Turbine Motor Performance. Using the turbine motor performance data in Table 4-110 and the scaling relationships in Equations 4-145 through 4-151, the performance graph for the turbine motor operating with a circulation flowrate of 340 gal/min and mud weight of 16.2 lb/gal can be prepared. This is given in Figure 4-200.

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*This safety factor is not necessary for new, well-maintained equipment.

**The volumetric efficiency factor is about 0.95 for precharged pumps.

Table 4-113
Drill String Component Pressure Losses at Various Circulation Flowrates for Example 2

	Pressure (psi)			
Components	**200 gpm**	**300 gpm**	**400 gpm**	**500 gpm**
Surface Equipment	4	11	19	31
Drill Pipe Bore	460	878	1401	2021
Drill Collar Bore	60	117	118	272
Turbine Motor	536	1207	2145	3352
Drill Bit	729	486	364	292
Drill Collar Annulus	48	91	144	207
Drill Pipe Annulus	133	248	391	561
Total Pressure Loss	1970	3038	4652	6736

Total Flow Area for Bit. Knowing the optimal circulation flowrate, the actual pressure loss across the bit can be found as before in the above. This is

$$\Delta p_b = \frac{85.05(1,714)}{340}$$

$$= 429 \text{ psi}$$

With the flowrate and pressure loss across the bit, the total flow area of the diamond bit A_{tf} (in.^{2}) can be found using [88]

$$\Delta p_b = \frac{q^2 \bar{\gamma}_m}{8795\left(A_{tf} e^{-0.832} \dfrac{ROP}{N_b}\right)^2} \tag{4-153}$$

where ROP = rate of penetration in ft/hr
N_b = bit speed in rpm

The bit speed will be the optimum speed of the turbine motor, 685 rpm. The total flow area A_{tf} for the diamond bit is

$$A_{tf} = \left[\frac{(340)^2 16.2}{8795(429)}\right]^{1/2} \frac{1}{0.9879}$$

$$= 0.713 \text{ in.}^2$$

(text continued on page 882)

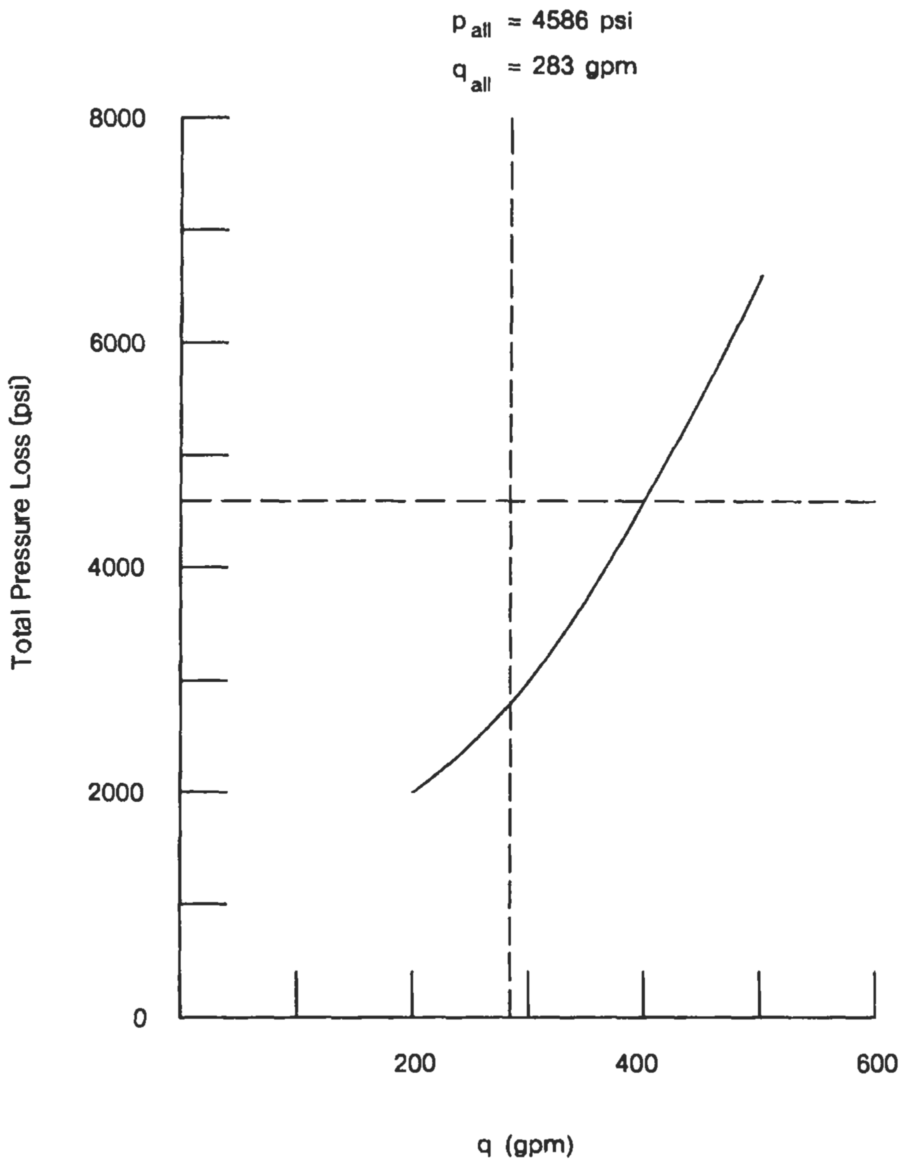


Figure 4-195. 5¼-in. liner, total pressure loss versus flowrate, Example 2. *(Courtesy Smith International, Inc.)*

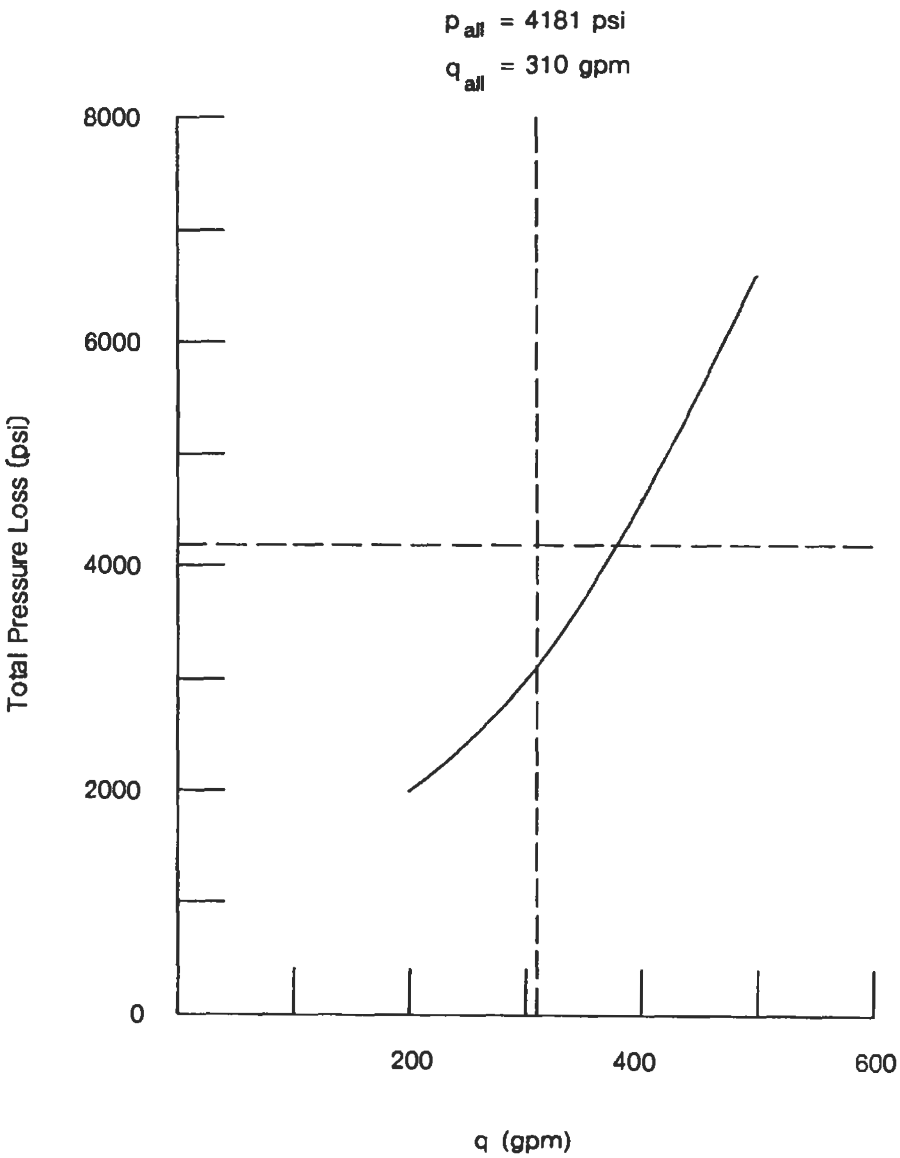


Figure 4-196. 5½-in. liner, total pressure loss versus flowrate, Example 2. *(Courtesy Smith International, Inc.)*

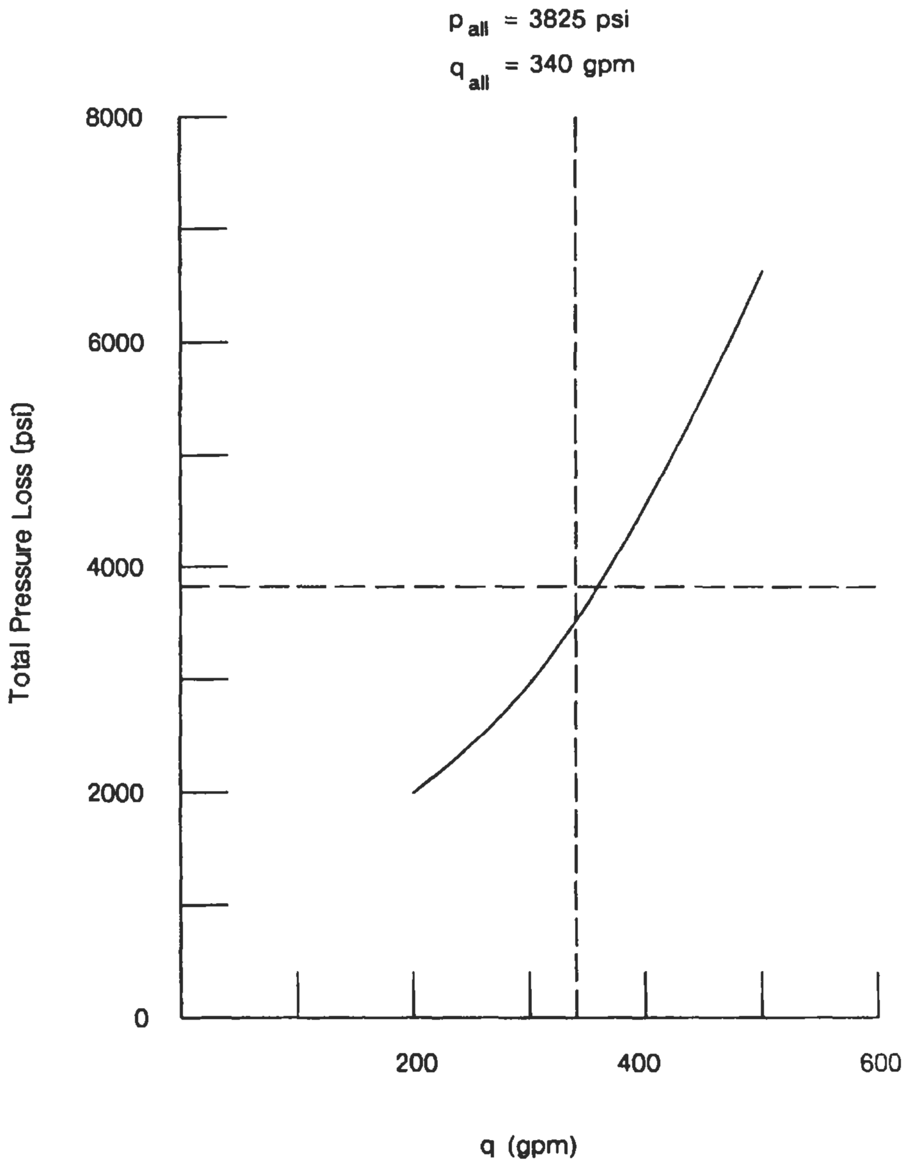


Figure 4-197. 5¾-in. liner, total pressure loss versus flowrate, Example 2. *(Courtesy Smith International, Inc.)*

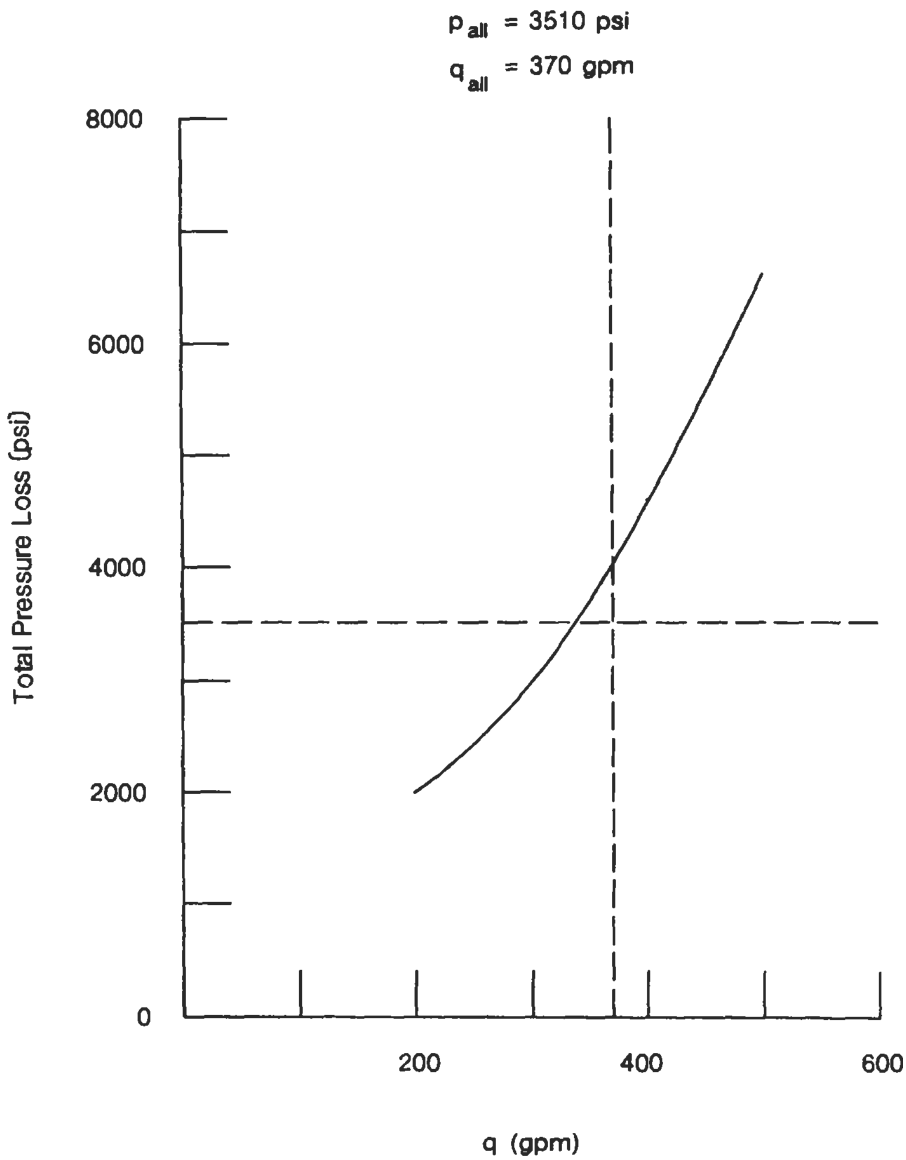


Figure 4-198. 6-in. liner, total pressure loss versus flowrate, Example 2. *(Courtesy Smith International, Inc.)*

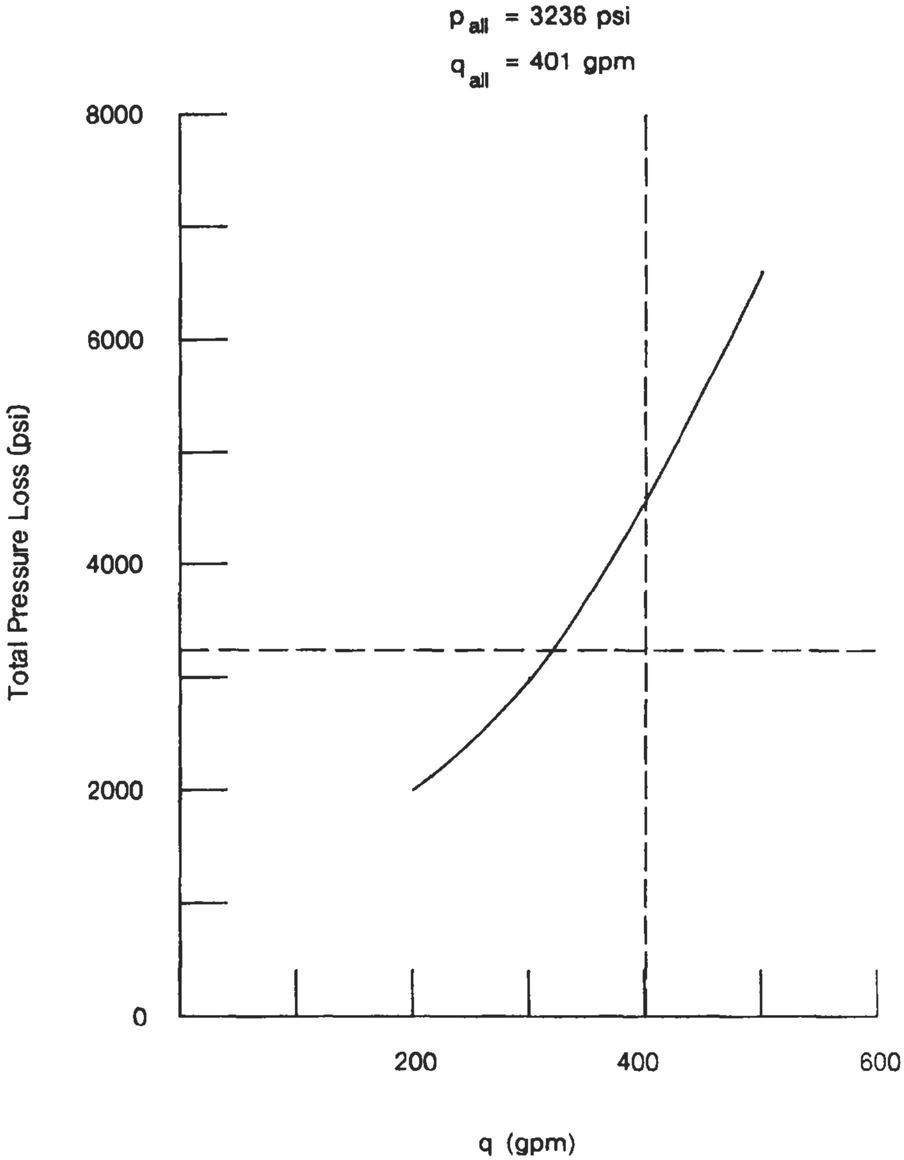


Figure 4-199. 6¼-in. liner, total pressure loss versus flowrate, Example 2. *(Courtesy Smith International, Inc.)*

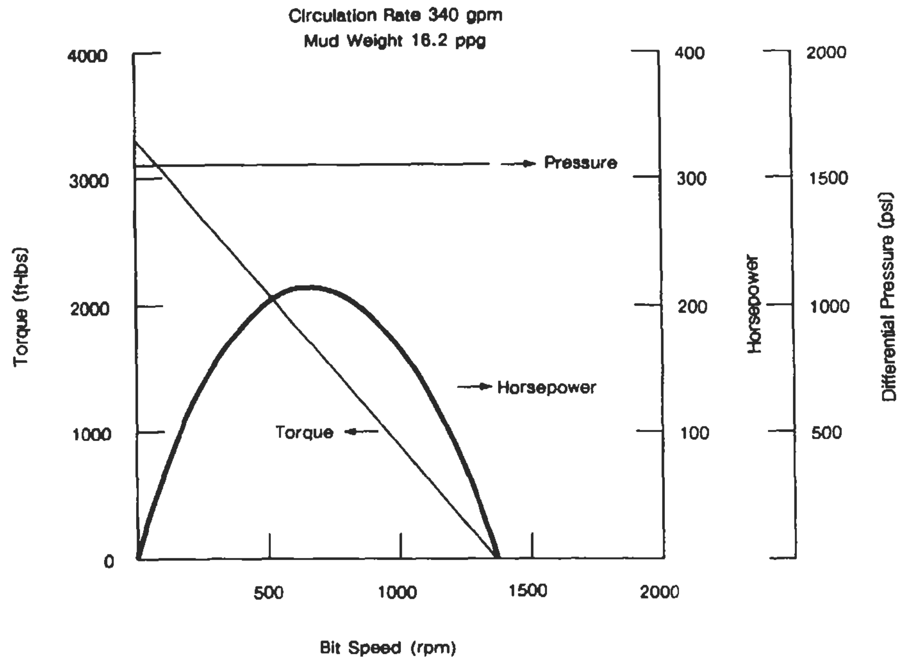


Figure 4-200. Turbine motor, 6¾-in. outside diameter, two motor sections, 212 stages, 340 gal/min, 16.2-lb/gal mud weight, Example 2. *(Courtesy Smith International, Inc.)*

(text continued from page 876)

Positive Displacement Motor

Figure 4-201 shows the typical rigid rotor and flexible elastomer stator configuration for a single chamber of a multichambered downhole positive displacement motor section. All the positive displacement motors presently in commercial use are of Moineau type, which uses a stator made of an elastomer. The rotor is made of a rigid material such as steel and is fabricated in a helical shape. The activating drilling mud, freshwater, aerated mud, foam or misted air is pumped at rather high velocity through the motor section, which, because of the eccentricity of the rotor and stator configuration, and the flexibility of the stator, allows the hydraulic pressure of the flowing fluid to impart a torque to the rotor. As the rotor rotates the fluid is passed from chamber to chamber (a chamber is a lengthwise repeat of the motor). These chambers are separate entities and as one opens up to accept fluid from the preceding, the preceding closes up. This is the concept of the positive displacement motor.

Design

The rotational energy of the positive displacement motor is provided by the flowing fluid, which rotates and imparts torque to the drill bit. Figure 4-202 shows the typical complete downhole positive displacement motor.

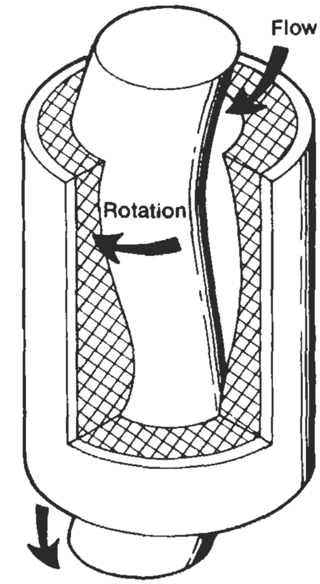


Figure 4-201. Basic positive displacement motor design principle. *(Courtesy Smith International, Inc.)*

In general, the downhole positive displacement motor constructed on the Moineau principle is composed of four sections: (1) the dump valve section, (2) the multistage motor section, (3) the connecting rod section and (4) the thrust and radial-bearing section. These sections are shown in Figure 4-202. Usually the positive displacement motor has multichambers, however, the number of chambers in a positive displacement motor is much less than the number of stages in a turbine motor. A typical positive displacement motor has from two to seven chambers.

The dump valve is a very important feature of the positive displacement motor. The positive displacement motor does not permit fluid to flow through the motor unless the motor is rotating. Therefore, a dump valve at the top of the motor allows drilling fluid to be circulated to the annulus even if the motor is not rotating. Most dump valve designs allow the fluid to circulate to the annulus when the pressure is below a certain threshold, say below 50 psi or so. Only when the surface pump is operated does the valve close to force all fluid through the motor.

The multichambered motor section is composed of only two continuous parts, the rotor and the stator. Although they are continuous parts, they usually constitute several chambers. In general, the longer the motor section, the more chambers. The stator is an elastomer tube formed to be the inside surface of a rigid cylinder. This elastomer tube stator is of a special material and shape. The material resists abrasion and damage from drilling muds containing cuttings and hydrocarbons. The inside surface of the stator is of an oblong, helical shape. The rotor is a rigid steel rod shaped as a helix. The rotor, when assembled into the stator and its outside rigid housing, provides continuous seal at contact points between the outside surface of the rotor and the inside surface of the stator (see Figure 4-201). The rotor or driving shaft is made up of n_r lodes. The

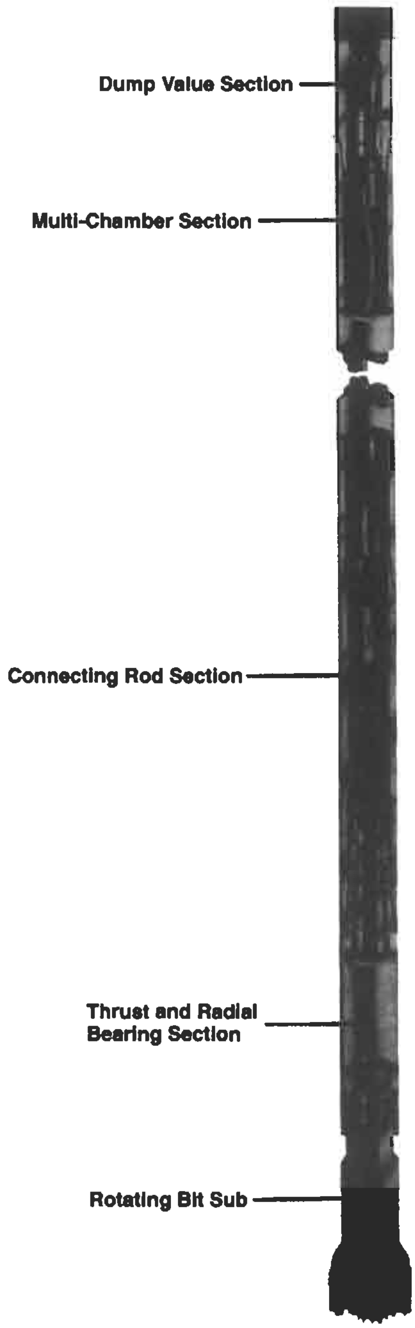


Figure 4-202. Downhole positive displacement motor design. *(Courtesy Smith International, Inc.)*

stator is made up of n_s lodes, which is equal to one lobe more than the rotor. Typical cross-sections of positive displacement motor lobe profiles are shown in Figure 4-203. As drilling fluid is pumped through the cavities in each chamber that lies open between the stator and rotor, the pressure of the flowing fluid causes the rotor to rotate within the stator. There are several chambers in a positive displacement motor because the chambers leak fluid. If the first chamber did not leak when operating, there would be no need for additional chambers.

In general, the larger lobe profile number ratios of a positive displacement motor, the higher the torque output and the lower the speed (assuming all other design limitations remain the same).

The rotors are eccentric in their rotation at the bottom of the motor section. Thus, the connecting rod section provides a flexible coupling between the rotor and the main drive shaft located in the thrust and radial bearing section. The main drive shaft has the drill bit connected to its bottom end.

The thrust and radial-bearing section contains the thrust bearings that transfer the weight-on-bit to the outside wall of the positive displacement motor. The radial support bearings, usually located above the thrust bearings, ensure that the main drive shaft rotates about a fixed center. As in most turbine motor designs, the bearings are cooled by the drilling fluid. There are some recent positive displacement motor designs that are now using grease-packed, sealed bearing assemblies. There is usually a smaller upper thrust bearing that allows rotation of the motor while pulling out of the hole. This upper thrust bearing is usually at the upper end of thrust and radial bearing section.

There are, of course, variations on the downhole positive displacement motor design, but the basic sections discussed above will be common to all designs.

The main advantages of the downhole positive displacement motor are:

1. Soft, medium and hard rock formations can be drilled with a positive displacement motor using nearly any type of rock bit. The positive displacement motor is especially adaptable to drilling with roller rock bits.
2. Rather moderate flow rates and pressures are required to operate the positive displacement motor. Thus, most surface pump systems can be used to operate these downhole motors.
3. Rotary speed of the positive displacement motor is directly proportional to flowrate. Torque is directly proportional to pressure. Thus, normal surface instruments can be used to monitor the operation of the motor downhole.
4. High torques and low speeds are obtainable with certain positive displacement motor designs, particularly, the higher lobe profiles (see Figure 4-203).
5. Positive displacement motors can be operated with aerated muds, foam and air mist.

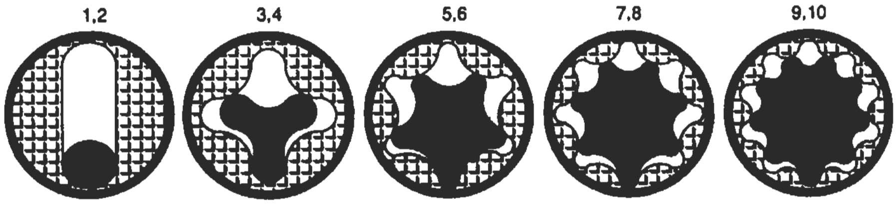


Figure 4-203. Typical positive displacement motor lobe profiles. *(Courtesy Smith International, Inc.)*

The main disadvantages of the downhole positive displacement motors are:

1. When the rotor shaft of the positive displacement motor is not rotating, the surface pump pressure will rise sharply and little fluid will pass through the motor.
2. The elastomer of the stator can be damaged by high temperatures and some hydrocarbons.

Operations

Figure 4-204 gives the typical performance characteristics of a positive displacement motor. The example in this figure is a $6\frac{3}{4}$-in. outside diameter positive displacement motor having five chambers activated by a 400-lb/gal flowrate of drilling mud.

For this example, a pressure of about 100 psi is required to start the rotor shaft against the internal friction of the rotor moving in the elastomer stator (and the bearings). With constant flowrate, the positive displacement motor will run at or near constant speed. Thus, this 1:2 lobe profile example motor has an

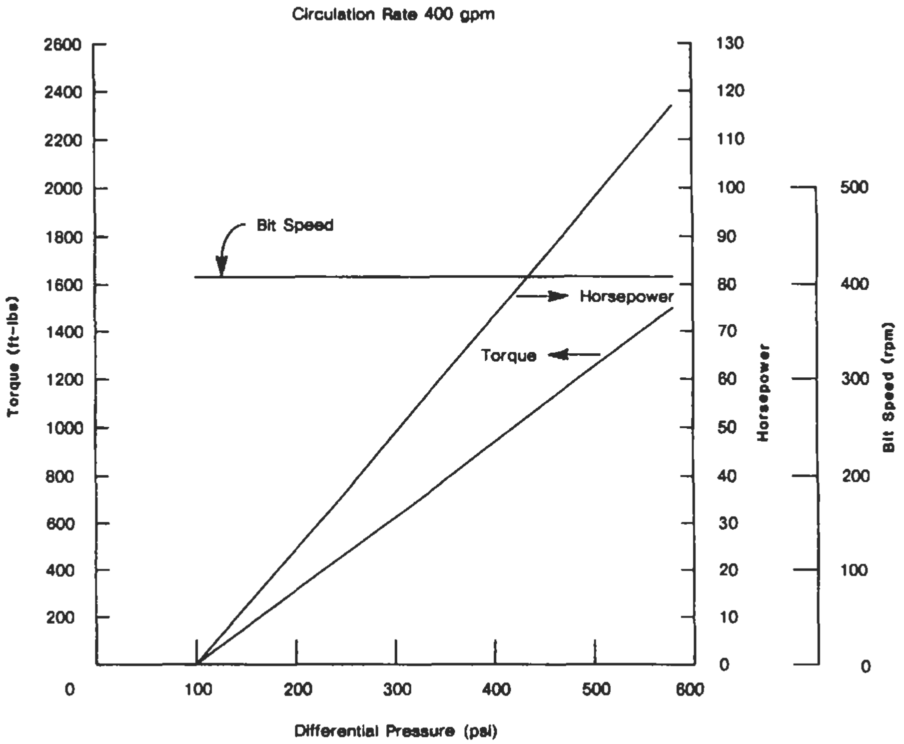


Figure 4-204. Positive displacement motor, 6¾-in. outside diameter, 1:2 lobe profile, 400 gal/min, differential pressure limit 580 psi. *(Courtesy Smith International, Inc.)*

operating speed of 408 rpm. The torque and the horsepower of the positive displacement motor are both linear with the pressure drop across the motor. Therefore, as more weight is placed on the drill bit (via the motor), the greater is the resisting torque of the rock. The mud pumps can compensate for this increased torque by increasing the pressure on the constant flowrate through the motor. In this example the limit in pressure drop across the motor is about 580 psi. Beyond this limit there will be either extensive leakage or damage to the motor, or both.

If the positive displacement motor is lifted off the bottom of the borehole and circulation continues, the motor will simply continue to rotate at 408 rpm. The differential pressure, however, will drop to the value necessary to overcome internal friction and rotate, about 100 psi. In this situation the motor produces no drilling torque or horsepower.

As the positive displacement motor is lowered and weight is placed on the motor and thus the bit, the motor speed continues but the differential pressure increases, resulting in an increase in torque and horsepower. As more weight is added to the positive displacement motor and bit, the torque and horsepower will continue to increase with increasing differentiated pressure (i.e., standpipe pressure). The amount of torque and power can be determined by the pressure change at the standpipe at the surface between the unloaded condition and the loaded condition. If too much weight is placed on the motor, the differential pressure limit for the motor will be reached and there will be leakage or a mechanical failure in the motor.

The rotor of the Moineau-type positive displacement motor has a helical design. The axial wave number of the rotor is one less than the axial wave number for the stator for a given chamber. This allows the formation of a series of fluid cavities as the rotor rotates. The number of stator wave lengths n_s and the number of rotor wave lengths n_r per chamber are related by [79,86]

$$n_s = n_r + 1 \tag{4-154}$$

The rotor is designed much like a screw thread. The rotor pitch is equivalent to the wavelength of the rotor. The rotor lead is the axial distance that a wave advances during one full revolution of the rotor. The rotor pitch and the stator pitch are equal. The rotor lead and stator lead are proportional to their respective number of waves. Thus, the relationship between rotor pitch t_r (in.) and stator pitch, t_s (in.) is [86]

$$t_r = t_s \tag{4-155}$$

The rotor lead L_r (in.) is

$$L_r = n_r t_r \tag{4-156}$$

The stator lead L_s (in.) is

$$L_s = n_s t_s \tag{4-157}$$

The specific displacement per revolution of the rotor is equal to the cross-sectional area of the fluid multiplied by the distance the fluid advances. The specific displacement s (in.^{3}) is

$$s = n_r n_s t_r A \tag{4-158}$$

where A is the fluid cross-sectional area (in.^{2}). The fluid cross-sectional area is approximately

$$A \approx 2ne_r^2(n_r^2 - 1) \tag{4-159}$$

where e_r is the rotor rotation eccentricity (in.). The special case of a 1:2 lobe profile motor has a fluid cross-sectional area of

$$A \approx 2e_r d_r \tag{4-160}$$

where d_r is the reference diameter of the motor (in.). The reference diameter is

$$d_r = 2e_r n_s \tag{4-161}$$

For the 1:2 lobe profile motor, the reference diameter is approximately equal to the diameter of the rotor shaft.

The instantaneous torque of the positive displacement motor M (ft-lb) is

$$M = 0.0133s\ \Delta p\eta \tag{4-162}$$

where Δp = differential pressure loss through the motor in psi
η = total efficiency of the motor. The 1:2 lobe profile motors have efficiencies around 0.80. The higher lobe profile motors have efficiencies that are lower (i.e., of the order of 0.70 or less)

The instantaneous speed of the positive displacement motor N (rpm) is

$$N = \frac{231.016q}{s} \tag{4-163}$$

where q is the circulation flowrate (gal/min).

The positive displacement motor horsepower HP (hp) for any speed is

$$HP = \frac{q\Delta p}{1,714}\eta \tag{4-164}$$

The number of positive displacement motor chambers n_c is

$$n_c = \frac{L}{t_s} - (n_s - 1) \tag{4-165}$$

where L is the length of the actual motor section (in.).

The maximum torque M_{max} will be at the maximum differential pressure Δp_{max}, which is

$$M_{max} = 0.133s\ \Delta p_{max}\eta \tag{4-166}$$

The maximum horse power HP_{max} will also be at the maximum differential pressure Δp_{max}, which is

$$HP_{max} = \frac{q\Delta p_{max}}{1,714}\eta \tag{4-167}$$

It should be noted that the positive displacement motor performance parameters are independent of the drilling mud weight. Thus, these performance parameters will vary with motor design values and the circulation flowrate.

If the above performance parameters for a positive displacement motor design are known for a given circulation flowrate (denoted as 1), the performance parameters for the new circulation flowrate (denoted as 2) can be found by the following relationships:

Torque

$$M_2 = M_1 \tag{4-168}$$

Speed

$$N_2 = \left(\frac{q_2}{q_1}\right)N_1 \tag{4-169}$$

Power

$$HP_2 = \left(\frac{q_2}{q_1}\right)HP_1 \tag{4-170}$$

Table 4-114 gives the performance characteristics for various circulation flowrates for the 1:2 lobe profile, $6\frac{3}{4}$-in. outside diameter positive displacement motor. Figure 4-204 shows the performance of the 1:2 lobe profile positive displacement motor at a circulation flowrate of 400 gal/min.

Table 4-114
Positive Displacement Motor, 6¾-in. Outside Diameter, 1:2 Lobe Profile, Five Motor Chambers

Circulation Rate (gpm)	Speed (rpm)	Maximum Differential Pressure (psi)	Maximum Torque (ft-lbs)	Maximum Horsepower
200	205	580	1500	59
250	255	580	1500	73
300	306	580	1500	87
350	357	580	1500	102
400	408	580	1500	116
450	460	580	1500	131
500	510	580	1500	145

Courtesy Eastman-Christensen

Table 4-115 gives the performance characteristics for various circulation flowrates for the 5:6 lobe profile, $6\frac{3}{4}$-in. outside diameter positive displacement motor. Figure 4-205 shows the performance of the 5:6 lobe profile positive displacement motor at a circulation flow rate of 400 gal/min.

The positive displacement motor whose performance characteristics are given in Table 4-114 is a 1:2 lobe profile motor. This lobe profile design is usually used for deviation control operations. The 1:2 lobe profile design yields a downhole motor with high rotary speeds and low torque. Such a combination is very desirable for the directional driller. The low torque minimizes the compensation that must be made in course planning which must be made for the reaction torque in the lower part of the drill string. This reactive torque when severe can create difficulties in deviation control planning. The tradeoff is, however, that higher speed reduces the bit life, especially roller rock bit life.

The positive displacement motor whose performance characteristics are given in Table 4-115 is a 5:6 lobe profile motor. This lobe profile design is usually used for straight hole drilling with roller rock bits, or for deviation control operations where high torque polycrystalline diamond compact bit or diamond bits are used for deviation control operations.

Example 3

A $6\frac{3}{4}$-in. outside diameter positive displacement motor of a 1:2 lobe profile design (where performance data are given in Table 4-114) has rotor eccentricity of 0.60 in., a reference diameter (rotor shaft diameter) of 2.48 in. and a rotor pitch of 38.0 in. If the pressure drop across the motor is determined to be 500 psi at a circulation flowrate of 350 gal/min with 12.0 lb/gal, find the torque, rotational speed and the horsepower of the motor.

Torque. Equation 4-160 gives the fluid cross-sectional area of the motor, which is

$$A = 2(0.6)(2.48)$$

$$= 2.98 \text{ in.}^2$$

Equation 4-159 gives the specific displacement of the motor, which is

Table 4-115
Positive Displacement Motor, 6¾-in. Outside Diameter, 5:6 Lobe Profile, Five Motor Chambers

Circulation Rate (gpm)	Speed (rpm)	Maximum Differential Pressure (psi)	Maximum Torque (ft-lbs)	Maximum Horsepower
200	97	580	2540	47
250	122	580	2540	59
300	146	580	2540	71
350	170	580	2540	82
400	195	580	2540	94

Courtesy Eastman-Christensen

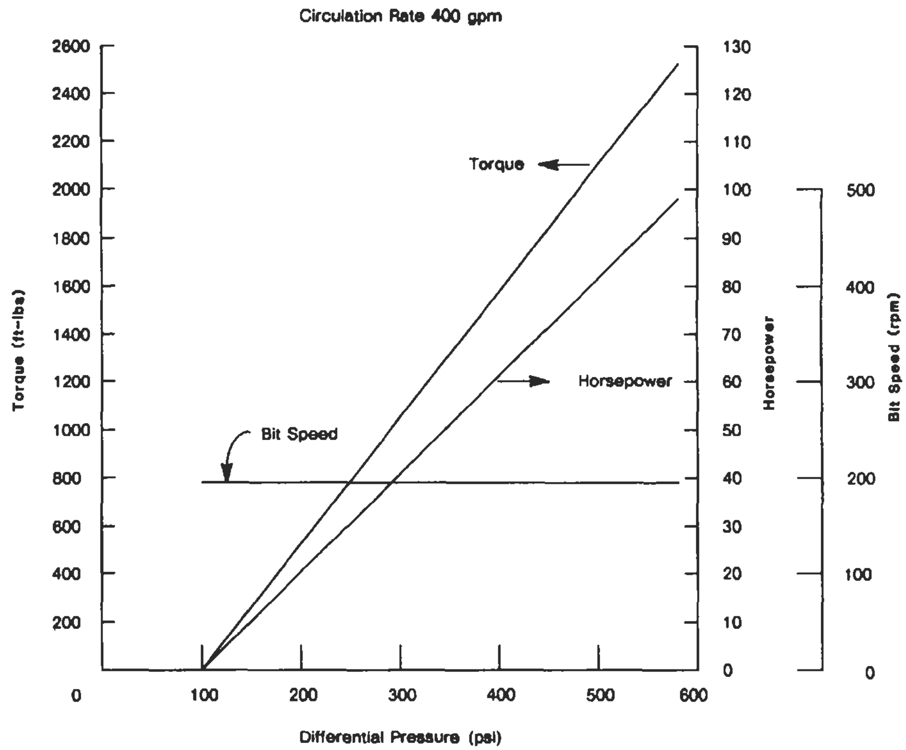


Figure 4-205. Positive displacement motor, 6¾-in. outside diameter, 5:6 lobe profile, 400 gal/min, differential pressure limit 580 psi. *(Courtesy Smith International, Inc.)*

$$s = (1)(2)(38.0)(2.98)$$

$$= 226.5 \text{ in.}^3$$

The torque is obtained from Equation 4-162, assuming an efficiency of 0.80 for the 1:2 lobe profile motor. This is

$$M = 0.0133(226.5)(500)(0.80)$$

$$= 1205 \text{ ft-lb}$$

Speed. The rotation speed is obtained from Equation 4-163. This is

$$N = \frac{231.016(350)}{226.5}$$

$$= 357 \text{ rpm}$$

Horsepower. The horsepower the motor produces is obtained from Equation 4-164. This is

$$HP = \frac{350(500)}{1714}(0.80)$$

$$= 82$$

Planning for a positive displacement motor run and actually drilling with such a motor is easier than with a turbine motor. This is mainly due to the fact that when a positive displacement motor is being operated, the operator can know the operating torque and rotation speed via surface data. The standpipe pressure will yield the pressure drop through the motor, thus the torque. The circulation flowrate will yield the rotational speed.

Example 4

A $6\frac{3}{4}$-in. outside diameter positive displacement motor (whose performance data are given in Tables 4-114 and 4-115) is to be used for a deviation control direction drill operation. The motor will use an $8\frac{1}{2}$-in. diameter roller rock bit for the drilling operation. The directional run is to take place at a depth of 10,600 ft (measured depth). The rock formation to be drilled is classified as medium, and it is anticipated that 30 ft/hr will be the maximum possible drilling rate. The mud weight is to be 11.6 lb/gal. The drilling rig has a National Supply Company duplex mud pump Model E-700 available. The details of this pump are given in Table 4-116 (also see the section titled "Mud Pumps"). Because this is a deviation control run, the 1:2 lobe profile positive displacement motor will be used since it has the lowest torque for a given circulation flowrate (see Table 4-114). Determine the appropriate circulation flowrate to be used for the roller rock bit, positive displacement motor combination and the appropriate liner size to be used in the duplex pump. Also, prepare the positive displacement motor performance graph for the chosen circulation flowrate. Determine the bit nozzle sizes.

Bit Pressure Loss. It is necessary to choose the bit pressure loss such that the thrust load created in combination with the weight on bit will yield an on-bottom load on the motor thrust bearings, which is less than the maximum allowable load for the bearings. Since this is a deviation control run and, therefore, the motor will be drilling only a relatively short time and distance, the motor thrust bearings will be operated at their maximum rated load for on-bottom operation. Figure 4-206 shows that maximum allowable motor thrust bearing load is about

Table 4-116
Duplex Mud Pump, Model E-700, National Supply Company, Example 4

Input Horsepower 825, Maximum Strokes per Minute, 65 Length of Stroke, 16 Inches

	5¾	6	6¼	6½	6¾	7
Output per Stroke (gals)	6.14	6.77	7.44	8.13	8.85	9.60
Maximum Pressure (psi)	3000	2450	2085	1780	1535	1260

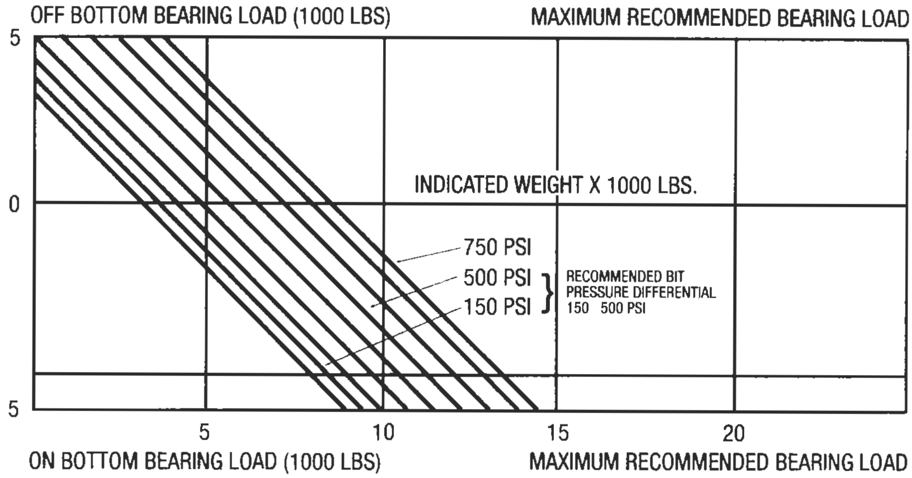


Figure 4-206. Hydraulic thrust and indicated weight balance for positive displacement motor. *(Courtesy Smith International, Inc.)*

6,000 lb. To have the maximum weight on bit, the maximum recommended bit pressure loss of 500 psi will be used. This will give maximum weight on bit of about 12,000 lb. The higher bit pressure loss will, of course, give the higher cutting face cleaning via jetting force (relative to the lower recommended bit pressure losses).

Total Pressure Loss. Since bit life is not an issue in a short deviation control motor run operation, it is desirable to operate the positive displacement motor at as high a power level as possible during the run. The motor has a maximum pressure loss with which it can operate. This is 580 psi (see Table 4-114). It will be assumed that the motor will be operated at the 580 psi pressure loss in order to maximize the torque output of the motor. To obtain the highest horsepower for the motor, the highest circulation flowrate possible while operating within the constraints of the surface mud pump should be obtained. To obtain this highest possible, or optimal, circulation flowrate, the total pressure losses for the circulation system must be obtained for various circulation flowrates. These total pressure losses tabulated in the lower row of Table 4-117 represent the surface standpipe pressure when operating at the various circulation flowrates.

Pump Limitations. Table 4-116 shows there are six possible liner sizes that can be used on the Model E-700 mud pump. Each liner size must be considered to obtain the optimum circulation flowrate and appropriate liner size. The maximum pressure available for each liner size will be reduced by a safety factor of 0.90. The maximum volumetric flowrate available for each liner size will also be reduced by a volumetric efficiency factor of 0.80 and an additional safety factor of 0.90. Thus, from Table 4-116, the allowable maximum pressures and allowable maximum volumetric flowrates will be those shown in Figures 4-207 through 4-212, which are the liner sizes $5\frac{3}{4}$, 6, $6\frac{1}{4}$, $6\frac{1}{2}$ and 7 in., respectively. Plotted on each of these figures are the total pressure losses for the various circulation flowrates considered. The horizontal straight line on each figure is

Table 4-117
Drillstring Component Pressure Losses at Various Circulation Flowrates for Example 4

	Pressure (psi)			
Components	**200 gpm**	**300 gpm**	**400 gpm**	**500 gpm**
Surface Equipment	5	11	19	30
Drill Pipe Bore	142	318	566	884
Drill Collar Bore	18	40	71	111
PDM	580	580	580	580
Drill Bit	500	500	500	500
Drill Collar Annulus	11	25	45	70
Drill Pipe Annulus	32	72	128	200
Total Pressure Loss	1288	1546	1909	2375

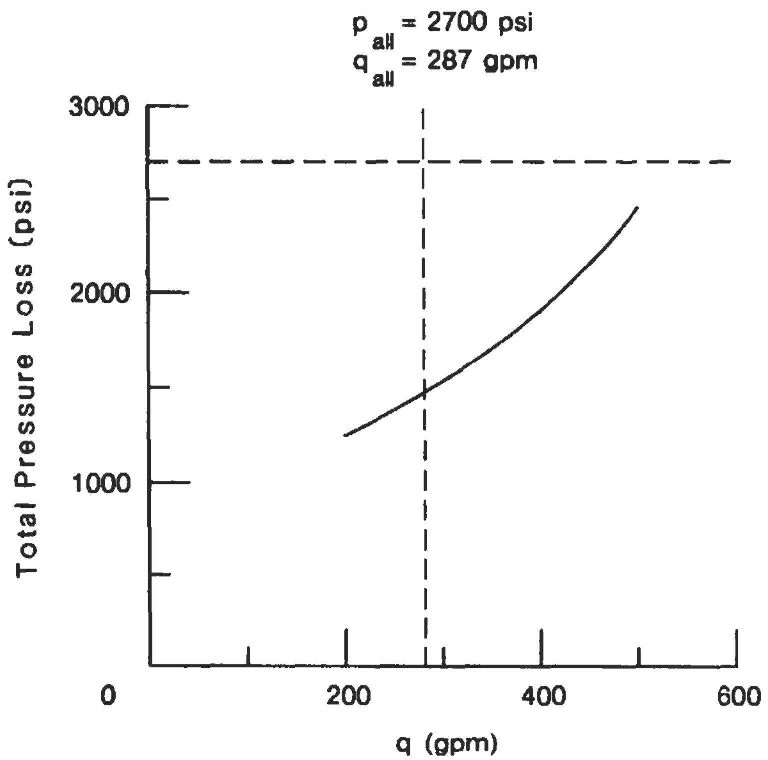


Figure 4-207. 5¾-in. liner, total pressure loss vs. flowrate, Example 4. *(Courtesy Smith International, Inc.)*

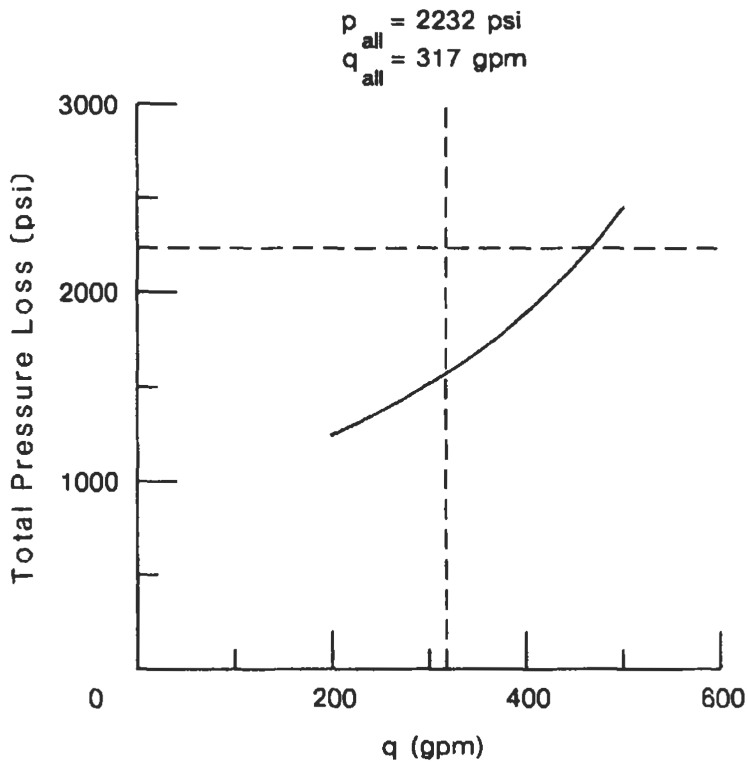


Figure 4-208. 6-in. liner, total pressure loss vs. flowrate, Example 4. *(Courtesy Smith International, Inc.)*

the allowable maximum pressure for the particular liner size. The vertical straight line is the allowable maximum volumetric flowrate for the particular liner size. Only circulation flowrates that are in the lower left quadrant of the figures are practical. The highest circulation flowrate (which produces the highest positive displacement motor horsepower) is found in Figure 4-209, the $6\frac{1}{4}$-in. liner. The optimal circulation flow rate is 348 gal/min.

Positive Displacement Motor Performance. Using the positive displacement motor performance data in Table 4-114 and the scaling relationships in Equations 4-168 through 4-170, the performance graph for the positive displacement motor operating with a circulation flowrate of 348 gal/min can be prepared. This is given in Figure 4-213.

Bit Nozzle Sizes. The pressure loss through the bit must be 500 psi with a circulation flowrate of 348 gal/min with 11.6-lb/gal mud weight. The pressure loss through a roller rock bit with three nozzles is (see the section titled "Drilling Bits and Downholes Tools")

(text continued on page 898)

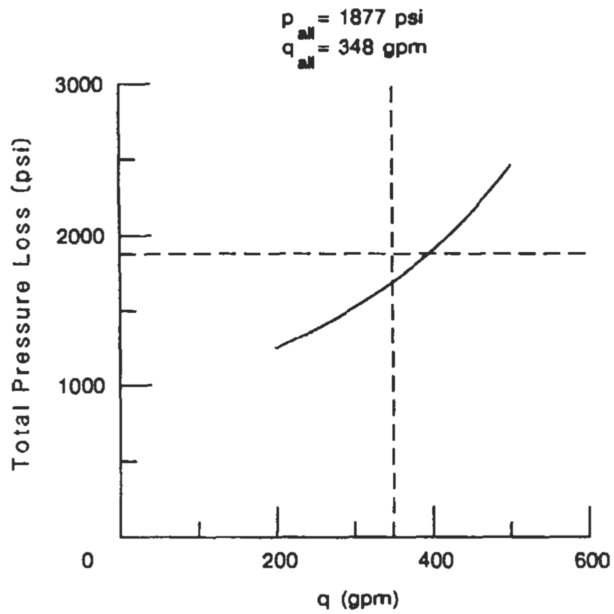


Figure 4-209. 6¼-in. liner, total pressure loss vs. flowrate, Example 4. *(Courtesy Smith International, Inc.)*

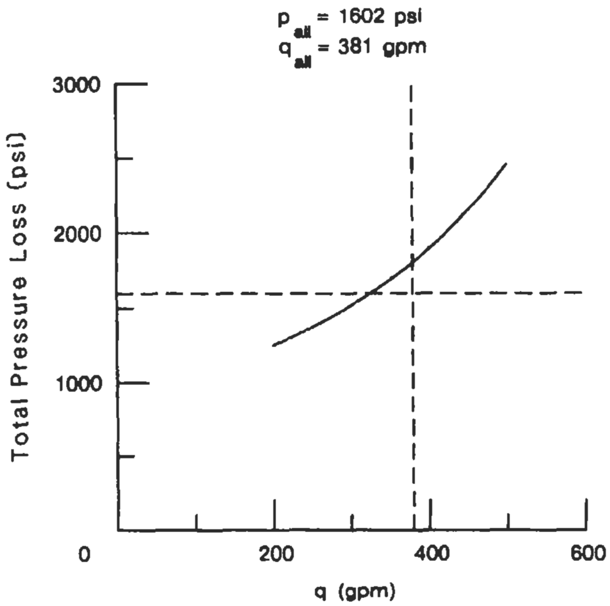


Figure 4-210. 6½-in. liner, total pressure loss vs. flowrate, Example 4. *(Courtesy Smith International, Inc.)*

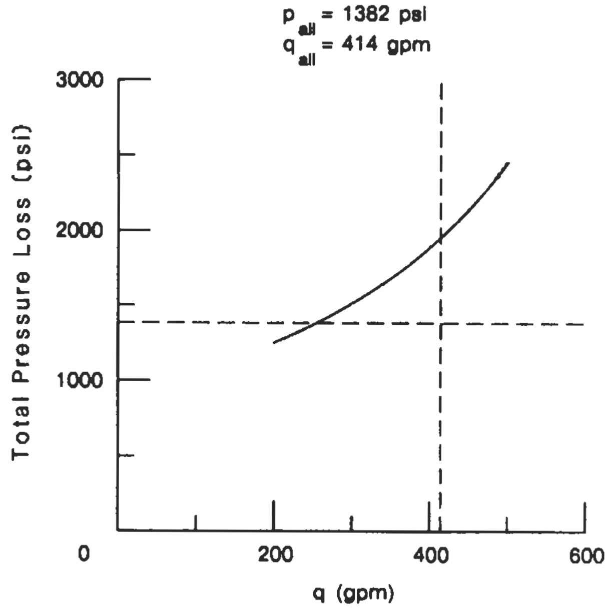


Figure 4-211. 6¾-in. liner, total pressure loss vs. flowrate, Example 4. *(Courtesy Smith International, Inc.)*

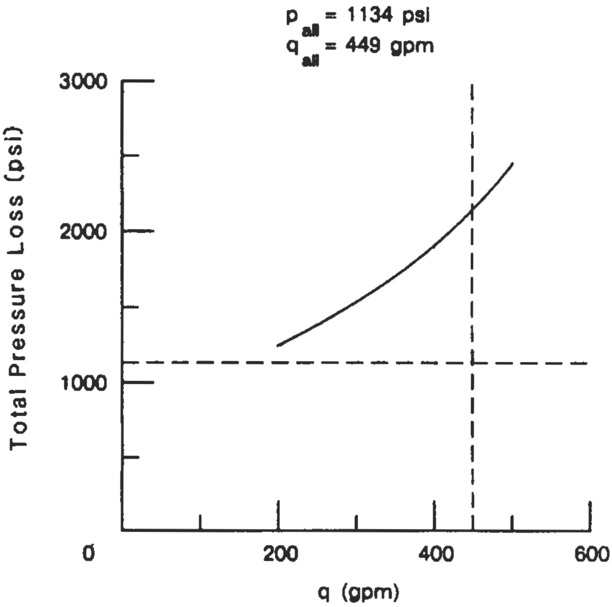


Figure 4-212. 7-in. liner, total pressure loss vs. flowrate, Example 4. *(Courtesy Smith International, Inc.)*

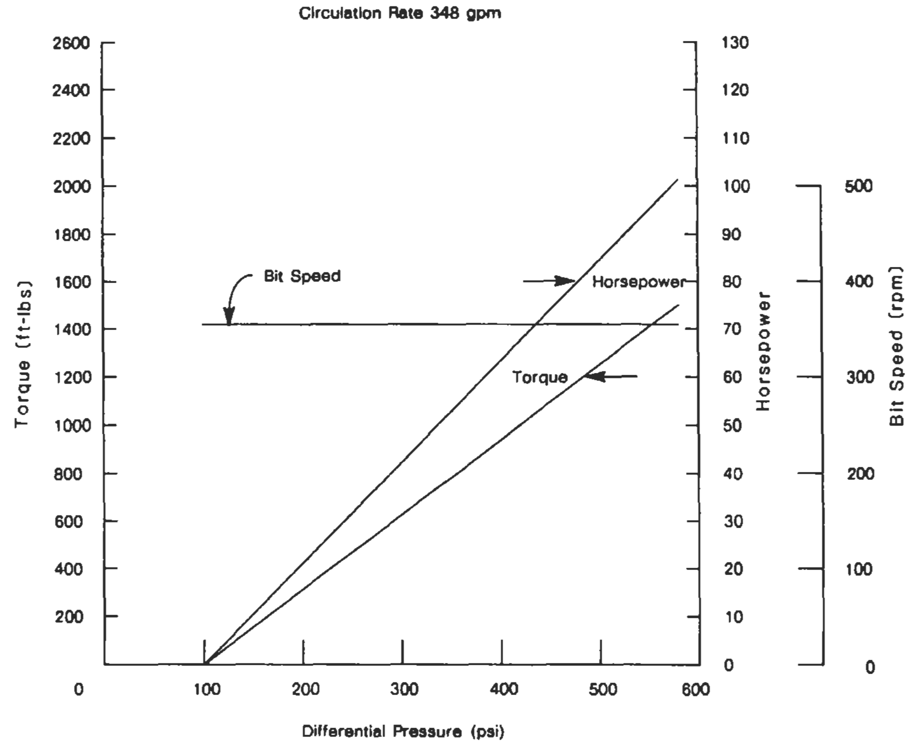


Figure 4-213. Positive displacement motor, 6¾-in. outside diameter, 1:2 lobe profile, 348 gal/min, differential pressure limit 580 psi, Example 4. *(Courtesy Smith International, Inc.)*

(text continued from page 895)

$$\Delta p_b = \frac{q^2 \bar{\gamma}_m}{7430 C^2 d^4} \tag{4-171}$$

where C = nozzle coefficient (usually taken to be 0.95)
d_e = hydraulic equivalent diameter in in.

Therefore, Equation 4-171 is

$$500 = \frac{(348)^2 (11.6)}{7,430 (0.98)^2 de^4}$$

which yields

d_e = 0.8045 in.

The hydraulic equivalent diameter is related to the actual nozzle diameters by

$$d_e = [ad_1^2 + bd_2^2 + cd_3^2]^{1/2}$$

where
a = number of nozzles with diameter d_1
b = number of nozzles with diameter d_2
c = number of nozzles with diameter d_3
d_1, d_2 and d_3 = three separate nozzle diameters in in.

Nozzle diameters are usually in 32nds of an inch. Thus, if the bit has three nozzles with $\frac{15}{32}$ of an inch diameter, then

$$d_e = [(3)(0.4688)^2]^{1/2}$$

$$= 0.8120 \text{ in.}$$

The above hydraulic equivalent diameter is close enough to the one obtained with Equation 4-171. Therefore, the bit should have three $\frac{5}{32}$-in. diameter nozzles.

Special Applications

As it becomes necessary to infill drill the maturing oil and gas reservoirs in the continental United States and elsewhere in the world, the need to minimize or eliminate formation damage will become an important engineering goal. To accomplish this goal, air and gas drilling techniques will have to be utilized (see the section titled "Air and Gas Drilling"). It is very likely that the future drilling in the maturing oil and gas reservoirs will be characterized by extensive use of high-angle directional drilling coupled with air and gas drilling techniques.

The downhole turbine motor designed to be activated by the flow of incompressible drilling mud cannot operate on air, gas, unstable foam or stable foam drilling fluids. These downhole turbine motors can only be operated on drilling mud or aerated mud.

Recently, a special turbine motor has been developed to operate on air, gas and unstable foam [82]. This is the downhole pneumatic turbine motor. This motor has been tested in the San Juan Basin in New Mexico and the Geysers area in Northern California. Figure 4-214 shows the basic design of this drilling device. The downhole pneumatic turbine motor is equipped with a gear reduction transmission. The compressed air or gas that actuates the single stage turbine motor causes the rotor of the turbine to rotate at very high speeds (i.e., ~20,000 rpm). A drill bit cannot be operated at such speeds; thus it is necessary to reduce the speed with a series of planetary gears. The prototype downhole pneumatic turbine motor has a gear reduction transmission with an overall gear ratio of 168 to 1. The particular version of this motor concept that is undergoing field testing is a 9-in. outside diameter motor capable of drilling with a $10\frac{5}{8}$-in.-diameter bit or larger. The downhole pneumatic turbine motor will deliver about 40 hp for drilling with a compressed air flowrate of 3,600 scfm. The motor requires very little additional pressure at the surface to operate (relative to normal air drilling with the same volumetric rate).

The positive displacement motor of the Moineau-type design can be operated with unstable foam (or mist) as the drilling fluid. Some liquid must be placed in the air or gas flow to lubricate the elastomer stator as the metal rotor rotates against the elastomer. Positive displacement motors have been operated quite

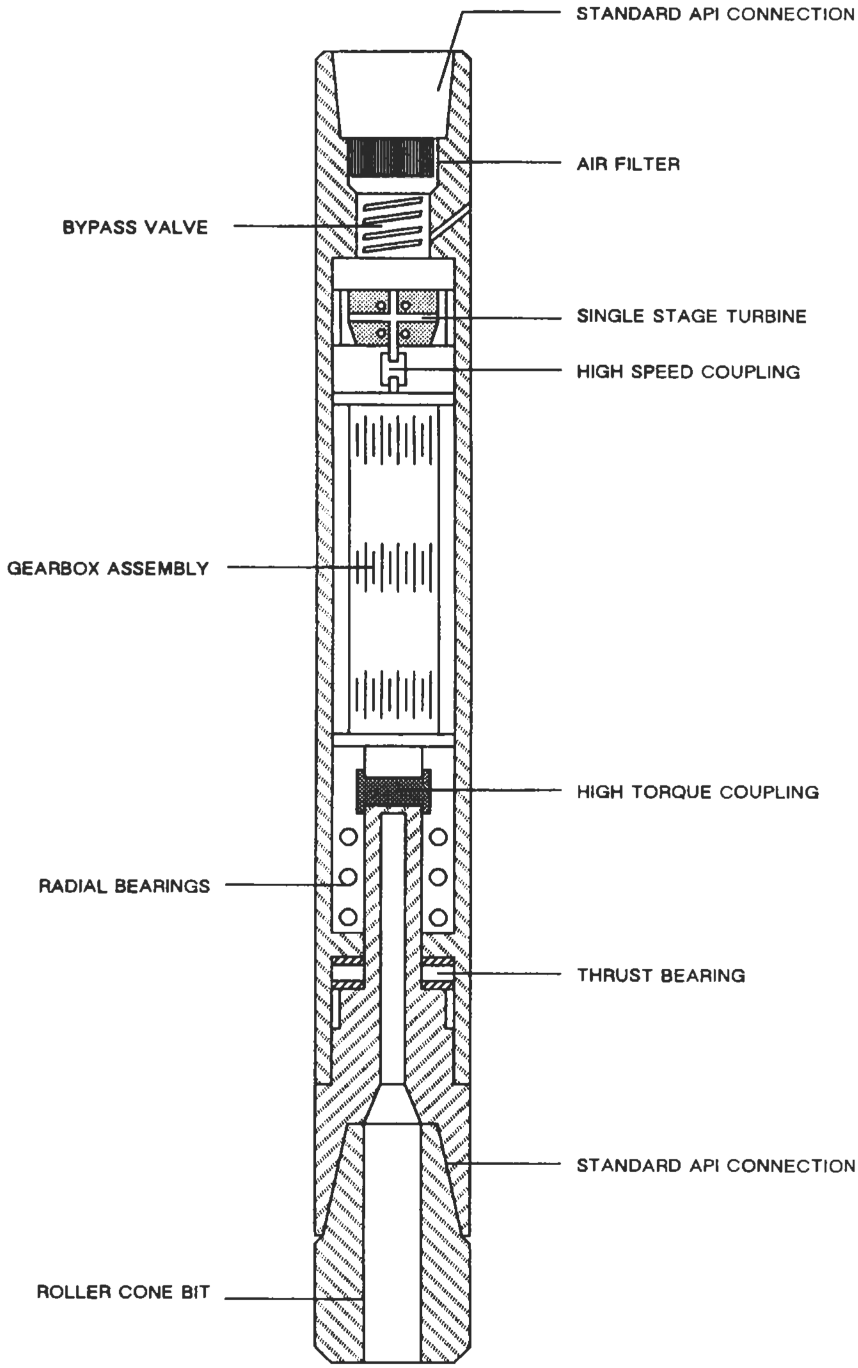


Figure 4-214. Downhole pneumatic turbine motor design. *(Courtesy Pneumatic Turbine Partnership.)*

successfully in many air and gas drilling situations. The various manufacturers of these motors can give specific information concerning the performance characteristics of their respective motors operated with air and gas drilling techniques. The critical operating characteristic of these motors, when operated with unstable foam, is that these motors must be loaded with weight on bit when circulation is initiated. If the positive displacement motor is allowed to be started without weight on bit, the rotor will speed up quickly to a very high speed, thus burning out the bearings and severely damaging the elastomer stator.

MWD AND LWD

Most of the cost in a well is expanded during the drilling phase. Any amount of information gathered during drilling can be used to make decisions regarding the efficiency of the process. But the scope and ultimate cost to gather and analyze such information must be offset by a decrease in drilling expenditures, an increase in drilling efficiency and an increase in safety.

As drilling technology moved the pursuit of hydrocarbon resources into higher-cost offshore and hostile environments, intentionally deviated boreholes required information such as azimuth and inclination that could not be derived by surface instruments. Survey instruments, either lowered on a sand line or dropped into the drill pipe for later retrieval, to some degree satisfied the requirements but consumed expensive rig time and sometimes produced questionable results.

For many years researchers have been looking for a simple, reliable *measurement while drilling* technique, referred to by its abbreviation MWD. As early as 1939, a logging while drilling (LWD) system, using an electric wire, was tested successfully but was not commercialized [89,90]. Mud pulse systems were first proposed in 1963 [91,92]. The first mechanical mud pulse system was marketed in 1964 by Teledrift for transmitting directional information [93]. In the early 1970s, the steering tool, an electric wire operated directional tool, gave the first real-time measurements while the directional buildup was in progress. Finally, the first modern mud pulse data transmission system was commercialized in 1977 by Teleco [94]. State-of-the-art surveys of the technology were made in 1978 [95], in 1988 [96–98], and in 1990 [99].

A problem with the early MWD mud pulse systems was the very slow rate of data transmission. Several minutes were needed to transmit one set of directional data. Anadrill working with a Mobil patent [100] developed in the early 1980s a continuous wave system with a much faster data rate. It became possible to transmit many more drilling data, and also to transmit logging data making LWD possible. Today, as many as 16 parameters can be transmitted in 16 s. The dream of the early pioneers has been more than fulfilled since azimuth, inclination, tool face, downhole weight-on-bit, downhole torque, shocks, caliper, resistivity, gamma ray, neutron, density, Pe, sonic and more can be transmitted in real-time to the rig floor and the main office.

MWD Technology

Steering Tool

Up until 1970 all directional drilling was conducted using singleshot and multishot data. The normal procedure was:

a. drill vertically in rotary to the kick-off depth;
b. kick-off towards the target using a downhole motor and a bent sub to an inclination of approximately 10°;

c. resume rotary drilling with the appropriate bottomhole assembly to build angle, hold, or drop.

The kick-off procedure required numerous single-shot runs to start the deviation in the correct direction. Since, during this phase, the drillpipe was not rotating a steering tool was developed to be lowered on an electric wireline instead of the single shot. The measurements were then made while drilling.

Measurements by electric cable are possible only when the drillstem is not rotating, hence with a turbine or downhole motor. The logging tool is run in the drillstring and is positioned by a mule shoe and key. The process is identical to the one used in the single-shot measurements. The magnetic orientation sensor is of the flux-gate type and measures the three components of the earth's magnetic field vector in the reference space of the logging tool. Three accelerometers measure the three components of the gravity vector still in the same reference space. These digitized values are multiplexed and transmitted by an insulated electric conductor and the cable armor toward the surface. On the surface a minicomputer calculates the azimuth and the drift of the borehole as well as the angle of the tool face permanently during drilling. In the steering-tool system, the computer can also determine the azimuth and slant of the downhole motor underneath the bent sub and thus anticipate the direction that the well is going to take. It can also determine the trajectory followed. Figure 4-215 shows the steering tool system.

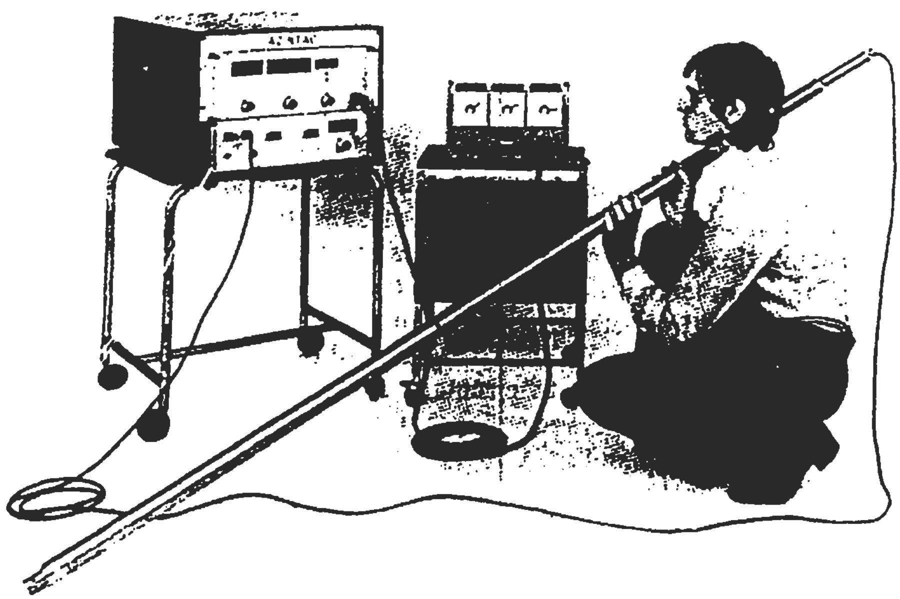


Figure 4-215. Typical steering tool unit with surface panel and driller readout. *(Courtesy of Institut Francois du Petrole.)*

Naturally for operating the tool, the seal must be maintained at the point where the cable enters the drillstring:

a. either at the top of the drillpipes, in which case the logging tool is pulled out every time a new drillpipe length is added on;
b. or through the drillpipe wall in a special sub placed in the drillstring as near the surface as possible, in which case new lengths are added on without pulling out the logging tool.

Figure 4-216 shows the typical operation of a steering tool for orienting the drill bit. The electric wireline goes through a circulating head located on top

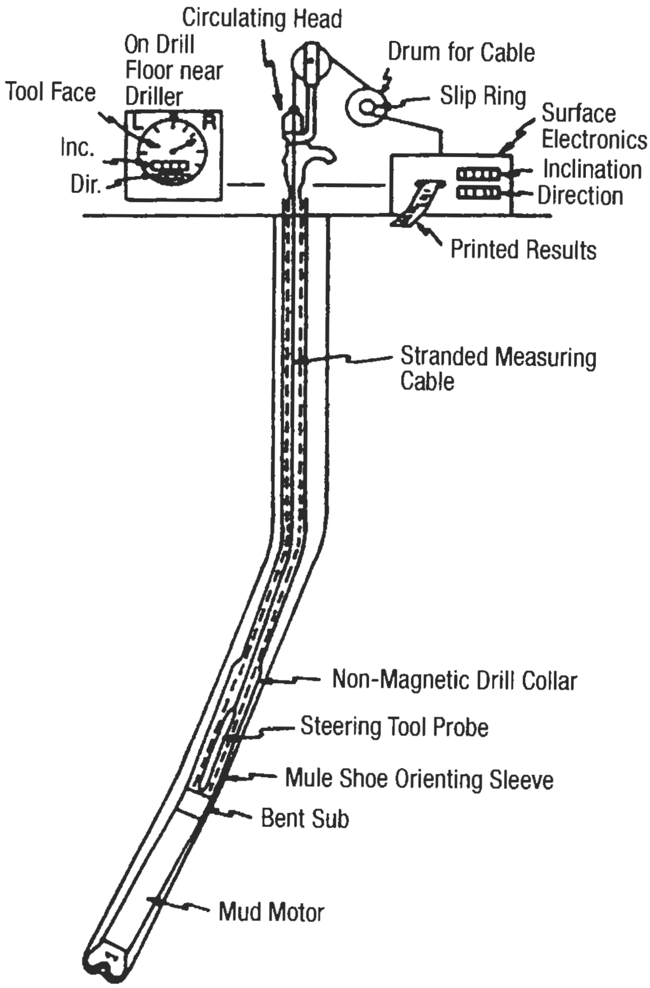


Figure 4-216. Typical operation of a steering tool for orienting the drill bit using a circulating head on the swivel. *(Courtesy SPE* [101].*)*

of the swivel. As mentioned previously, the tool has to be pulled out when adding a single.

Figure 4-217 shows the same operation using a side-entry sub. With this sub, the electric wireline crosses over from inside the drill pipe to the outside. Consequently, singles may be added without pulling the steering tool out. On the other hand, there is a risk of damaging the cable if it is crushed between the drillpipe tool joints and the surface casing. The wireline also goes through the rotary table and special care must be taken not to crush it between the rotary table and the slips. Furthermore, in case of BHA sticking, the steering tool has to be fished out by breaking and grabbing the electric wireline inside the drillpipes.

Figure 4-218 shows the arrangement of the sensors used in a steering tool. Three flux-gate-type magnetometers and three accelerometers are positioned

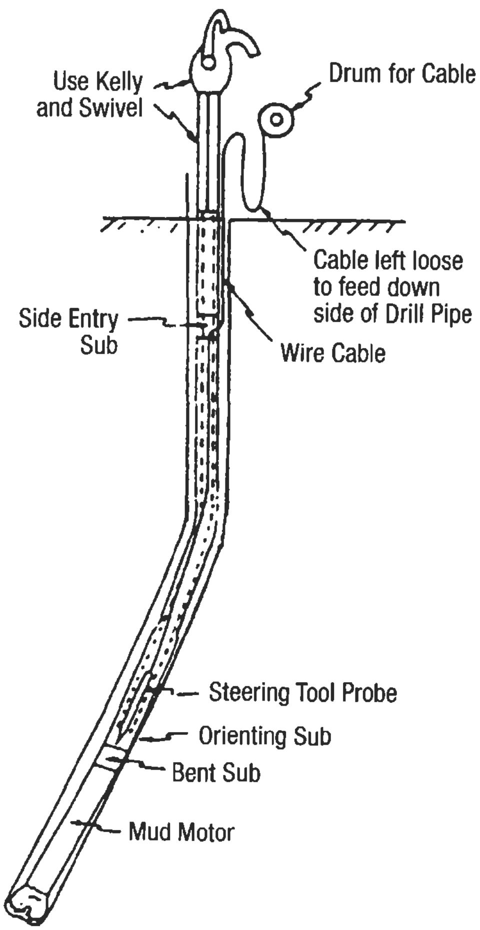


Figure 4-217. Typical operation of a steering tool for orienting the drill bit using a side-entry sub. *(Courtesy SPE* [101].*)*

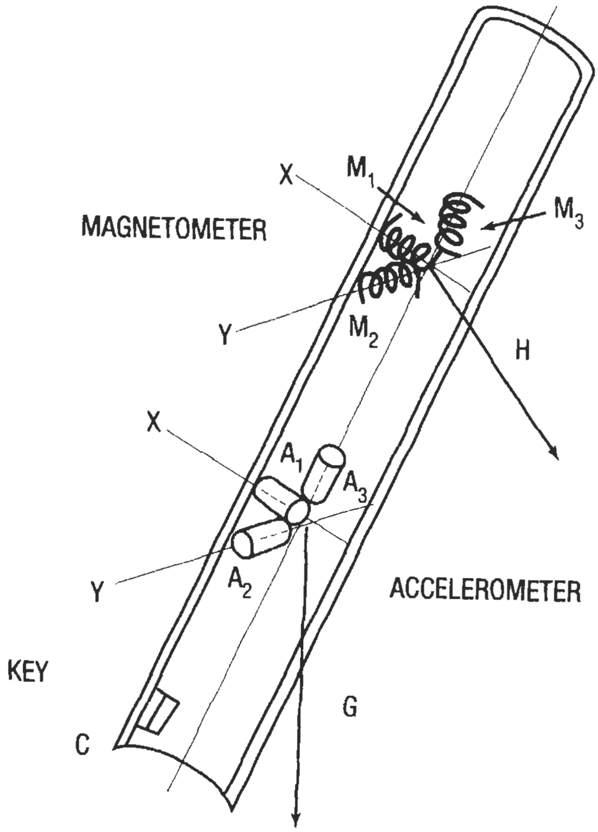


Figure 4-218. Sketch of the principle of the sensor arrangement in a steering tool and any magnetic directional tool. *(Courtesy SPE* [101].*)*

with their sensitive axis along the principal axis of the tool. O_x is oriented toward the bent sub in the bent sub/tool axis plane. O_y is perpendicular to O_x and the tool axis. O_z is oriented along the tool axis downward.

The arrangement of Figure 4-218 is common to all directional tools based on the earth's magnetic field for orientation: MWD tools or wireline logging tools.

The steering tools have practically been abandoned and replaced by MWD systems, mostly because of the electric wireline. However, the high data rate of the electric wireline (20–30 kbits/s) compared to the low data rate of the MWD systems (1–10 bits/s) make the wireline tools still useful for scientific work.

Accelerometers. Accelerometers measure the force generated by acceleration according to Newton's law:

$$F = ma \tag{4-172}$$

where F = force in lb
m = mass in $(lb - s^2)/ft$ (or slugs)
a = acceleration in ft/sec^2

If the acceleration is variable, as in sinusoidal movement, piezoelectric systems are ideal. In case of a constant acceleration, and hence a force that is also constant, strain gages may be employed. For petroleum applications in boreholes, however, it is better to use servo-controlled accelerometers. Reverse pendular accelerometers and "single-axis" accelerometers are available.

Figure 4-219 shows the schematic diagram of a *servo-controlled inverted pendular dual-axis accelerometer.* A pendulum mounted on a flexible suspension can oscillate in the direction of the arrows. Its position is identified by two detectors acting on feedback windings used to keep the pendulum in the median position. The current required to achieve this is proportional to the force ma_x, and hence to a_x.

This system can operate simultaneously along two axes, such as x and y, if another set of detectors and feedback windings is mounted in the plane perpendicular to xO_z, such as yO_z. The corresponding accelerometer is called a *two-axis accelerometer.*

Figure 4-220 shows the schematic diagram of a *servo-controlled single-axis accelerometer.* The pendulum is a disk kept in position as in the case of the reverse pendulum. Extremely efficient accelerometers can be built according to this principle in a very limited space. The Sunstrand accelerometer is seen in Figure 4-221.

Every accelerometer has a response curve of the type shown schematically in Figure 4-222. Instead of having an ideal linear response, a nonlinear response is generally obtained with a "skewed" acceleration for zero current, a scale factor error and a nonlinearity error. In addition, the skew and the errors vary with temperature. If the skew and all the errors are small or compensated in the accelerometer's electronic circuits, the signal read is an ideal response and can be used directly to calculate the borehole inclination. If not, "modeling" must be resorted to, i.e., making a correction with a computer, generally placed at the surface, to find the ideal response. This correction takes account of the skew,

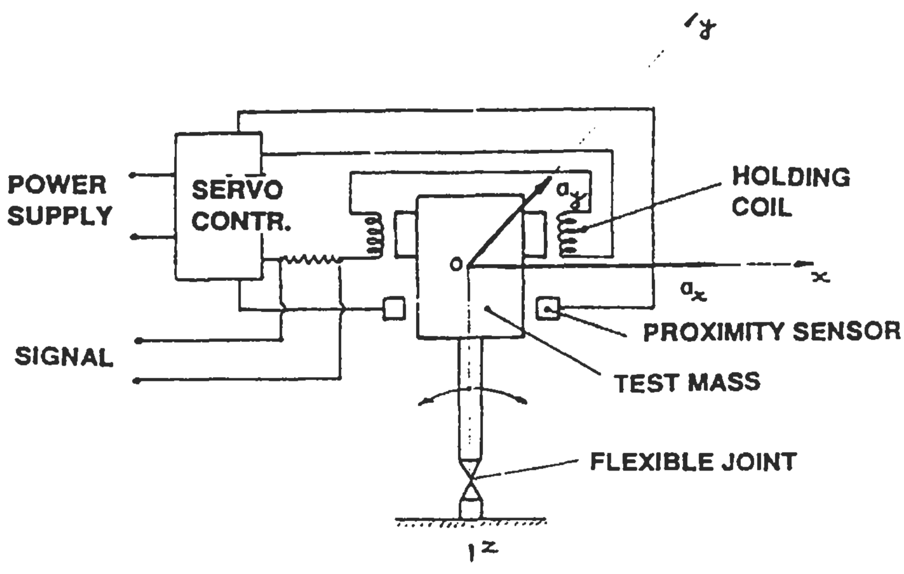


Figure 4-219. Sketch of principle of a servo-controlled inverted pendular dual-axis accelerometer.

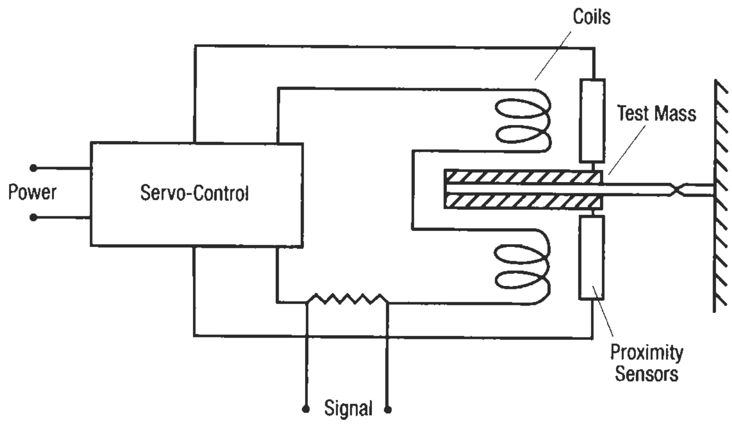


Figure 4-220. Schematic diagram of a servo-controlled single-axis accelerometer.

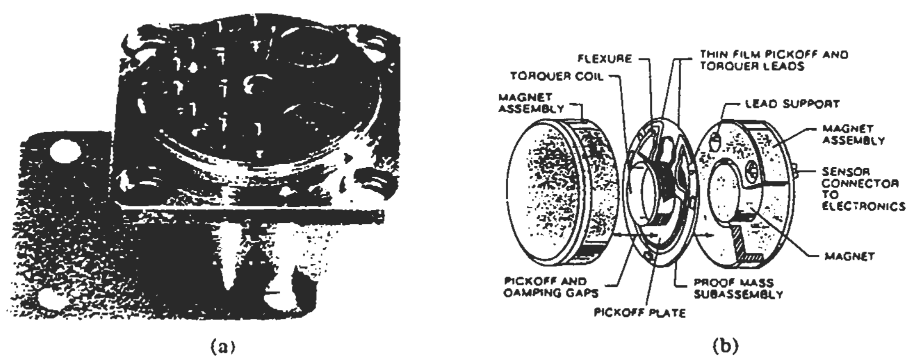


Figure 4-221. Servo-controlled "single-axis" Sunstrand accelerometer: (a) accelerometer photograph; (b) exploded view of the accelerometer. *(Courtesy Sunstrand* [102].*)*

all the errors, and their variation with temperature. In this case, the accelerometer temperature must be known. The maximum current feedback defines a measurement range beyond which the accelerometer is saturated. Vibrations must be limited in order not to disturb the accelerometer response.

Assume that the accelerometer has the ideal response shown in Figure 4-223, with a measurement range of 2 g (32.2 ft/s^2). We want to measure 1 g, but the ambient vibration level is ±3 g. In this case, the accelerometer's indications are shaved and the mean value obtained is not 1 g but 0.5 g. The maximum acceleration due to vibrations which are not filtered mechanically, plus the

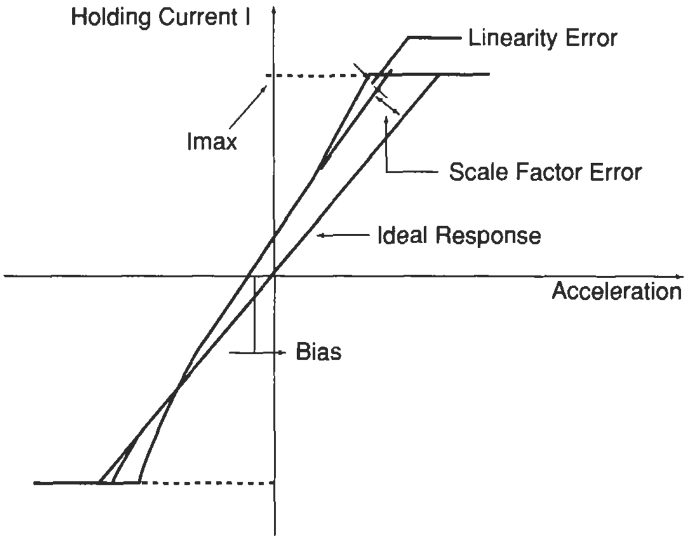


Figure 4-222. Accelerometer response.

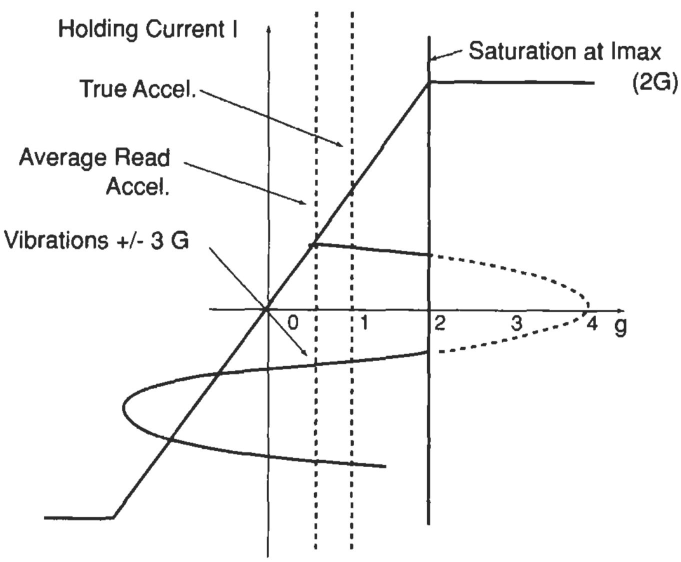


Figure 4-223. Effect of vibrations on an accelerometer response.

continuous component to be measured, must be less than the instrument's measurement range.

These instruments serve to measure the earth's gravitational field with a maximum value of 1 g. The typical values of the characteristics are:

- scale factor, 3 mA/g
- resolution, 10^{-6} g
- skew, 10^{-3} g
- service temperature, –55 to +150°C

For measurement ranges from 0 to 180°, three accelerometers mounted orthogonally must be used as shown in Figure 4-224. The x and y accelerometers are mounted with their sensitive axis perpendicular to the tool axis. The z accelerometer is mounted with its sensitive axis lined up with the tool axis.

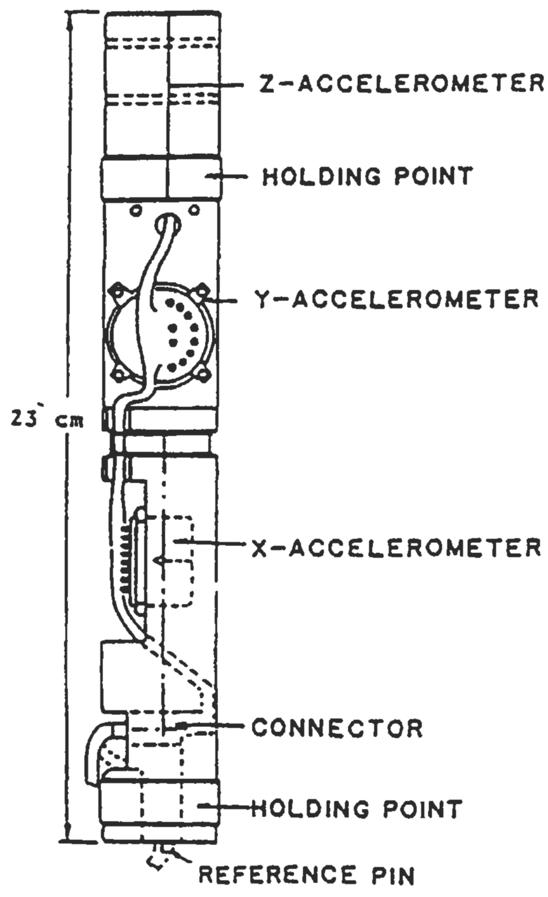


Figure 4-224. Mechanical drawing of the accelerometer section of a directional tool. *(Courtesy Sunstrand* [102].*)*

Figure 4-225 shows the inclination measurement using a triaxial sensor featuring three accelerometers. The three coordinates of the earth's gravitational acceleration vector serve to define this vector in the reference frame of the probe. The earth's acceleration is computed as

$$G = \sqrt{G_x^2 + G_y^2 + G_z^2} \tag{4-173}$$

It must be equal to the 32.2 ft/s²; otherwise the accelerometers are not working correctly. When the readings are in g units, G must be equal to one.

For the best accuracy, inclination less than 60° is computed with

$$i = \arcsin \frac{\sqrt{G_x^2 + G_y^2}}{G} \tag{4-174}$$

and for i greater than 60°

$$i = \arccos \frac{G_z}{G} \tag{4-175}$$

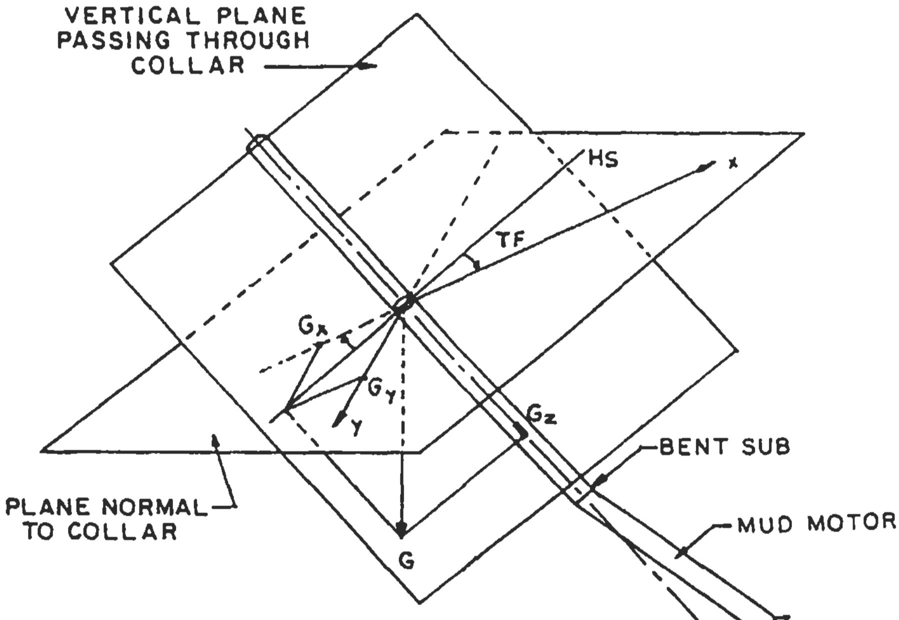


Figure 4-225. Vector diagram of the inclination measurement with three accelerometers.

The *gravity tool face angle,* or angle between the plane defined by the borehole axis and the vertical and the plane defined by the borehole axis and the BHA axis below the bent sub, can also be calculated. Figure 4-226 shows the gravity tool face angle. It is readily calculated using the equation

$$TF = \arctan \frac{-G_y}{G_x} \tag{4-176}$$

The gravity tool face angle is used to steer the well to the right, TF > 0, or to the left, TF < 0.

Typical specifications for a gravity sensor are as follows:

Temperature
- Operating = 0 to 200°C
- Storage = −40 to 200°C
- Scale factor = 0.01 V/°C

Power requirement
- 24 V nominal
- Less than 100 mA

Output impedance
- 10 Ω

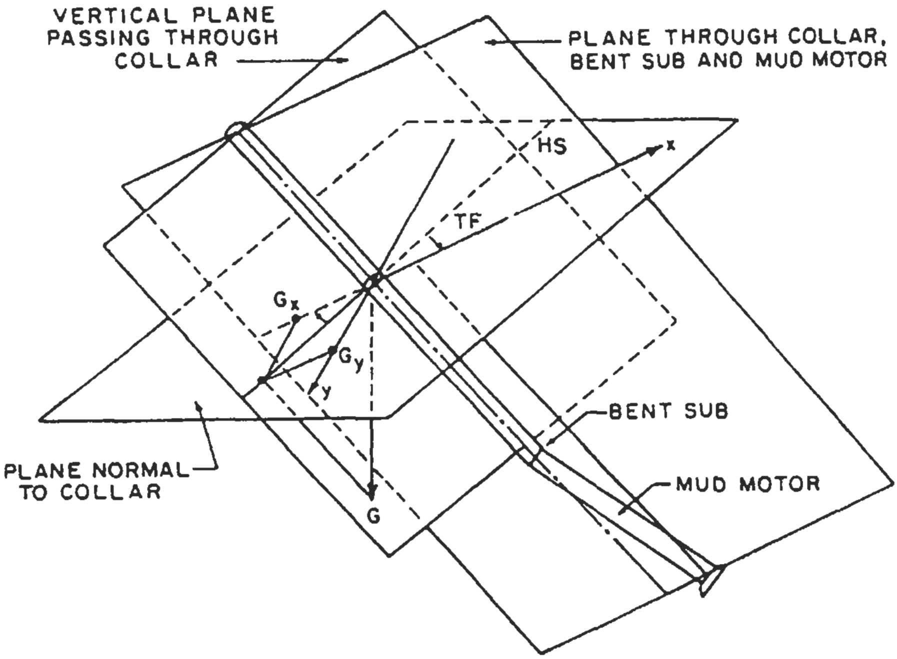


Figure 4-226. Solid geometry representing the gravity tool face angle concept.

Mechanical characteristics
- Length = 60 cm (24 in.)
- Diameter = 3.75 cm (1.5 in.)
- Mass = 2 kg (4 lb)
- Alignment = ±0.4°

Electrical characteristics
- Scale factor = 5 V/g ±1% (g = 32.2 ft/s^2)
- Bias = ±0.005 g @ 25°C
- Linearity = ±0.1% full scale

Environmental characteristics
- Vibrations = 1.5 cm p-p (peak-to-peak), 10 to 50 Hz
50g, 50 to 2000 Hz
- Shock = 2000 g, 0.5 ms, 0.5 sine

Magnetometers. Magnetometers used in the steering tools or MWD tools are of the flux-gate type.

The basic definition of a magnetometer is a device that detects magnetic fields and measures their magnitude and/or direction. One of the simplest types of magnetometers is the magnetic compass. However, due to its damping problems more intricate designs of magnetometers have been developed. The "Hall effect" magnetometer is the least sensitive. The "flux-gate" magnetometer concept is based on the magnetic saturation of an iron alloy core.

If a strip of an iron alloy that is highly "permeable" and has sharp "saturation characteristics" is placed parallel to the earth's magnetic field, as in Figure 4-227, some of the lines of flux of the earth's field will take a short cut through the alloy strip, since it offers less resistance to their flow than does the air. If we place a coil of wire around the strip, as in Figure 4-228 and pass enough electrical current through the coil to "saturate" the strip, the lines of flux due to earth's field will no longer flow through the strip, since its permeability has been greater reduced.

Therefore, the strip of iron alloy acts as a "flux gate" to the lines of flux of the earth's magnetic field. When the strip is not saturated, the gate is open and the lines of flux bunch together and flow through the strip. However, when the strip is saturated by passing and electric current through a coil wound on it, the gate closes and the lines of flux pop out and resume their original paths.

One of the basic laws of electricity, Faraday's law, tells us that when a line of magnetic flux cuts or passes through an electric conductor a voltage is produced in that conductor. If an AC current is applied to the drive winding A-A, of Figure 4-228, the flux gate will be opening and closing at twice the frequency of the AC current and we will have lines of flux from the earth's field moving in and out of the alloy at a great rate. If these lines of flux can be made to pass through

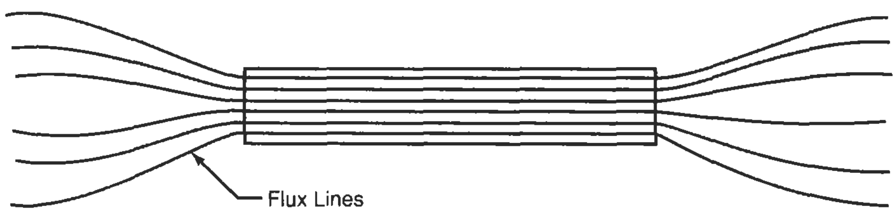


Figure 4-227. Magnetic flux-lines representation in a highly permeable iron alloy core.

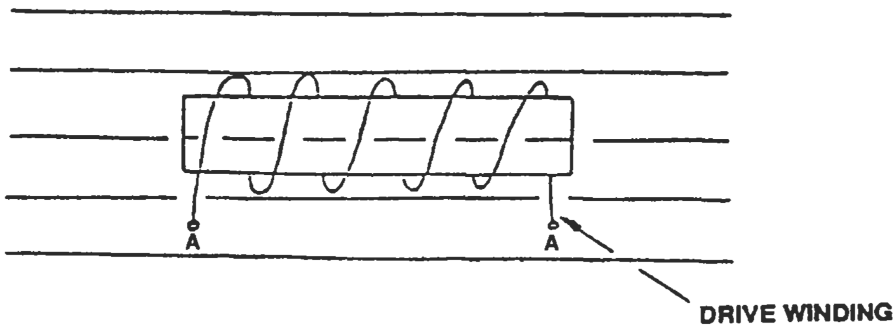


Figure 4-228. Magnetic flux-lines representation in a highly permeable iron alloy core saturated with an auxiliary magnetic field.

an electrical conductor ("sense winding"), a voltage will be induced each time they pop in or out of the alloy strip. This induced voltage in the sense windings is proportional to the number of lines of flux cutting through it, and thus proportional to the intensity of that component of the earth's magnetic field that lies parallel to the alloy strip.

When the alloy strip is saturated, a lot of other lines of flux are created that are not shown in Figure 4-228. The lines of flux must be sorted out from the lines of flux due to the earth's field to enable a meaningful signal to be produced. A toroidal core as shown in Figure 4-229 will enable this separation of lines of flux to be accomplished. The material used for the toroidal core is usually mu metal.

Each time the external lines of flux are drawn into the core, they pass through the sense windings B-B to generate a voltage pulse whose amplitude is proportional to the intensity of that component of the external field that is parallel to the centerline of the sense winding. The polarity, or direction of this pulse, will be determined by the polarity of the external field with respect to the sense windings. When the flux lines are expelled from the core they cut the sense

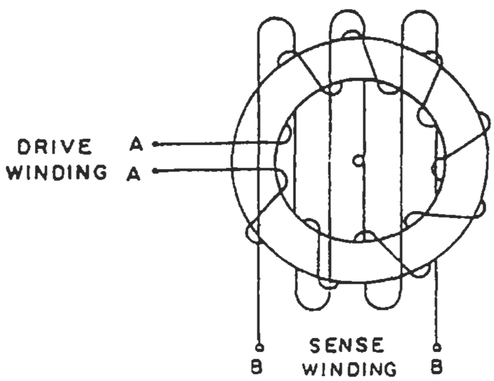


Figure 4-229. Sketch of principle of a single-axis flux-gate magnetometer.

windings in the opposite direction and generate another voltage pulse of the same amplitude but of opposite polarity or sign. Because of this voltage pulse occurring at twice the driving voltage frequency, the flux gate is sometimes known as "second harmonic magnetometer."

The main advantages of the flux gate magnetometers are that they are solid-state devices much less sensitive to vibration than compasses, they have uniaxial sensitivity, and they are very accurate.

Typical specifications are:

Temperature
- Operating = 0 to 200°C
- Storage = -20 to 200°C

Power requirement, output in impedance, and *mechanical characteristics* are similar to the acceleromete sensors.

Electrical characteristics
- Alignment = ±0.5°
- Scale factor = 5 V/G ±5%
- Bias = ±0.005 G @ 25°C
- Linearity = ±2% full scale

Note: 1 gauss = 1G = 10^{-4} tesla

Environmental characteristics
- Vibrations = 1.5 cm p-p, 2 to 10 Hz
  20 g, 10 to 200 Hz
- Shock = 1000 g, 0.5 ms, 0.5 sine

Figure 4-230 shows the photograph of a Develco high-temperature directional sensor. For all the sensor packages, calibration data taken at 25, 75, 125, 150, 175 and 200°C are provided. Computer modeling coefficients provide sensor accuracy of ±0.001 G and ±0.1° alignment from 0 to 175°C. From 175 to 200°C the sensor accuracy is ±0.003 G and ±0.1° alignment.

Example 1: Steering Tool Measurements—Tool Face, Deviation

Single-axis accelerometer systems are used in the steering tools and MWD tools for inclination and tool face data acquisition. Using a spreadsheet, compute the current values for each accelerometer in the following cases:

Use a spreadsheet for a three single-axis accelerometer system mounted in a steering tool or a MWD tool and compute the output current values for each accelerometer in the following cases:

1. Tool-face angle: 0°
   Hole deviation: 0, 15, 30, 45, 60, 75, 90°
2. Hole deviation: 30°
   Tool-face angle: −180, −135, −90, −45, 0, 45, 90, 135, 180°

The usual conventions as shown in Figure 4-231 are:

- Axis x lines up with the mule shoe key and the tool face.
- Axis y is perpendicular to O_x and O_z.
- Axis z is the same as tool axis or borehole axis, oriented downward.

Figure 4-230. Photograph of a high-temperature directional sensor with three accelerometers and three magnetometers. *(Courtesy Develco* [103].*)*

- The tool face angles are counted looking downward, clockwise positive and counterclockwise negative.
- We will assume a perfect accelerometer calibration line that reads 3 mA for 1 g of acceleration.

Solution

Tables 4-118 and 4-119 give answers to Example 1 in tabular form.

Example 2: Steering Tool Measurements—Tool Face, Deviation, and Azimuth

The following set of data have been recorded with a MWD directional package:

$G_x = -0.2$ mA
$G_y = 0.1$ mA
$G_z = 2.99$ mA
Accelerometer sensitivity: 3 mA = 1 g

$H_x = 0.1$ G
$H_y = -0.2$ G
$H_z = 0.484$ G
Earth magnetic field amplitude: 0.52 G
Earth magnetic field inclination/vertical: 32°

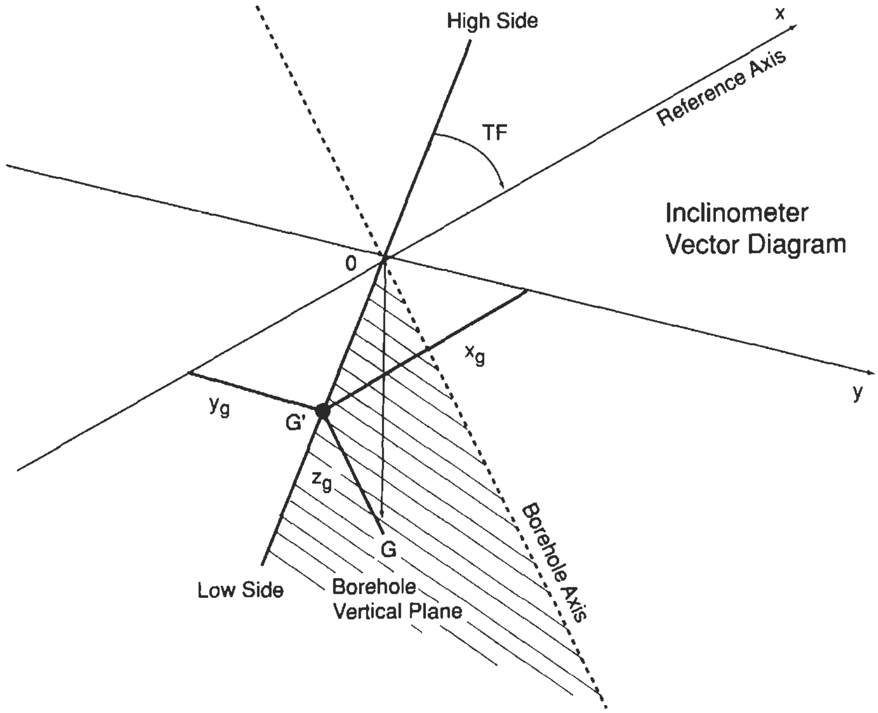


Figure 4-231. Vector diagram for the tool-face determination.

Table 4-118
Accelerometer Output for 0° Tool-Face Angle and Various Tool-Face Angles

Hole Deviation (°)	Tool Face (°)	Axis x (mA)	Axis y (mA)	Axis z (mA)
0	0	0.00	0.00	3.00
15	0	0.78	0.00	2.90
30	0	1.50	0.00	2.60
45	0	2.12	0.00	2.12
60	0	2.60	0.00	1.50
75	0	2.90	0.00	0.78
90	0	3.00	0.00	0.00

Table 4-119
Accelerometer Output for 30° of Hole Deviation Angle and Various Tool-Face Angles

Hole Deviation (°)	Tool Face (°)	Axis x (mA)	Axis y (mA)	Axis z (mA)
30	−180	−1.50	0.00	2.60
30	−135	−1.06	−1.06	2.60
30	−90	0.00	−1.50	2.60
30	−45	1.06	−1.06	2.60
30	0	1.50	0.00	2.60
30	45	1.06	1.06	2.60
30	90	0.00	1.50	2.60
30	135	−1.06	1.06	2.60
30	180	−1.50	0.00	2.60

1. Compute the inclination of the borehole. Are the accelerometers working properly? Why?
2. Compute the tool face angle, clockwise and counterclockwise. If we drill ahead with this angle is the hole going to turn left or right?
3. Compute the field disturbance H_{dc} due to the drill collars.
4. Compute the inclination of the corrected magnetic field. Does it check with the local data? What could prevent this inclination from being correct?
5. Give the principle of one of the borehole azimuth calculation methods.

Solution

1. The inclination is i = 4.27°. Yes, the accelerometers work properly because

$$\sqrt{G_x^2 + G_y^2 + G_z^2} = 3\,\text{mA}$$

2. The tool-face angle is TF = 26.56°. The borehole is turning right.
3. H_z = 0.469 G, H_{DC} = 0.014 G.
4. Corrected field inclination: 25.69°. External disturbance: nearby casing or drill collar hot spots.
5. If Z is the borehole axis unit vector,
   compute $\mathbf{A} = \mathbf{G} \times \mathbf{Z}$ (vector product)
   $\mathbf{B} = \mathbf{G} \times \mathbf{H}$
   and

$$\cos\alpha = \frac{\mathbf{A} \bullet \mathbf{B}}{|\mathbf{A}| \bullet |\mathbf{B}|} \qquad \text{(scalar product)}$$

Example 3: Steering Tool Measurements—Tool Face, Deviation, and Azimuth

A steering tool is normally used during drilling with a mud motor and is connected to the surface with an electric wireline. The sensing devices shown in Figure 4-232 are also used in most MWD mud pulse systems. The coordinates

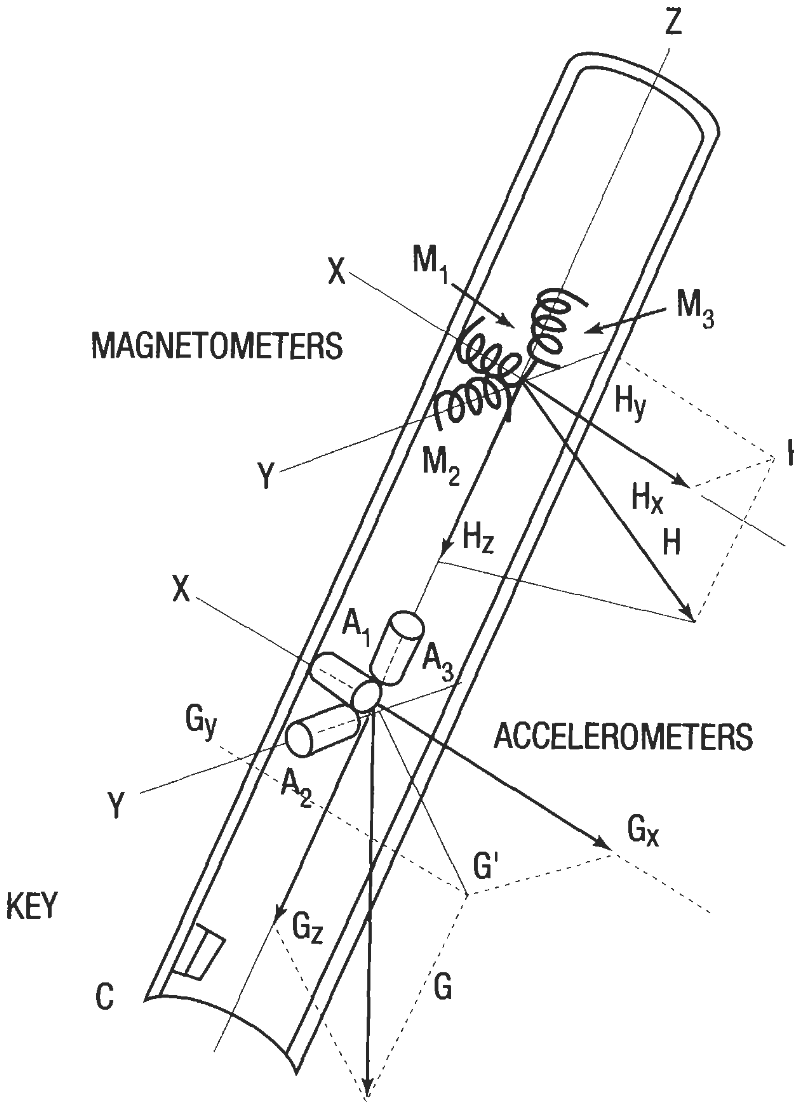


Figure 4-232. Schematic view of the sensor arrangement in a steering tool.

of two vectors are permanently measured during drilling in the frame of reference of the sonde. They are

- Vector gravity G, which represents the vertical direction and is defined by the coordinates G_x, G_y and G_z.
- Vector magnetic field, which is located in the north vertical plane and is defined by the coordinates H_x, H_y and H_z.

Also needed:

- Vector well direction located along the well (sonde axis) and is defined by the coordinates $Z_x = 0$, $Z_y = 0$ and $Z_z = 1$.
- O_x is lined up with the mule shoe key and the tool face direction.

For the numerical applications, we shall have:

- Accelerometer scale factor 3 mA/g, $I_x = -2$ mA, $I_y = 1$ mA, $I_z = 2$ mA at a given depth,
- Magnetometer readings: $H_x = -0.1077$ G, $H_y = 0.2$ G, $H_z = 0.45$ G at the same depth,
- Magnitude of the magnetic field: 0.52 G, magnetic field inclination: 30° with respect to the vertical.

1. Compute the borehole deviation. Show that a check of the accelerometer readings is possible if we assume that the **G** vector module is g.
2. Compute the tool face orientation. In the numerical application above, is the borehole going to turn right, left or go straight if we keep on drilling with this orientation?
3. Show that we can check the magnitude of the magnetic field vector and correct for an axial field due to the drill collars.
4. Compute the dip angle of the magnetic field vector after correction for the drill collar field, it should check with the local magnetic field data. What do you conclude if it does not?
5. Compute the orientation of the borehole with respect to magnetic north without axial field correction.
6. Write an interactive computer program for solving the above questions.

Solution

1. i = 48.2°; 3 mA.
2. TF = +26.5°; turning right.
3. Drill collar magnetic field = 0.0178 G; H_z corrected = 0.468 G.
4. λ = 30° from vertical.
5. One way of making the calculation is to use the three vectors:
   G (G_x, G_y, G_z)
   H (H_x, H_y, H_z)
   Z (0, 0, 1)
   (See Figure 4-233 a and b).
   a. Compute the coordinates of vector A normal to vector **G** and vector **H**. Vector **A** = cross-product of vector **G** by vector **H**.
   b. Compute the coordinates of vector **B** normal to vector **G** and vector **Z**. Vector **B** = cross-product of vector **G** by vector **Z**.
   c. Compute the angle between vector **A** and vector **B**. Being both normal to vector **G**, they are in the horizontal plane. The angle represents the azimuth. In some configurations 180° must be added. The angle is computed by making the scalar product of vector **A** by vector **B**.

   $$\mathbf{A} \bullet \mathbf{B} = |\mathbf{A}| \bullet |\mathbf{B}| \cos Az = A_x B_x + A_y B_y + A_z B_z$$

   Care must be exercised since cos(Az) = cos(−Az).
   d. Numerical results: Angle between vertical planes, 31.71°; azimuth, 328.29°.

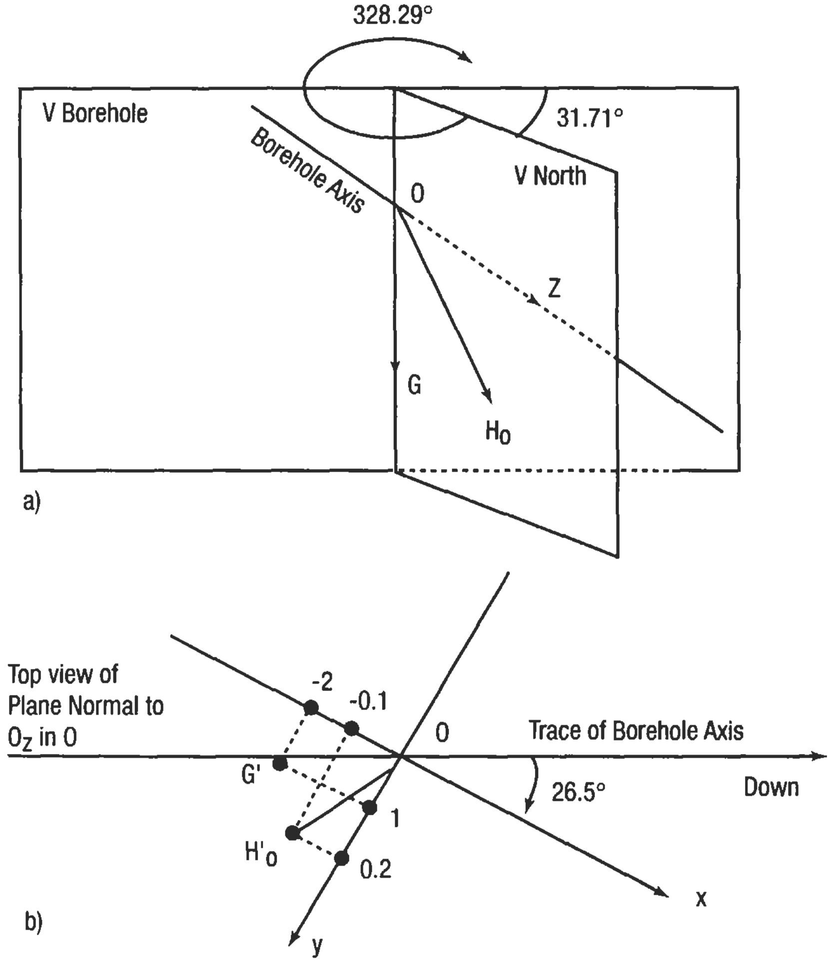


Figure 4-233. Representation of the three main vectors: (a) solid geometry view; (b) projection on plan normal to O_z.

Note: Another way, probably less ambiguous, to compute the azimuth is to make a rotation of coordinates around Oz to bring Ox in the vertical plane and Oy in the horizontal plane. Then, make another rotation around Oy to bring Oz vertical and Ox in the horizontal plane. The azimuth is the clockwise angle between the new OH_o' and Ox.

Example 4: Steering Tool Measurements—Trajectory Forecast

An interesting problem that can be solved with the steering tool or the MWD measurements is the trajectory forecast when drilling ahead with a given bent sub (constant angle), and a given tool-face angle.

1. After drilling the mud motor length with a given tool face angle and a given bent sub angle, what is the borehole deviation and orientation likely to be at the mud motor depth?

Using the drawing Figure 4-234 find the algorithm to compute these angles. Write a short computer program and use the data of Example 3 for a numerical application with a 2° sub.

2. If we change the tool face angle to -30° (turning left), what will be the probable borehole deviation and azimuth after drilling another motor length? Use the same computer program.

Note: We will assume that the borehole axis is the same as the drill collar axis at the steering tool depth and also that the borehole axis is the same as the mud motor axis at the mud motor depth.

Solution

The same algorithms are used as in Example 3. The vector **Z** (0, 0, 1) is replaced by vector **Z** (sin 2° • cos TF, sin 2° • sin TF, cos 2°). This new vector **Z** should be used to compute the new inclination, using the scalar product

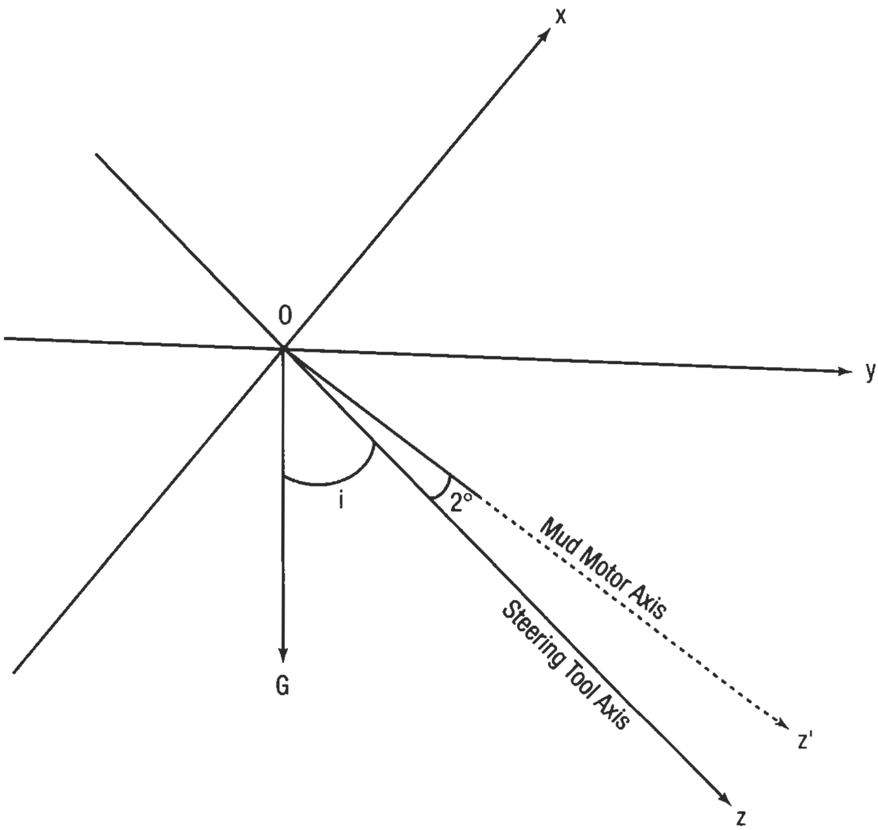


Figure 4-234. Vector diagram showing the mud motor axis as well as the steering tool axis.

between vector **G** and new vector **Z**. The new vector **Z** should also be used to compute the new azimuth.

Vibrations and Shocks

Measurements while drilling are made with sensors and downhole electronics that must operate in an environment where vibrations and shocks are sometimes extremely severe. A brief study of vibrations and shocks will be made to understand better the meaning of the specifications mentioned earlier.

The vibration frequencies encountered during drilling are well known. They correspond to the rotation of the drill bit, to the passing of the bit rollers over the same hard spot on the cutting face, and to the impact of the teeth. Figure 4-235 gives the order of magnitude for frequencies in hertz (60 rpm = 1 Hz). In each type the lowest frequencies correspond to rotary drilling and the highest ones correspond to turbodrilling. Three vibrational modes are encountered:

1. axial vibrations due to the bouncing of the drill bit on the bottom
2. transverse vibrations generally stemming from axial vibrations by buckling or mechanical resonance
3. angular vibrations due to the momentary catching of the rollers or stabilizers

In vertical rotary drilling, the drillpipes are almost axially and angularly free. Therefore, the highest level of axial and angular vibration is encountered for this type of drilling. In deviated rotary drilling, the rubbing of the drill string on the well wall reduces axial vibrations, but the stabilizers increase angular vibrations. In drilling with a downhole motor, the rubbing of the bent sub on the well wall reduces the amplitude of all vibrations.

Vibrations are characterized by their peak-to-peak amplitude at low frequencies or by their acceleration at high frequencies. Assuming that vibration is sinusoidal, the equation for motion is

$$x = \frac{A}{2} \sin 2\pi f \times t \qquad (4\text{-}177)$$

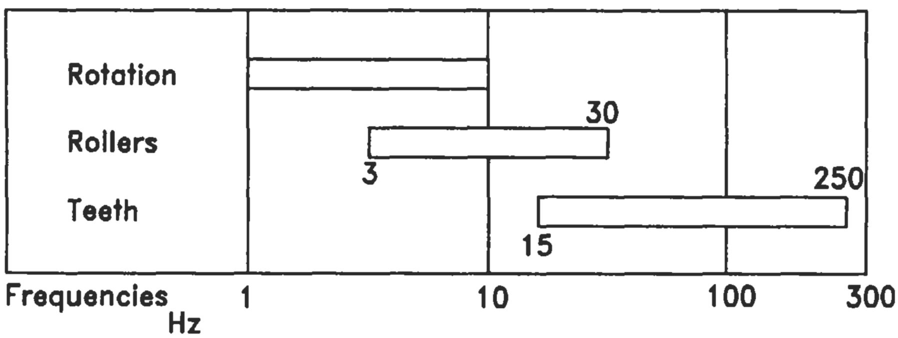


Figure 4-235. Main vibration frequencies encountered while drilling.

where x = elongation in m
A = peak-to-peak amplitude in m
f = frequency in Hz
t = time in s

By deriving twice, acceleration becomes

$$a = -\frac{A}{2}(2\pi f)^2 \sin 2\pi f \times t \qquad (4\text{-}178)$$

Maximum acceleration is thus $a_m = 2A\pi^2 f^2$. For example, peak-to-peak 12 mm at 10 Hz corresponds to $a_m = 11.8\ m/s^2 = 1.2\ g$. Acceleration of gravity is expressed as g.

Very few vibration measurements are described in the literature, but the figures in Figure 4-236 can be proposed for vertical rotary drilling. The lower limits correspond to soft sandy formations and the upper limits to heterogeneous formations with hard zones. Table 4-120 gives the specifications that the manufacturers propose for several tools.

The shocks that measuring devices are subjected to are generally characterized by an acceleration (or deceleration) and a time span. For example, a device is said to withstand 500 g (5,000 m/s^2) for 10 ms. This refers to a "half-sine." Shock testing machines produce a deceleration impulse having the form shown in Figure 4-237.

In the preceding example, $a_m = 500$ g, $t_2 - t_1 = 10$ ms. The impulse represented as a solid line is approximately equivalent to the rectangular amplitude impulse $0.66\ a_m$. This impulse can be used for calculating the deceleration distance, which is

$$d = \int\int_{t1}^{t2} a\,dt = \frac{1}{2} 0.66 a_m t^2$$

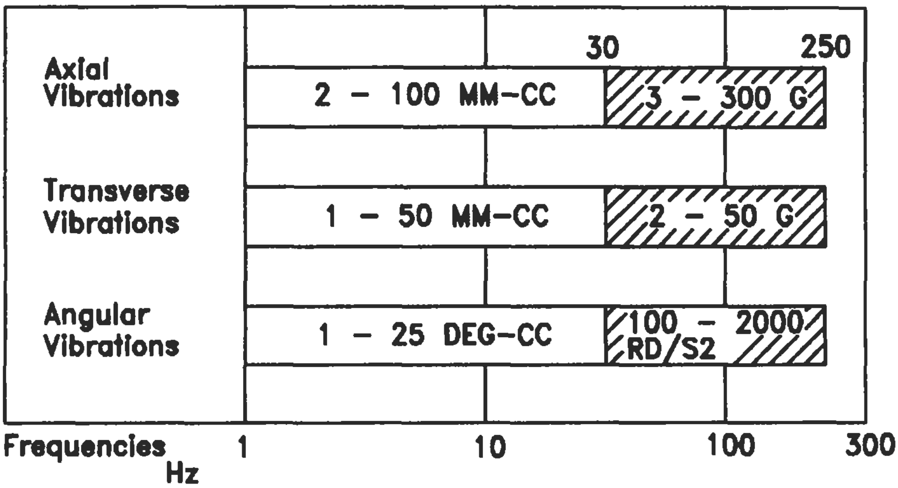


Figure 4-236. Order of magnitude of the vibration amplitudes encountered during drilling.

Table 4-120
Resistance of Some Directional Tools or Components to Vibrations

Tool	Low Frequencies	High Frequencies
Azintac (1)	1 mm cm^3 (5–50 Hz)	10 g (50–500 Hz)
Drill-director (2)	2 g (5–45 Hz)	5 g (45–400 Hz)
Q-Flex accelerometer		25 g
Develco accelerometer	12.7 mm cm^3 (20–40 Hz)	40 g (40–2000 Hz)
4-Gimbal gyroscope	2 g (10–200 Hz)	
2-Axis gyroscope	5 g (10–300 Hz)	
On-shore military specifications		14 g (50–2000 Hz)

(1) Iust. Fr. du Pet. trademark
(2) Humphrey Inc. trademark
(The acceleration values correspond to maximum amplitudes.)

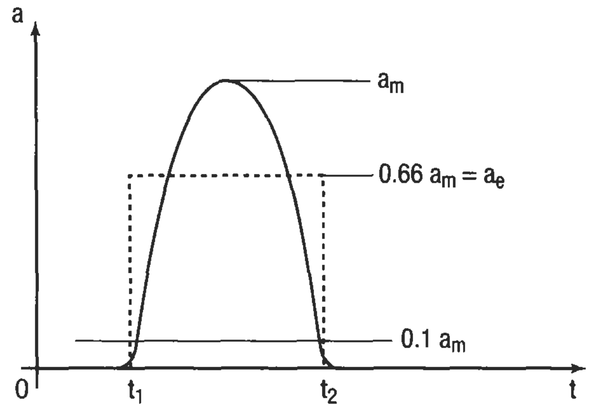


Figure 4-237. Theoretical deceleration variation during a shock or impact.

and the velocity at the beginning of the deceleration:

$$v = \int_{t1}^{t2} a\,dt = 0.66 a_m t$$

For 500 g, 10 ms, we would have

$$d = 0.33 \times 5000 \times (0.01)^2 = 0.16 \text{ m}$$

$$v = 0.66 \times 5000 \times 0.01 = 33 \text{ m/s}$$

assuming that 1 g = 10 m/s^2.

Assuming the deceleration constant, as in the square approximation of Figure 4-237, we have a constant braking force, which is

$$F = ma = 0.66\ m\,a_m \qquad (4\text{-}179)$$

where m is the decelerated mass. For a 2-kg mass and a_m = 500 g (5,000 m/s²)

$$F = 2 \times 0.66 \times 5000 = 660 \text{ daN}$$

Note: 1 daN = 2.25 lb

Measuring devices run inside the drillstring are mainly subject to axial impacts. These shocks come from sudden halts in the mule shoe or from an obstruction in the string. The measuring devices used while drilling are generally subjected to axial and angular impacts caused by the bouncing of the bit on the bottom and by the catching of the rollers and stabilizers on the borehole walls. There is very little information in the literature about measuring impacts while drilling.

Table 4-121 gives the specifications compiled by manufacturers for several measuring devices. It can thus be seen that such devices must be equipped with an axial braking system capable of having a stroke of 10 and even 20 cm.

Future Developments. Orientation measurements while drilling are practically impossible with gimbal gyroscopes. Two-axis flexible-joint gyroscopes should be able to withstand vibrations and impacts while maintaining a sufficiently accurate heading provided that periodic recalibration is performed by halting drilling and switching on the north seeking mode. In the more distant future, laser or optical-fiber gyroscopes that have been suitably miniaturized should provide a solution.

Example 5: Vibration and Shock Analysis—Measurement Package Design

An MWD sensor package has the following specifications:

- package mass: 0.906 kg (weight: 2 lb)
- maximum vibrations allowable (all axes)
  0.5-in p-p for 2 to 10 Hz
  20 g for 10 to 200 Hz
- shocks: 1000 g, 0.5 ms (all axes) ($g = 9.81$ m/s² = 32.17 ft/s²)

Table 4-121
Resistance of Some Directional Tools or Components to Axial Shocks or Impacts

Tool	Deceleration (g, G)	Time (ms)	Braking Distance (m)	Initial Velocity (m/s)
Azintac	60	11	0.024	4.35
Drill-director	700	10	0.23	46.2
Q-Flex accelerometer	250	11	0.10	18.15
Develco accelerometer	400	1	0.0013	2.64
4-Gimbal accelerometer	50	10	0.016	3.3
2-Axis gyroscope	100	10	0.033	6.6
Military specifications	30 to 100	10		

1. *Vibrations:*
   a. Compute the maximum acceleration that the instrument will accept at 2 and at 10 Hz.
   b. Compute the peak to peak motion which can be applied to the instrument at 10 and 200 Hz.
   c. Compare the 10-Hz values. At what frequency will the peak to peak data of the low frequency be consistent with the acceleration data of the high frequency?
   d. The package is held in a housing with several rubber rings laterally assumed to behave like perfect springs. Two different ring stiffnesses are available with total values of:
      - 100 lb/in
      - 10,000 lb/in

   Compute the resonant frequencies for lateral vibrations. In the frequency range usually encountered in drilling, which one should be used?
2. *Shocks along the borehole axis* (sensor package axis):
   Assume that the shock specification refers to the maximum deceleration (a_{max}) of a half sine wave impact. The mean deceleration will be taken equal to 0.66 a_{max}.
   a. Assuming a dampener in the tool housing exerting a constant force and the housing stopping abruptly, compute according to the specifications:
      - the distance of deceleration
      - the velocity at the beginning of the deceleration
      - the braking force applied to the sensor package
   b. Now if the braking force is supplied with a coil spring of 5,000 lb/in. compute:
      - the braking length for the velocity calculated in a.
      - the maximum deceleration, is it acceptable?
      - will such a coil spring be suitable for the vertical vibrations generated in rotary drilling?

Solution

1. a. 0.1 and 2.55 g
   b. 3.91 and 0.039 in.
   c. At 10-Hz amplitudes: 0.5 and 3.91 in.; 28 Hz
   d. 22 and 221 Hz; 10,000-lb/in. ring more suitable since frequency further from drilling frequencies
2. a. d = 0.03 in.; v = 10.6 ft/s; F = 1320 lb
   b. x = 0.13 in.; a = 324 g (a_m = 491 g); acceptable; f = 156 Hz; acceptable

Example 6: Vibration and Shock Analysis—Mule Shoe Engaging Shock

A steering tool sensor and electronic package is mounted in a housing in Figure 4-238 with a shock absorber and a spring to decrease the value of deceleration when engaging the mule shoe.

The package has a mass of 2 kg or a weight of 4.415 lb. Assume a downward velocity of about 10 ft/s. The shock absorber develops a constant force (independent of the relative velocity) of 10 lb (44.48 N). The spring stiffness is 57.10 lb/in. The potential energy due to gravity will be neglected.

1. Taking into account *only* the shock absorber, compute the distance x traveled by the instrument package with respect to the housing when the

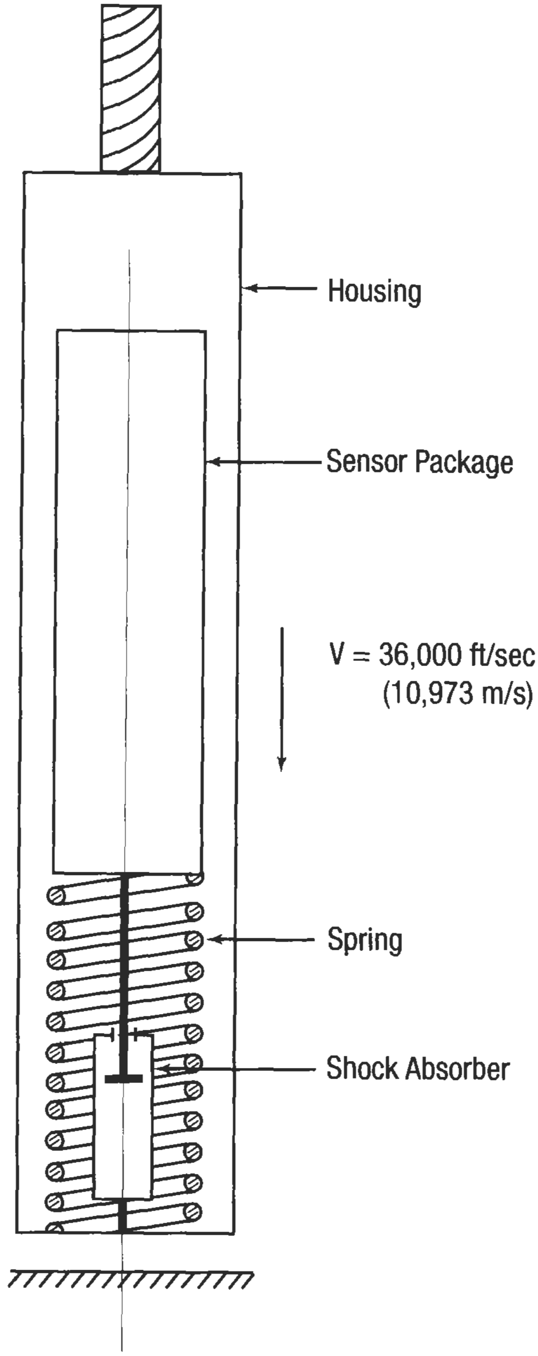


Figure 4-238. Schematic representation of a shock dampening mechanism.

housing stops abruptly. Compute the maximum deceleration x in g's and the deceleration time t. All calculations can be made in English or metric units.
2. Taking into account the spring *only* compute the distance traveled x and the maximum deceleration a in g's when the housing stops abruptly.
3. Now if both the spring and the shock absorber are acting together, what will be the distance traveled x and the maximum deceleration a in g's when the housing stops abruptly?
4. What is the advantage in adding a shock absorber to the spring in such a system?

Solution

1. Neglect the effect of gravity. Energy balance: $Fx = \frac{1}{2} mv^2$ ($v = 3.048$ m/s)
   $x = mv^2/2F = 0.209$ m
   $a = F/m = 22.24\ m/s^2 = 2.26$ g
   $x = \frac{1}{2}at^2 \rightarrow t = (2x/a)^{1/2} = 0.137$ s

   $x = 0.209$ m $= 0.685$ ft $= 8.23$ in. $= 20.9$ cm
   $a = 2.26$ g
   $t = 0.137$ s $= 137$ ms
2. Before the tool hits, going down at a constant velocity, the spring is already compressed by the weight of the instrument package, so the effect of gravity can be neglected. Energy balance:
   $\frac{1}{2}mv^2 = \frac{1}{2}kx^2$
   $x = (mv^2/k)^{1/2} = 0.043$ m $= 0.141$ ft $= 1.69$ in.
   $F_{max} = kx = 430$ N
   $a = F/m = 215\ m/s^2 = 705.38\ ft/s^2 = 21.9$ g

   $x = 0.043$ m $= 0.141$ ft $= 4.3$ cm $= 1.69$ in.
   $a = 21.9$ g
3. Still neglecting gravity. Energy balance: $Fx + \frac{1}{2}kx^2 = \frac{1}{2}mv^2$
   $kx^2 + 2Fx - mv^2 = 0$
   $x = [-F + (F^2 + kmv^2)^{1/2}]/k = 0.0388$ m $= 0.127$ ft $= 1.53$ in. $= 3.88$ cm
   $F_T = F_s + 1/2\ kx = 432.5$ N
   $a = F/m = 216.2\ m/s^2 = 709.5\ ft/s^2 = 22.03$ g

   $x = 3.88$ cm $= 1.53$ in.
   $a = 22.03$ g
4. Oscillations will be dampened; x slightly decreased: 3.88 cm versus 4.3 cm.

Teledrift and Teleorienter

The first transmission of data during drilling using mud pulses was commercialized by B.J. Hughes Inc. in 1965 under the name of *teledrift* and *teleorienter*. Both tools are purely mechanical. A general sketch of principle is given in Figure 4-239. The tool is now operated by Teledrift Inc.

The tool generates at bottom positive pulses by restricting momentarily the flow of mud each time that the mud flow (pumps) is started. The pulses are detected at surface on the stand pipe and recorded as a function of time.

Figure 4-240 shows the sketch of principle of the teledrift unit which is measuring inclination.

A pendulum hangs in a conical grooved bore. A spring tends to move the pendulum and the poppet valve upwards when the circulation stops. If the tool

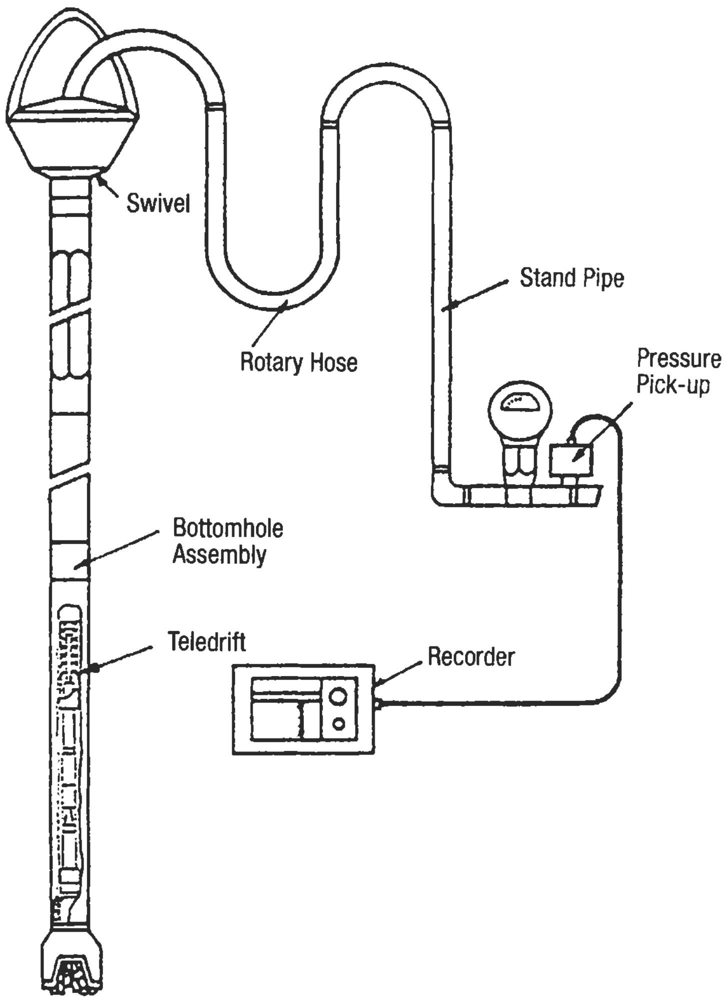


Figure 4-239. Sketch of principle of the teledrift tool or teleorienter tool attached to the drillstring. *(Courtesy Teledrift, Inc.* [104].*)*

is inclined, the pendulum catches the grooves at different levels according to the inclination and stops there. For example, for minimum inclination (Figure 4-240b) it stops the poppet valve past the first restriction. In Figure 4-240d, the poppet valve stops past the seventh restriction due to the high inclination.

When the circulation is started, the poppet valve travels slowly down, generating one pressure pulse when passing each restriction. The measurement range in the standard tool is of 2.5° (also 7° ranges, 1° increments, max. 17°).

Table 4-122 gives the inclination angles corresponding to one to seven pulses with three cones. Fifteen cones are available. The maximum measurable angle is 10°. The range must be selected before lowering the drillstring.

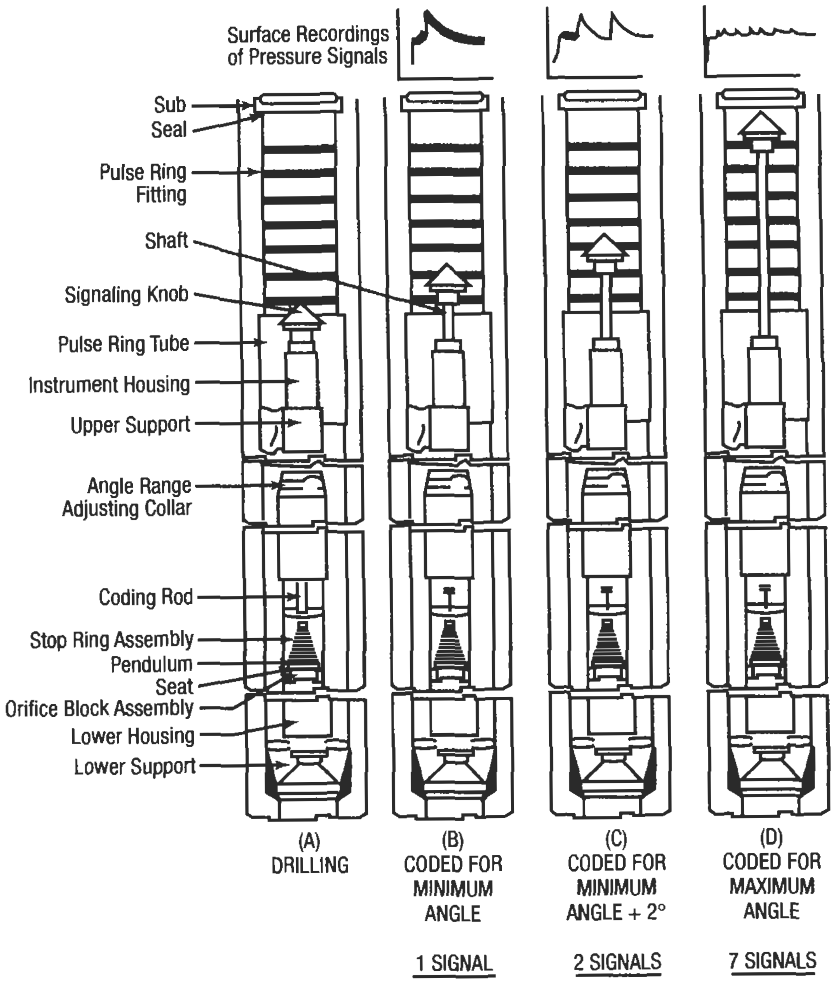


Figure 4-240. Teledrift mechanism in various coding positions. *(Courtesy Teledrift, Inc.* [104].*)*

Table 4-122
Teledrift Angle Range Settings

Angle Range	Deviation Angle in Degrees						
	1 signal	2 signals	3 signals	4 signals	5 signals	6 signals	7 signals
0.5–3.0	0.5	1.0	1.5	2.0	2.5	3.0	3+
1.0–3.5	1.0	1.5	2.0	2.5	3.0	3.5	3.5+
7.5–10.0	7.5	8.0	8.5	9.0	9.5	10.0	10.0+

The cone can be replaced by a mechanism that senses the angular position rather than the inclination of the drillstring.

The tool is then sensitive to the tool face and is called the *teleorienter.*

Figure 2-241 shows the read-out display of the driller in a zero tool-face position. Four mud pressure pulses will be recorded each time the pumps are started. In Figure 4-242a, a tool-face value of 20° is indicated by three pulses, turning to the right. In Figure 4-242b, a tool-face value of −20° is indicated by five pulses, and the borehole is turning left.

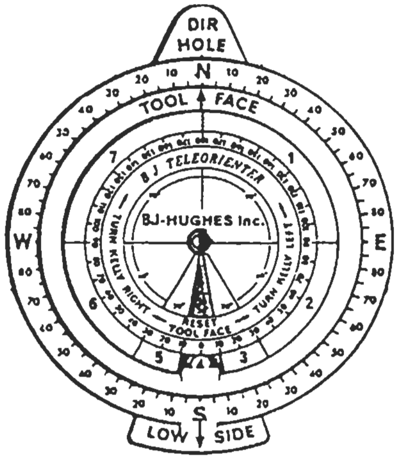
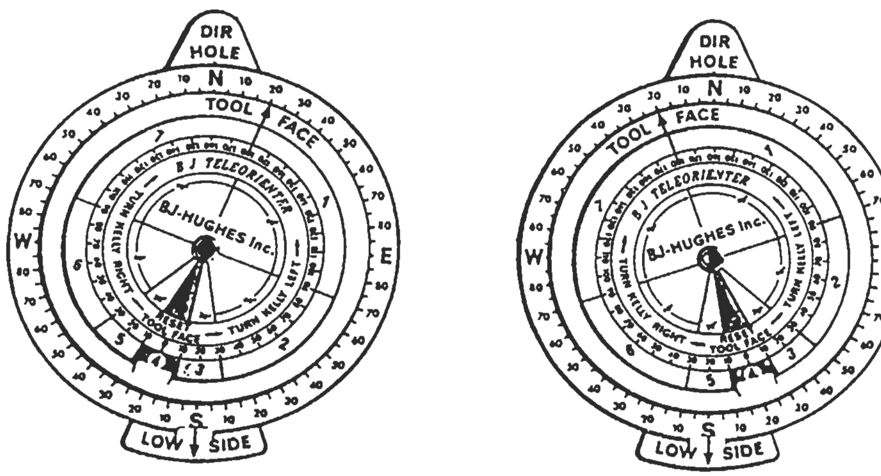


Figure 4-241. Driller read-out display of the teleorienter in a zero tool face angle "go straight, build angle" position. *(Courtesy Teledrift, Inc.* [104].*)*



(a) (b)

Figure 4-242. Driller read-out display of the teleorienter: (a) +20° tool-face angle, turning right position; (b) −20° tool-face angle, turning left position. *(Courtesy Teledrift, Inc.* [104].*)*

These tools have been widely used in the past by the cost conscious operators. The tool could be rented and operated by the rig floor personnel. However, the inclination ranges are limited, only one tool, teledrift or teleorienter, can be used during a trip, and only the tool-face angle is read by the teleorienter, not the azimuth.

These tools are still available but tend to be replaced by the MWD systems.

Mud Pressure and EM Telemetry

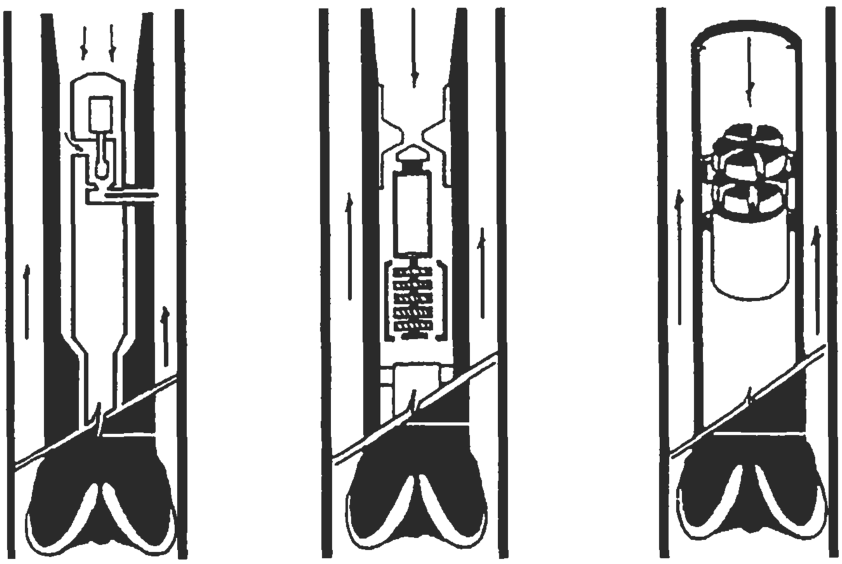
Two methods are currently used to transmit data from downhole to surface: mud pressure telemetry and electromagnetic earth transmission.

There are three principles for transmitting data by drilling mud pressure:

1. positive pulses obtained by a momentaneous partial restriction of the downhole mud current;
2. negative pulses obtained by creating a partial and momentaneous communication between the drill string internal mud stream and the annular space at the level of the drill collars;
3. phase changes of a low-frequency oscillation of the drilling mud pressure induced downhole in the drillstring.

Figure 4-243 shows sketches of the three systems.

Transmission by Positive Pulses. This system is used by Inteq/Teleco. It is placed in a nonmagnetic drill collar containing sensors of the flux-gate type



(a) (b) (c)

Figure 4-243. Telemetry systems using mud pressure waves: (a) negative pulse system; (b) positive pulse system; (c) continuous wave system.

for measuring the direction of the earth's magnetic field and accelerometers for measuring the gravity vector. An electromagnetic and electronic unit, every time rotation is halted, calculates and memorizes the azimuth, drift and tool face angles. Bottomhole electric power is supplied by an AC generator coupled with a turbine situated on the mud stream in the drill collar.

In rotary drilling a rotation detector triggers angle measurements when the string stops rotating with circulation maintained. With a downhole motor the measurements are repeated as long as mud continues to circulate. The transmission uses a ten-bit digital coding. Figure 4-244 gives a schematic diagram of the positive pulse generator.

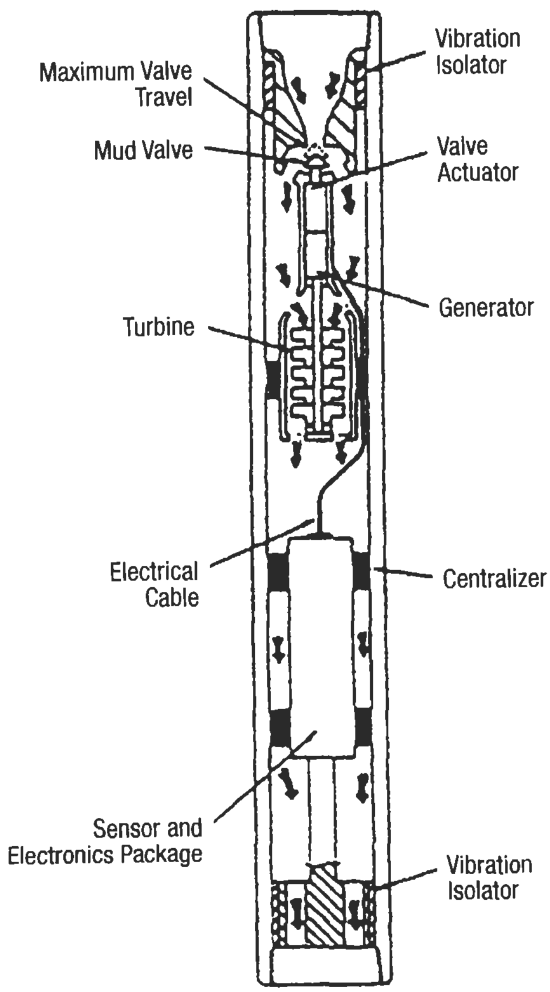


Figure 4-244. Schematic diagram of the positive pulse system. *(Courtesy Inteq-Teleco* [105].*)*

The coding principle is given in Figure 4-245. Each angular value of azimuth, drift and tool face is represented by ten bits. The practical "positive pulse" system is slightly different. The "1" bits correspond to incomplete strokes of the poppet valve as shown in Figure 4-246, making the system a slow phase-shift-keying system. Transmission rates of 0.2 or 0.4 bit/s are commonly used.

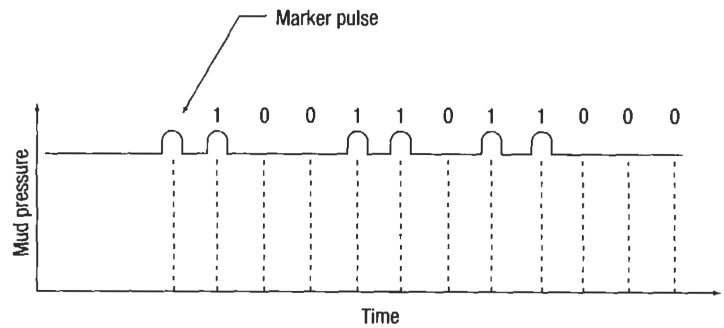


Figure 4-245. Principle of the coding of the positive pulse system. *(Courtesy Inteq-Teleco* [105].*)*

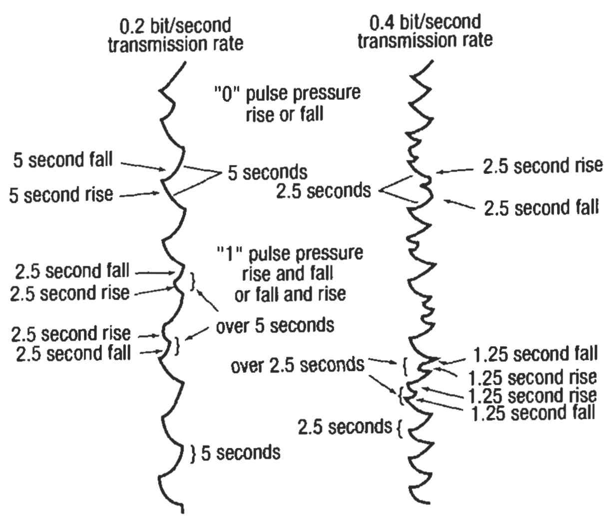


Figure 4-246. Pressure waves used in the practical application of the positive pulse system. *(Courtesy Inteq-Teleco* [105].*)*

The calculation of the amplitude of pressure variation at bottom can be done assuming that the restriction behaves as a choke. The pressure loss can be estimated using the relations

$$\Delta P = \frac{Q^2 \bullet \gamma \bullet 144}{2 \bullet g_c \bullet c^2 \bullet A_0^2} \tag{4-180}$$

where ΔP = pressure loss in psi
Q = flowrate in ft^3/s
γ = fluid specific weight in lb/ft^3
c = coefficient assumed to be one
A_0 = cross-sectional area of the restriction in $in.^2$
g_c = acceleration of gravity (32.2 ft/s^2)

When using a mud motor, the ΔP due to the restriction must be added to the ΔP due to the motor and the bit nozzles.

The mud motor pressure loss is given by

$$\Delta P = \frac{1714 \bullet W}{\eta \bullet Q} \tag{4-181}$$

where ΔP = pressure loss in psi
W = motor power in HP
η = motor efficiency
Q = mud flowrate in gal/min

Formula 4-180 will apply to the bit nozzle pressure loss.

Transmission by Negative Pulses. Drilling with a nozzle bit or with a downhole motor introduces a differential pressure between the inside and the outside of drill collars. This differential pressure can be changed by opening a valve and creating a communication between the inside of the drill string and the annular space. In this way, negative pulses are created that can be used to transmit digital data in the same way as positive pulses. Halliburton and other companies are marketing devices using this transmission principle.

Equation 4-180 can be used to calculate the pressure change inside the drill collars by changing the cross-sectional area A_0 from bit nozzles only to bit nozzles plus the pulser nozzle.

Continuous-Wave Transmission. Anadrill, a subsidiary of Schlumberger, markets a tool which produces a 12-Hz sinusoidal wave downhole. Ten-bit words representing data are transmitted by changing or maintaining the phase of the wave at regular intervals (0.66 s). A 180° phase change represents a 1, and phase maintenance represents a 0.

Figure 4-247 shows a sketch of principle of the system and of the phase-shift-keying technique. Frames of data are transmitted in a sequence. Each frame contains 16 words, and each word has 10 bits. Some important parameters may be repeated in the same frame, for example, in Figure 2-248, the torque Tp, the resistivity R and the gamma ray GR, are repeated four times. The weight on bit WOB is repeated twice, and the alternator voltage V_{alt} one time. Note that a synchronization pulse train starts the frame.

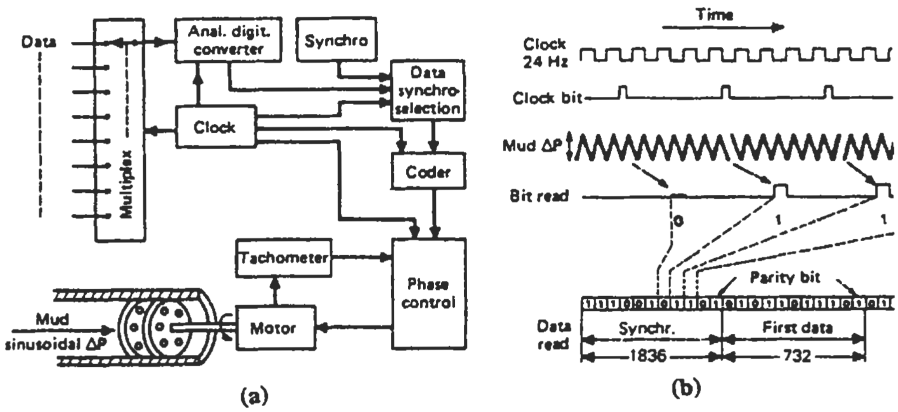


Figure 4-247. Principle of the continuous wave system: (a) sketch of the siren and electronic block diagram; (b) principle of the coding by phase shift keying. *(Courtesy Anadrill* [106].*)*

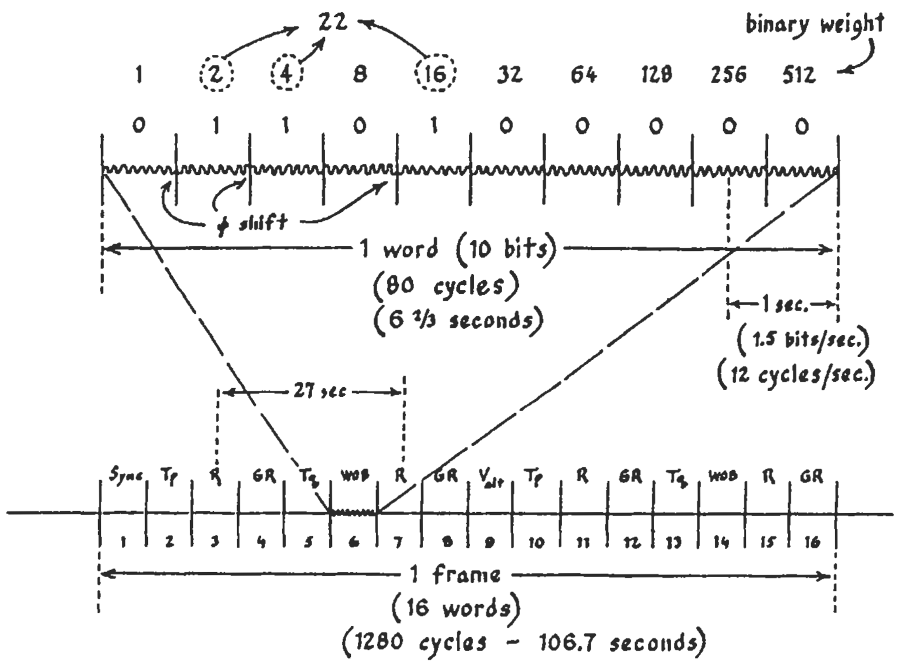


Figure 4-248. Example of a frame of data transmitted by the continuous wave system. *(Courtesy Anadrill* [106].*)*

The early system was transmitting 1.5 bits/s (4 sine waves to identify one bit). Later systems went to 3 bits/s. Now with a 24-Hz carrier frequency, 6 bits/s can be transmitted.

Then, with data compression techniques (sending only changes for most of the words in the frame and rotating the data), an effective transmission rate of 10 bits/s can be achieved.

The continuous wave technique has a definite advantage over the other techniques: a very narrow band of frequencies is needed to transmit the information. The pulse techniques, on the contrary, use a large band of frequencies, and the various noises, pump noises in particular, are more difficult to eliminate.

In principle, several channels of information could be transmitted simultaneously with the continuous wave technique. In particular, a downward channel to control the tool modes and an upward channel to bring up the information.

Fluidic Pulser System. A new type of pulser is being developed at Louisiana State University. It is based on a patent by A. B. Holmes [107]. The throttling of the mud is obtained by creating a turbulent flow in a chamber as shown in Figure 4-249.

A vortex is generated by momentarily introducing a dissymetry in the chamber. The resulting change in pressure loss can be switched on and off very rapidly. The switching time is approximately 1 ms and the amplitude of the pressure loss change can be as high as 145 psi (10 bars). The prototype tool can operate up to 20 Hz. Using a continuous wave with two cycles per bit could lead to a rate of 10 bits/s. With a data compression technique, 15 effective bits per second could be transmitted, corresponding to 1.5 data per second.

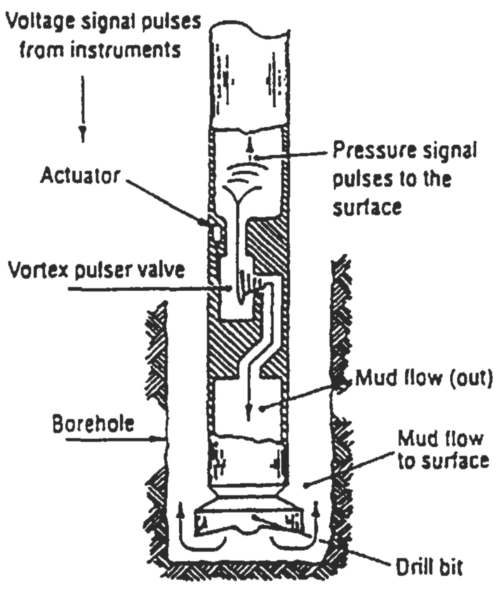


Figure 4-249. Fluidic mud pulser principlet. *(Courtesy Louisiana State University* [107].*)*

Surface Detection of the Mud Pressure Signals. The pulse or wave amplitude varies largely according to depth, frequency, mud type and pulse generator device. A typical mud surface pulse amplitude is 1 bar (14.5 psi). In a sine wave transmission the surface amplitude may go as low as 0.1 bar (1.5 psi) rms. The pump noise must be lowered to minimum by the use of properly adjusted dampeners and triplex instead of duplex pumps, the pulse amplitude being about twice as large for the duplex pumps. The pump noise amplitude varies from 0.1 to 10 or more bars (1.5 to 145 psi) with dominant frequencies ranging from 2 to 10 Hz. The rotation speed of the pumps may have to be changed so the noise frequency does not interfere with the measurements. The pressure sensors are generally of the AC-type, which sense only the pressure variations. A common sensor is of the piezoelectric type with a crystal transducer. Generally a built-in constant current follower amplifier converts the signal to a low impedance voltage. A typical sensitivity is 5 V per 1,000 psi (70 bars) with a maximum constant pressure of 10,000 psi (700 bars). The filtering can be done with digital filters or Fourier transform analyzers. For a Fourier transform processing, the signal must be properly analog-filtered and then digitized. Two pressure transducers can be used at different locations on the standpipe, as shown in Figure 4-250, to take advantage of the phase shift that is opposite for pump noise and downhole signal. Sophisticated digital cross correlation techniques can then be used.

Downhole Recording. Most MWD service companies offer the possibility of recording the data versus time downhole. The memories available may reach several megabytes, allowing the recording of many parameter values during many hours. This information is particularly valuable when the mud pulse link breaks down. The data can be dumped in a computer, during the following drillpipe trip.

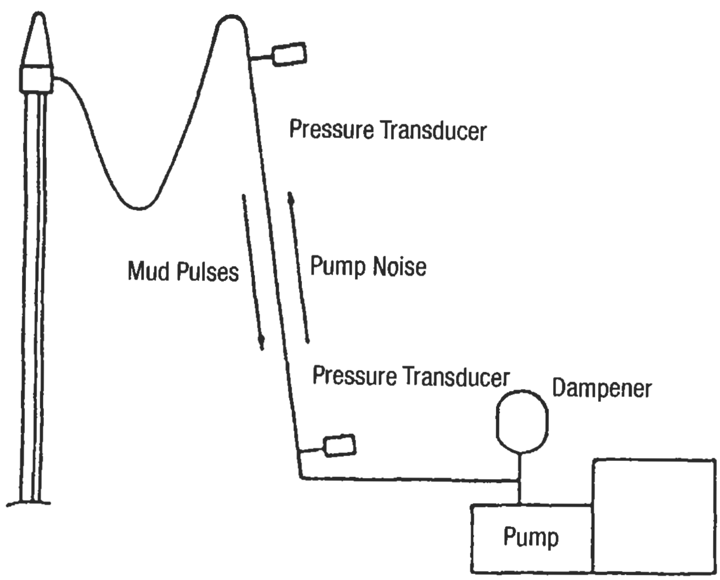


Figure 4-250. Surface pressure transducers location for pump noise elimination.

Retrievable Tools. Retrievable MWD tools similar to the steering tools are available from several service companies. They are generally battery powered and generate coded positive pressure mud pulses or continuous pressure waves. The lower part of the tool has a mule shoe that engages in a sub for orientation. Currently, tools are available for measuring directional parameters and gamma rays. A typical retrievable tool is shown in Figure 4-251.

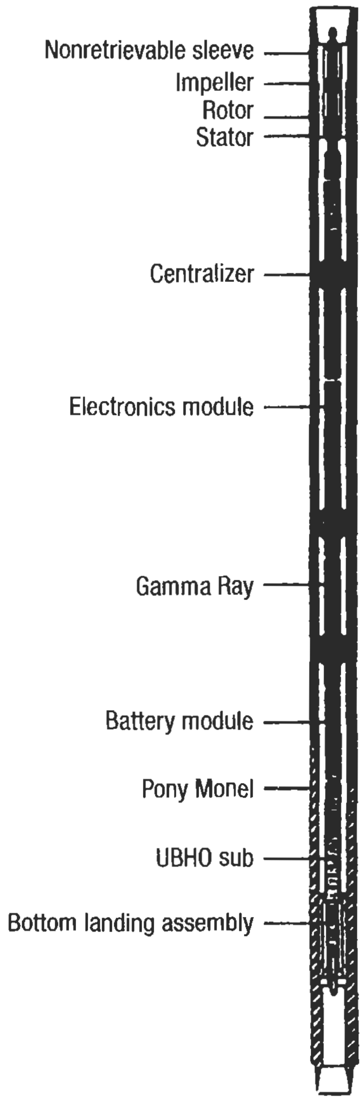


Figure 4-251. Retrievable MWD tool. *(Courtesy Anadrill* [106].*)*

The benefits of such a tool are apparent in the following instances:

- kickoffs and sidetracks
- correction runs
- high stuck-pipe risk
- high temperature
- slim hole
- low-budget drilling

Velocity and Attenuation of the Pressure Waves. The velocity and attenuation of the mud pulses or waves have been studied theoretically and experimentally. The velocity depends on the mud weight, mud compressibility, and on the drillpipe characteristics, and varies from 4920 ft/s for a light water-base mud to 3,940 ft/s for a heavy water-base mud. An oil-base mud velocity will vary from 3,940 ft/s for a light mud to 3,280 ft/s for a heavy mud.

The propagation velocity can be calculated using the equation

$$V = \sqrt{\frac{g_c \bullet 144 \bullet B \bullet M}{\gamma(B + M)}} \tag{4-182}$$

and

$$M = E\frac{(a^2 - b^2)}{4 \bullet b^2\left(\frac{5}{4} - \lambda\right) + 2(1 + \lambda)(a^2 + b^2)} \tag{4-183}$$

where V = pressure wave velocity in ft/s
g_c = acceleration due to gravity: 32.17 ft/s^2
B = mud bulk modulus in psi (inverse of compressibility)
E = steel Young modulus of elasticity in psi
a = OD of the pipe in in.
b = ID of the pipe in in.
λ = steel Poisson ratio
γ = mud specific weight in lb/ft^3

For example, in a 9 lb/gal water-base mud, and a $4\frac{1}{2}$-in. steel drillpipe, the pressure wave velocity is 4,793 ft/s.

The attenuation of the pressure waves increases with depth and with the mud pressure wave velocity. More attenuation is observed with oil-base muds, which are mostly used in deep or very deep holes, and can be calculated with the mud and pipe characteristics [108] according to the equations

$$P(x) = P(0)(e^{-\frac{x}{L}}) \tag{4-184}$$

$$L = d_i \bullet V \bullet \sqrt{\frac{2}{\eta \bullet \omega}} \tag{4-185}$$

where $P(x)$ = pressure wave amplitude at distance x in psi
$P(0)$ = pressure wave amplitude at distance 0 in psi
η = kinematic viscosity in ft^2/s ($1\ cSt \times 1.075 \times 10^{-5} = 1\ ft^2/s$)
ω = angular frequency in rad/s ($\omega = 2\pi f$)
with f = frequency in Hz
d_i = pipe internal diameter in ft
V = wave velocity in ft/s

Figure 4-252a and b gives the pressure wave amplitude versus the distance for various typical muds.

Electromagnetic Transmission Systems. One system uses a low-frequency antenna built in the drill collars. This system is a two-way electromagnetic arrangement allowing communication from bottom to surface for data transmission and from surface to bottom to activate or modify the tool mode. At any time the sequence of the transmitted parameters, as well as the transmission rate, can be modified. The tool is battery powered and can work without mud circulation. The principle of the system is shown in Figure 4-253. The receiver is connected between the pipe string and an electrode away from the rig for the bottom to surface mode. This system can be used on- or off-shore. Two tools are available: the directional tool, which transmits inclination, azimuth, gravity tool face or magnetic tool face, magnetic field inclination and intensity; and the formation evaluation tool, which measures gamma ray and resistivity. The formation evaluation data are stored downhole in a memory that can be interrogated from the surface or transferred to a computer when pulling out.

Figure 4-254a gives the attenuation per kilometer as a function of frequency for an average formation resistivity of 10 and 1 Ω • m.

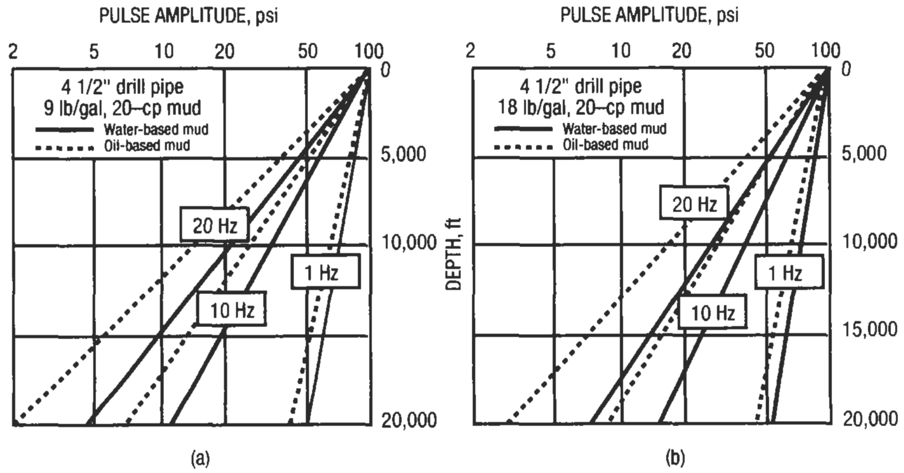


Figure 4-252. Wave amplitude variation as a function of distance in water-base mud and in oil-base mud: (a) mud weight, 9 lb/gal; (b) mud weight, 17.9 lb/galt. *(Courtesy Petroleum Engineer International* [108].*)*

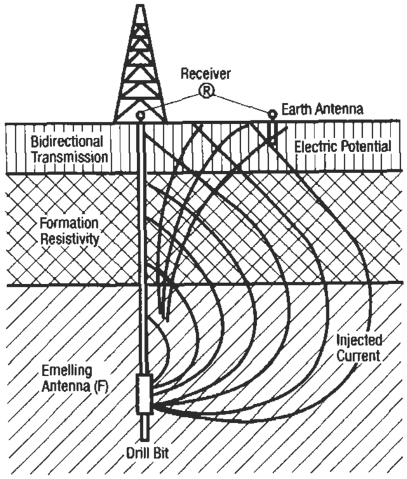


Figure 4-253. Principle of the electromagnetic MWD transmission. *(Courtesy Geoservices* [109].*)*

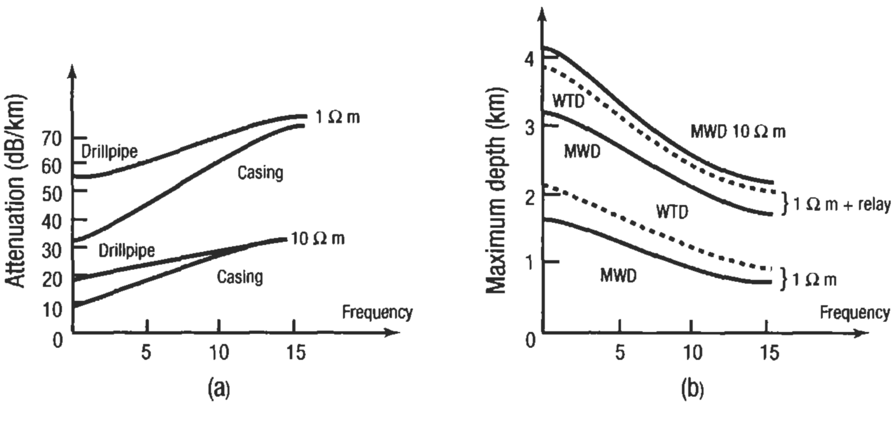


Figure 4-254. Attenuation of electromagnetic signals for 1 and 0 Ω • m average earth resistivity: (a) attenuation as a function of frequency; (b) maximum depth reached versus frequency. *(Courtesy Geoservices* [109].*)*

With the downhole power available and the signal detection threshold at surface, Figure 4-254b gives the maximum depth that can be reached by the technique as a function of frequency. Assuming that phase-shift keying is used with two cycles per bit, in a 10 Ω • m area (such as the Rocky mountains) a depth of 2 km (6,000 ft) could be reached while transmitting 7 bits/s.

Coding and Decoding. Ten-bit binary codes are used to transmit the information in most techniques. In one technique, the maximum reading to be transmitted is divided ten times. In a word, each bit has the value corresponding to its rank.

Demonstration. Transmit a range of values between 0 and 90°.

Bit	1	1	1	1	1	1	1	1	1	1
Value	45	22.5	11.25	5.62	2.81	1.40	0.70	0.35	0.17	0.08789

Word 1111111111 = 89.91°
Word 1011011001 = 64.06°
Word 0001100111 = 9.04°

In another technique, each bit represents a power of two in a given word. The highest number that can be transmitted is

$$2^9 + 2^8 + 2^7 + 2^6 + 2^5 + 2^4 + 2^3 + 2^2 + 2^1 + 2^0 = 1023$$

as well as zero.

The smallest value that can be transmitted for a full scale of 90° is

$$90/1024 = 0.08789°$$

Each bit has the following numerical value:

Bit	2^9	2^8	2^7	2^6	2^5	2^4	2^3	2^2	2^1	2^0
Value	512	256	128	64	32	16	8	4	2	1

For example, to transmit 64.06°, the numerical value is

$$64.06/0.08789 = 729$$

We will have one "2^9" bit: 729 − 512 = 217
one "2^7" bit: 217 − 128 = 89
one "2^6" bit: 89 − 64 = 25
one "2^4" bit: 25 − 16 = 9
one "2^3" bit: 9 − 8 = 1
one "2^0" bit: 1 − 1 = 0.

The word would be 1011011001 = 729. This is the same binary word as found previously. In each technique the accuracy is 0.08789° for a range of 0 to 90°. The rounding must be done the same way when coding and decoding.

Example 7: Mud Pulse Telemetry—Positive Pulse Calculations

A positive pulsing device has been designed as shown in Figure 4-255.

1. Compute the pressure loss when the puppet valve travels from 0.2 to 0.5 in., where 0.0 in. is the fully closed position, for each 0.05 in. for a flowrate of 400, 500 and 600 gal/min, and for a mud weight of 10 lb/gal. The nozzle equation is

$$Q = C \bullet A \bullet \sqrt{\frac{2 \bullet 144 \bullet g_c \bullet dP}{\gamma}} \tag{4-186}$$

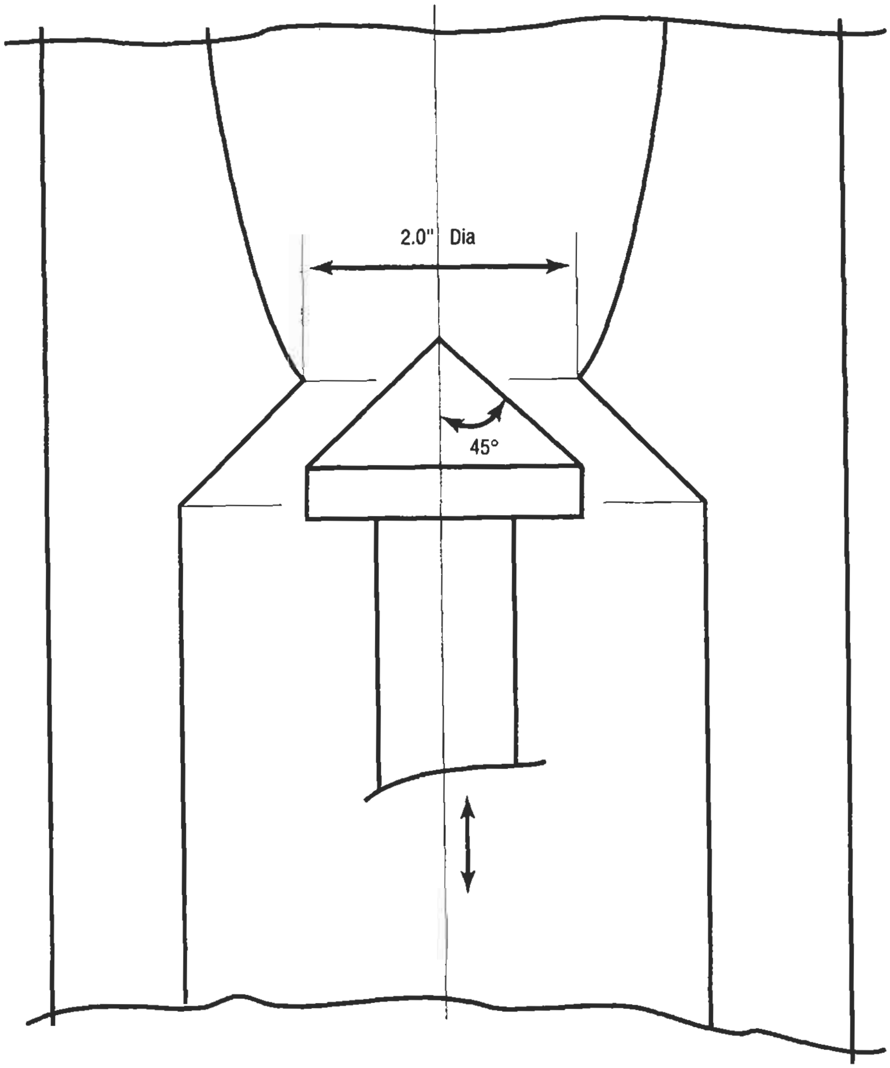


Figure 4-255. Typical positive pulse valve design.

where Q = flowrate in ft^3/s
A = flow area in ft^2
C = flow factor (C = 1.0)
dP = pressure loss in psi
γ = fluid specific weight in lb/ft^3
g_c = 32.18 ft/s^2

Trace the curve representing dP versus the puppet valve displacement for the 500-gal/min flowrate.

2. What is the pulse amplitude (pressure surge) for the various flowrates when the puppet valve travels from 0.2 to 0.5 in.? At what position will we have a half-height pulse?
3. We are using a mud motor which rotates at 500 rpm and develops a useful power of 100 hp. Assuming a constant flowrate of 500 gal/min, no pressure loss in the bit nozzles and 80% motor efficiency, compute the total bottomhole assembly ΔP at 0.2- and 0.5-in. valve opening.
4. The pulses are used to transmit deviation data from 0° to 90° with a 45, 22.5, 11.25, etc., sequence binary code of 10 bits. What is the transmission accuracy? Give the binary number for 27.4°.
5. Assuming the pressure pulse travels with the sound velocity of the mud, how long will it take to reach the rig floor in a 12,000-ft borehole? The sound velocity in the mud is given by

$$v = (g_c \times B \times 144/\gamma)^{1/2}$$

where v = sound velocity in ft/s
B = mud bulk modulus (3.3×10^5 psi)
γ = mud specific weight in lb/ft^3

Solution

1.

Table 4-123
Typical Positive Pulse Amplitude Generated at Bottomhole

	0.2 in.	0.5 in.	Amplitude
400 gal/min	187 psi	35 psi	152 psi
500 gal/min	290 psi	55 psi	235 psi
600 gal/min	420 psi	79 psi	341 psi

2.

Table 4-124
Half-Height Stroke of the Valve

	Amplitude	Half Height
400 gal/min	152	0.264 in.
500 gal/min	235	0.264 in.
600 gal/min	341	0.264 in.

3. ΔP motor: 429 psi
   Bottomhole ΔP: 0.2 in., 719 psi
   0.5 in., 484 psi
4. 27.4°: 0100110111
   Accuracy: 0.08789°
   Average error: 0.043945°
5. Sound velocity: 4510 ft/s
   Travel time: 2.67 s

Example 8: Mud Pulse Telemetry—Negative Pulse Calculations

The negative mud pulse system works with a nozzle which periodically opens in the wall of the drill collar to lower the pressure in the pipe string. The following data will be used:

- Bit nozzles: $3 \times \frac{16}{32}$ or $3 \times \frac{15}{32}$ or $3 \times \frac{14}{32}$ in.
- Pulse nozzle sizes: 0.3 or 0.4 or 0.5 in. diameter
- Mud flowrates: 400 or 500 or 600 gal/min
- Mud weight: 12 lb/gal

1. Compute the pressure change inside the drillpipe at bottom when the pulse nozzle opens in each case. Give the optimal combinations for getting 200 to 250 psi pulses.
2. A 10-bit digital system uses the sequence 180, 90, 45, 22.5, etc., to transmit the azimuth value. What is the accuracy of the transmission? Give the binary numbers for S-23-E by excess, default and nearest.
3. A positive displacement mud motor is included in the downhole assembly between the bit and the MWD system. It develops a true power of 100 hp when the pulse nozzle is closed. What is the true power obtained when a 200-psi pulse is created? The bit nozzle pressure loss will be neglected. Use
   - pulse nozzle diameter: 0.5 in.
   - mud flowrate: 400, 500, 600 gal/min
   - mud motor efficiency: 80%
4. Same question taking into account the pressure drop in the bit nozzles with $3 \times \frac{16}{32}$ in. nozzles. Solving for ΔP in Equation 4-186 gives

$$\Delta P = \frac{Q^2 \bullet \gamma \bullet 144}{2 \bullet g_c \bullet C^2 \bullet A^2}$$

where Q = flowrate in ft^3/s
A = nozzle area in $in.^2$
C = nozzle factor (C = 1.0)
ΔP = pressure drop across the nozzles in psi
γ = mud specific weight in lb/ft^3
g_c = 32.2 ft/s^2
HP = eff × ΔP × Q/1714

Solution

1.

Table 4-125
Optimum Nozzles Combination for Generating 200 to 250 psi Pulses

	Bit Nozzles	Pulse Nozzles
400 gal/min	15/32 in.	0.4 in.
400 gal/min	16/32 in.	0.5 in.
500 gal/min	15/32 in.	0.3 in.
500 gal/min	16/32 in.	0.4 in.
600 gal/min	16/32 in.	0.3 in.

2. S-23-E = 157°
   Default: 0110111110 156.796°
   Excess: 0110111111 157.148°
   Nearest: 0110111111 157.148°
3. Motor hp at 400 gal/min: 44.8 hp
   Motor hp at 500 gal/min: 43.3 hp
   Motor hp at 600 gal/min: 38.3 hp
4.

Table 4-126
Typical Conditions Encountered for Various Flowrates

		400 gal/min	500 gal/min	600 gal/min	Units
Pulse	ΔP motor 100 hp	536	428	357	psi
nozzle	ΔP bit nozzle	460	719	1035	psi
closed	Total ΔP	996	1147	1392	psi
	ΔP pulse	–200	–200	–200	psi
Pulse	ΔP open	796	947	1192	psi
nozzle	Flow pulse nozzle	175	191	215	gal
open	Flow motor/bit	225	309	385	gal
	New ΔP bit	145	274	427	psi
	New ΔP motor	651	673	765	psi
	New hp motor	68	97	138	hp

Example 9: Mud Pulse Telemetry—Pressure Wave Attenuation

An MWD system is lowered at the end of a 4.5-in. drillstring in an 8-in. borehole. Neglect the drill collar section of the string. The following data are available:

- total depth: 10,000 ft
- borehole average diameter: 8 in.
- drillpipe OD: 4.5 in.
- drillpipe ID: 3.64 in.
- drillpipe Young modulus: 30×10^6 psi
- drillpipe Poisson ratio: 0.3
- mud specific weight: 12 lb/gal
- mud compressibility: 2.8×10^{-6} psi^{-1}
- mud viscosity: 12 cp
- mud flow rate: 400 gal/min
- bit nozzles: $3 \times \frac{13}{32}$ in.

1. Compute the bottomhole hydrostatic pressure with no flow.
2. Compute the pressure drop in the drill pipe while circulating.
3. Compute the pressure drop in the annulus while circulating.
4. What is the pressure drop in the bit nozzles?
5. What is the pump pressure at surface?
6. Make a graph of the pressure variation with depth without circulation in the drillpipe and annulus.
7. Compute the velocity of the pressure wave in free mud (not in a drillpipe).

8. Compute the velocity of the pressure wave in the drillpipes.
9. Compute the amplitude of a pressure wave at surface of a wave generated at bottom with an amplitude of 200 psi at frequencies of 0.2, 6, 12 and 24 Hz.

Pressure loss in pipe (turbulent flow) is

$$dP = \frac{dL \bullet \gamma^{0.75} \bullet v^{1.75} \bullet \mu^{0.25}}{1800 \bullet d^{1.25}} \quad (4\text{-}187)$$

Pressure loss in annulus (turbulent flow) is

$$dP = \frac{dL \bullet \gamma^{0.75} \bullet v^{1.75} \bullet \mu^{0.25}}{1396 \bullet (d_2 - d_1)^{1.25}} \quad (4\text{-}188)$$

where dP = pressure loss in psi
dL = pipe or annulus length in ft
γ = fluid specific weight in lb/gal
v = fluid velocity in ft/s
d = ID pipe diameter in in.
d_1 = OD pipe diameter in in.
d_2 = external annulus diameter in in.
μ = fluid viscosity in cp

Solution

1. Bottomhole pressure, no flow: 6,240 psi
2. Drillpipe pressure loss: 1,076 psi
3. Annulus pressure loss: 113 psi
4. Bit nozzle pressure loss: 1,055 psi
5. Pump pressure: 2,244 psi
6. Graph (see Figure 4-256)
7. Wave velocity in free mud: 4,294 ft/s
8. Wave velocity in drill pipes: 4,064 ft/s
9. Wave amplitude at surface (Equations 4-184 and 4-185):
   0.2 Hz, L = 86,744 ft, 178 psi
   6 Hz, L = 15,837 ft, 106 psi
   12 Hz, L = 11,198 ft, 81 psi
   24 Hz, L = 7,918 ft, 56 psi

Example 10: Mud Pulse Telemetry—Pulse Velocity and Attenuation

Assume a well 10,000-ft deep, mud weight of 12 lb/gal, mud viscosity of 12 cp, $4\frac{1}{2}$-in drillpipes (3.640 in. ID), mud flowrate of 400 gal/min, steel Young modulus of 30×10^6 psi, and steel Poisson ratio of 0.3.

1. Compute the pressure at bottom inside the drill collars:
   a. with no flow and no surface pressure,
   b. with no flow and 2,500 psi surface pressure,
   c. while pumping 400 gal/min with 2,500 psi at surface.

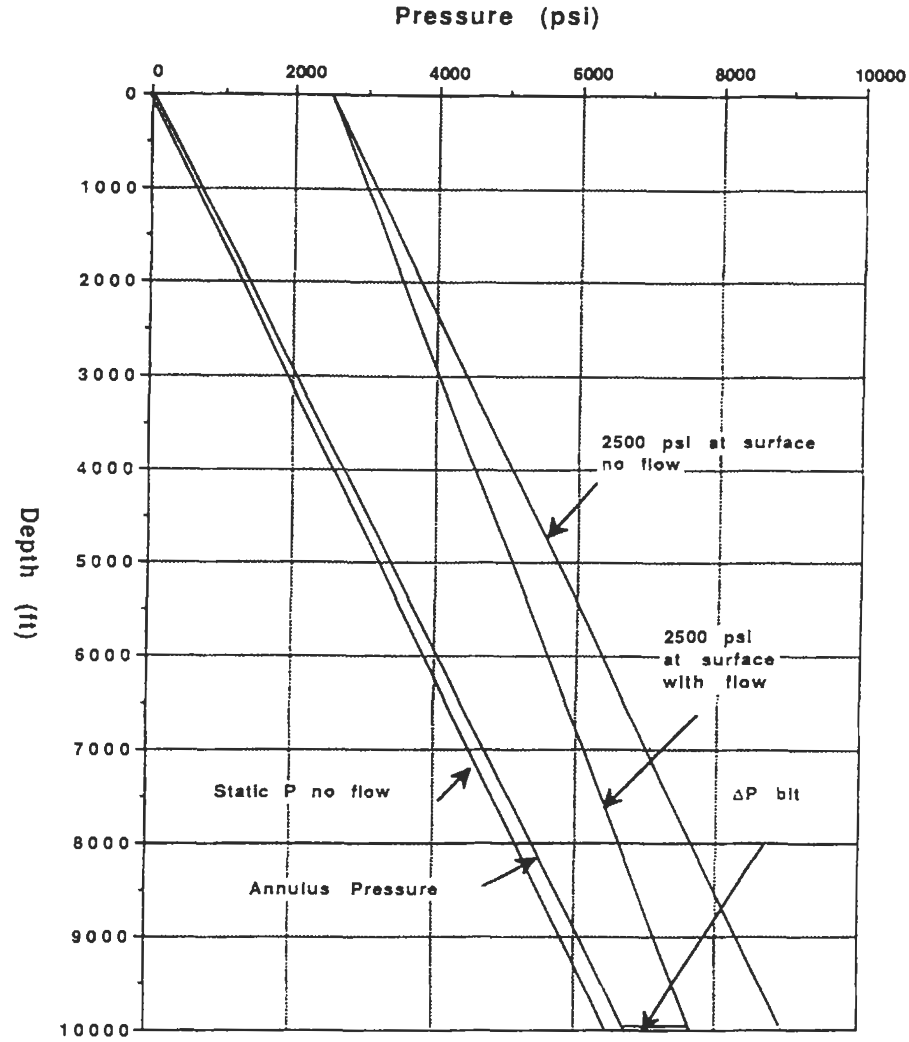


Figure 4-256. Pressure variation with depth: grid with the solution.

Draw a pressure traverse for each case in the attached graph, use pressure loss equation given in Data Sheet below.

2. A pressure pulse is generated at bottom. Compute the pulse velocity in the pipe at bottomhole and at surface, while circulating, assuming a surface temperature of 25°C and a bottomhole temperature of 85°C. The mud compressibility is assumed equal to the water compressibility given in Figure 4-257. Compare to the free mud pressure pulse velocity.

3. For the average pressure wave velocity in the pipe, compute the distance at which the amplitude falls to 1/e of its original value, the distance at which it falls at one-half of its original value (half depth) and the attenuation in dB/1,000 ft. Compute also the amplitude at surface. Bottomhole amplitude peak to peak: 200 psi; frequencies: 0.2, 12 and 24 Hz.

Data Sheet. Pressure loss (Equation 4-187) is given by

$$dP = \frac{dL \bullet \gamma^{0.75} \bullet v_m^{1.75} \bullet \mu^{0.25}}{1800 \bullet d^{1.25}}$$

where dP = pressure loss in psi
dL = pipe or annulus length in ft
γ = mud specific weight in lb/gal
v_m = mud velocity in ft/s
d = ID pipe diameter in in.
μ = mud viscosity in cp

Pressure wave velocity is

$$V_w = \sqrt{\frac{B + M}{\gamma} \bullet g_c}$$

where V_w = pressure wave velocity in ft/s
γ = mud specific weight in lb/ft^3
B = fluid bulk modulus in lb/ft^2
M = drill pipe modulus in lb/ft^2

$$B = 1/K$$

where K = mud compressibility in ft^2/lb
g_c = gravity acceleration 32.2 ft/s^2

and

$$M = \frac{E(D_0^2 - D_i^2)}{2(1 - \nu)(D_0^2 + D_i^2) - (2 \bullet \nu \bullet D_i^2)}$$

where E = steel Young modulus in lb/ft^2
D_0 = external drill pipe diameter in ft
D_i = internal drill pipe diameter in ft
ν = steel Poisson ratio

Pressure wave attenuation is

$$P(x) = P(0) \bullet e^{-x/L}$$

where P(x) = wave amplitude at distance x
P(0) = wave amplitude at origin

x = distance in ft
L = distance at which the amplitude falls to 1/e of its original value

Thus, the length can be expressed as

$$L = 0.5 \bullet D_i \bullet V_w \bullet \sqrt{\frac{2}{\eta \bullet \omega}}$$

where D_i = internal pipe diameter in ft
V_w = wave velocity in ft/s
ω = angular frequency in rad/s ($\omega = 2\pi f$)
F = wave or pulse frequency
η = kinematic viscosity in ft^2/s

Also,

$$P(x) = P(0) \bullet 2^{-x/D}$$

where D = distance at which the amplitude falls to $\frac{1}{2}$ of original value (half depth)

and

$$P(x) = P(0) \bullet 10^{-x/B}$$

where B = distance at which amplitude falls to $\frac{1}{10}$ of its original value (attenuation of 2 bel or 20 dB)

Attenuation at distance x in dB is

$$x(dB) = 20 \bullet \log \frac{P(x)}{P(0)} = -\frac{(20 \bullet x)}{B}$$

Kinematic viscosity (in consistent units) is

$$\eta = \mu/\rho \qquad (4\text{-}189)$$

where η = kinematic viscosity
μ = absolute viscosity
ρ = fluid density

Conversion equations are

$$\eta\left(\frac{ft^2}{s}\right) = \frac{\mu(cp) \bullet 2.09 \times 10^{-5} \bullet g_c}{\gamma}$$

where g_c = 32.18 ft/s^2
γ = mud specific weight in lb/ft^3

and

$$\eta\left(\frac{ft^2}{s}\right) = 1.075 \times 10^{-5} \bullet \eta(cst).$$

Water isothermal compressibility is

$$K = (A + BT + CT^2) \bullet 10^{-6}$$

where $A = 3.8546 - 0.000134\ P$
$B = -0.01052 + 4.77 \times 10^{-7}\ P$
$C = 3.9267 \times 10^{-5} - 8.8 \times 10^{-10}\ P$

and

P = pressure in psig
T = temperature in °F

See Figure 4-257 [110].

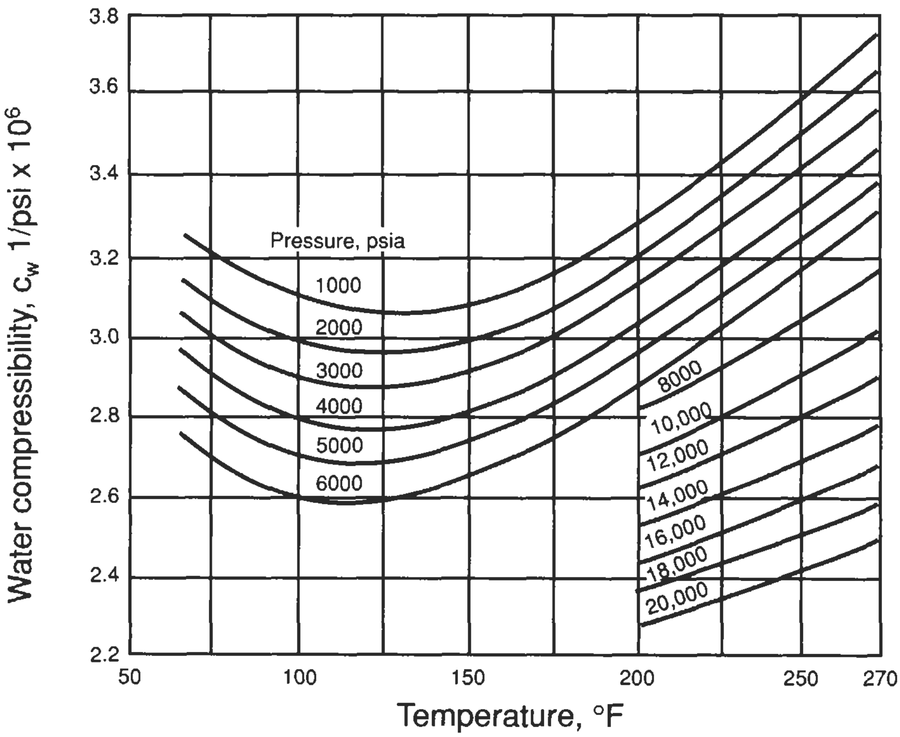


Figure 4-257. Chart showing the variation of the coefficient of isothermal compressibility of water versus pressure and temperature [110].

Solution

1. a. 6,255 psia
   b. 8,740 psia
   c, 7,664 psia
2. Free fluid velocity:
   Surface: 4,168.6 ft/s
   Bottom: 4,356 ft/s
   Velocity in drillpipe:
   Surface: 4,044 ft/s
   Bottom: 4,215 ft/s
   Average velocity in pipe: 4,130 ft/s
3.

Table 4-127
Amplitudes at Surface for a 10,000-ft Well, 200-psi Downhole Pulses, for Various Frequencies

	0.2 Hz	12 Hz	24 Hz
L	83,357 ft	10,761 ft	7,609 ft
D	57,766 ft	7,457 ft	5,273 ft
B	191,971 ft	24,782 ft	17,523 ft
Attenuation	0.104 dB/1,000 ft	0.807 dB/1,000 ft	1.141 dB/1,000 ft
Pulse p-p amplitude at surface	177.4 psi	78.96 psi	53.73 psi

Example 11: Mud Pulse Telemetry—Fluidic Pulser Calculations

We have built a fluidic pulser system that can generate approximately 100 psi peak to peak with 500 gal/min mud flowrate. It is to be used down to 15,000 ft. The surface detector needs a 5-psi peak to peak sine wave for proper phase detection. The following oil-base mud is used:

- density: 12 lb/gal
- viscosity: 25 cp
- pressure wave velocity in the drill pipe: 3685 ft/s
- drillpipe diameter: 4.5 in. OD, 3.64 in. ID.

The system transmits 5 bits/s with a phase-shift-keying system. Four sine waves are necessary to define the phase with a negligible chance of error. Assume a perfect pipe, drill collar ID same as the drill pipe and no wave reflections at the drill pipe ends.

1. What frequency(s) should be used?
2. What peak-to-peak amplitude (psi) of the pressure wave is necessary at bottom to get the required peak to peak value at surface? Is our pulsing device suited for this job?
3. The pump noise frequency is varying around 8 Hz with a peak-to-peak amplitude of 20 psi. Can the signal still be detected? Explain.
4. If we generate a 12-Hz wave at surface to transmit instructions downhole to the instrument package, what amplitude should it have at surface to reach bottom with 5 psi peak to peak?
5. Can both channels work simultaneously with proper filtering? Explain.

6. The mud flow rate is 500 gal/min and the fluidic pulser has 4 × 24/32 in. diameter nozzles in parallel. Compute the pressure loss in the fluidic pulser in the minimum loss mode (C = 1).
7. What is the equivalent diameter of each nozzle in the maximum loss mode to produce the peak to peak wave value computed in question 2 (C = 1)?

Solution

1. 5 bit/s × 4 cycles = 20 Hz (cycles/s).
2. $P(x) = P(0) \bullet e^{-x/D}$
   $P(x)/P(0) = 0.0544$, $x = 4572$ m
   $P(0) = 91.9$ psi
3. Yes, by filtering only the wave amplitude corresponding to 20 Hz can be measured, thus eliminating the noise.
4. $P(x)/P(0) = 0.1049$
   $P(0) = 47.6$ psi
5. Yes, each detector will "see" only the wave amplitudes corresponding to 20 and 12 Hz. They will be sensitive to "their" signal only.
6. 78.8 psi
7. 0.619 in.

Directional Drilling Parameters

With the modern accelerometers and solid-state magnetometers, a complete set of data is available for inclination, tool face and azimuth calculation. Magnetic corrections can be done. Inclination can be calculated with Equations 4-174 and 4-175. The gravity tool face angle can be calculated with Equation 4-176.

Azimuth calculation can be done by using vector analysis. In Figure 4-258 the vector **Z** represents the borehole axis, vector **H** the earth magnetic field and vector **G** the vertical or gravity vector. The azimuth is the angle between the vertical planes V_H and V_Z counted clockwise starting at V_H. This angle is the same as the angle between vectors **A** and **B**, respectively, perpendicular to V_H and V_Z. We know that

$$\mathbf{A} = \mathbf{G} \times \mathbf{H} \text{ (vector product)} \tag{4-190}$$

$$\mathbf{B} = \mathbf{G} \times \mathbf{Z}$$

The components of **H** and **G** are measured in the referential of the MWD tool, and **Z** is the vector (0, 0, 1) in the same referential. Now the azimuth α of the borehole can be computed with the scalar product **B** • **A**; thus

$$\alpha = \text{arc cos}\left(\frac{\mathbf{A} \bullet \mathbf{B}}{|\mathbf{A}| \bullet |\mathbf{B}|}\right) \tag{4-191}$$

Some precautions must be taken to be sure that the correct angle is computed since $\cos(\alpha) = \cos(-\alpha)$.

The MWD sensors are located in a nonmagnetic part of the drill collars. The magnetic collars located several meters away still have an effect by creating a perturbation in the direction of the borehole axis. This introduces an error that is

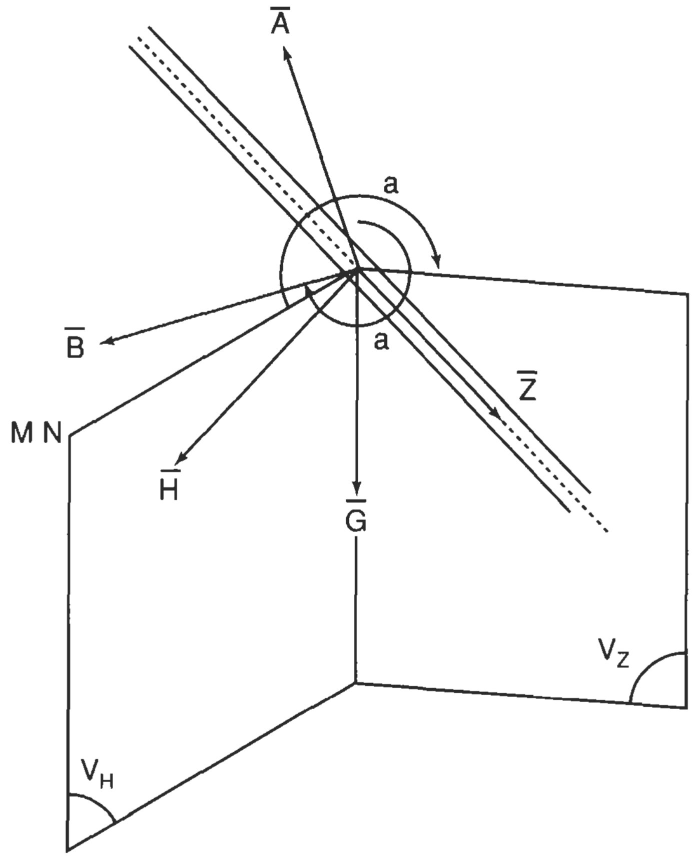


Figure 4-258. Solid geometry sketch of the planes defining the azimuth angle.

empirically corrected with the single-shot instruments. Since the three components of H are measured, the magnitude of the error vector can be calculated if the module of the nonperturbed earth magnetic vector is known. The corrected dip angle vector can also be computed and compared to the non-perturbed dip angle. The computation should match; if it does not, then a nonaxial perturbation is present. This perturbation may be due to "hot spots," points in the nonmagnetic drill collar that have developed some magnetism, or to external factors such as a casing in the vicinity. Correction techniques have been introduced for the hot spots.

External magnetism due to casing or steel in the well vicinity is used in passive ranging tools for blowout well detection from a relief well.

The accuracy of MWD directional measurements is generally much better than the single- or multishot-type measurements since the sensors are more advanced and the measurements more numerous. The azimuth measurement is made with the three components of the earth magnetic field vector and only with the horizontal component in the case of the single shot or multishot. The accelerometer measurements of the inclination are also more accurate whatever the value of the inclination. The average error in the horizontal position varies from

6 ft per 3,000 ft drilled at no deviation to 24 ft per 3,000 ft drilled at 55° of deviation. The reference position is given by the inertial Ferranti platform FINDS [111]. A large dispersion is noted on the 102 wells surveyed.

When the borehole is vertical and a kickoff must be done, a mud motor and bent sub are generally used. To orient the bent sub in the target direction, the gravity toolface is undetermined according to Equation 4-176. Up to about 5° or 6° of deviation the *magnetic tool face* is used. The magnetic tool face is the angle between the north vertical plane and the plane defined by the borehole (vertical) and the mud motor or lower part of the bent sub if the bent sub is located below the mud motor. After reaching 5° or 6° of inclination the surface computer is switched to the gravity tool face mode.

Drilling Parameters

The main drilling parameters measured downhole are:

- weight-on-bit
- torque
- bending moment
- mud pressure
- mud temperature

Strain gages are usually used for the first four measurements.

Strain Gages. Strain gages are used to measure the strain or elongation caused by the stress on a material. They are usually made of a thin foil grid laid on a plastic support as shown in Figure 4-259. They are the size of a small postal stamp and are glued to the structure to be stressed.

The sensitive axis is along the straight part of the conducting foil. When elongated, this conducting foil increases in resistance. The change in resistance is very low. Two gages are usually used and mounted in a Wheatstone bridge. Two more gages not submitted to the strain are also used to compensate for temperature variation. The change in resistance for one gage is given by

$$\Delta R = \frac{F \bullet R \bullet \sigma}{E} \tag{4-192}$$

where ΔR = resistance change in Ω
F = gage factor
R = gage resistance in Ω
E = Young modulus in psi or Pa
σ = stress in psi or Pa

Demonstration. The gage is a platinum gage with $F = 4$, $R = 50\ \Omega$. The structure measured is steel with $E = 30 \times 10^6$ psi. If the stress is 1000 psi, the resistance change is

$\Delta R = 0.0067\ \Omega$

For constantan, the gage constants are usually $F = 2$, $R = 100\ \Omega$.

Weight-on-Bit. Weight-on-bit is usually measured with strain gages attached to a sub subjected to axial load. The axial load is composed of three parts:

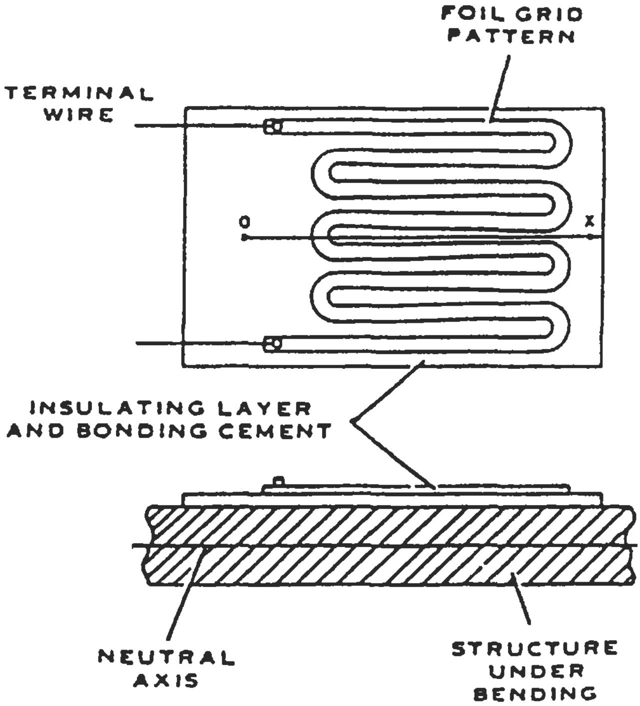


Figure 4-259. Sketch of principle of glued foil strain gage transducers.

1. weight on bit proper
2. end effect due to the differential internal pressure in the drill collar
3. hydrostatic pressure effects

Hydraulic lift must also be taken into account when using diamond bits and PDC bits. The weight-on-bit varies between 0 and 100,000 lb or 0 and 50 ton-force. The end effect is due to the differential pressure between the drill collar internal pressure and the external hydrostatic pressure. This differential pressure acts on the sub internal cross-sectional area.

Demonstration. The WOB sub internal diameter is 3.5 in.; the differential pressure is 1,000 psi; the downward force acting to elongate the sub is

$$F_d = \frac{\pi}{4}(3.5)^2 \times 1{,}000 = 9.621 \text{ lb}$$

The hydrostatic pressure has two effects: an upward force acting on the wall cross-section of the sub, and a downward stress due to the lateral compression of the subwall.

Demonstration. The sub has an ID of 3.5 in. and OD of 6 in. The area of the wall is 18.65 in.^{2} For a 10,000-ft well with a 10-lb/gal mud

$P_H = 0.052 \times 10{,}000 \times 10 = 5{,}200$ psi

The upward force acting on the sub is

$F_U = 96{,}980$ lb

The downward stress due to the mud pressure (neglecting the differential pressure) is

$$\sigma = 2 \times \nu \times P_H \tag{4-193}$$

where σ = stress downward (trying to elongate the sub)
ν = Poisson ratio of the sub
P_H = hydrostatic pressure

Demonstration. For steel, $\nu = 0.3$ and $P_H = 5{,}200$ psi and $\sigma = 2 \times 0.3 \times 5{,}200 = 3{,}120$ psi.

This stress is to be accounted for when computing the total stress in the sub wall. The calculation can be done easily in real time with a computer; however, it is easier and probably more accurate to measure a difference in strain (or stress) in the sub between the off-bottom position and while drilling. This value should be related closely to the true weight-on-bit.

Torque. The torque is measured by placing two strain gages at 45° on the sub as shown in Figure 4-260. Stress at 45° due to torque is

$$\sigma = \frac{T \bullet R_o}{\pi(R_o^4 - R_i^4)} \tag{4-194}$$

where σ = stress in psi
R_o = outer radius in in.
R_i = inner radius in in.
T = torque in in. • lb

Strain is

$$\varepsilon = \pm\frac{T}{G} \times \frac{R_o}{\pi(R_o^4 - R_i^4)} \tag{4-195}$$

where G is shear modulus (usually 12×10^6 psi) and

$$\Delta R = \pm\frac{T}{G} \times \frac{R_o}{\pi(R_o^4 - R_o^4)} \times F \times R \tag{4-196}$$

Demonstration. Platinum gages on steel collar:

$T = 2{,}000$ ft • lb $\qquad F = 2$

$R_o = 2.5$ in. $\qquad R = 100\ \Omega$

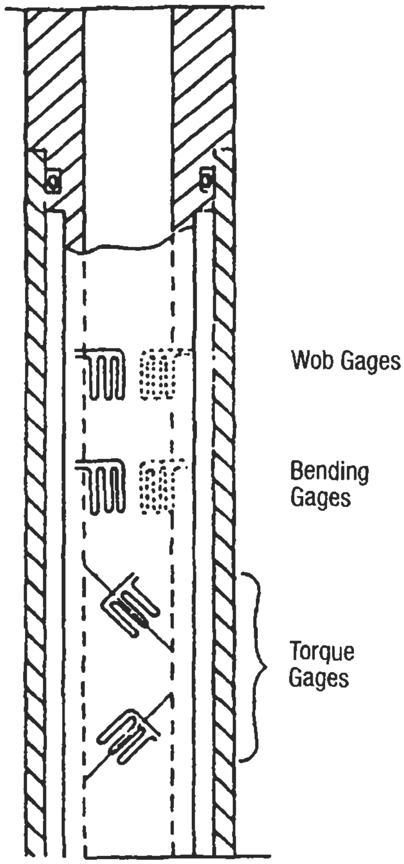


Figure 4-260. Sketch of theoretical strain gage position in a sub to read WOB, torque, and bending moment.

R_i = 1.5 in.

$\Delta R = 9.36 \times 10^{-3}\ \Omega$

Using two gages in opposite legs of the bridge will double the sensitivity.

The axial load (compression) gives a uniform stress and strain in the absence of a bending moment. If a bending moment exists, then one side is extended while the other is compressed.

During the rotation an alternative signal for the axial load is superimposed on the DC signal. By filtering, both the axial load and the bending moment can be measured. In practice, the strain gages are placed in holes drilled in the measuring sub as shown in Figure 4-261.

Mud Pressure. Internal and external mud pressures are usually measured with strain gages mounted on a steel diaphragm. Figure 4-262 shows a sketch of principle.

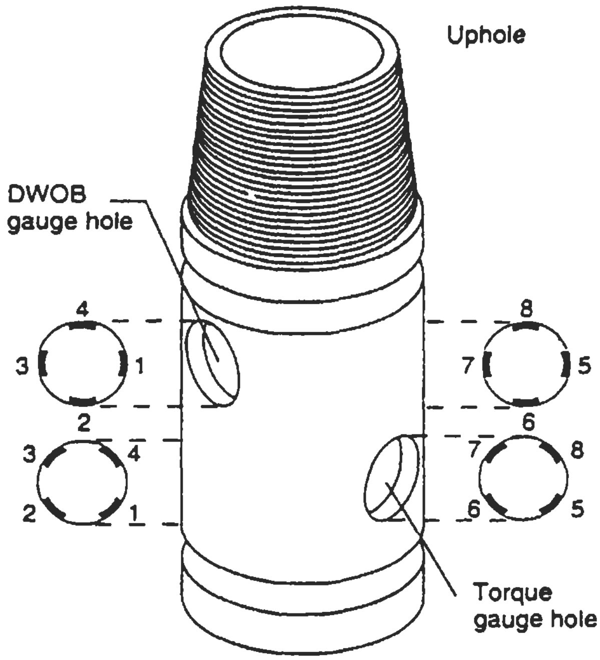


Figure 4-261. Practical design of a drilling parameter sub. *(Courtesy Anadrill* [106].*)*

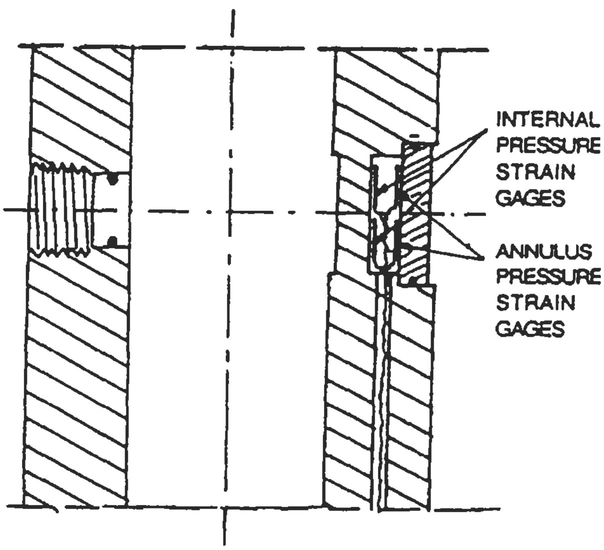


Figure 4-262. Sketch of principle of downhole pressure measurements.

One steel diaphragm is exposed to the internal pressure, the other is exposed to the external pressure. Four gages are normally used. Two of them are sensitive to pressure and temperature, and two are sensitive to the temperature. A Wheatstone bridge is used for detection of the pressure.

Downhole Shocks Measurements. An accelerometer in the MWD telemetry tool measures transverse accelerations, or shocks, that may be damaging for the bottomhole assemblies. When acceleration exceeds a certain threshold, the event is signaled to the surface as being a shock. These events versus time or depth are displayed as shock count. This information is used as a warning against excessive downhole vibrations and to alert the driller to change the rpm or weight on the bit [106].

A simple circuit has been designed to count the number of shocks that the tool experiences above a preset "g" level. The transverse shocks are measured in the range of 2 to 1,000 Hz in excess of the preset level. The level is adjustable and defaults at 25 g's (when no preset level is specified).

Downhole shock measurements are used to:

- send alarms of excessive downhole vibration in real-time so that action can be taken to reduce damage to the MWD tools, drill bits, and bottomhole assemblies;
- reduce costly trips to replace damaged equipment;
- improve drilling rate by eliminating counter-productive BHA vibration motion.

Downhole Flowrate Measurement. Anadrill's basic MWD tool can be set up to monitor the alternator voltage being produced by the mud flowing across the MWD turbine downhole. By comparing this voltage to the standpipe pressure and the pump stroke rate, the surface system shows that a washout in the drill string is occurring much quicker than with conventional methods [106].

The downhole flowrate monitoring and washout detection system is used to

- avoid potential twist-offs from extensive drill string washouts;
- determine if the washout is above or below the MWD tool, thus saving rig time when searching for the failure.

Safety Parameters

One area where MWD would be most useful is drilling safety and, particularly, early gas kick detection and monitoring. Conventional kick monitoring is based on pit gain measurements and all other available surface indication such as drilling rate break, injection pressure variation, etc.

Using the probable detection threshold achievable and a gas kick model applied to a typical 10,000-ft drill hole, an early alarm provided by MWD systems decreases significantly the amount of gas to be circulated as compared to using conventional methods of kick detection.

Dissolved Gas. Gas which enters the borehole when penetrating a high pressure zone may not dissolve immediately in the mud. The free gas considered here is the gas entering the borehole minus the dissolved gas. Table 4-128 indicates the maximum volume of dissolved gas at bottomhole conditions expressed in percent of annulus mud volume. Thus, when entering a high pressure permeable formation this much gas will dissolve first before free gas appears in the mud.

Table 4-128
Maximum Dissolved Gas Content of Drilling Muds

Mud	Density (lb/gal)	Pressure (psi)	Gas Volume* (scf/STB)	% of Injected Mud Volume*
Water base	9	4,680	16	0.9
Water base	18	9,360	13.7	0.6
Oil base	9	4,680	760	38
Oil base	18	9,360	7,200	77

Conditions:
Depth = 10,000 ft
Temperature = 150°F
Filtrate salinity = 20 kppm
Gas density (air = 1.0) = 0.7
Oil S.G. = 0.83 (39°API)
Brine density = 1.175 g/cm³
Brine-oil ratio = 16:84

*At bubble point in the mixture

Note that from Table 4-128 the very large volumes that can dissolve in oil-base muds. For the water-base muds, 0.6 to 0.9% of gas will dissolve and not appreciably change the density or compressibility of the mud. It will be difficult to detect these low concentrations with downhole physical measurements. Free gas will be easily detected as shown hereafter. For the oil-base muds we will assume no free gas is present at bottomhole and the mud properties are changed only due to the dissolved gas. The detection will be more difficult than with free gas.

Bottomhole Gas Detection. Many techniques could be used for bottomhole gas detection:

- mud acoustic velocity
- mud acoustic attenuation
- mud specific weight
- mud resistivity
- mud temperature
- annulus noise-level

Figure 4-263 shows the various sensors that could, schematically, be installed in the annulus. Cuttings, turbulent flow, vibration and shock may render some measurements difficult. We shall study those that can be related easily to gas content.

Mud Acoustic Velocity. Acoustic velocity can be accurately predicted. The measurements could be made over a short distance in the annulus of the order of 1 to 2 ft. The "free" mud formula can be used. This is

$$V = \sqrt{\frac{144 g_c}{K\gamma}} \tag{4-197}$$

where V = acoustic wave velocity in ft
K = gas cut mud compressibility in psi^{-1}
γ = gas cut mud specific weight in lb/ft^3

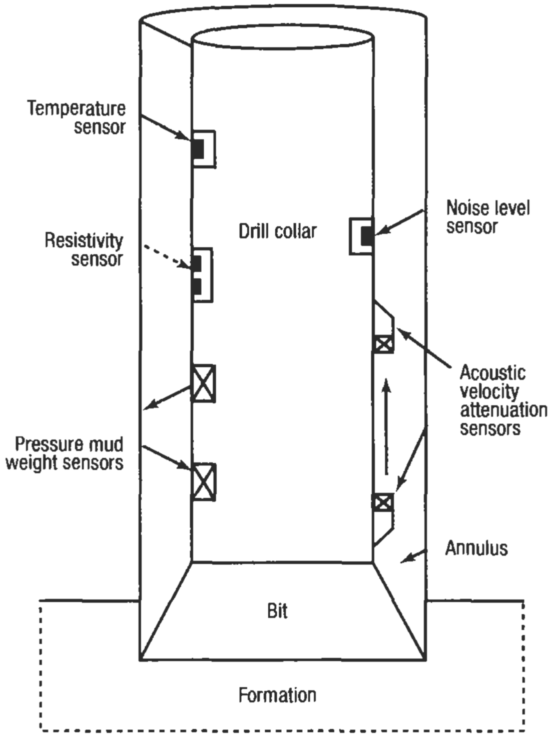


Figure 4-263. Schematic representation of bottomhole kick detection sensorst. *(Courtesy Petroleum Engineer International* [96].*)*

with

$$K = (1 - f_b)K_m + \frac{f_g}{P^{1.4}} \tag{4-198}$$

where K_m = gas free mud compressibility in psi^{-1}
P = mud pressure in psi
f_g = free gas content (fraction)

and

$$\gamma = (1 - f_g)\gamma_m + f_g \frac{0.763\gamma_g}{B_g} \tag{4-199}$$

where γ = gas cut mud specific weight in lb/ft^3
γ_m = gas free mud specific weight in lb/ft^3
γ_g = gas specific gravity (air = 1.0)
B_g = gas volume factor

For oil-base muds Equation 4-197 can be applied, but K and ρ must be calculated for an average natural gas using tables or the corresponding algorithms.

Table 4-128 shows maximum dissolved gas concentrations in drilling muds at the bottom of the hole. Figure 4-264 shows the variation of the acoustic velocity for two water-base muds and two oil-base muds of 9 and 18 lb/gal at pressures of 5,000 and 10,000 psi.

A sharp velocity decrease is seen for the water-base muds. Assuming a threshold detection of 500 ft/s, the alarm could be given for 0.5% of free gas or 1.1 to 1.4% of total gas (dissolved and free).

The oil-base muds having no free gas behave differently and the 500-ft/s threshold is not reached before approximately 5% of gas is dissolved. Then the velocity decrease is almost as fast as with the water-base mud.

Mud Specific Weight. The water-base mud specific weight can be calculated readily using Equation 4-199. The oil-base mud specific weight requires the use of tables.

The variations are shown in Figure 4-265 for the same 9- and 18-lb/gal muds. Specific weight-wise, the muds behave in a similar manner. Assuming that a density measurement with the gradiomanometer can be made accurately, the specific weight threshold would be 0.15 lb/gal. The gas content of the mud would be 2 to 5% according mainly to the density, the greater sensitivity being for the heavier mud.

Mud Resistivity. The mud resistivity can be measured only with the water-base muds. It is measured easily with a small microlog-type sensor embedded in the outer wall of the drill collar. Assuming the free gas is dispersed in small bubbles in the mud, the resistivity of the gas cut mud is

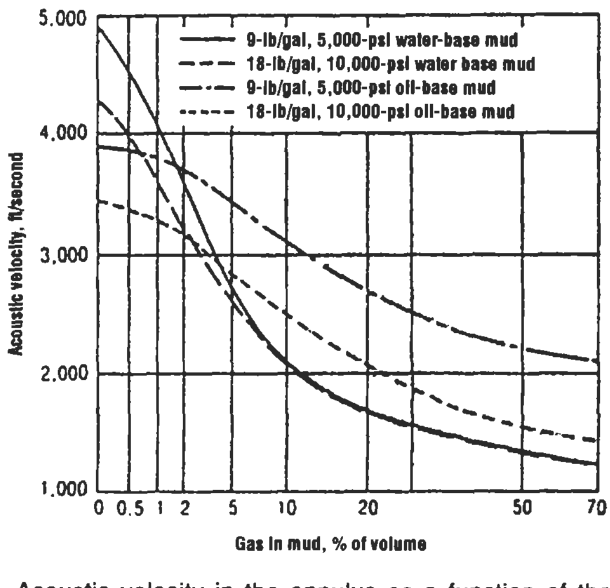


Figure 4-264. Acoustic velocity in the annulus as a function of the gas content in the mud. (*Courtesy Petroleum Engineer International* [96].)

$$R_{gcm} = \frac{R_m}{(1 - f_g)^2} \qquad (4\text{-}200)$$

where R_{gcm} = gas cut mud resistivity in $\Omega \bullet m$
R_m = gas free mud resistivity in $\Omega \bullet m$
f_g = volumetric gas content (fraction)

The variation is independent of the mud weight, pressure, or temperature, but is sensitive to fluids other than gas, such as oil or saltwater. Figure 4-266 shows the resistivity variations for a 1-$\Omega \bullet m$ mud. If we assume that a change of 10% can be detected, then the alarm could be given again for a free gas or oil volumetric concentration of 2 to 5%.

Mud Temperature. One can attempt to calculate the variation of the temperature of the mud when it mixes with a gas stream cooled by expansion.

Calculations were made with a 500-gal/min mud flowrate, an expansion from 10,500 to 10,000 psi with an 18-lb/gal mud and also an expansion from 5,500 to 5,000 psi with a 9-lb/gal mud. The temperature decrease of the mud was a few °F up to 50% gas by volume in the mud.

Temperature measurements do not seem to be good gas indicators.

Mud acoustic attenuation and annulus noise level are being investigated. It is expected that attenuation would be very sensitive to free gas concentration.

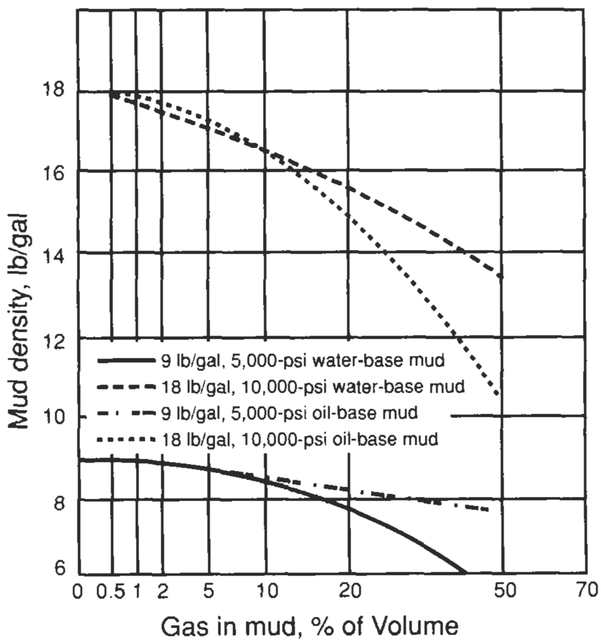


Figure 4-265. Bottomhole mud density in the annulus as a function of the gas content of the mud. (*Courtesy Petroleum Engineer International* [96].)

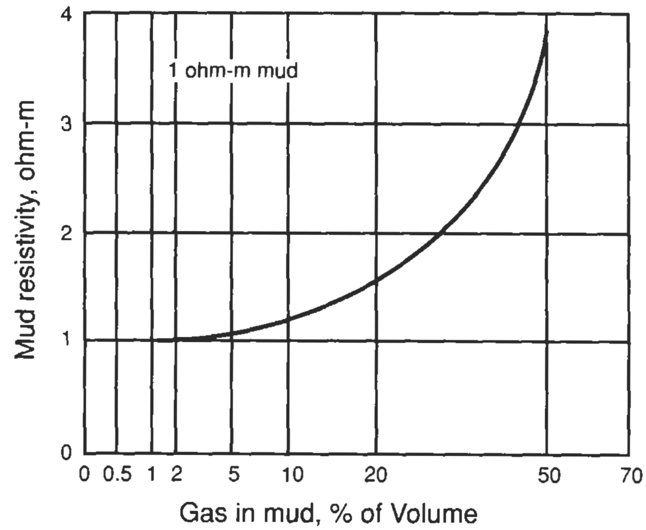


Figure 4-266. Bottomhole mud resistivity in the annulus as a function of the gas content of the mud. (*Courtesy Petroleum Engineer International* [108].)

Example 12: Drilling Parameters—Downhole Weight-on-Bit and Torque

We want to measure the bottomhole weight on bit and torque with a sensing collar sub 5-in. OD and 3-in. ID. The differential pressure across the three $\frac{16}{32}$-in. bit nozzles is 1,000 psi. We want to use platinum strain gages with a resistance of 50 Ω and a gage factor of 4. The Young modulus of the steel sub is 30×10^6 psi, the shear modulus is 12×10^6 psi.

1. Compute the end effect due to the internal differential mud pressure. Should we correct for this effect?
2. Compute the force acting on the drill collar for a weight-on-bit of 0, 30,000 and 100,000 lb.
3. The gages are stuck on the sub and connected to a Wheatstone bridge. What is the change in resistance from no load and no pressure for a weight-on-bit of 0, 30,000 and 100,000 lb?
4. Show that with two gages conveniently placed on the sub the bending strain compensates.
5. The bridge is supplied with 10 V, balanced for 0 lb. What is the unbalance for 100,000 lb?
6. Trace the response versus the weight-on-bit with
   (a) no differential pressure,
   (b) 1,000 psi differential pressure.
7. The maximum torque to be measured is 20,000 ft-lb. Using the same type of gages, properly placed on the sub, show that with two gages conveniently oriented the differential pressure strain and weight-on-bit strain do not register on the bridge.
8. Compute the resistance variation for each gage due to torque.
9. Compute the maximum signal using a 10-V supply.

Gage response to axial load is

$$\Delta R = \frac{(F \bullet R \bullet L)}{(E \bullet A)} \tag{4-201}$$

where R = gage resistance in Ω
ΔR = resistance variation in Ω
F = gage factor
L = load in lb
E = Young modulus
A = sub cross-section or area in in.^{2}

Gage response to torque is

$$\Delta R = \pm \frac{(T \bullet F \bullet R \bullet R_o)}{G \bullet \pi \bullet (R_o^4 - R_i^4)} \tag{4-202}$$

where T = torque in in • lb
F = gage factor
R = gage resistance in Ω
R_o = sub internal radius in in.
R_i = sub external radius in in.
G = shear modulus

Solution

1. 7.070 lb; yes, we should correct for this effect.
2. 7,070 lb downward, 22,930 lb upward, 92,930 lb upward.
3. -0.004, 0.012 and 0.049 Ω.
4. Typical Wheatstone bridge as shown in Figure 4-267. Connect the sensitive gages (1) and (4) on opposite sides of the sub.
5. Current in each leg: 0.1 A. For 100,000 lb (92,930 lb effective), ΔR = 0.049 Ω/gage. For two gages: ΔR = 0.098 Ω. ΔV = 0.0098 V = 9.8 mV.
6. Response equation: $\Delta V = 10.6 \times 10^{-5}$ L V.
7. Place the sensitive gages on opposite sides and opposite directions as shown in Figure 4-268. Connect the gages in (1) and (2), bridge will not be sensitive to axial load and ΔP.

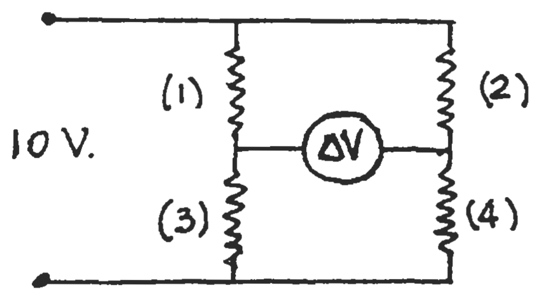


Figure 4-267. Sketch of a Wheatstone bridge for small resistance variations.

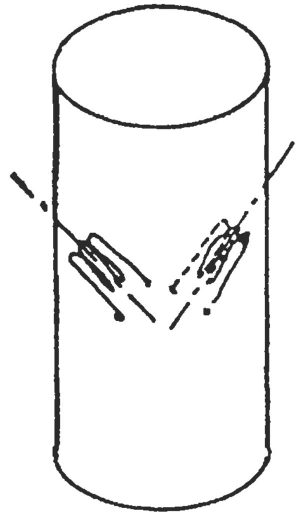


Figure 4-268. Sketch showing the theoretical position of strain gages for torque measurement.

8. ΔR torque = 0.01 Ω/gage.
9. Signal due to torque: ΔV = 0.002 V = 2 mV.

Example 13: Drilling Parameters—Annular Temperature

Bottomhole annulus mud temperature is recorded during drilling for mechanical problems and for fluid entry diagnosis.

Borehole depth: 10,000 ft, deviated hole
Drill pipe rotation rate: 10 rpm
Mud heat capacity: 0.77 cal/g
Hole diameter: $12\frac{1}{4}$ in.
Drainage radius: 660 ft
Mud specific weight: 12 lb/gal
Mud flowrate: 500 gal/min
Gas gravity: 0.7
z: 0.9

1. The surface measured torque is 2 kft-lb and the downhole torque is 1 kft-lb. Assuming the heat generated is entirely transferred to the descending mud stream, what is the temperature rise due to the pipe friction?
2. A water inflow occurs suddenly at the rate of 1,000 bbl/day. Water heat capacity is 1 cal/g; water density is 1,00 kg/m^3. The formation temperature is 200°F and the mud reaches the drill collars at a temperature of 160°F. Compute the annular temperature rise.
3. A gas inflow occurs suddenly when entering an abnormal pressure zone. Compute the flowrate of gas if the formation pressure is 7,000 psi, 1 ft has been penetrated in a 50-ft zone with 500 md, gas viscosity is 0.035 cp. Assume no annulus pressure drop, no cutting. Compute the annular temperature drop.

$$Q_b = 19.88 \bullet \frac{k \bullet f \bullet h\left(P_e^2 - P_w^2\right)}{\mu \bullet P_w \bullet \ln\left(\frac{r_e}{r_w}\right)}\left[1 + 7\cos\left(f \bullet \frac{\pi}{2}\right)\sqrt{\frac{r_w}{2 \bullet f \bullet h}}\right] \tag{4-203}$$

$$Q_s = \frac{P_w \bullet T_s \bullet Q_b}{z \bullet P_s \bullet T_w} \tag{4-204}$$

Solution

1. Dissipated energy: 852,000 J/min
   Massic mud flow: 2721 kg/min
   Calories required to raise temperature by 1°C: 2,095,170 cal/min
   Calories available: 203,828 cal/min
   Temperature rise: $\Delta T = 0.1°C = 0.17°F$
2. Heat given up by the inflowing water equals heat received by the mud. $\Delta T = 2.5°F$
3. Bottomhole pressure: 6240 psi
   Gas flow at downhole conditions: 286,000 ft³/d = 8,099 m³/d = 5.62 m³/min
   Gas pressure decrease: 760 psi = 5,239,440 Pa
   a. Energy absorbed by the gas if E = VdP = 29,468,211 J/min = 7,049,811 cal/min
   Temperature decrease of the mud: 3.36°C = 6°F
   b. Energy absorbed by the gas in isentropic process = 600 Btu/lb mole (See [113], p. 96)
   When converted and for massic flowrate of 443.6 lbm/min = 2,501,393 cal/min
   Temperature decrease of the mud: 1.19°C = 2.15°F
   (second calculation is probably more correct)

Example 14: Drilling Parameters—Drill Collar Pressure Drop

The following data characterize a well during drilling:

depth: 10,000 ft
$4\frac{1}{2}$-in. drillpipes (ID = 3.64 in.)
mud specific weight: 12 lb/gal
flowrate: 500 gal/min
three-bit nozzles: $\frac{16}{32}$-in. diameter
mud viscosity: 12 cp
nozzle factor: C = 1.0
hole diameter: 8.5 in.

1. Assuming no cutting in the annulus, compute the pressure recorded inside the drill collars downhole and the pressure in the standpipe at surface using the formula given hereafter.
2. A leak develops in the pipe string. The standpipe pressure reading drops to 1,896 psi with the same mud flowrate and the downhole drill collar inside pressure drops to 6,700 psi. What is the flowrate of the leak? What is the area of the leaking hole assuming it is located at 3,000 or 5,000 or 7,000 ft? (Assume that ΔP annulus does not change.)

Equations

1. Hydrostatic pressure is

$$P_H = 0.052 \bullet \gamma \bullet z \tag{4-205}$$

where P_H = hydrostatic pressure in psi
γ = mud specific weight in lb/gal
z = depth in ft

2. Turbulent flow pressure loss in pipe (Equation 4-187) is

$$\Delta P = \frac{\Delta L \bullet \gamma^{0.75} \bullet v^{1.75} \bullet \mu^{0.25}}{1800 \bullet d^{1.25}}$$

where ΔP = pressure loss in psi
ΔL = pipe length in ft
γ = fluid specific weight in lb/gal
v = average fluid velocity in ft/s
m = fluid viscosity in cp
d = pipe ID in in.

with

$$v = Q/(2.448 \times d^2)$$

where Q = flowrate in gal/min
d = pipe ID in in.

3. Turbulent flow pressure loss in annulus (Equation 4-188) is

$$\Delta P = \frac{\Delta L \bullet \gamma^{0.75} \bullet v^{1.75} \bullet \mu^{0.25}}{1396(d_2 - d_1)^{1.25}}$$

where (notations same as above) d_2 = borehole or casing diameter in in.
d_1 = pipe OD in in.

4. Flowrate through a choke, or nozzle, or leak (Equation 4-186) is

$$Q = C \bullet A \bullet \sqrt{\frac{2 \bullet g_c \bullet \Delta P}{144 \bullet \gamma}}$$

where Q = flowrate in ft³/s
C = coefficient (0.95 to 1.0)
g_c = acceleration of gravity (32.17 ft/s²)
ΔP = pressure loss in psi
γ = fluid specific weight in lb/ft³
A = area in ft²

Solution

1. ΔP drillpipes: 1,590 psi
ΔP bit nozzles: 718 psi
ΔP annulus: 74 psi

Hydrostatic pressure: 6,240 psi
P inside DC: 7,032 psi
P standpipe: 2,382 psi

2. Total DP in pipe (friction plus leak): 1,436 psi
   Pressure drop in DC due to leak: 332 psi
   New ΔP across nozzles: 386 psi
   Q through nozzles: 366.5 gal/min
   Q through drillpipes below leak: 366.5 gal/min
   Q through leak: 133.5 gal/min

If leak at 3000 ft:

Pipe ΔP above leak: 477 psi
Pipe ΔP below leak: 646.5 psi
ΔP across leak: 312.5 psi
Leak cross-section: 0.24 in.^{2}

If leak at 5000 ft:

Pipe ΔP above leak: 745 psi
Pipe ΔP below leak: 462 psi
ΔP across leak: 229 psi
Leak cross-section: 0.28 in.^{2}

If leak at 7,000 ft:

Pipe ΔP above leak: 1,272 psi
Pipe ΔP below leak: 277 psi
Sum is more than total ΔP
The leak must be above 7,000 ft

LWD Technology

Logging while drilling has been attempted as early as 1939. The first commercial logs were run in the early 1980s. First gamma ray logs were recorded downhole and transmitted to the surface by mud pulses. Then came the resistivity logs of various types that were also recorded downhole and/or transmitted to the surface. Now, neutron-density and Pe logs are also available. Soon, sonic logs will be offered commercially.

Gamma Ray Logs

Gamma rays of various energy are emitted by potassium-40, thorium, uranium, and the daughter products of these two last elements contained in the earth formations surrounding the borehole. These elements occur primarily in shales. The gamma rays reaching the borehole form a spectrum typical of each formation extending from a few keV to several MeV.

The gamma rays are detected today with sodium iodide crystals scintillation counters. The counters, 6 to 12 in. long (15 to 30 cm) are shock mounted and housed in the drill collars. Several types of measurements can be made: total gamma rays, direction-focused gamma rays, spectral gamma rays.

Total Gamma Rays. Total gamma ray logs have been run on electric wireline since 1940. The sondes are rather small in diameter (1.5 to 4 in. or 37 to 100 mm).

The steel housing rarely exceeds 0.5 in. (12 mm) and a calibration is done in terms of API units, arbitrary units defined in a standard calibration pit located at the University of Houston.

The MWD total gamma ray tools cannot be calibrated in the standard pit, since they are too large. Their calibration in API units is difficult because it varies with the spectral content of the radiation. By spectral matching the MWD logs can be made to closely resemble the wireline logs. The logs which were recorded by the MWD companies in counts per second (cps) are now recorded in API units.

Another difference between the wireline logs and the MWD logs is the logging speed. With a wireline, the sonde is pulled out at a speed of 500 to 2,000 ft/min (150 to 600 m/min). The time constant used to optimize the effect of the statistical variations of the radioactivity emission, varied from 2 to 6 s. Consequently, the log values are somewhat distorted and inaccurate.

In MWD, the recording speed is the rate of penetration which rarely exceeds 120 to 150 ft/hr or 2 to 2.5 ft/min, two orders of magnitude less than the logging speed. Counters can be made shorter and time constant longer (up to 30 s or more). This results in a better accuracy and a better bed definition. Figure 4-269 shows an example of comparison between an MWD gamma ray log and the wireline log ran later.

To summarize, the total gamma ray measurements are used for real-time correlation, lithology identification, depth marker and kick-off point selection.

Direction-Focused Gamma Rays. It is important to keep the trajectory of horizontal or nearly horizontal wells in the pay zone. By focusing the provenance of the gamma rays it is possible to determine if a shale boundary is approached from above or from below.

The tool shown in Figure 4-270 has its scintillation detector inserted in a beryllium-copper housing, fairly transparent to gamma rays. A tungsten sleeve surrounds the beryllium-copper housing, with a 90° slot or window running from top to bottom. Figure 4-270 is a sketch of the tool cross-section. The center of the window is keyed to the reference axis of the directional sensor. Consequently the directional sensor indicates if the window is pointing up or down.

By rotating the tool, one can differentiate between the level of gamma rays entering from the top and the lower part of the borehole. A sinusoidal response is recorded which depends on the following:

- distance from the bed boundary.
- gamma ray intensity of the bed in which the tool is in.
- the contrast of radioactivity at the boundary.
- the shielding efficiency of the tungsten sleeve.

An example of the log ran is a horizontal borehole as shown in Figure 4-271. The depths on the log are along the hole depths. Vertical depths are shown in the higher part of the log with a representation of the true radioactivity of each bed. The following observations can be made:

- Approaching formation bed boundaries are detected by concurrent separation and displacement of the high and low gamma counts. These are shown in Figure 4-271 at measured depth intervals (7970–7980 ft) and (8010–8020 ft).
- Radioactive events occur in the measured depth interval (8,100–8,200 ft) with no displacement of the low/high side gamma ray logs. The radioactive events must be perpendicular to the gamma detector and could be indications of vertical natural fractures in the formation.

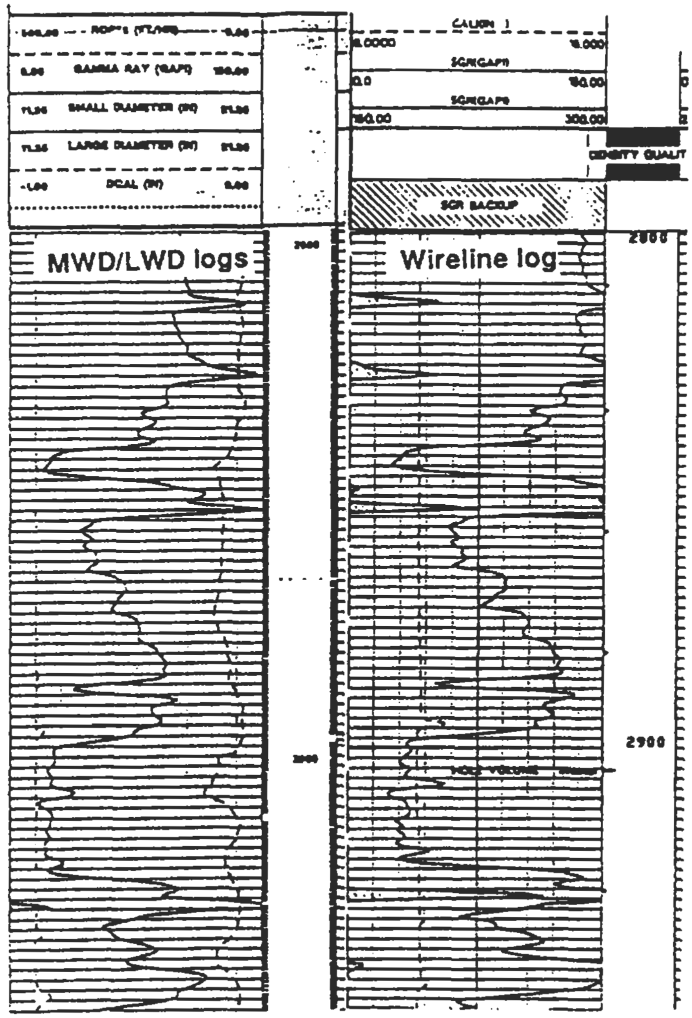


Figure 4-269. Example of good similarity displayed between the MWD gamma ray log and the wireline log.

Spectral Gamma Ray Log. This log makes use of a very efficient tool that records the individual response to the different radioactive minerals. These minerals include potassium-40 and the elements in the uranium family as well as those in the thorium family. The GR spectrum emitted by each element is made up of easily identifiable lines. As the result of the Compton effect, the counter records a continuous spectrum. The presence of potassium, uranium and thorium can be quantitatively evaluated only with the help of a computer that calculates in real time the amounts present. The counter consists of a crystal optically coupled to a photomultiplier. The radiation level is measured in several energy windows.

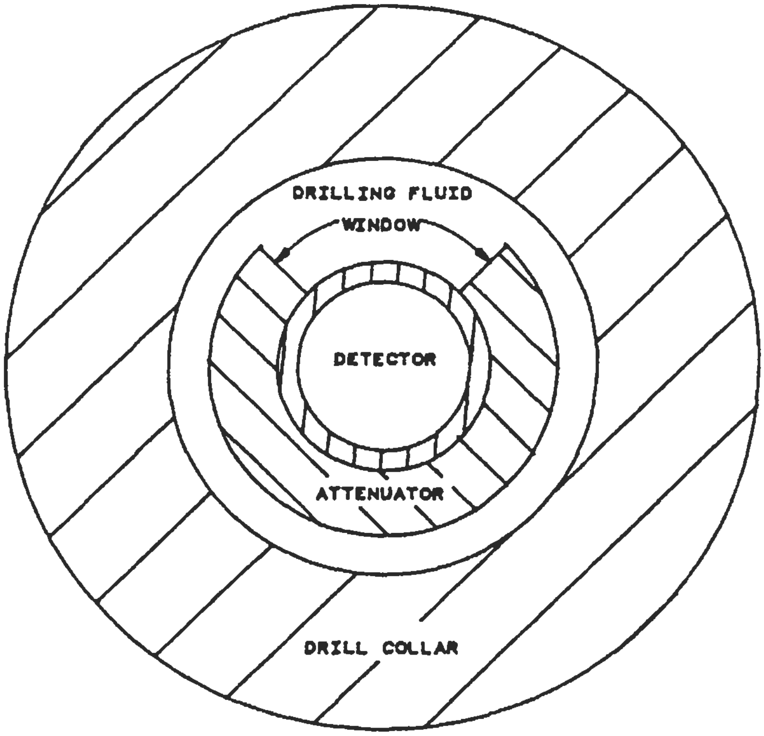


Figure 4-270. Cross-section of an MWD focused gamma ray tool. *(Courtesy SPWLA* [112].*)*

Figure 4-272 shows an example of a MWD spectral GR log. On the left track, SGR is the total GR count, and CGR is this total count minus the uranium count. On the right side of Figure 4-272 the wireline spectral gamma ray in the same interval is displayed. The curves are similar but some differences occur in the amplitude of the three curves.

The main field applications of this log are:

1. Clay content evaluation: Some formations may contain nonclayey radioactive materials. Then the curve GR-U or GR-K may give a better clay content estimate.
2. Clay type identification: A plot of thorium versus potassium will indicate what type of clay is present. The thorium/potassium ratio can also be used.
3. Source rock potential of shale: A relation exists between the uranium-to-potassium ratio and the organic carbon content. The source rock potential of shale can thus be evaluated.

Resistivity Logs

Four types of resistivity logs are currently run while drilling:

1. short normal resistivity
2. focused current resistivity

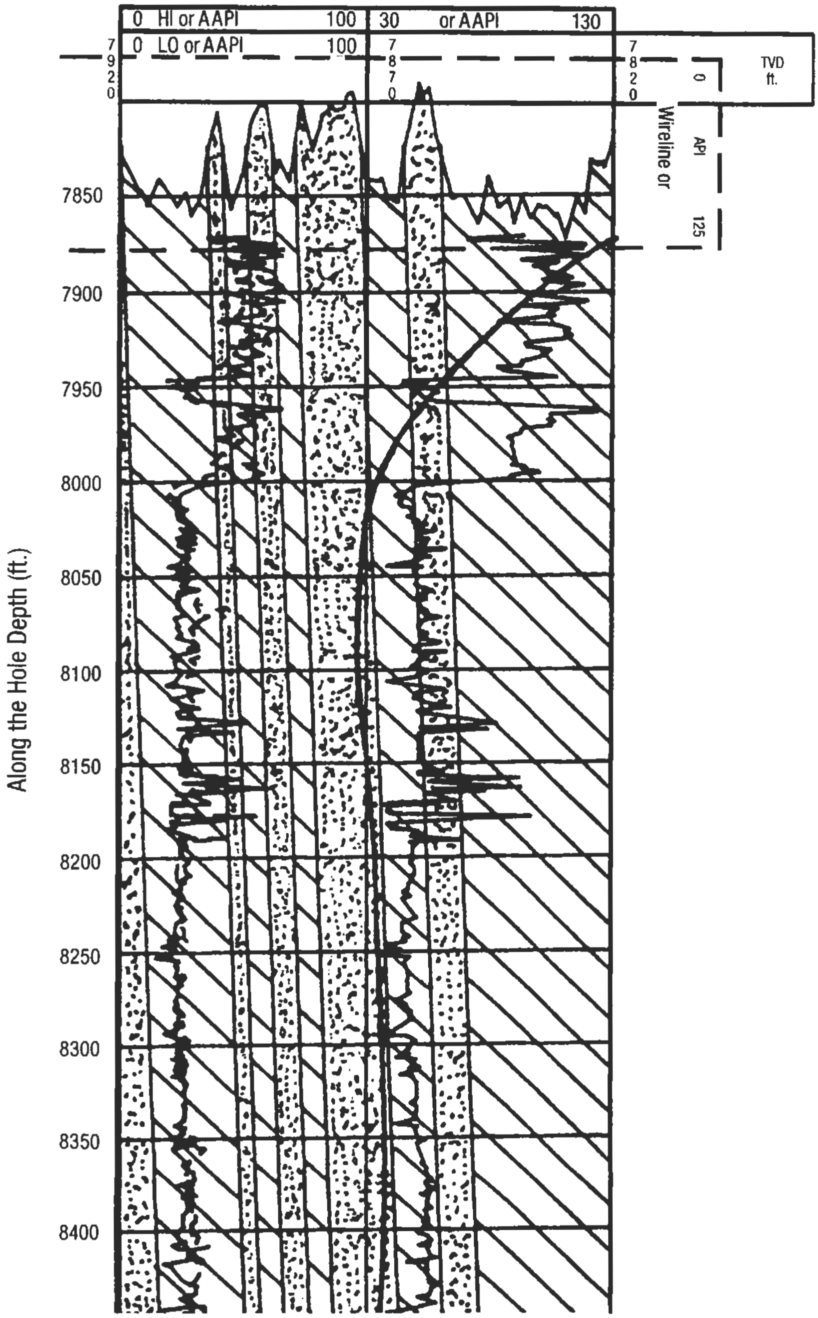


Figure 4-271. MWD focused gamma ray composite log in a horizontal borehole. *(Courtesy SPWLA* [112].*)*

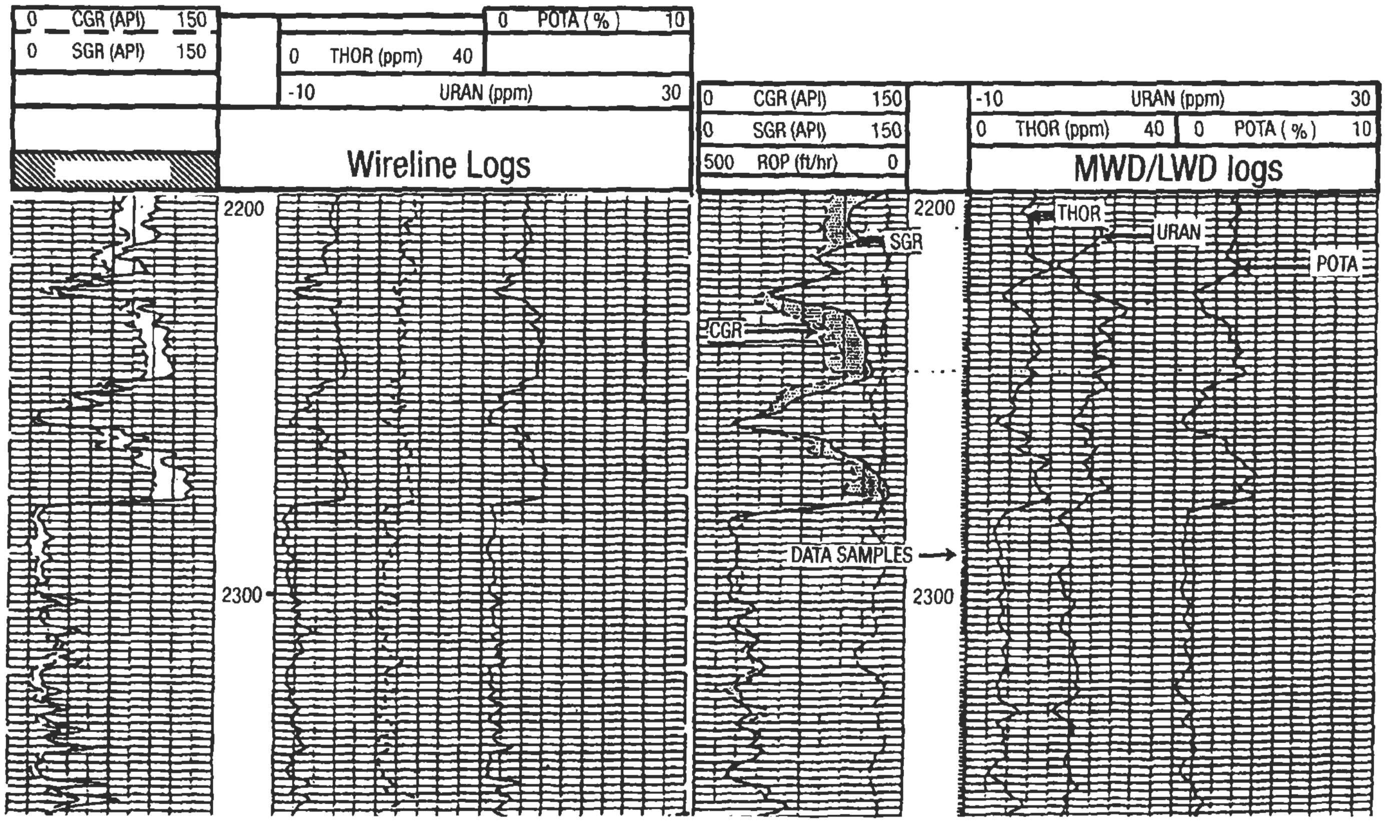


Figure 4-272. Example of natural gamma ray spectral logs recorded while drilling and with a wireline.

3. electromagnetic resistivity
4. toroidal system resistivity

Short Normal Resistivity (after Anadrill). The short normal (SN) resistivity sub provides a real-time measurement of formation resistivity using a 16-in. electrode device suitable for formations drilled with water-base muds having a moderate salinity. A total gamma ray measurement is included with the resistivity measurement; an annular bottomhole mud temperature sensor is optional. The short normal resistivity sub schematically shown in Figure 4-273 must be attached to the MWD telemetry tools and operates in the same conditions as the other sensors.

Due to the small invasion and the large diameter of the sonde body, a resistivity near the true resistivity of the formation is generally measured. This is particularly true in shale where no invasion takes place. The main applications are:

- real-time correlation and hydrocarbon identification
- lithology identification for casing point and kick-off point selection
- real-time pore pressure analysis based on resistivity trend in shales
- resistivity range: 0.2 to 100 Ω • m

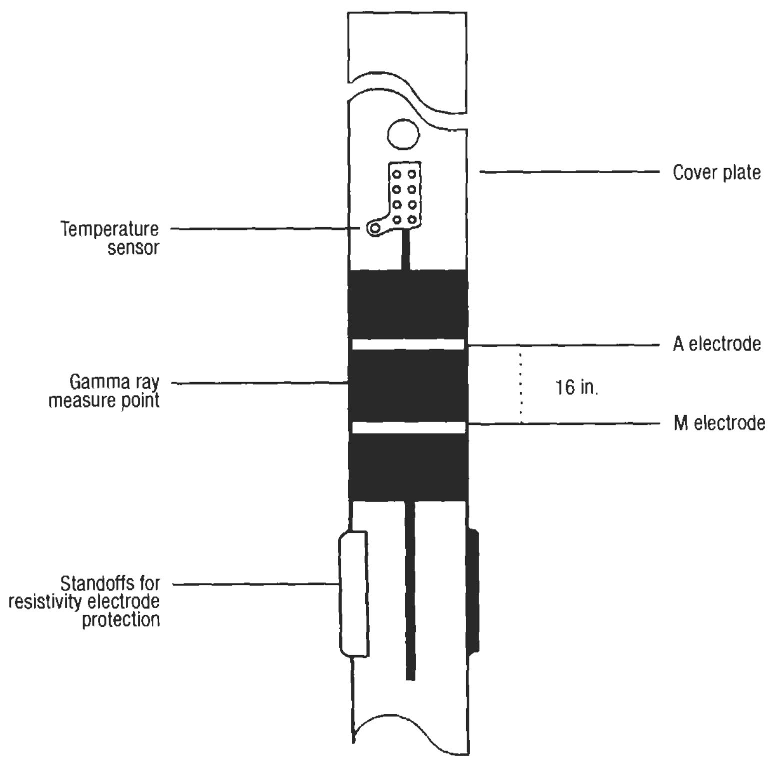


Figure 4-273. Short normal resistivity sub. *(Courtesy Anadrill* [113].*)*

Focused Current Resistivity. Focused resistivity devices are particularly suited for wells where highly conductive drilling muds are used, where relatively high formation resitivities are encountered and where large resistivity contrasts are expected.

The focused current system employs the guarded electrode design shown in Figure 4-274.

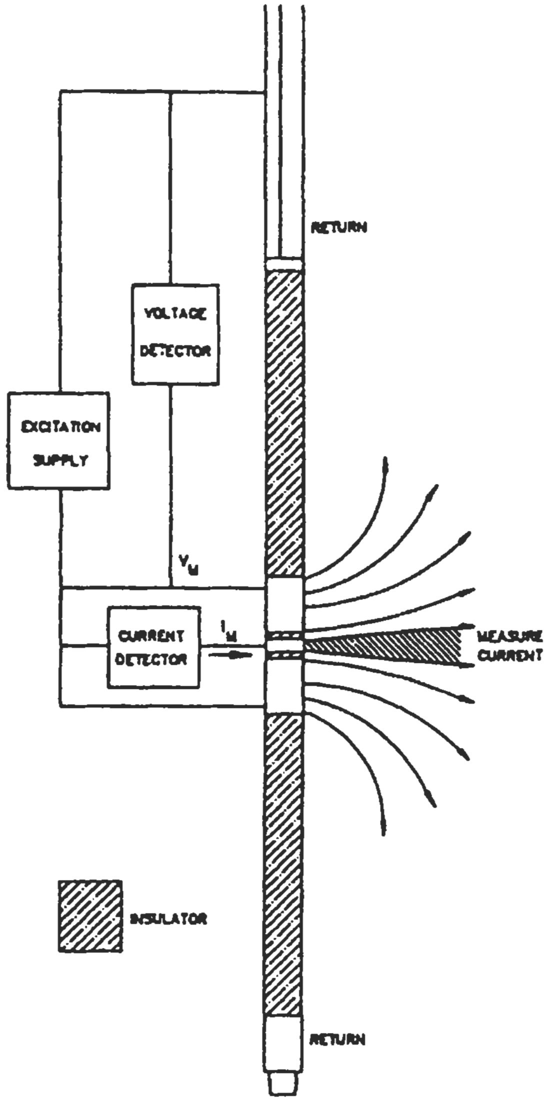


Figure 4-274. Block diagram of an LWD focused current system. *(Courtesy SPE* [114].*)*

The system is similar to the laterolog 3 used in wireline logging. A constant 1-k Hz AC voltage is maintained for all electrodes. The current flowing through the center electrode is measured.

The resistivity range is 0.1 to 1000 Ω • m. Beds as thin as 6 in. (15 cm) can be adequately delineated.

Electromagnetic Resistivity. The measurement in electromagnetic resistivity systems is similar to the wireline induction sonde resistivity. The frequency used is 2 MHz instead of 20 kHz. This is due to the drill collars steel that would completely destroy a 20-kHz signal. Early systems had one transmitter coil and two receiver coils. Systems presently in use have two to four transmitters allowing the recording of many curves with different depths of investigation. Figure 4-275a shows the CDR, compensated dual resistivity tool of Anadrill.

Figure 4-275b is a schematic of the operating principle. Two signals are measured: the wave amplitude reduction and the wave phase shift.

Two values of the resistivity can be calculated. The wave amplitude resistivity (R_{ad}) appears to have a deep investigation radius: 35 to 65 in. according to the formation resistivity. The phase shift resistivity (R_{ps}) appears to have a shallow investigation radius: 20 to 45 in. An example of tool response is given in Figure 4-276.

The deep penetration curve reads a value close to the noninvaded zone resistivity and the shallow penetration curve reads a value much lower than the invaded zone resistivity. The resistivity ranges for an acceptable accuracy are 0.15 to 50 Ω • m for the deep investigation radius (R_{ad}) and 0.15 to 200 Ω • m for the shallow investigation radius (R_{ps}). The vertical resolution is 6 in. (15 cm).

Toroidal System Resistivity (after Gearhart-Halliburton). The system uses one toroidal transmitter operating at 1 kHz and a pair of toroidal receiver coils mounted on the drill collars. Figure 4-277 shows a sketch of a toroid.

The winding of the toroid acts as a transformer primary and the drill collar as the secondary. The current lines induced by the drill collar are shown in Figure 4-278.

The drill collar acts as a series of elongated electrodes in a way similar to the laterolog 3 wireline sonde. The lower electrode, which is the drill bit, is used to get the "forward" resistivity curve. A lateral resistivity measurement is made between the two toroid receivers. An example of toroid logs is shown in Figure 4-279.

The readings of both toroid curves seem to follow closely the ILd and ILm curves.

Example 15: Gamma Ray and Resistivity Interpretation

A typical set of logs recorded while drilling is shown in Figure 4-280. The wireline caliper is shown in the gamma ray track. Displayed on this attachment are gamma ray, R_{wa} curve, Pe curve, neutron and density curve. The delta-rho curve is the quality curve check for the density log.

1. Draw a lithology description in the depth column.
2. Is the clean formation permeable? Why?
3. Does the porous zone contain hydrocarbons? What type? Give the boundaries.
4. Determine R_w.
5. Compute the hydrocarbon saturation at 8400 ft assuming $a = 1$ and $m = 2$.

(text continued on page 982)

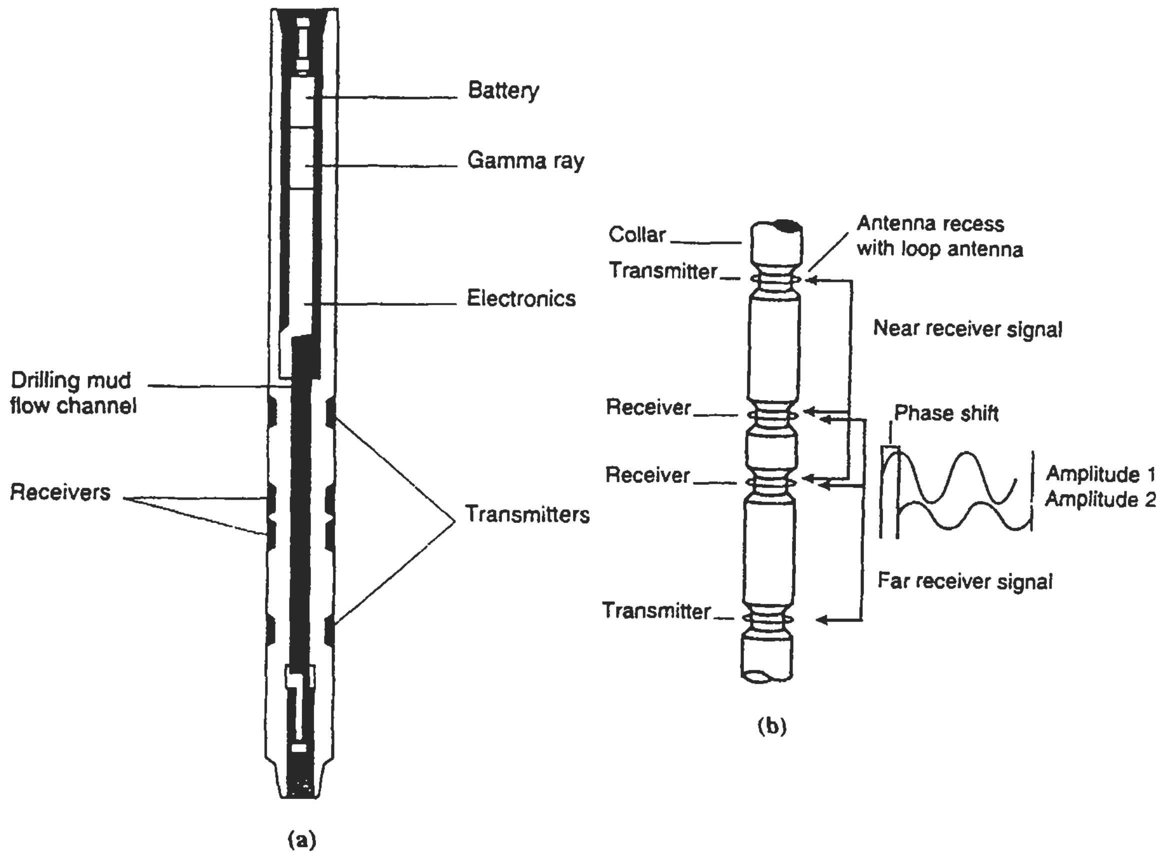


Figure 4-275. Compensated dual-resistivity tool; (a) sub design; (b) operating principle. *(Courtesy Anadrill* [113].*)*

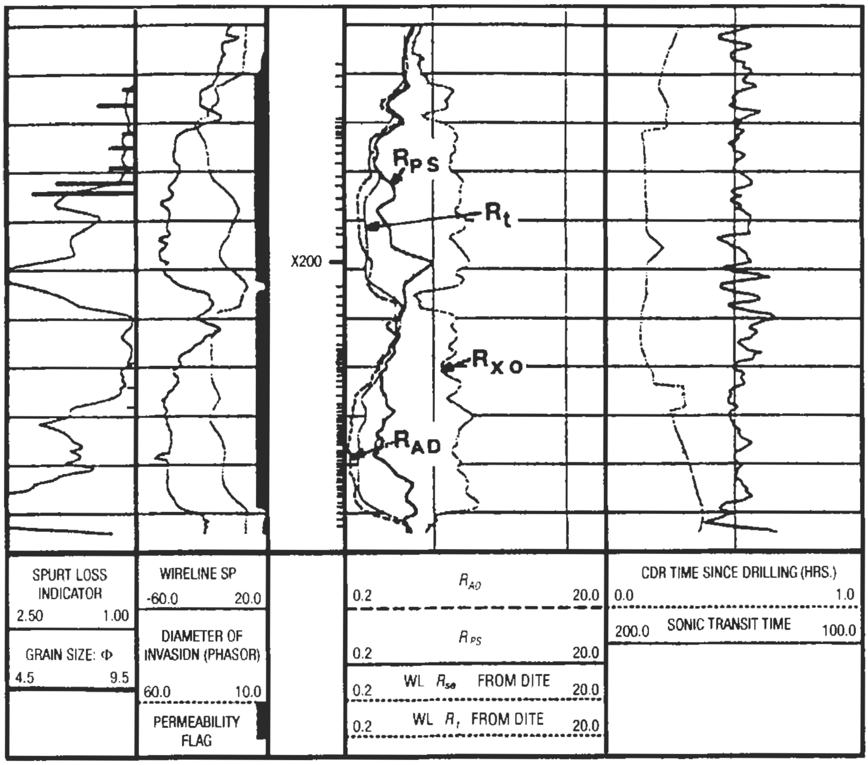


Figure 4-276. Comparison of the compensated dual-resistivity log resistivities run while drilling to the invaded and noninvaded resistivities calculated with wireline phasor induction data. The spurt loss is the ratio R_{ps}/R_{ad}. *(Courtesy Anadrill* [113].*)*

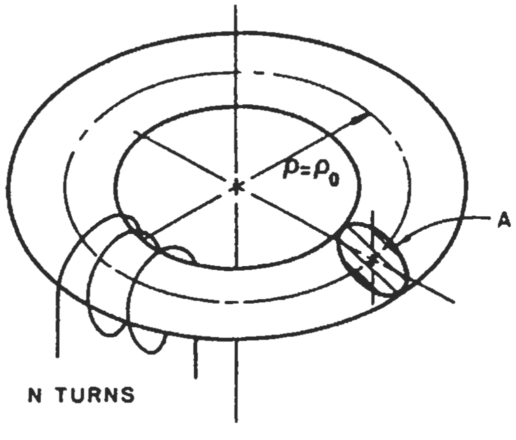


Figure 4-277. Toroid mounted on a drill collar. *(Courtesy SPWLA* [115].*)*

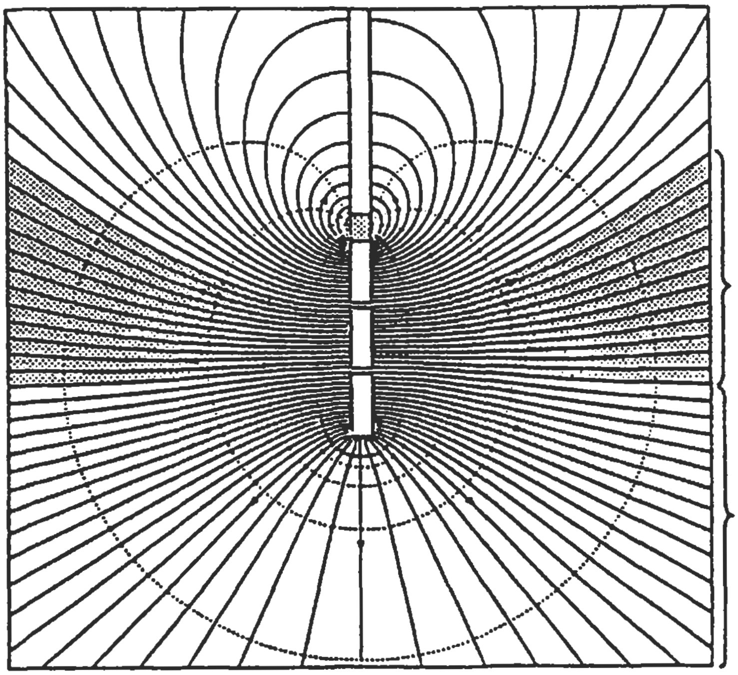


Figure 4-278. Computed current pattern in a homogeneous formation for the MWD toroid system. *(Courtesy SPWLA* [115].*)*

(text continued from page 979)

Solution

1. 8,450 to 8,434 ft dolomite
   8,434 to 8,430 ft shale
   8,430 to 8,426 ft dolomite
   8,426 to 8,423 ft shale
   8,423 to 8,374 ft dolomite
   8,374 to 8,350 ft shale
   Rock nature is read on the Pe log.
2. Yes, a mud cake is seen on the caliper log.
3. Yes, the R_{wa} curve increases sharply in the main zone at 8425 ft. Oil from 8425 to 8400 ft. Gas above 8,400 ft. Gas is indicated by a density porosity larger than the neutron porosity.
4. $R_w = 0.05\ \Omega \bullet m$; read on R_{wa} curve in the lower porous zone.
5. At 8400 ft, porosity = 20%, $R_{wa} = 0.45$, F = 25, $R_t = 11.25\ \Omega \bullet m$

$$S_w = \sqrt{\frac{25 \bullet 0.05}{11.25}} = 0.33 = 33\%$$

$$S_{hc} = 67\%$$

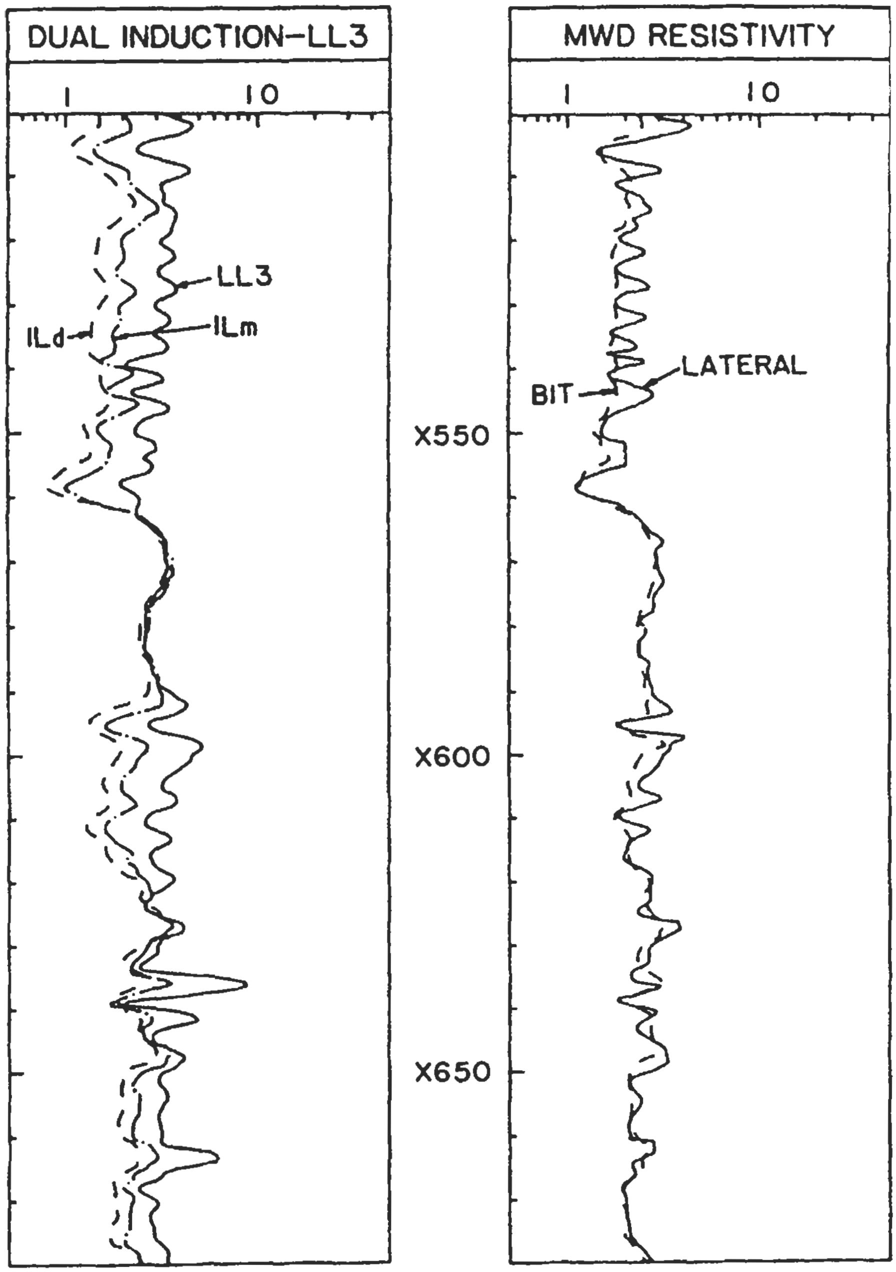


Figure 4-279. Comparison of toroid logs with dual induction logs. *(Courtesy SPWLA* [115].*)*

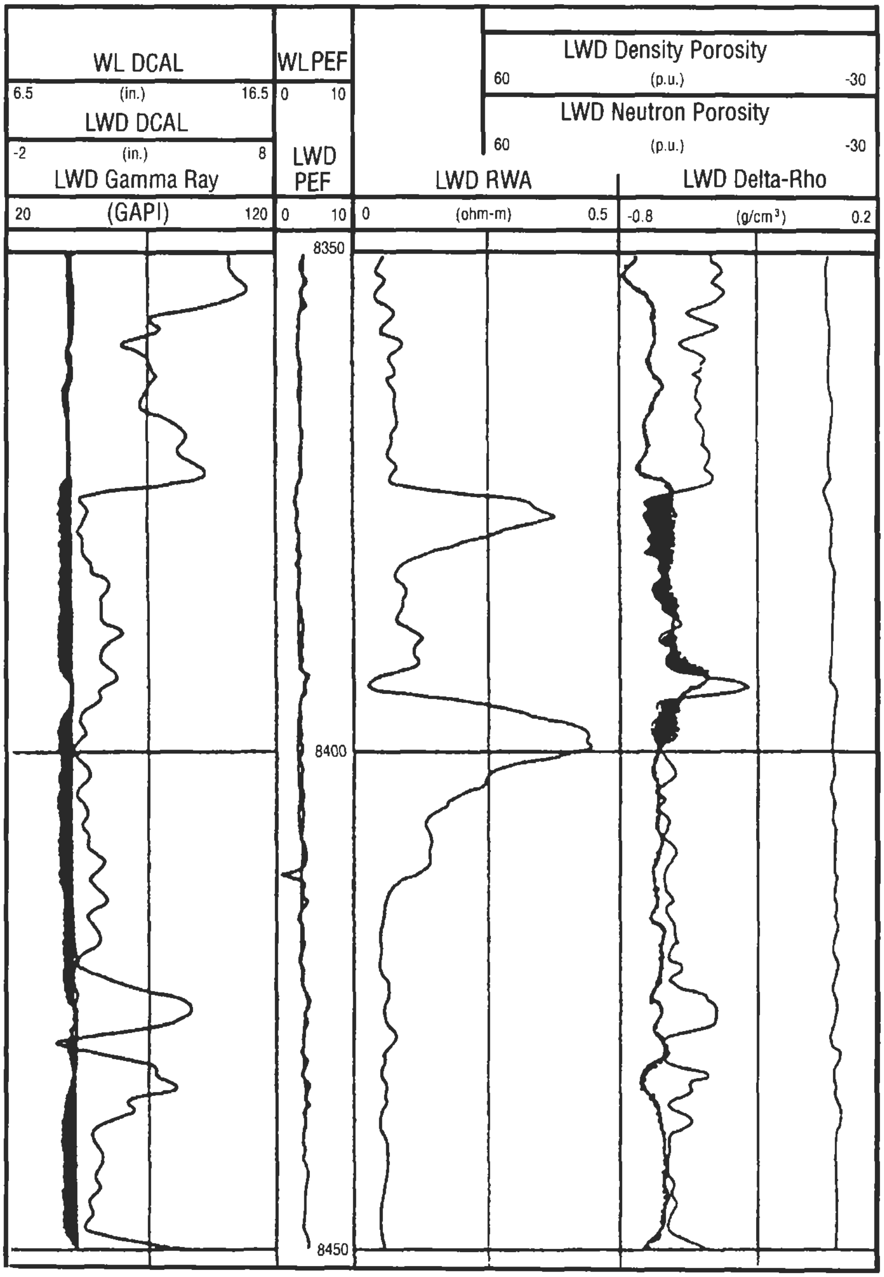


Figure 4-280. LWD logs recorded while drilling [113].

Neutron-Density Logs

The physics of the measurements made by the MWD neutron-density tools are similar to those of corresponding wireline sondes. A sketch of principle of the Anadrill tool is shown in Figure 4-281.

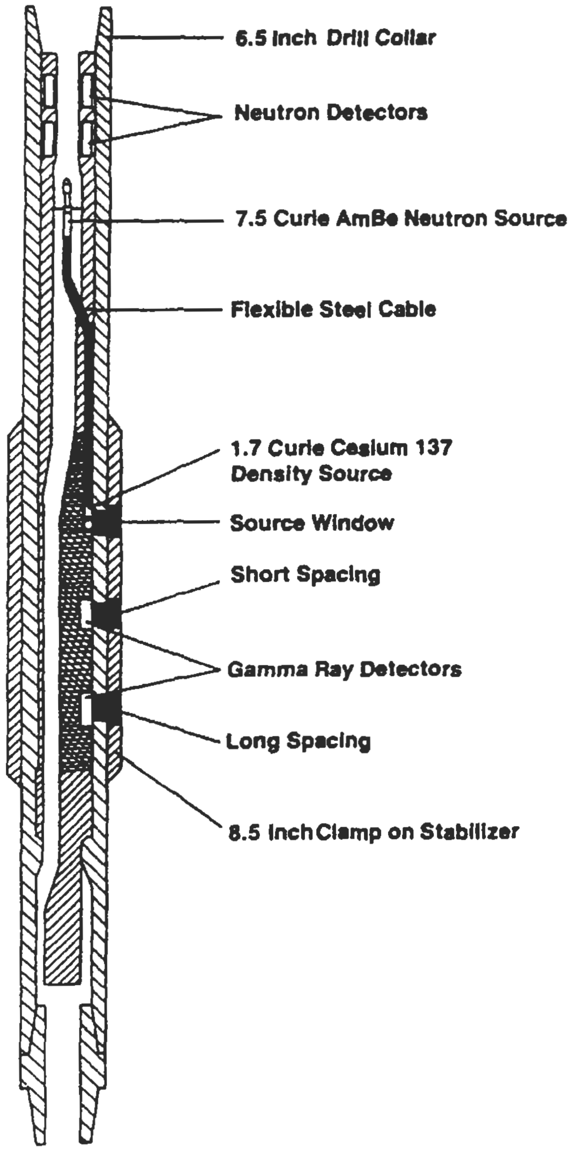


Figure 4-281. Neutron-density sub. *(Courtesy Anadrill* [113].*)*

For the neutron porosity measurement, fast neutrons are emitted from a 7.5-curie (Ci) americium–beryllium (Am–Be) source. The quantities of hydrogen in the formation, in the form of water or oil-filled porosity as well as crystallization water in the rock if any, primarily control the rate at which the neutrons slow down to epithermal and thermal energies. Neutrons are detected in near- and far-spacing detectors, located laterally above the source. Ratio processing is used for borehole compensation.

The energy of the detected neutrons has an epithermal component because a high percentage of the incoming thermal neutron flux is absorbed as it passes through a 1 in. of drill collar steel. Furthermore, a wrap of cadmium under the detector banks shields them from the thermal neutron arriving from the inner mud channel. This mainly epithermal detection practically eliminates adverse effects caused by thermal neutron absorbers in the borehole or in the formation, such as boron.

Figure 4-282 shows a typical comparison of wireline and MWD gamma ray and neutron logs in a borehole in excellent hole conditions. The MWD/LWD log matches the wireline log almost perfectly.

The density section of the tool, also seen in Figure 4-281, uses a 1.7 curie (Ci) of 137-cesium (Ce) gamma ray source in conjunction with two gain-stabilized scintillation detectors to provide a high-quality, borehole compensated density measurement.

The tool also measures the photoelectric effect Pe for lithology identification. The density source and detectors are positioned close to the borehole wall in the fin of a full-gage clamp-on stabilizer as seen in Figure 4-281. This geometry excludes mud from the path of the gamma rays, greatly reducing borehole effect. In deviated and horizontal wells, the stabilizer may be run under gage for directional drilling purposes. Rotational processing provides a correction in oval holes and yields a differential caliper. Figure 4-283 shows a schematic of the tool positions in a borehole.

In the top part of Figure 4-283 the stand-off is constant during the rotation. In the oval borehole represented in the lower part of the figure, the stand-off is excessive when the density system is oriented up and the normal $\Delta\rho$ correction is not enough.

Statistical methods are used to measure the density variation as the tool rotates, the stand-off can be estimated and the density corrected. A density caliper can be computed that works for cavings of 2 in. (5 cm) or less when the tool rotates at a speed ranging from 6 to 150 rpm.

Figure 4-284 shows a typical MWD density log compared to a wireline density log. The calipers are also shown. At 1620 ft, the wireline caliper detects a much larger caving because it was run several days later.

Photoelectric (Pe) Curve. The Compton effect (change in gamma ray energy by interaction with the formation electrons) is used for measuring the density of the formation. The energy range is 200 to 450 keV. The photoelectric effect (absorption of a low-energy gamma due to ejection of a low orbital electron from its orbit) is seen for gamma in an energy range of 35 to 100 keV.

By counting the gamma of low energy reaching the first counter a Pe curve sensitive to the nature of the formation can be recorded. A special counter protection fairly transparent to low-energy gamma ray (beryllium) is used. Table 4-129 shows the value of Pe for various lithologies.

Two systems are presently used by the MWD service companies concerning the radioactive source installation. One way is to lock the sources in holes in the drill collars. Thus, if the BHA is lost, the sources are left in the formation.

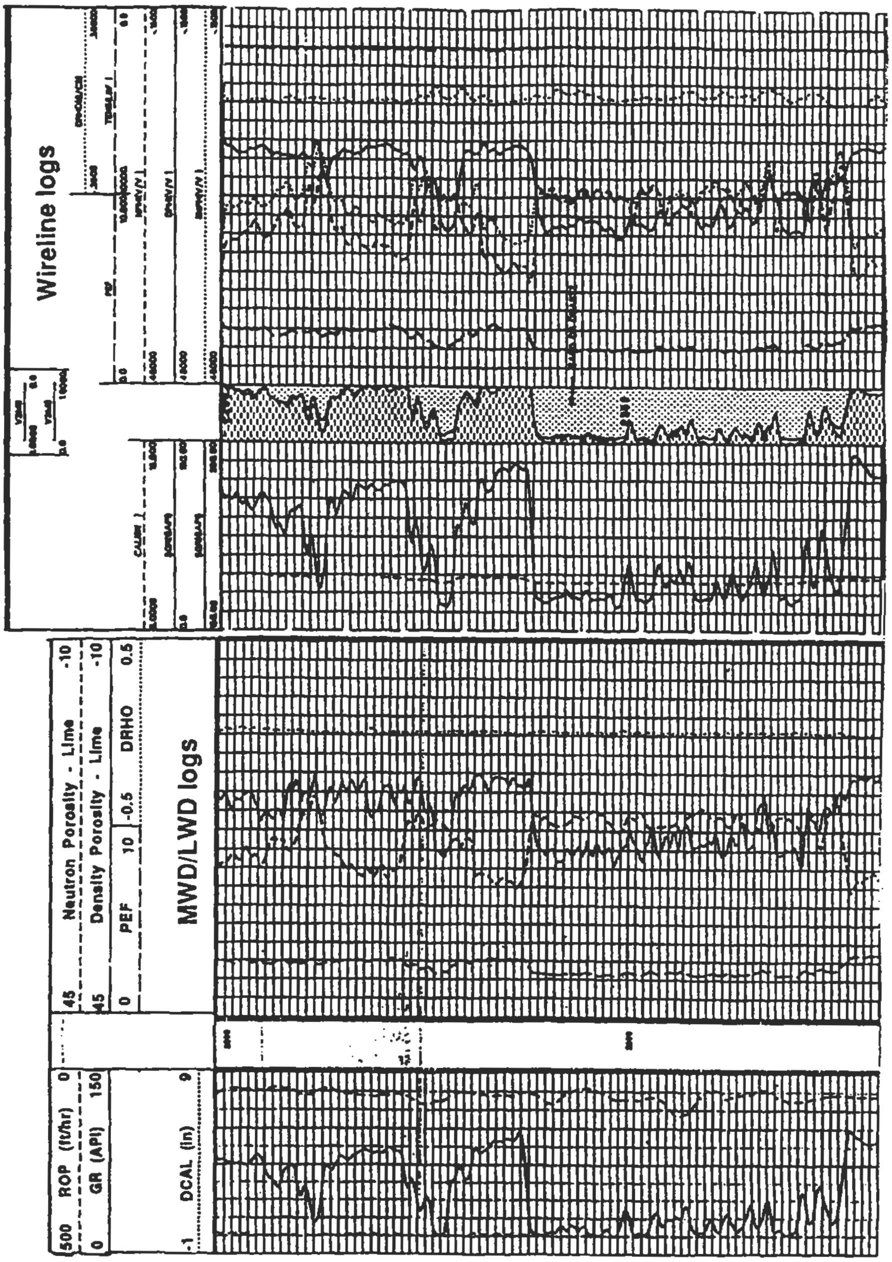


Figure 4-282. Comparison of wireline and LWD gamma ray and neutron logs in a borehole in good conditions.

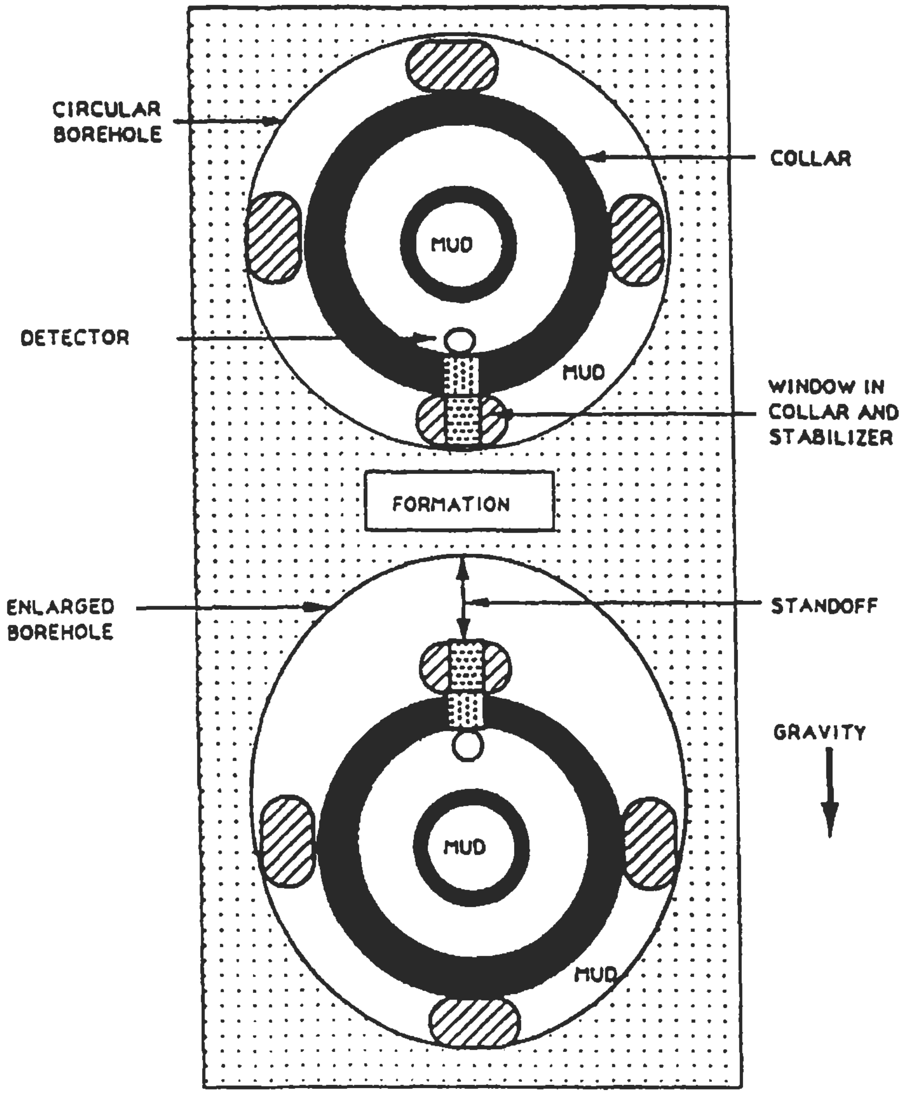


Figure 4-283. Schematic of the density tool position in a borehole. *(Courtesy Anadrill* [113].*)*

Another way (Anadrill) is to use removable sources. Figure 4-285 shows the sources being installed in the tool at surface. The two sources, neutrons and gammas, are mounted on the same flexible shaft. They are moved from the shield to the sub without being exposed. Furthermore, if the BHA becomes stuck, they can be fished out with an overshot that connects to the fishing head on top of the neutron source.

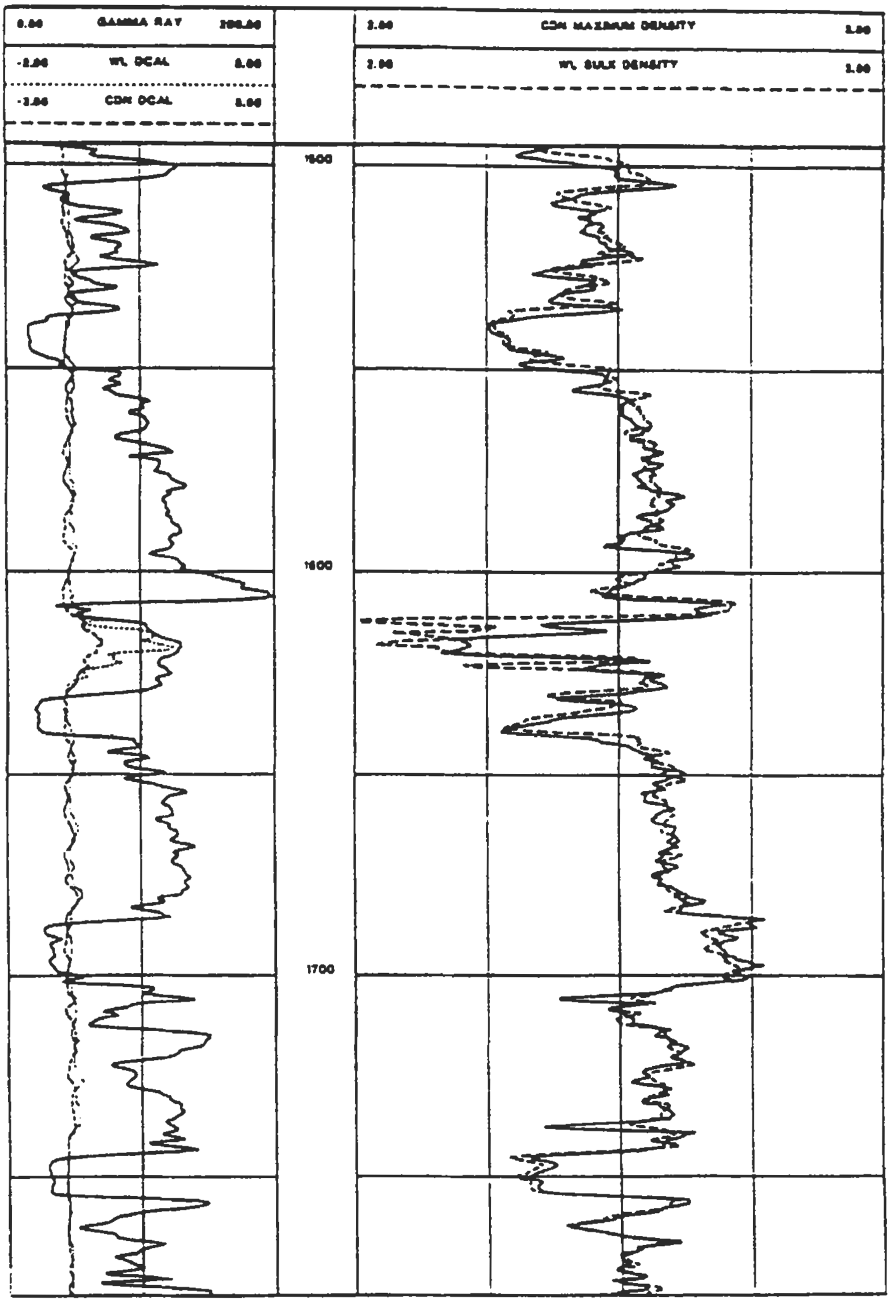


Figure 4-284. Density comparison in a deviated well. *(Courtesy Anadrill* [113].*)*

Table 4-129
Pe and Rock Matrix Densities for Various Lithologies

Lithologies	Pe	Rock Density (g/cm³)
Sandstone	1.81	2.65
Limestone	5.08	2.71
Dolomite	3.14	2.85
Illite	3.45	2.45–2.65
Kaolinite	1.83	2.35–2.50
Smectite	2.04	2.05–2.30

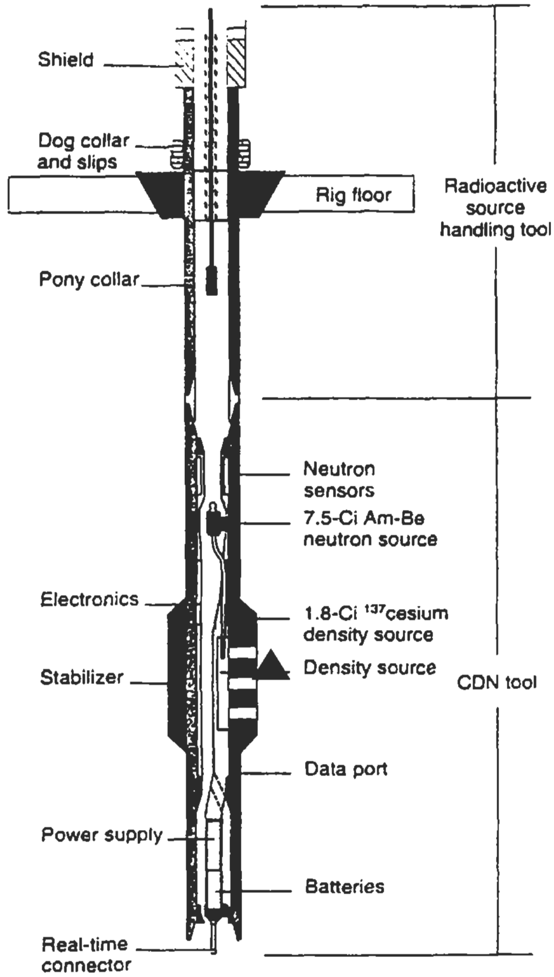


Figure 4-285. Radioactive sources being installed in the neutron-density sub. *(Courtesy Anadrill* [113].*)*

Example 16: Neutron-Density Logs Interpretation

The set of logs in Figure 4-286 have all been recorded while drilling in a Gulf Coast sand-shale sequence.

1. Give the boundaries of the clean sand. Are some zones shaly?
2. Give the probable hydrocarbon/water contact. Give the probable nature of the hydrocarbons and the gas/oil contact.
3. Compute the gas saturation at 9692 ft.
4. Can the oil saturation be computed at 9720 ft using the basic formula given in this chapter?

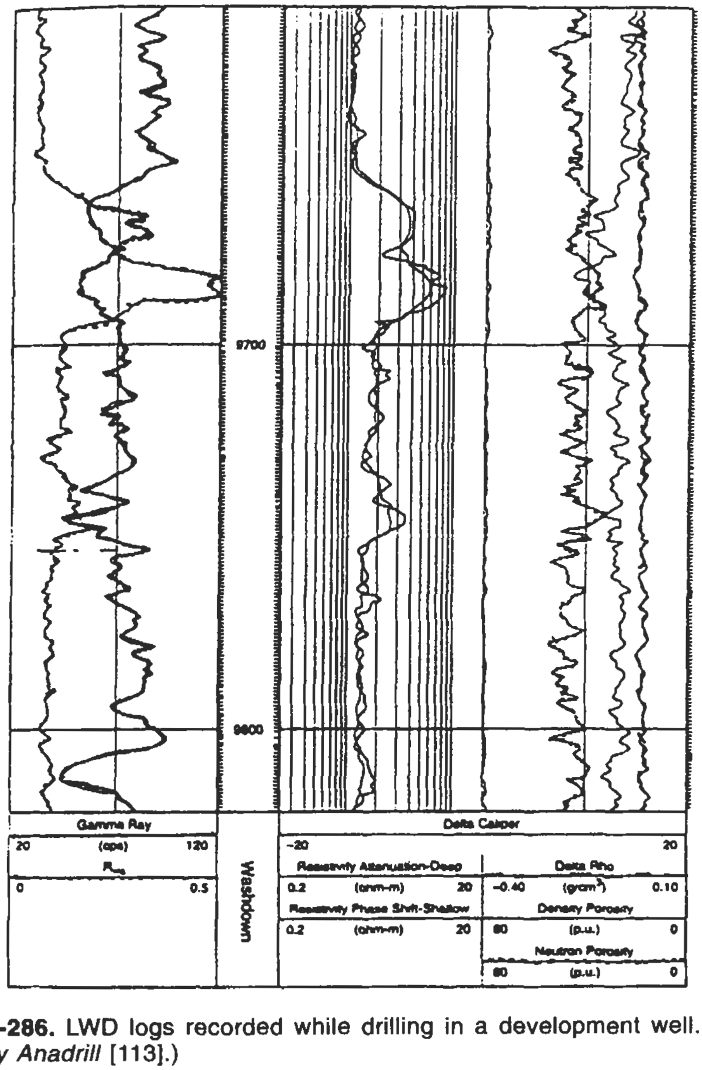


Figure 4-286. LWD logs recorded while drilling in a development well. (*Courtesy Anadrill* [113].)

Solution

1. Shaly sand. Cleanest parts: 9679 to 9696 and 9803 to 9808 ft.
2. Hydrocarbon/water contact at 9750 ft with R_{wa} curve. Oil to 9696 ft. Gas above with neutron density.
3. At 9692 ft, Φ = 30%, R_t = 8 $\Omega \bullet$ m, R_w = 0.1 $\Omega \bullet$ m, S_w = 3.3%, S_g = 6.7%.
4. No, shaly sand formula must be used.

Example 17: Neutron-Density Logs Interpretation

The Gulf Coast logs shown in Figure 4-287 have been run in the same interval with MWD/LWD sondes and wireline sondes.

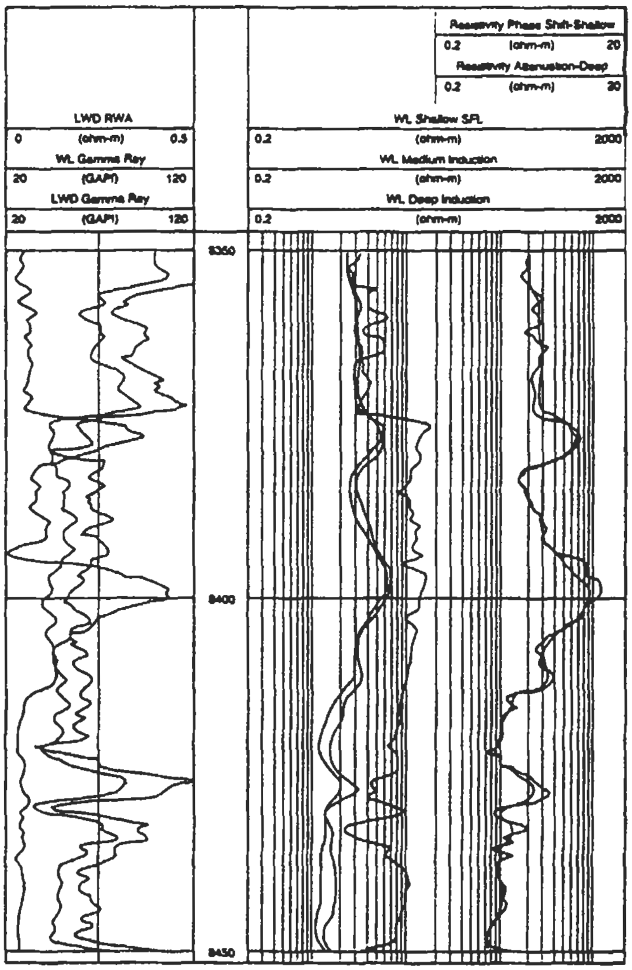


Figure 4-287. Wireline and LWD logs showing the effect of invasion. (*Courtesy Anadrill* [113].)

1. Draw the lithology description in the depth column.
2. Looking at the R_{wa} curve, where is the hydrocarbon/water contact?
3. Do we have enough information to know if we have oil or gas?
4. What is the invasion diameter at 8397 ft using the wireline logs? What is the invasion diameter using the MWD/LWD logs?
5. What is needed to compute the hydrocarbon saturation?

Solution

1. Shale-sand sequence:
   Top to 8,374 ft shale
   8,374 to 8,425 ft sand
   8,425 to 8,429 ft shale break
   8,429 to bottom sand
2. Hydrocarbon/water contact: 8,413 ft.
3. No, we need the neutron and density curves.
4. According to chart in Figure 4-304, d_i = 40 in. with the wireline logs. The d_i cannot be calculated with the MWD/LWD logs since we have only two resistivity curves.
5. We need the porosity. R_w is given by R_{wa} in the lower sand. We also need R_t.

Ultrasonic Caliper and Sonic Log while Drilling

In the ultrasonic caliper sub, two ultrasonic sensors are mounted 180° apart on stabilizer blades as shown in Figure 4-288.

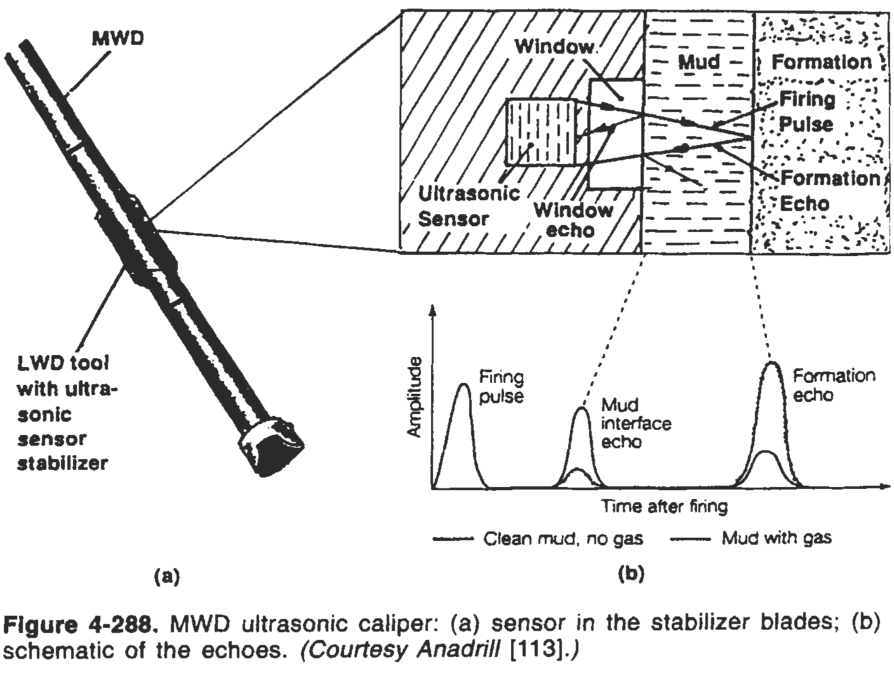


Figure 4-288. MWD ultrasonic caliper: (a) sensor in the stabilizer blades; (b) schematic of the echoes. *(Courtesy Anadrill* [113].*)*

The sensors function in a pulse-echo mode that allows the direct measurement of stand-off, from which short and long axes of the borehole diameter are computed. The vertical resolution is 1 in. (25 mm) and accuracy of the diameter measurement is ±0.1 in. (2.5 mm).

The caliper is used to correct the density and neutron porosity measurements for borehole effects and also can be used as a borehole stability indicator. Figure 4-289 shows an example of comparison between the MWD ultrasonic caliper and the four-arm wireline caliper run five days later.

The MWD caliper sub can also be used for downhole detection of free gas in the annulus (gas bubbles, not dissolved gas) through a combination of formation and "faceplate" echo signals. In Figure 4-288b a schematic of the system is represented. The faceplate is the interface between the window and the mud. The faceplate echo signal is the echo due to the impedance mismatch between the window and the mud. This echo is affected by the gas content of the mud, with echo amplitude increasing with the gas content. It can be seen in Figure 4-288b that concurrently the formation echo decreases.

The smallest amount of gas detectable is about 3% of free gas in volume. Real-time transmission of this information can shorten the time needed to detect gas influxes while drilling. It can help and simplify the kill operations.

Figure 4-290 shows an example of drilling in underbalance conditions. Gas influxes are very well outlined.

MWD Sonic. A new LWD tool developed by Anadrill provides sonic compressional Δt measurements in real time and recorded modes. The tool operates on the same general principles as modern wireline sonic tools. As the drilling operation progresses, the transmitter is actuated and acoustic waves propagating through the mud and formation are detected by the receiver array. Using a downhole processing algorithm, the compressional Δt of the formation is extracted from the waveforms and transmitted uphole in real time via mud telemetry. The compressional Δt and porosity logs are generated, providing an input for lithology identification and overpressure determination.

A successful sonic-while-drilling tool must overcome four major problems:

- suppressing collar arrivals
- transmitter and receiver mounting on drill collars
- interference of drilling noise
- processing sonic waveforms downhole

A diagram of the tool is shown in Figure 4-291. The array length and the use of four receivers give a good compromise for compatibility with wireline measurements and spatial aliasing properties. The choice of four receivers also minimizes memory and power requirements, which are both proportional to the number of receivers. The separation between transmitter and receivers is a compromise between a long distance for good signal amplitude and minimum tool cost. This distance is also similar to that used in wireline array tools. The receivers are small, wideband piezoceramic stacks, which have responses similar to wireline receivers.

The battery-powered electronics acquire and store sonic waveforms. Under the control of a downhole microprocessor, the transmitter is fired, and four receiver waveforms are simultaneously digitized at 12 bits and added to a signal stack. The transmitter firing is done in bursts at the rate of 10 Hz, which allows minimum movement while stacking.

The data acquisition rate is generally set so that the sample spacing of the sonic log (the distance between two acquired data points) ranges from 6 in. to 1 ft based on the anticipated drilling rate of penetration (ROP).

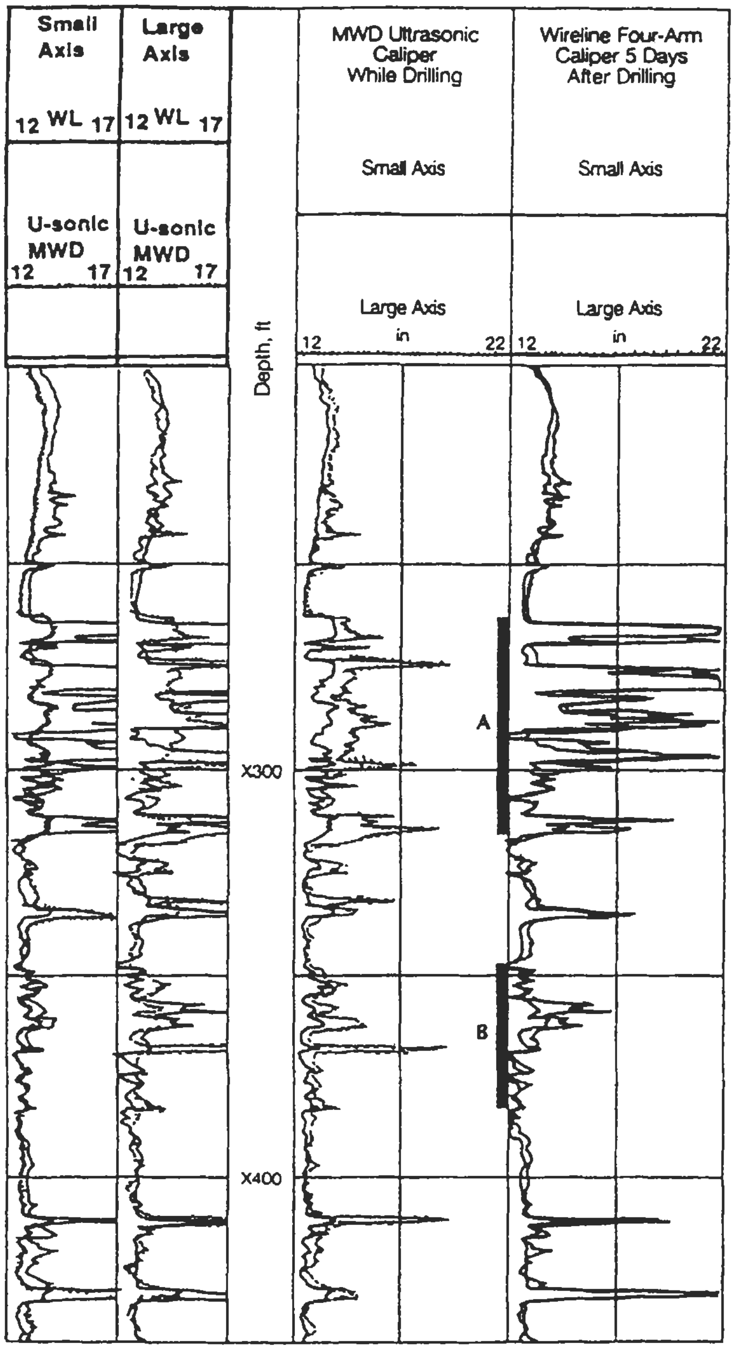


Figure 4-289. Comparison of MWD ultrasonic caliper logs with the 4-arm wireline caliper logs run five days later. *(Courtesy Anadrill* [113].*)*

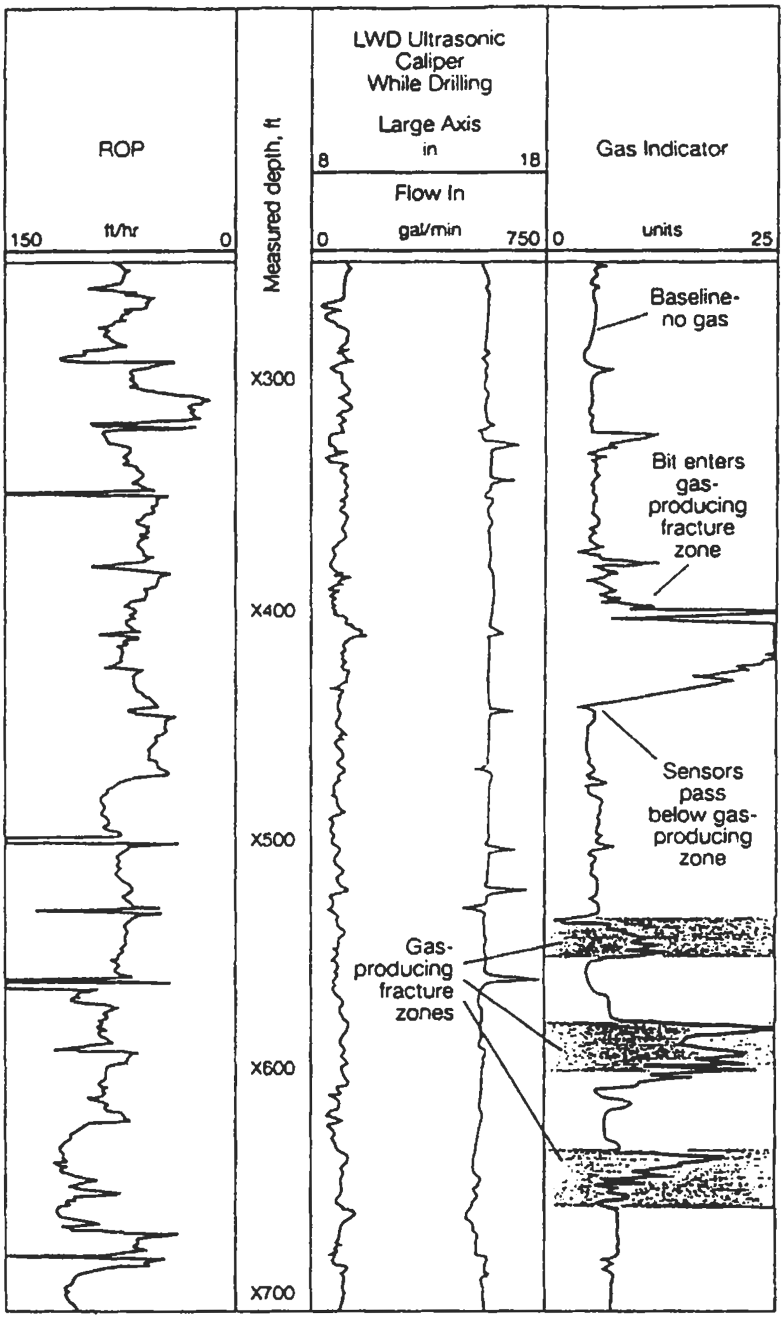


Figure 4-290. MWD ultrasonic caliper and gas influx log. *(Courtesy Anadrill [113].)*

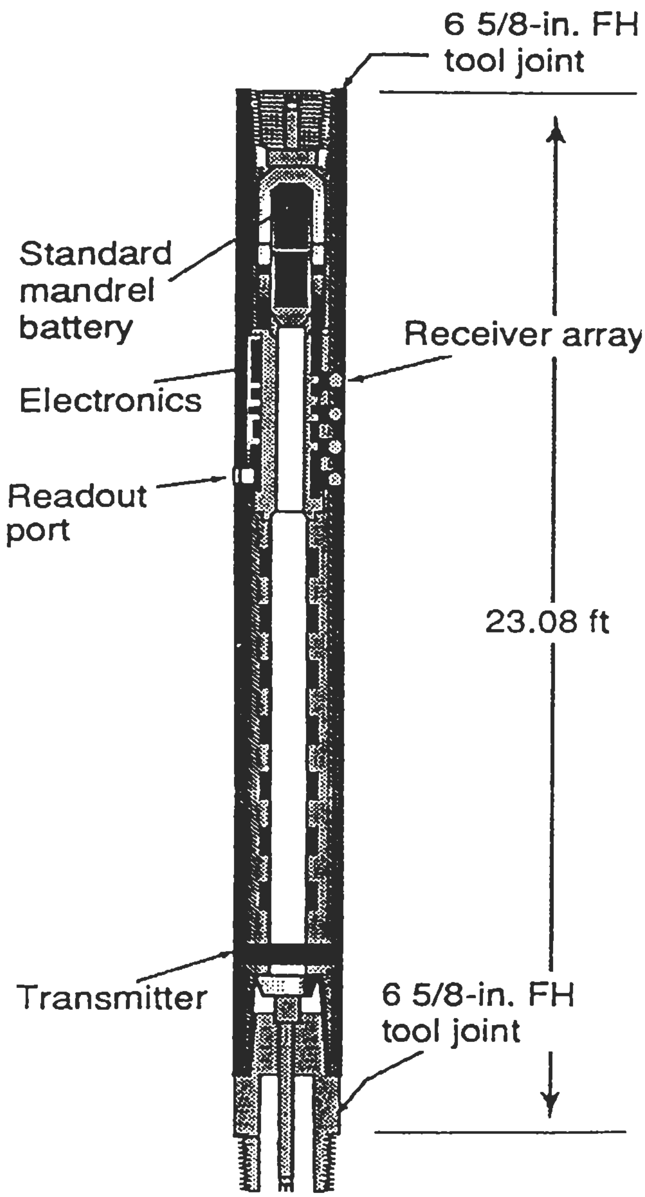


Figure 4-291. Schematic of the LWD sonic tool. *(Courtesy SPWLA* [116].*)*

For LWD/wireline sonic log comparison, we highlight the zone from 600 to 1000 ft in Figure 4-292. However, the two sonic measurements in the 840–860-ft interval show a major disagreement. This disagreement probably results from the deteriorated hole condition (two large washouts shown on the caliper logs) when the wireline logs were acquired (10 days after drilling).

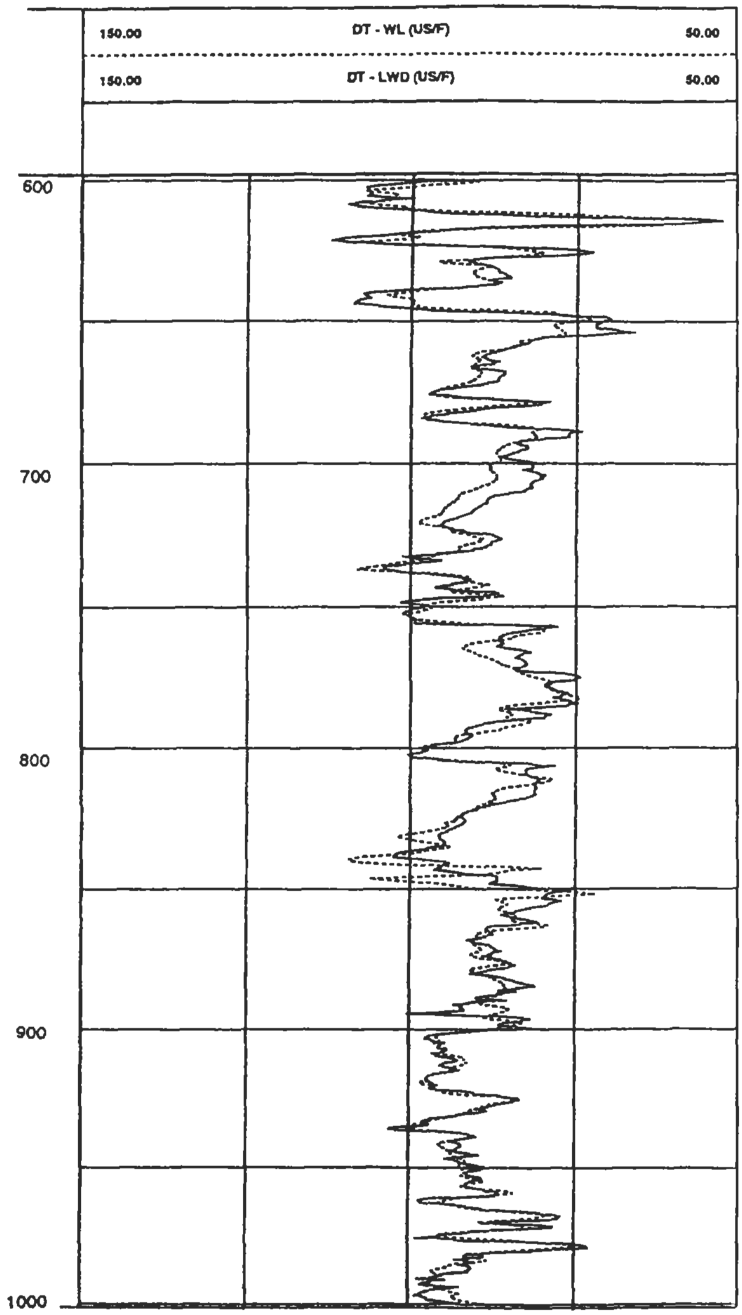


Figure 4-292. Comparison of a wireline sonic log with an LWD sonic log. *(Courtesy SPWLA* [116].*)*

Of particular interest in Figure 4-292 is the shaly sandstone in the 690–720-ft interval. In this zone, the LWD sonic measurements are consistently faster than the wireline measurements. Since the wireline logs were acquired 10 days after drilling, it is likely that shale swelling in the shaly sandstone has taken place. This phenomenon, known as formation alteration, causes the wireline sonic measurements to be slower. In this type of zone, LWD sonic yields a more correct Δt, which will better match surface seismic sections.

Measuring while Tripping: Wiper Logs

When the MWD systems are battery powered and have a downhole recording capability or use an electromagnetic telemetry, logging measurements can be repeated each time the bit is pulled out or run into the borehole. This new capability provides a way to map the progression of the filtrate front in the permeable formations.

Downhole Recording. When the logging measurements are battery powered the logging parameters can be recorded versus time while tripping the drill string. If the depth is simultaneously recorded versus time, the data can be plotted versus depth. Common memory capabilities are of the order of 2 to 10 megabytes. The recording rate is adjusted to obtain about two data sets per foot.

Electromagnetic Telemetry. The electromagnetic telemetry is usually powered downhole with batteries. Parameters such as gamma ray, resistivity and temperature, can be transmitted while tripping up or down. Since a two-way communication is possible, the system can be switched to a "logging only" mode to transmit only the logging information.

Invasion Diameter Versus Time. Many parameters determine the invasion diameter:

- formation porosity
- formation permeability
- mudcake permeability
- mudcake thickness
- differential pressure
- mud filtrate and formation fluid viscosity

Figure 4-293 shows two typical cases for a 1-µd, 0.25-in. (6-mm) mudcake, 500 psi (3450 kPa) differential pressure, 20% porosity (a), and 30% porosity (b).

The factor permeability is important only for the low permeabilities, below 1 md. The invasion diameter increases rapidly in the first few days, making the measurements during tripping particularly significant.

Example 18: Example of Wiper Logs

Figure 4-294 shows a set of resistivity logs run in a sand–shale sequence of the Gulf Coast. We have one wireline dual induction log, one MWD resistivity log, a wiper-MWD resistivity log and one gamma ray log.

1. Describe the lithology of this zone.
2. How do we know that the cleaner zones marked A and B are permeable?
3. Compare the invasion for the various logs.
4. What is the true resistivity of Zone A and Zone B?

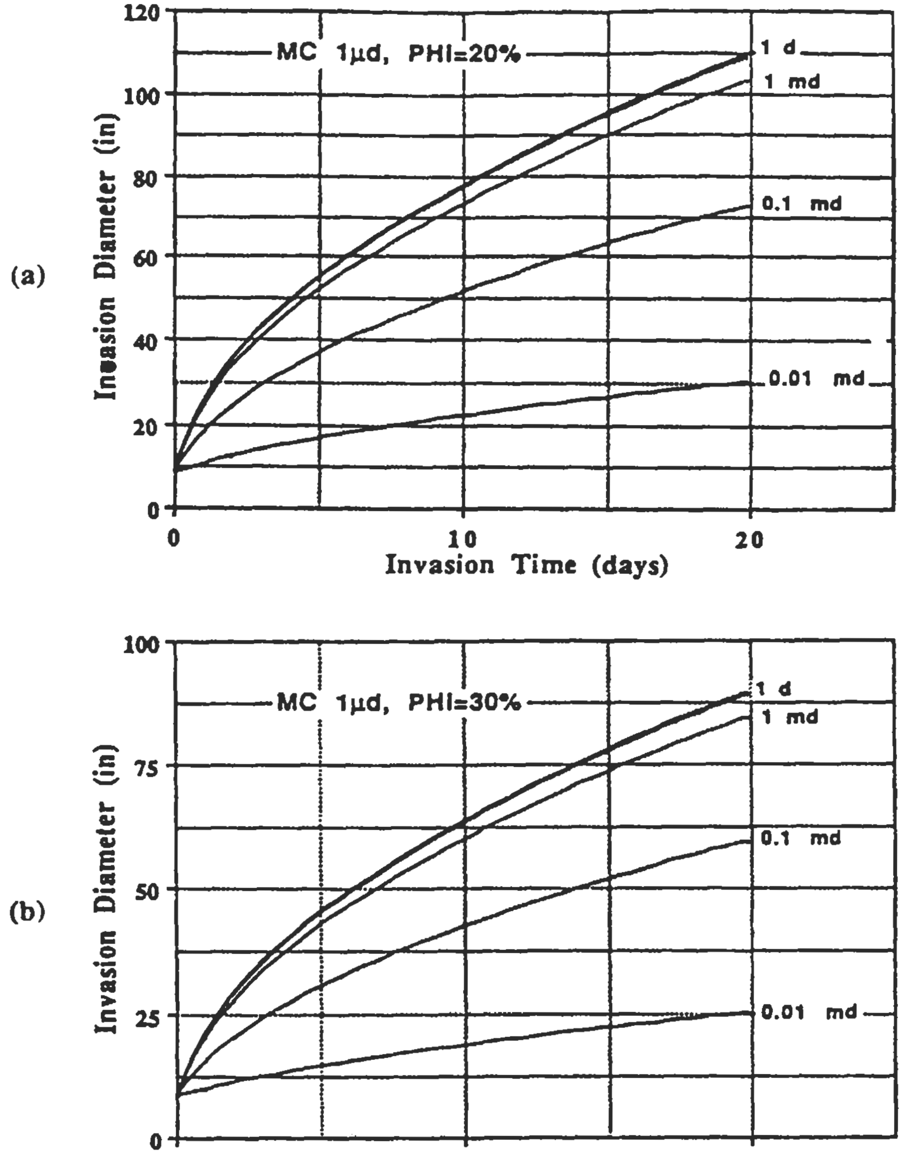


Figure 4-293. Invasion diameter versus invasion time for various formation permeabilities. (a) Filtrate invaded porosity: 20%; mudcake permeability: 1 μd; mudcake thickness: 0.25"; differential pressure: 500 psi. (b) Filtrate invaded porosity: 30%; mudcake permeability: 1 μd; mudcake thickness: 0.25"; differential pressure: 500 psi. *(Courtesy Louisiana State University* [99].*)*

Solution:

1. Laminated shaly zone; two fairly clean intervals marked A and B.
2. Invasion causes curve separation for wiper and wireline logs.
3. No invasion in the MWD/LWD log. In the wireline log, using the chart in Figure 4-304.
   Sand A: $d_i = 32$ in.
   Sand B: $d_i = 30$ in.
   Using the wiper log, we have invasion but it can not be determined quantitatively.

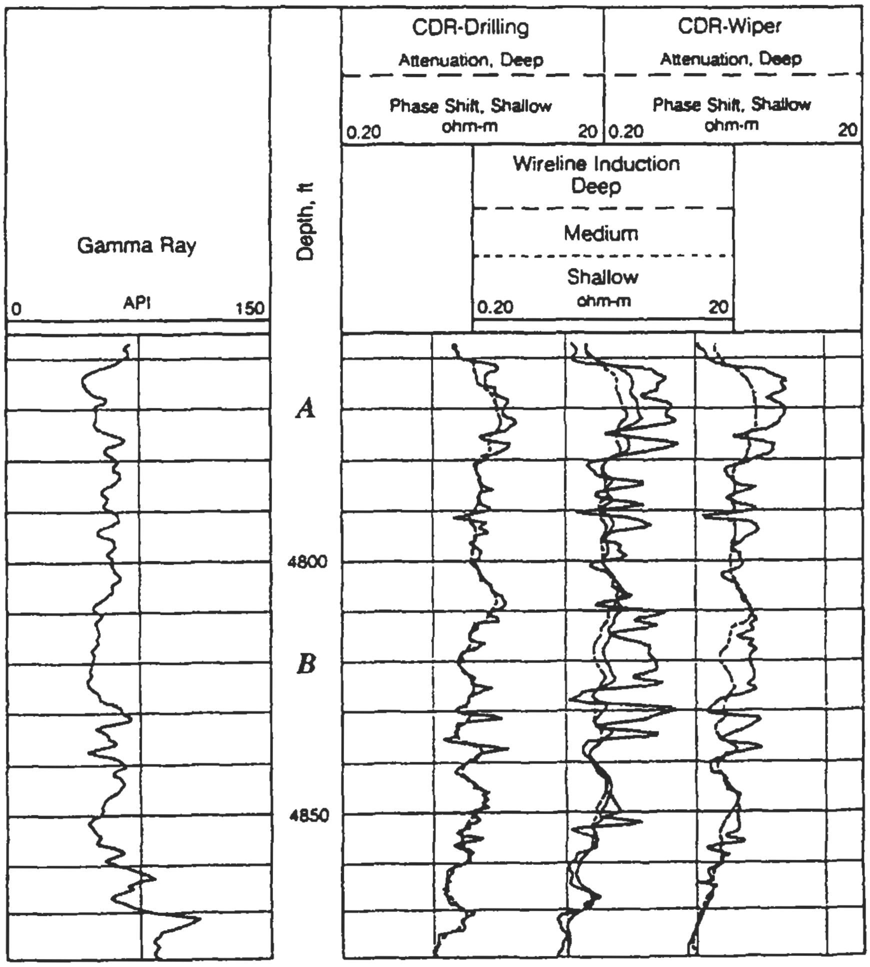


Figure 4-294. Comparison of compensated dual-resistivity logs run while drilling and in a wiper pass with the wireline induction logs. *(Courtesy Anadrill* [113].*)*

4. In the wireline log (R_{ld}),
   Sand A: R_t = 3.0 Ω • m
   Sand B: R_t = 1.8 Ω • m
   Using the wiper log (R_{att})
   Sand A: R_t = 3.0 Ω • m
   Sand B: R_t = 1.5 Ω • m

Example 19: Example of Wiper Logs

Figure 4-295 shows a set of MWD/LWD logs recorded in a Gulf Coast well.

1. What are the boundaries of the main sand? What curve is used for lithology?
2. Is there a hydrocarbon/water contact? Where? Which curve(s) will tell us?
3. Is the upper part gas or oil saturated?
4. Determine the porosity at 400 ft.
5. Determine the hydrocarbon saturation at 400 ft.

Solution:

1. Top of sand at 383 ft. Bottom of sand at 486 ft.
2. Gamma ray indicates shales and clean formations.
3. Yes, hydrocarbon/water contact at 410 ft. Resistivity and R_{wa}.
4. With chart in Figure 4-303, Φ = 28%. Slightly above sandstone line, probably gas.
5. Saturations: R_{wa} = 0.01; R_t = 6 Ω • m; Φ = 28%; S_w = 12%; S_g = 88%.

Measurements at the Bit

A typical MWD bottomhole assembly used in rotary drilling is as follows from bottom to top: drill bit, stabilizer, resistivity, WOB torque, directional and telemetry system, neutron-density Pe. The typical distances are seen in Figure 4-296a. When a mud motor is inserted between the lower stabilizer and the drill bit, the distances are increased as shown in Figure 4-296b.

The time elapsed between drilling and recording the data at a given depth can be read in Table 4-131 for various rates of penetration.

To meet the challenges posed by horizontal drilling in particular, a system has been developed to make the measurements at or near the bit and transmit them to the mud telemetry section of the MWD bottomhole assembly. The resistivity at the bit tool is similar to the toroidal resistivity tool described in the section titled "Resistivity Logs." As shown in Figure 4-297, the Anadrill geosteering package includes below the Power Pack mud motor:

- a surface adjustable bent housing
- a sub for measurement of the bit resistivity, azimuthal gamma ray, inclination, and bit rpm (the same sub contains an electromagnetic transmission system that sends the data to the mud telemetry above the mud motor)
- a $\frac{3}{4}$° fixed bent-housing section equipped with the bit resistivity toroid and the azimuthal resistivity button
- a stabilizer and bearing section, just above the drill bit

Figure 4-298a shows the sketch of principle of the resistivity measurement in water-base mud.

The drill bit resistivity is measured below toroid T_2. An average resistivity is measured between toroid T_1 and T_2. The azimuthal resistivity is measured with

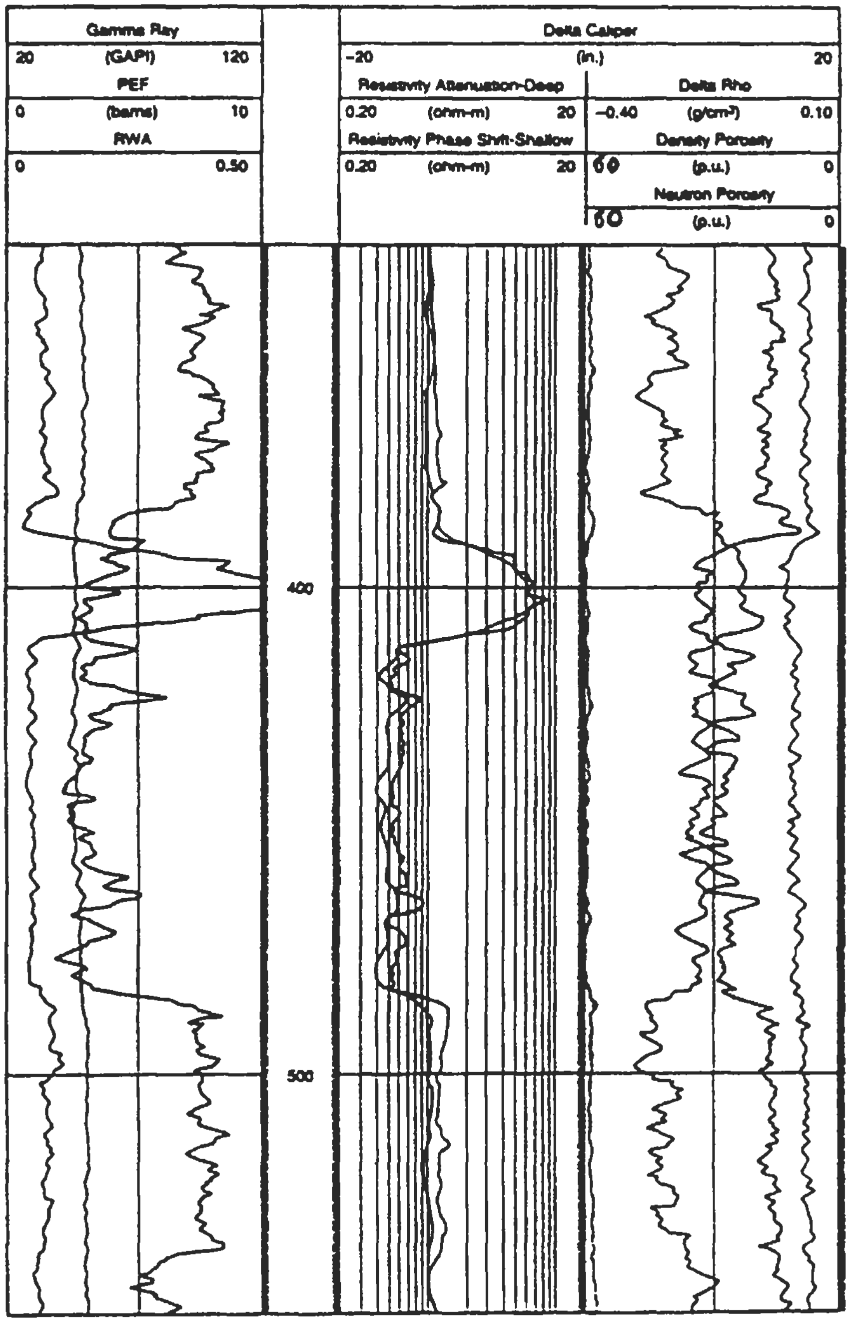


Figure 4-295. LWD resistivity logs recorded during a wiper pass. *(Courtesy Anadrill* [113].*)*

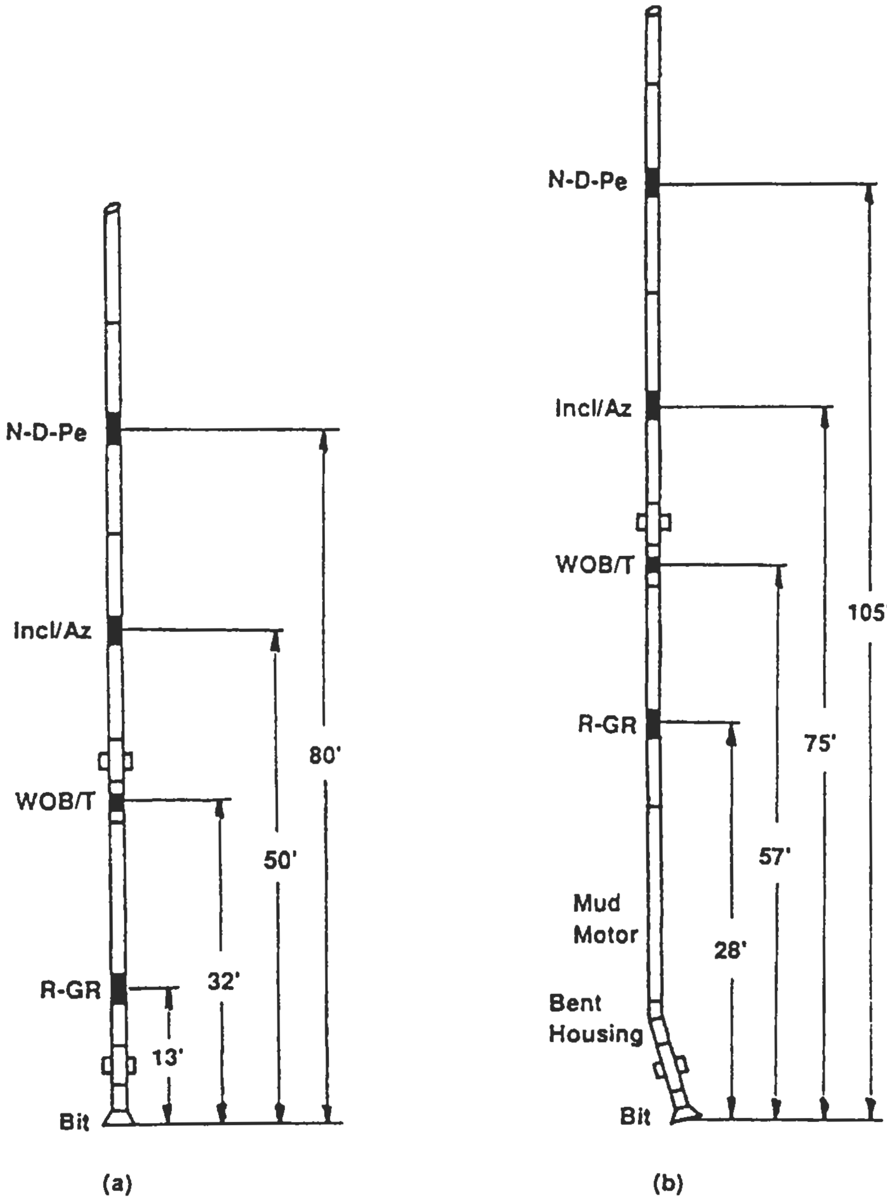


Figure 4-296. Typical MWD bottomhole assemblies: (a) rotary drilling; (b) mud motor drilling.

button R. In normal buildup drilling, the azimuthal resistivity measures the high side of the borehole and the gamma ray window sees the low side of the borehole. Measurements in other orientations are made by rotating the bottomhole assembly. In the geosteering configuration, the resolution of resistivity at

Table 4–130
"Time Since Drilling" for Various MWD/LWD Logs with Typical Spacings

(a) Rotary Drilling

ROP (ft/hr)	R/GR (hr/min)	WOB/T (hr/min)	INCL/AZ (hr/min)	ND Pe (hr/min)
1	13–0	32–0	50–0	80–0
5	2–36	6–24	10–0	16–0
10	1–18	3–12	5–0	8–0
20	0–39	1–36	2–30	4–0
50	0–15	0–38	1–0	1–36
100	0–8	0–19	0–30	0–48
	13	32	50	80

(b) Mud Motor Drilling

ROP (ft/hr)	R/GR (hr/min)	WOB/T (hr/min)	INCL/AZ (hr/min)	ND Pe (hr/min)
1	28–0	57–0	75–0	105–0
5	5–36	11–24	15–0	21–0
10	2–48	5–42	7–30	10–30
20	1–24	2–51	3–45	5–15
50	0–34	1–8	1–30	2–6
100	0–17	0–34	0–45	1–3
	28	57	75	105

the bit is 6 ft (1.8 m). When drilling in rotary, the resistivity at the bit (RAB) is also measured with a similar toroidal system as shown in Figure 4-298b. The RAB tool has three button electrodes spaced to give azimuthal resistivities with three depths of investigation: 3, 6 and 9 in. (7.6, 15 and 23 cm). A ring electrode is also provided to yield an axial focused resistivity 5 ft above the drill bit. Focusing is improved by a near-bit transmitter. The radius of investigation of the ring electrode is about 12 in. (30 cm) and the vertical resolution about 2 in. (5 cm). Figure 4-299 is a sketch of the ring electrode measurement. The button electrode measurement is similar with a button instead of the ring.

Both rotary and mud motor systems use an electromagnetic wireline telemetry to relay the data from the near-bit sub to the mud telemetry sub.

Basic Log Interpretation

The log interpretation using logging while drilling logs is very similar to the interpretation made with wireline logs. One major difference is that the invasion is usually less important due to the short time elapsed between drilling and logging.

Lithology. Gamma ray is used to differentiate between shales and clean formations. Pe is used to determine the nature of the clean formations: sandstone, limestone, or dolomite.

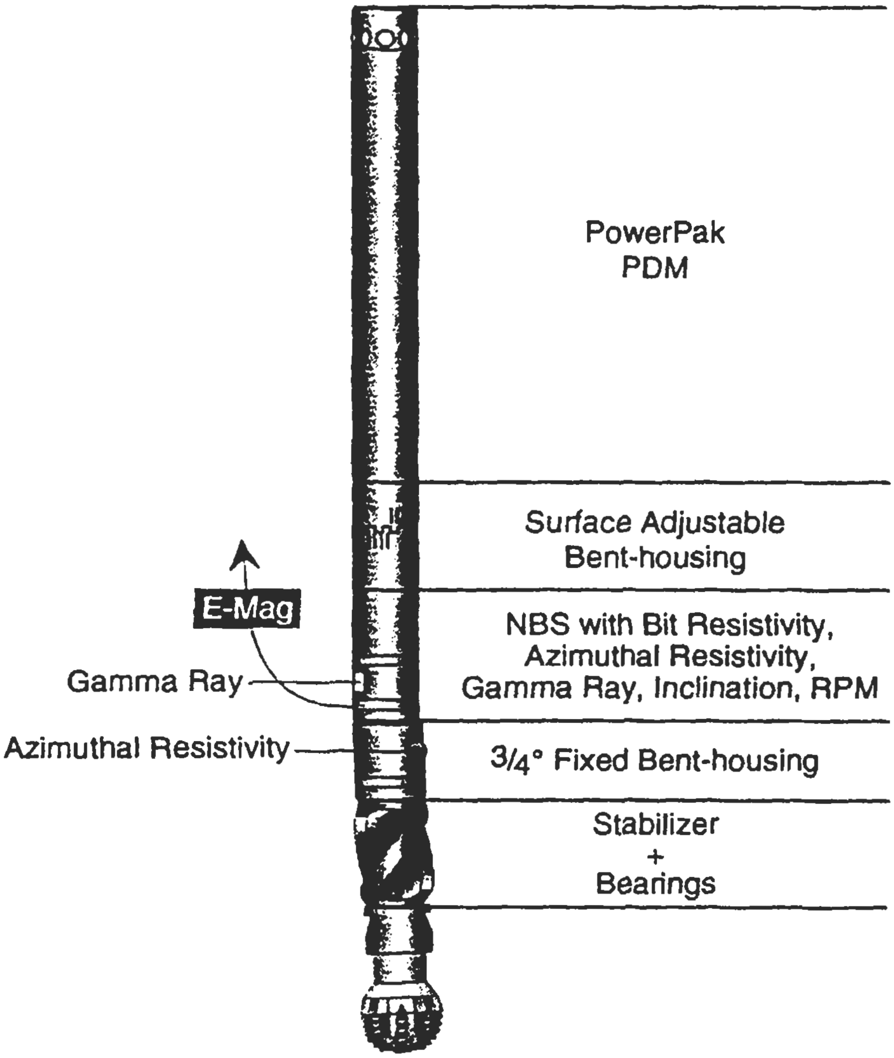


Figure 4-297. Anadrill geosteering tool. *(Courtesy Anadrill* [113].*)*

Porosity. Two porosity logs are run today (1994); namely, neutron and density. Soon the sonic will also be available.

With the lithology matching the log scale, and assuming the formation fully invaded by mud filtrate, a neutron porosity and a density porosity can be determined.

- If the two porosities match, the formation is saturated with a liquid, water or oil, and the porosities are true porosities.
- If the neutron porosity is low and the density porosity is high, the formation contains gas and the true porosity can be determined with charts.

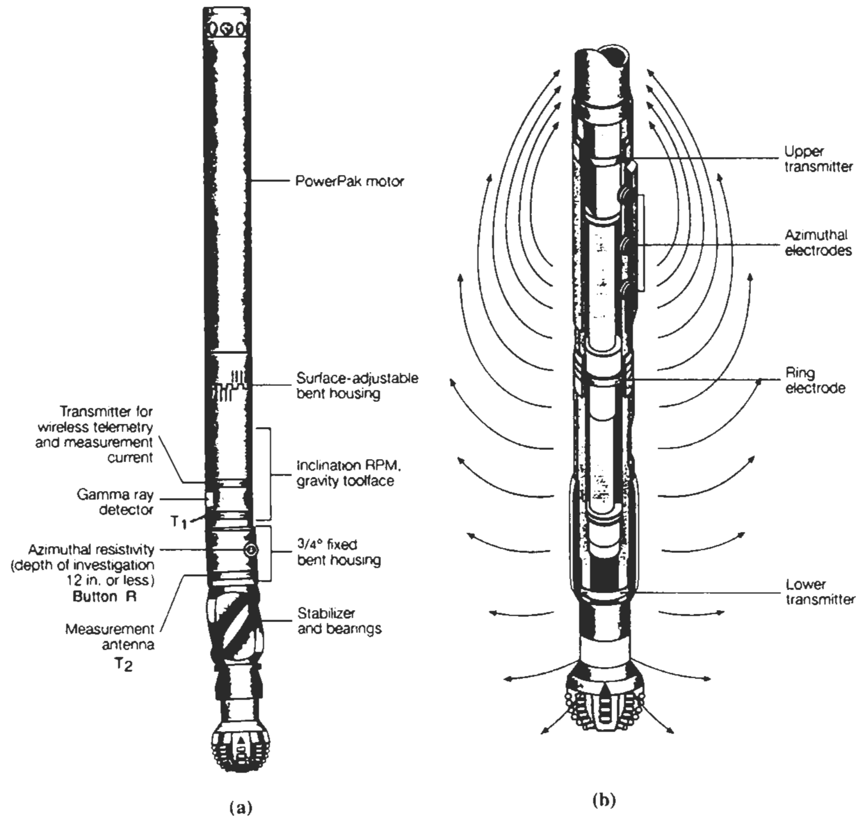


Figure 4-298. Measurement at the drill bit. *(Courtesy Anadrill* [113].*)*

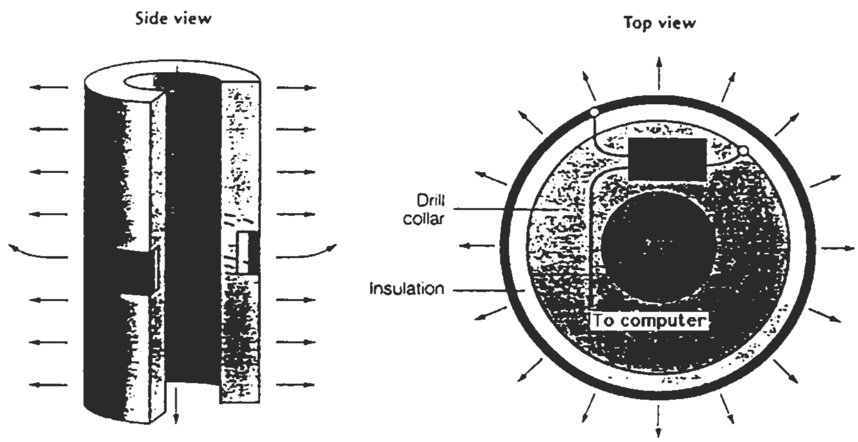


Figure 4-299. Principle of the ring resistivity measurement in the resistivity-at-the-bit tool. *(Courtesy Anadrill* [113].*)*

Saturation. Resistivity logs are used for saturation estimates. One, two or more resistivity logs with different investigation depths are usually available. The true resistivity of the formation can be estimated. If a water sand has been drilled near the interval of interest, the formation water R_w can be determined by

$$R_w = \frac{R_0}{F} \tag{4-206}$$

where R_0 = true resistivity of the water sand
F = formation factor

and

$$F = \frac{1}{\Phi^2} \tag{4-207}$$

or

$$F = \frac{0.62}{\Phi^{2.15}} \tag{4-208}$$

where Φ = porosity in fraction

Equation 4-207 is used in carbonates (limestone and dolomite) and Equation 4-208 is used in unconsolidated to medium consolidated sandstones. A third Equation 4-209 can be used in highly consolidated sandstones. This is

$$F = \frac{0.81}{\Phi^2} \tag{4-209}$$

If no water sand exists, then an estimate of R_w must be made based on regional knowledge.

Since the true porosity has been determined, formation water saturation can be determined by

$$S_w = \sqrt{\frac{F \bullet R_w}{R_t}} \tag{4-210}$$

where S_w = formation water saturation
F = formation factor
R_w = formation water resistivity
R_t = true resistivity of the formation

The oil or gas saturation is

$$S_{hc} = 1 - S_w \tag{4-211}$$

where S_{hc} = hydrocarbon saturation

The volume of hydrocarbons in place at reservoir conditions is

$$Q_{IP} = 7758 \bullet \Phi \bullet S_{hc} \bullet h \tag{4-212}$$

where Q_{IP} = volume of hydrocarbons in bbl/acre
Φ = porosity in fraction
S_{hc} = hydrocarbon saturation in fraction
h = formation thickness in ft

Apparent Water Resistivity R_{wa}. Some MWD/LWD log sets display a curve labeled R_{wa}. R_{wa} is computed using Equation 4-210 assuming that $S_w = 1$ (100%). Consequently we have

$$R_w = R_{wa} = \frac{R_t}{F} \tag{4-213}$$

Since

$$F = \frac{a}{\Phi^m} \tag{4-214}$$

where F = formation factor
a = constant depending on the formation, generally 1 or 0.81 or 0.62
m = cementation factor, generally 2 or 2.15

Finally,

$$R_{wa} = \frac{R_t \bullet \Phi^m}{a} \tag{4-215}$$

The true porosity Φ is determined with the neutron-density Pe logs. R_t is generally given by the deep investigation resistivity curve. R_{wa} equals R_w in the water formations. It increases rapidly in hydrocarbon saturated formations.

Permeability. Permeable zones can be identified with the resistivity measurements made with different radius of investigation. A departure between the curves of deep and shallow investigation is a qualitative indication of permeability.

The charts mentioned in the section titled "Measuring While Tripping: Wiper Logs" can be used to estimate quantitatively the permeability if several measurements during tripping are made with resistivity devices that can give the invasion diameter.

Surface measurements on the mud can be used to estimate the mudcake characteristics. If the formation pressure is known, the differential pressure can be calculated, and a chart similar to Figures 4-293a and b can be plotted.

The invasion diameters at various times should follow one of the permeability curves. Note that the permeability effect is seen only for formations with 1 md or less permeability. Above 1 md, the invasion diameter is dependent mostly on porosity.

Log Samples. Figure 4-300 shows samples of gamma ray and spectral gamma ray logs. The boundaries of the clean (not shaly) zone can be seen very clearly:

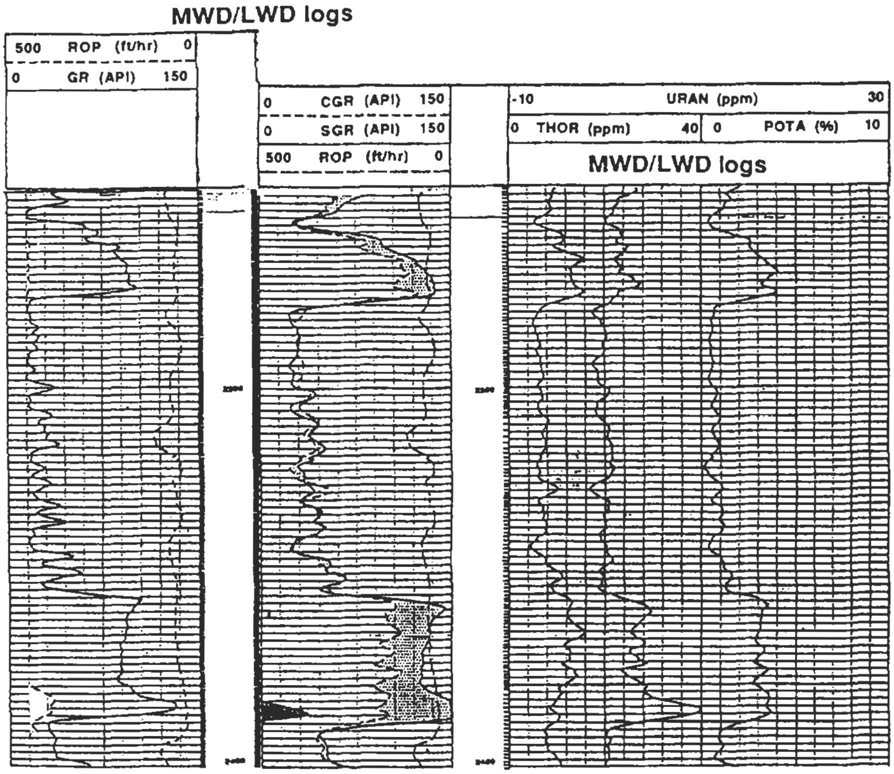


Figure 4-300. Sample of MWD/LWD logs.

2,256–2,274 ft	shale
2,274–2,356 ft	clean zone
2,356–2,388 ft	shale

Shale streaks are visible in most of the lower part of the clean section.

A high radioactivity peak can be seen at 2388 ft. Looking at the spectral log it can be seen that this peak is due to a high uranium content. Other radioactive elements' concentration is normal.

Figure 4-301 shows sample neutron-density Pe log over the same interval. Figure 4-302 shows sample MWD resistivity (left) and wireline dual induction (right) for the same interval.

A high neutron porosity and low density porosity occur in the shale zones: 2256–2274 and 2356–2388 ft. In the clean zone, around 2280–2290 ft, the cleanest part reads Φ_n = 18% and Φ_d = 13%. Plotting the point in the CNL chart of Figure 4-303, the rock matrix appears to be a dolomitized limestone and the true porosity is 17%.

Since the deep and shallow curve of the MWD log and the deep, shallow and guard (laterolog) of the wireline log shows a departure, the zone 2274–2356 ft is invaded, consequently permeable. Using the dual induction chart of Figure 4-304, we can plot the point at 2290 ft:

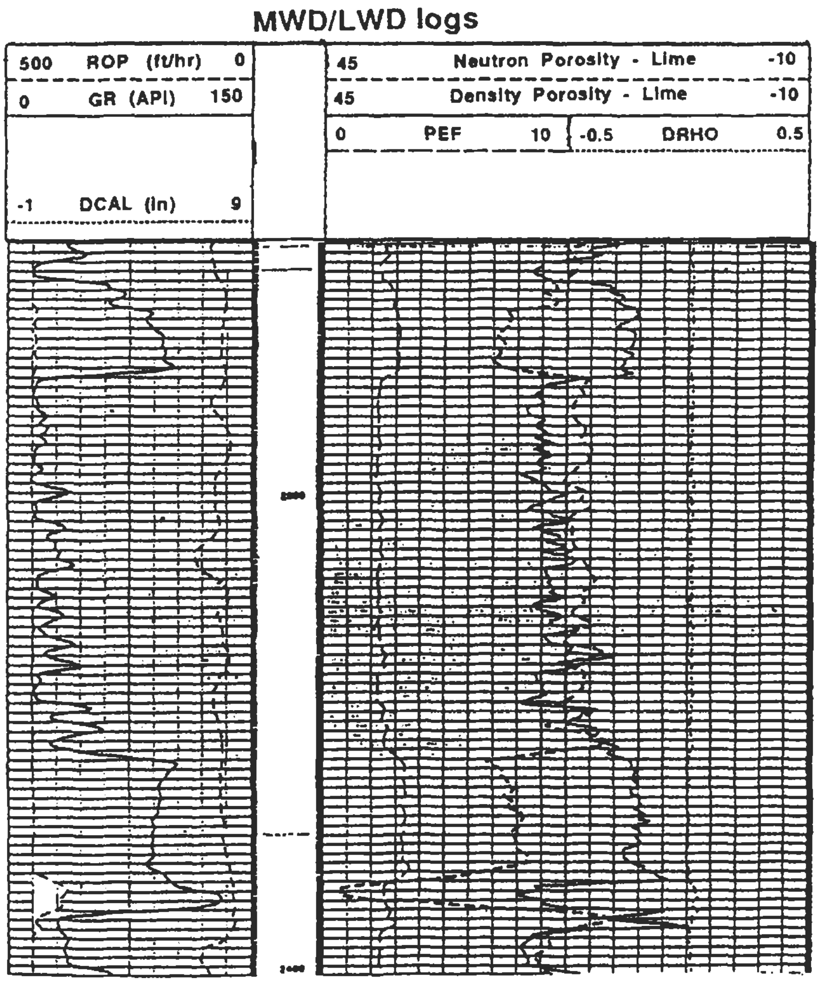


Figure 4-301. Sample of MWD/LWD logs.

$R_{IM}/R_{ID} = 1.2/1 = 1.2$

$R_{LL}/R_{ID} = 7/1 = 7$

The invasion diameter is 36 in.

where

$R_{x0}/R_t = 11$

$R_t/R_{ID} = 0.98$

(text continued on page 1014)

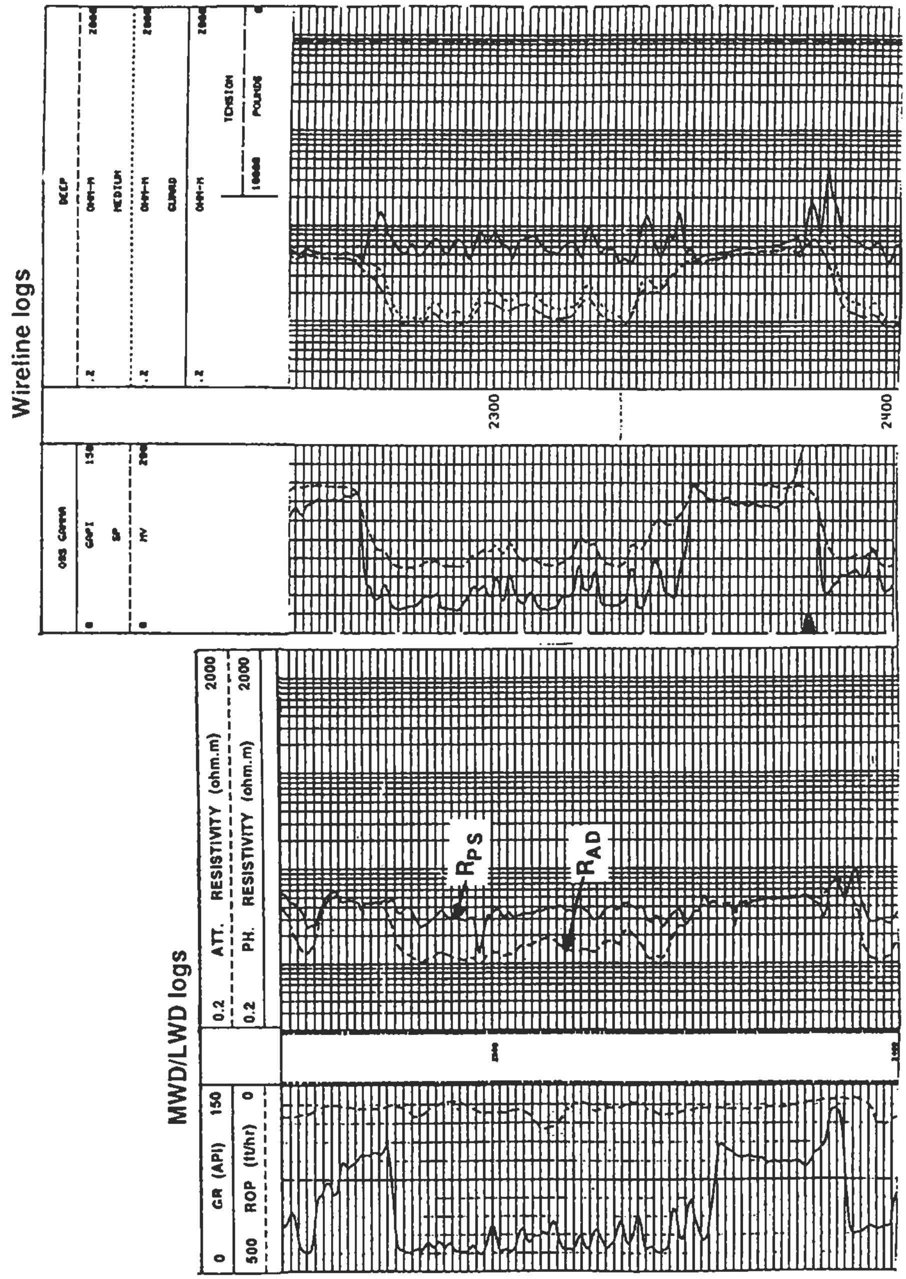


Figure 4-302. Sample of MWD/LWD logs.

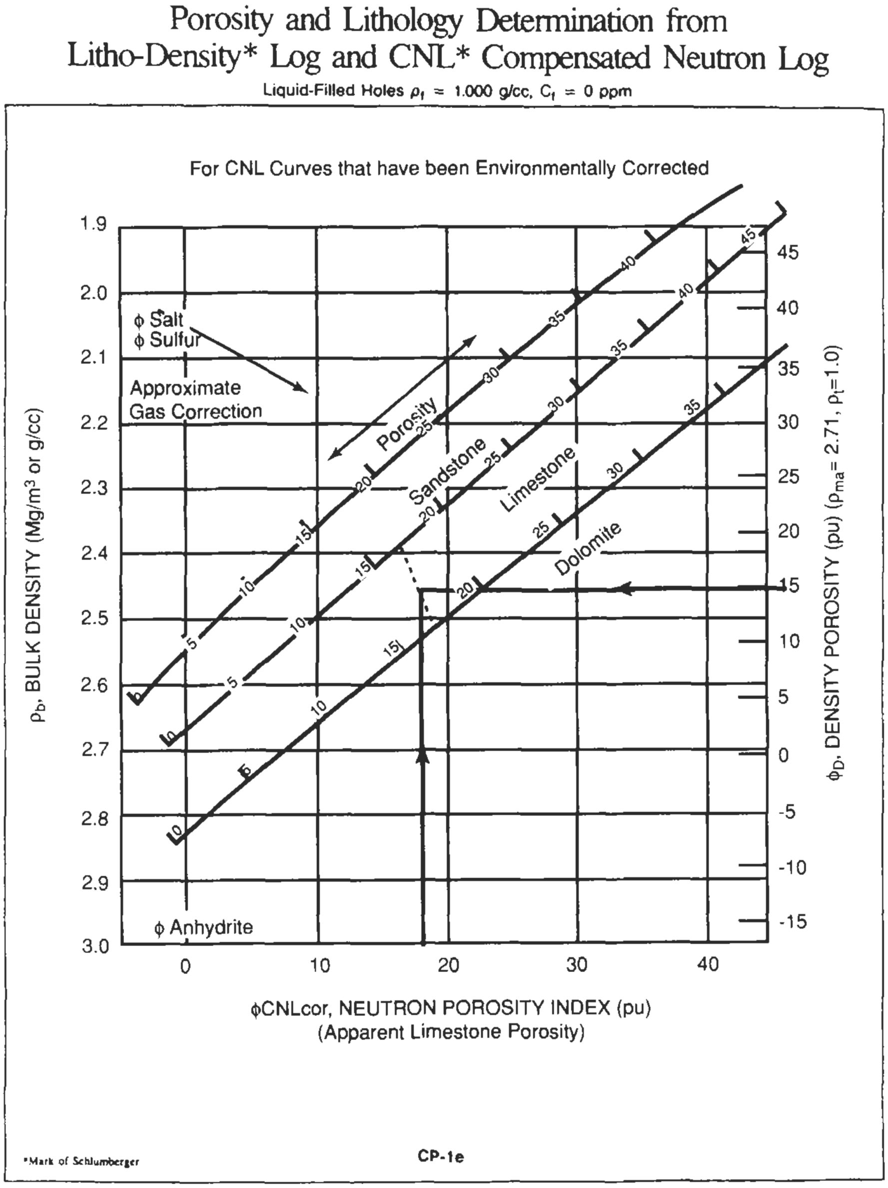


Figure 4-303. Porosity and lithology determination from the Litho-Density* logs and the CNL* Compensated neutron log. (*Courtesy Schlumberger.*) (*trade mark of Schlumberger.)

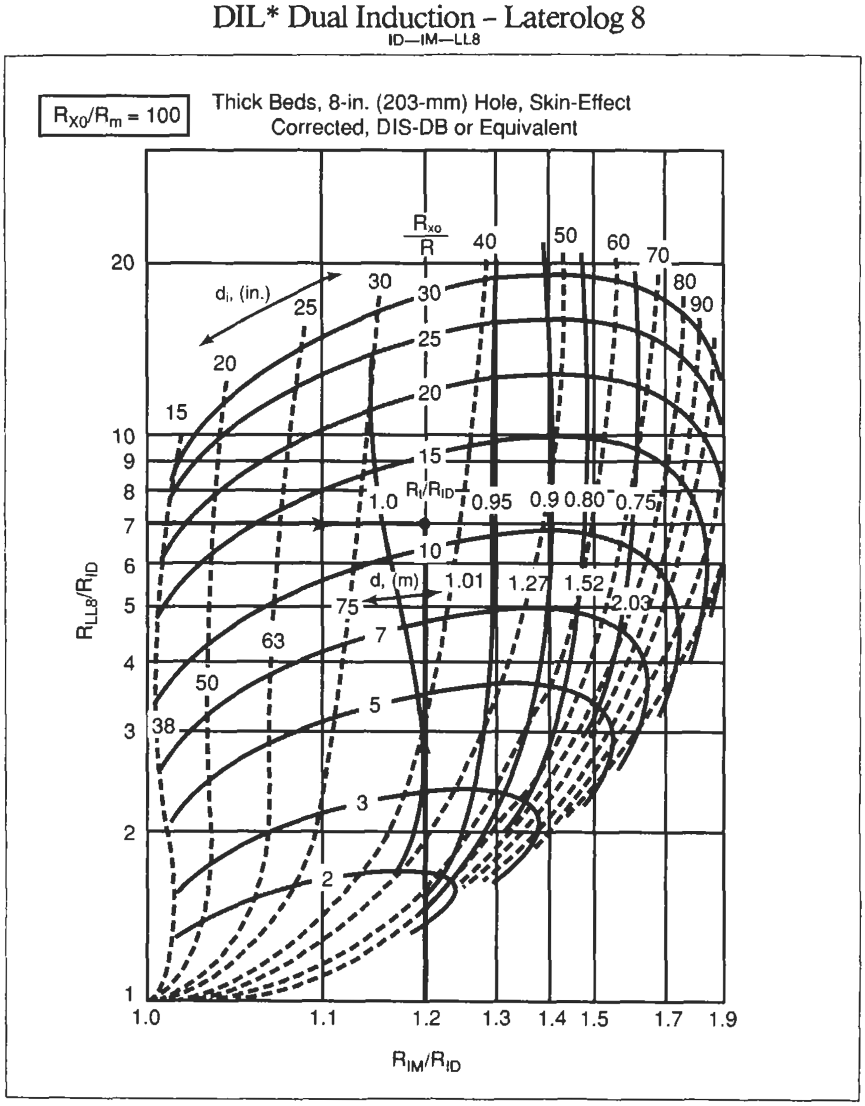


Figure 4-304. Resistivities and invasion determination with the Dual Induction log (DIL*) and the laterolog-8. (*Courtesy Schlumberger.*) (*trade mark of Schlumberger.)

(text continued from page 1011)

Consequently:

R_t = 0.98 Ω • m ≅ 1 Ω • m

R_{x0} = 11 Ω • m

The formation factor is

$$F = 1/\Phi^2 = 1/0.172 = 34.6$$

If we assume a 100% water saturation, the water resistivity is

$$R_w = R_t/F = 0.029\ \Omega \bullet m$$

If an oil or gas zone is present above this water sand, R_t should be computed the same way and S_w calculated.

MWD/LWD Applications

The progresses that have been made in the last few years in MWD and LWD have been described in the first part of this section. These progresses could not have been accomplished without a strong support from the oil industry. The support is motivated by the numerous, important and sometimes indispensable applications of the new technologies. In this chapter we will review drilling mechanics, properties determination, abnormal pressure detection, horizontal well steering and geosteering, and short radius well technology.

Drilling Mechanics

We will discuss four main topics: stuck pipe prevention, drillstring failure prevention, ROP optimization and real-time use of drilling mechanics.

Stuck Pipe Prevention. Drillpipes get stuck for many reasons.

Differential sticking occurs when the pipe is standing still and is pressed against the mud cake of a highly overbalanced formation.

The *formation* may cave into the wellbore if the pressure exceeds the hydrostatic mud pressure in shale.

The formation may be reactive and swell. It may be unconsolidated and collapse on the tool joints or drill collars. We may have "mobile" formations such as "gumbo" shales or salt beds in a plastic condition.

Key seats occur in dog legs of the borehole. The drill pipes cut a groove in the wall at their dimension and when tool joints or drill collars are pulled up, the drillstring gets stuck.

Wellbore geometry is another reason. In kick-off or well curvature, the drillstring has more flexibility when tripping in than when tripping out and the friction may increase to the point of getting stuck.

Undergage hole, due to the wearing out of the previous bit, may cause the new bit to get jammed and stuck.

Finally, *inadequate hole cleaning* results in an overloading of the annulus with cuttings, especially in very high penetration rate, poor mud properties, and insufficient annular velocity or circulation time. Inadequate hole cleaning can also be experienced in deviated wells with the formation of cutting beds on the low side migrating in a "sand dune" fashion.

The most important factor to avoid pipe sticking is the friction factor computed both when sliding and rotating. The rotating friction factor is usually called "friction" or FRIC; the sliding friction factor is called "drag" or DRAG.

Referring to Figure 4-305a and b, the buoyant weight of an element of pipe resting on a curved borehole exerts a side force that generates weight loss

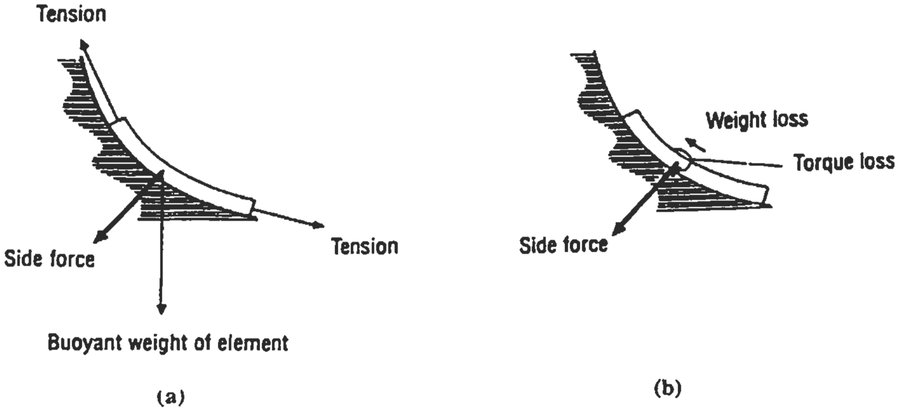


Figure 4-305. Forces acting on a drill pipe element: (a) side force; (b) weight and torque loss.

tripping in or over-pull tripping out. Also, a torque loss is generated by the side force during rotation.

For one element, the DRAG and FRIC are given by

$$\text{DRAG} = \frac{F_1 - F_2}{W_{pm} \bullet \sin i} \tag{4-216}$$

$$\text{FRIC} = \frac{T_1 - T_2}{W_{pm} \bullet r \bullet \sin i} \tag{4-217}$$

where F_1, F_2 = forces acting on element ends
T_1, T_2 = torque acting on element ends
W_{pm} = weight of the element in mud
r = radius of the drillpipe
i = average inclination of the element

The F_1, F_2, T_1, T_2 are not known for each element of the drillstring. Average friction factors are calculated for the whole string using the surface measurements of WOB and torque and the downhole measurements of WOB and torque. Consequently, DRAG and FRIC are calculated using the following equations:

$$\text{DRAG}(\%) = 100 \times \frac{\Delta\text{WOB}}{W_{pm} \bullet \sin i} \tag{4-218}$$

$$\text{FRIC}(\%) = 100 \times \frac{\Delta\text{TOR}}{W_{pm} \bullet r \bullet \sin i} \tag{4-219}$$

where ΔWOB = surface WOB minus downhole WOB
ΔTOR = surface torque minus downhole torque
W_{pm} = weight of drillstring in the mud

r = drillpipe radius
i = average inclination of the borehole

The sliding drag (DRAG) and rotary friction (FRIC) vary between 5 and 40%. According to Anadrill, the values listed in Table 4-131 are normally encountered in the field.

An example of a DRAG and FRIC log is given in Figure 4-306.

Figure 4-307 shows the raw data collected during drilling the interval shown in Figure 4-306.

By monitoring the weight loss between the MWD and the rotary table (surface weight - downhole weight) and the torque loss between the MWD tool and rotary table (surface torque - downhole torque), one can effectively look at the sticking forces on the drill string separated from the forces acting on the bit. The following comments can be made concerning Figure 4-306.

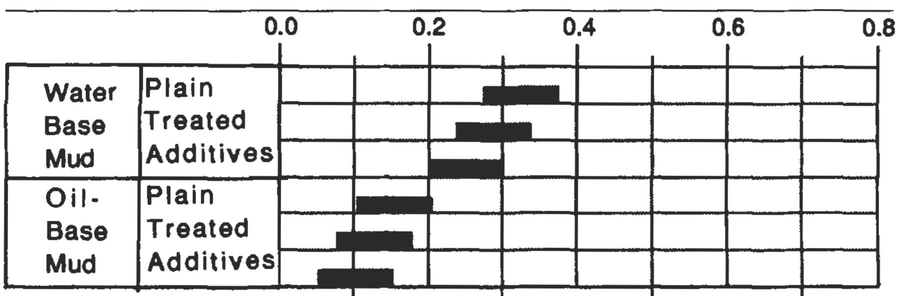
At X380 m: Drag and rotary friction increase due to the third and fourth stabilizers hanging and digging in sand at X310. ROP decreases from 50 to 10 ft/hr. Work and ream pipe, drag decreases, ROP increases to 70 ft/hr. Rotary friction does not go down until stabilizers are out of the sand.
At X520 m: Drag and rotary friction increase as the third stabilizer enters sand at X480. ROP decreases from 80 to 15 ft/hr. Work and ream pipe with no success. Five-stand wiper trip. Drag and rotary friction decrease to normal levels, ROP increases to 50 ft/hr.
At X730 m: Rotary friction increases sharply at X710 as top stabilizer enters sand at X570. Drag starts to increase as stabilizer begins to hand in the sand. This decreases the ROP from 60 to 10 ft/hr. Three-stand wiper trip decreases both drag and rotary friction and increases ROP from 10 to 80 ft/hr.
At X150 m: Connection, work and ream pipe. Stabilizers now out of sandy sections. Rotary friction factor decreases.
At X230 m: Drag friction increases due to top stabilizer hanging in sand at X080. Three-stand wiper trip. Drag decreases back to normal.

The DRAG-FRIC technique can be used to:

- detect the onset of sticking problems and avoid stuck pipe;
- optimize bit performance and avoid unnecessary trips;
- optimize hole conditioning techniques.

(text continued on page 1020)

Table 4-131
Common Friction Factor Values



Courtesy Anadrill [113]

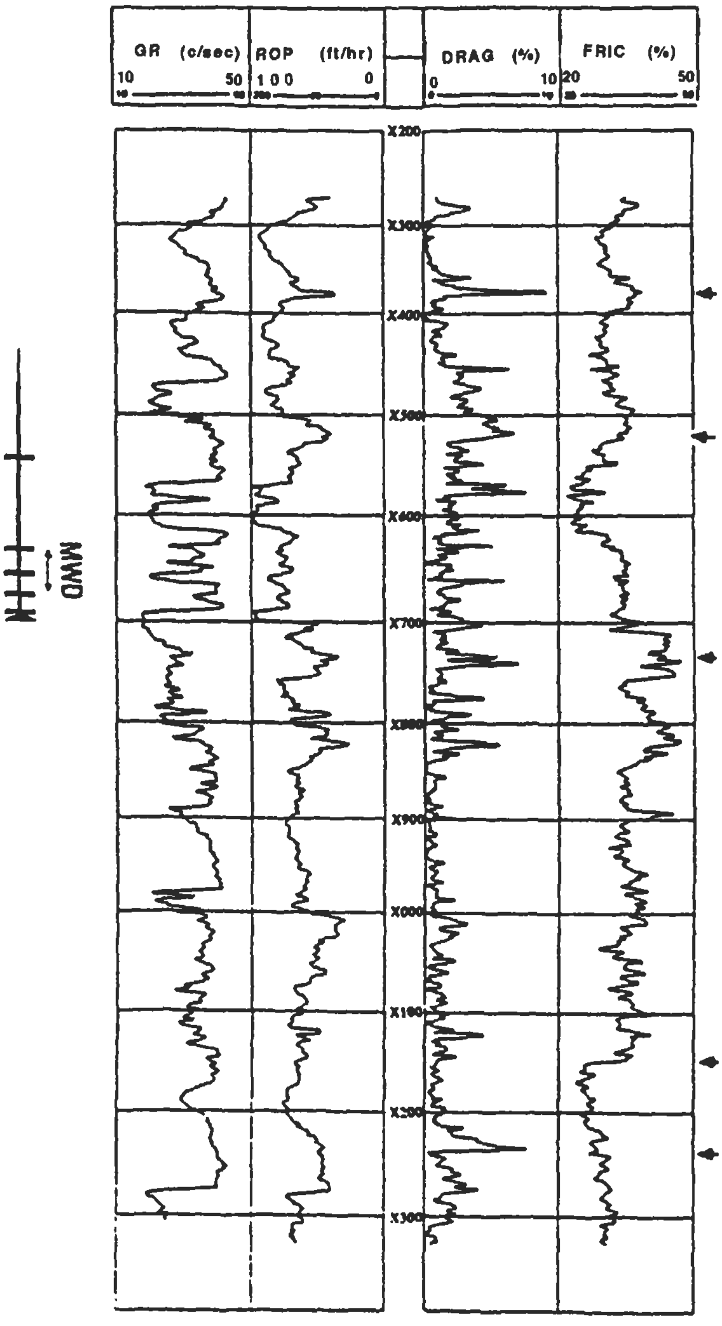


Figure 4-306. Example of sticking pipe indicators. (*Courtesy Petroleum Engineer International* [117].)

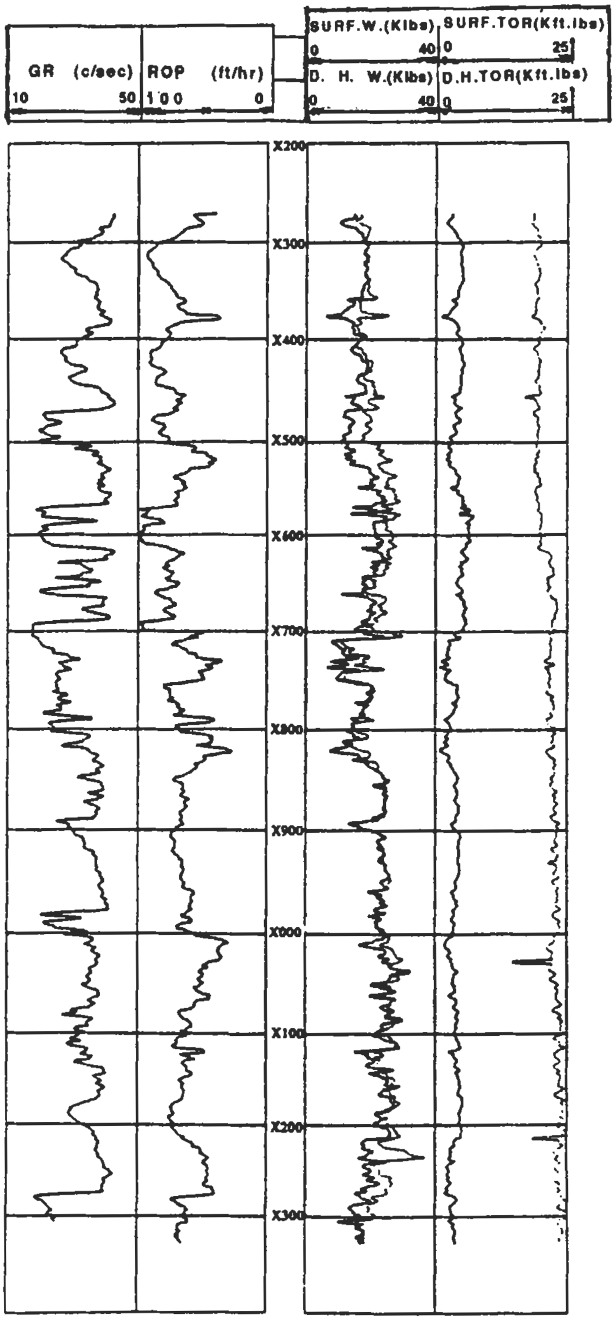


Figure 4-307. Raw data used for computing the log represented in Figure 4-306. (*Courtesy Petroleum Engineer International* [117].)

(text continued from page 1017)

Drillstring Failure Prevention. Drillstring failures are mainly due to vibrations, shocks and neutral point positioned too close to the drill pipes. They result in drillstring washouts and twist-offs.

Axial vibrations usually occur in vertical boreholes and hard rock drilling with tricone bits. They can cause top drive shaking, Kelly bouncing, and induce downhole shocks. Axial vibrations can be minimized by changing the WOB and rotary RPM after rotation comes to a complete stop. Change the bit type may also help.

Lateral vibrations result from the coupling with axial mode of buckling. The lateral vibrations may also result from a coupling with the torsional vibrations.

Torsional vibrations are due to the "stick-slip" effect of the stabilizers in deviated boreholes. They can be seen at surface as large torque oscillations with a period of 3 to 10 s. Figure 4-308 shows a near-bit stabilizer in a deviated borehole. The stick-slip effect increases with WOB and RPM.

A torque feedback system has been developed to dampen the surface torque oscillations and consequently the stick-slip motion at the bit. The system consists of (see Figure 4-309)

- motor current sensor
- shaft encoder coupled to the rotary drive motor
- microprocessor
- on/off switch for the driller to activate the feedback system

The system of Figure 4-309 reduces drillstring vibrations by 90%. Note that when deactivating the system, several minutes elapse before the vibrations start again. Similarly, when the system is turned on, several minutes will elapse before the vibrations are dampened.

The *lateral shocks*, the most damaging for the MWD/LWD equipment, are more severe in vertical holes than in horizontal holes. In a vertical hole, the collars or stabilizers hit the borehole wall hard because the gravity does not pull the collar on the low side as in deviated or horizontal boreholes.

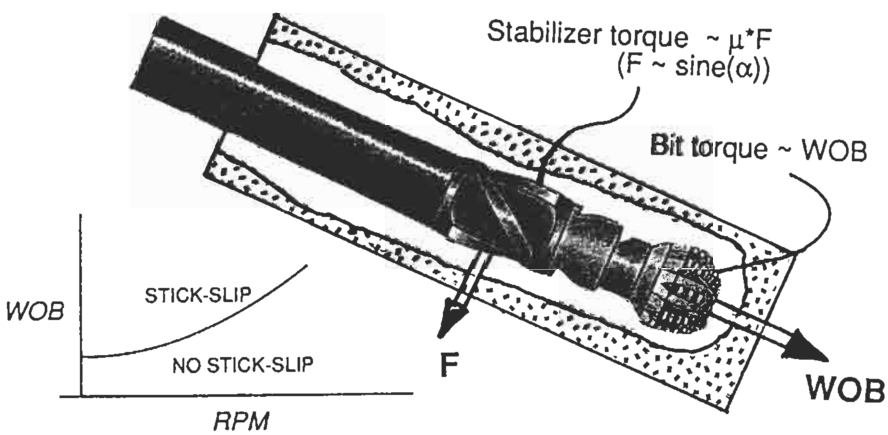


Figure 4-308. Stick-slip effect due to near-bit fraction in a deviated borehole. *(Courtesy Anadrill* [113].*)*

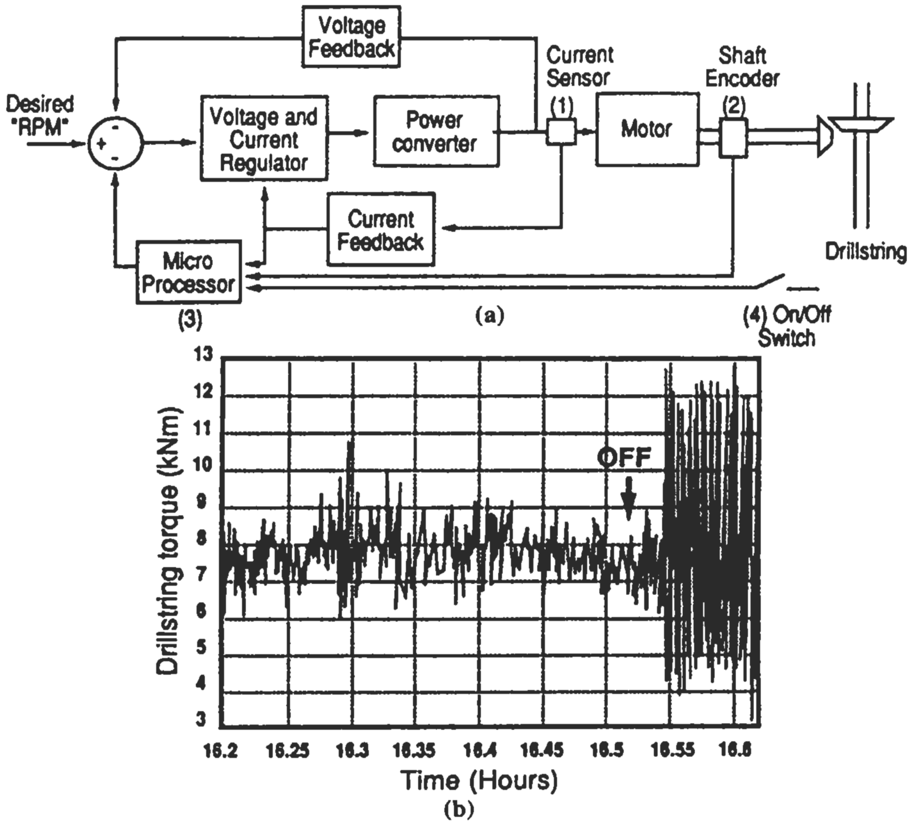


Figure 4-309. Torque feedback system: (a) block diagram; (b) effect on drill string torque. *(Courtesy Anadrill* [113].*)*

Bottomhole assembly whirling is also a cause of lateral shocks. Whirling is the rotation of a bent or buckled BHA or drillpipes. It results in extreme damage to the drillstring and bit. One-sided wear appears on the tool joints. Whirling can be minimized by stopping the rotation and then resuming it with a lower RPM and higher WOB.

Twist-offs, a rupture of the drillstring, mainly occur at tool joints or connections between drillpipe and drill collar. The main causes for twist-offs are

- bending fatigue due to lateral vibrations
- torsional vibrations overstress the tool joints
- neutral point cycling up and down the connection drill collar/drillpipe due to axial vibrations
- drill pipe washouts

Drillpipe washouts are cracks in the drill pipes, causing a mud leak. The detection of drillpipe washouts is done by recording in normal drilling:

- the standpipe pressure (SPP)
- the pump stroke number per minute (SPM)
- the MWD generator turbine speed in rpm (TRPM)

These parameters are related by the following relations:

$$SPP = k \times SPM^2 \tag{4-220}$$

$$SPM = m \times TRPM \tag{4-221}$$

where k and m are proportionality constants depending on the units.

If a drillstring washout occurs then

$$\frac{SPP}{SPM^2} = k \text{ (decreases)} \tag{4-222}$$

and

$$\frac{TRPM}{SPM} = m \text{ (decreases)} \tag{4-223}$$

An example of drill pipe washout is shown in Figure 4-310.

In the interval XX800 to XX980 ft the normal correlation between the total flowrate (strokes per minute) and the MWD flowrate (alternator voltage) can be seen. At the depth XX060, a sharp drop in MWD flowrate indicates that a leak is occurring somewhere in the drillstring. The string was pulled out and a washout was found in the second heavyweight above the MWD tool.

Note that in the example the torque friction was above 4% and the pipe drag about 10%, which is fairly low for a borehole deviation of 28°.

Rate of Penetration Optimization. One important procedure for the ROP optimization is the *drill-off* test. A drill-off test is conducted as follows:

A. Choose a starting weight-on-bit and an RPM. (The RPM will be maintained constant during the test.)
B. Drill at starting weight for a few minutes to establish the bottomhole conditions.
C. Lock the brake and let the bit "drill-off" the weight.
D. Record the time (ΔT) to drill a predetermined amount of weight (ΔWOB), usually 2,000 to 4,000 lb.
E. Calculate the average ROP over the WOB change using

$$ROP = L \times \frac{1}{E \bullet A} \times \frac{\Delta WOB}{\Delta T} \tag{4-224}$$

where ROP = rate of penetration in ft/min
L = drillpipe length in ft
E = Young modules of the drillpipes in psi
A = cross-sectional area of the drillpipe in in.^{2}

F. Repeat steps D and E as the drill bit continues to drill-off weight and graph.
G. Repeat drill-off test with different RPM.

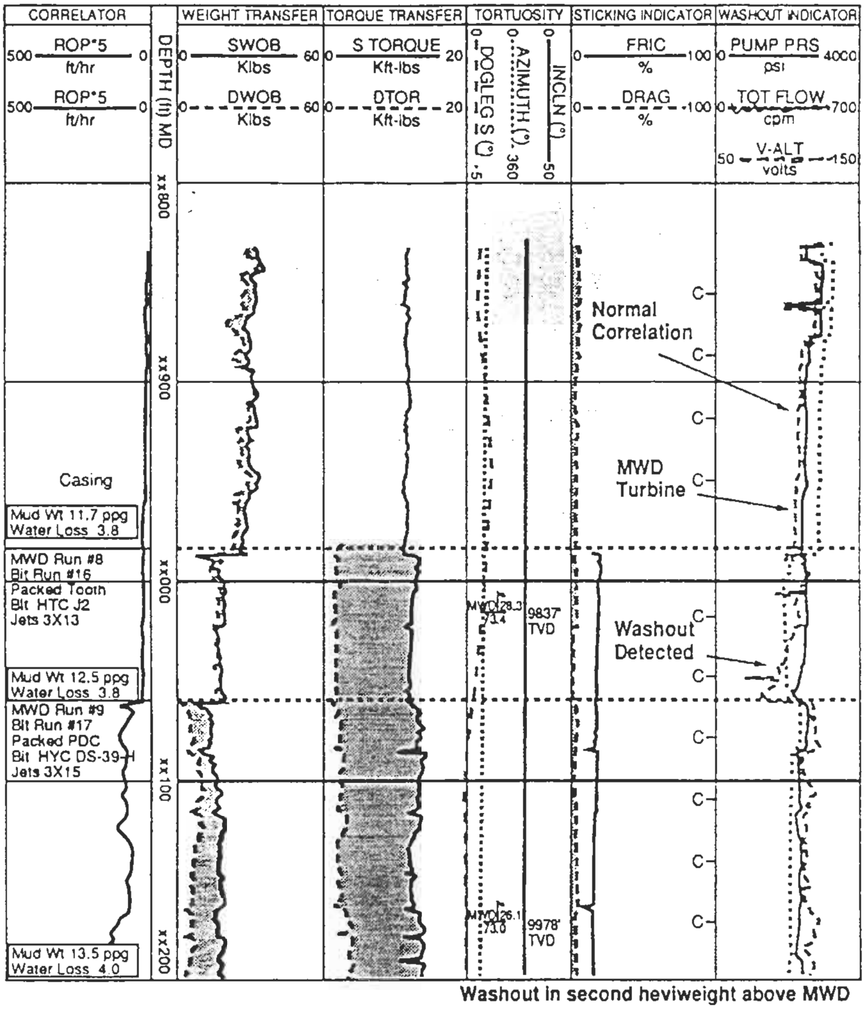


Figure 4-310. Example of drill pipe washout. *(Courtesy Anadrill.)*

Drill-off tests are tedious to perform by hand. Computerized drill-off tests are now available. ROP is directly calculated as a function of WOB. Figure 4-311 is an example of computer generated drill-off curves.

The driller uses the curves by selecting the WOB and RPM corresponding to the highest ROP. The curves can also be used to quantify the advantages of running unconventional weights-on-bit and rotary speeds.

The weight transfer is also an important parameter that can decrease the rate of penetration. In deviated wells, up to 20,000 lb of weight and 25,000 ft-lb of torque can be lost to drag and friction. Figure 4-312 shows the improvement that can be caused by an 11-stand short trip and circulating a mud pill of 15 lb/bbl of Barafiber.

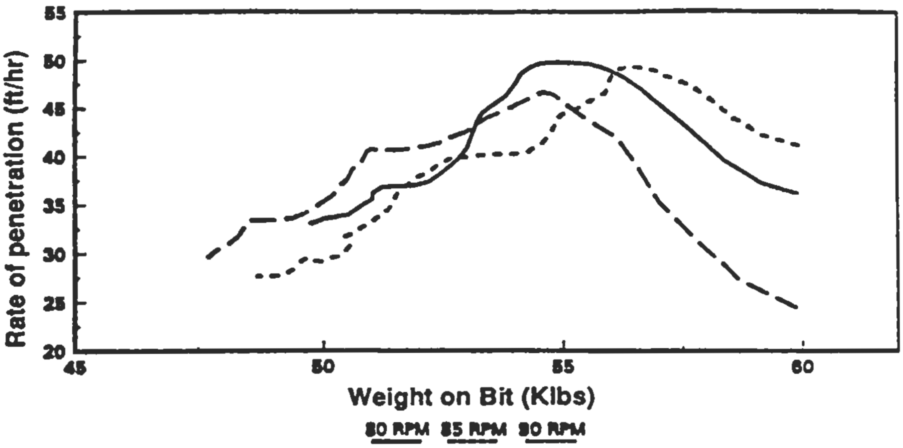


Figure 4-311. Drill-off test curves generated by a computerized system. *(Courtesy Anadrill.)*

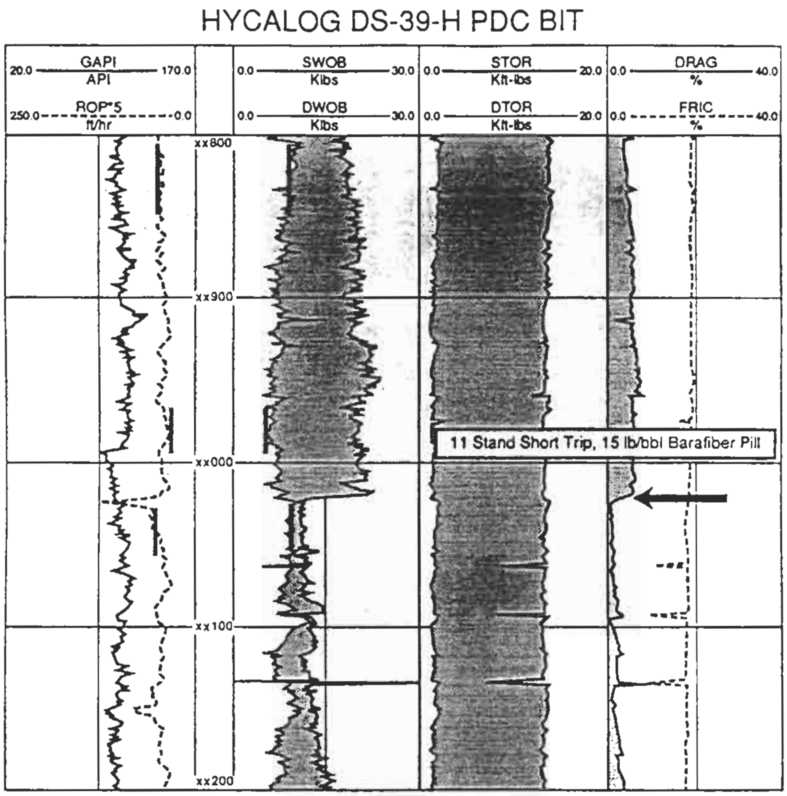


Figure 4-312. Example of poor weight transfer caused probably by cutting buildup. *(Courtesy Anadrill.)*

Bit Wear and Rock Type. Using downhole weight-on-bit and torque the *roller cone bit wear* can be estimated while drilling. The well site computer can be used to compute in real-time a dimensionless torque (T_D) and a dimensionless rate of penetration (R_D) using the following equation:

$$T_D = \frac{DTOR}{DWOB \bullet BD} \times 12 \tag{4-225}$$

$$R_D = \frac{ROP}{RPM \bullet BD} \times \frac{12}{60} \tag{4-226}$$

where DTOR = downhole torque in klb-ft
DWOB = downhole weight-on-bit in klb
ROP = rate of penetration ft/hr
RPM = bit rpm
BD = bit diameter in in.

For a sharp bit drilling in shale [118] the following expression is used:

$$T_D = A_1 + E_d + A_2 \times \sqrt{R_D} \tag{4-227}$$

where E_d = bit efficiency (one for sharp bit; zero for worn out bit)
A_1 = dimensionless gouging coefficient (A_1 = 0.07 to 0.12 for milled tooth bit; A_1 = 0.2 to 0.35 for PDC bits)
A_2 = crushing coefficient (A_2 = 0.07 to 0.15 for milled tooth bit)

Figure 4-313 shows schematically the variation of T_D versus $(R_D)^{0.5}$ in shales, as well as other zones in the diagrams such as porous zones and tight zones.

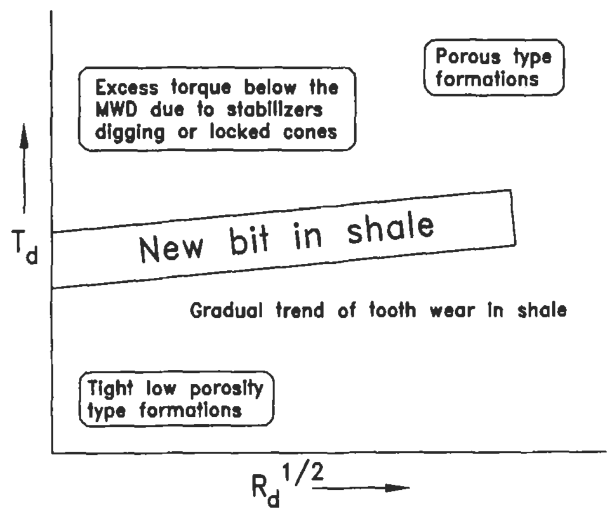


Figure 4-313. Variation of the dimensionless torque versus the dimensionless ROP.

Assuming that A_1 does not change, E_D can be calculated in real-time as shown in the example of Figure 4-314. The effect of a locked cone can be seen clearly at XX610 ft.

When the bit was pulled out and examined, one cone was hanging, one was locked, and one was okay (T_4).

Downhole Mud Motor Optimization. Downhole mud motors in use today are practically all of the positive displacement type (sometimes called the "moving cavity"). A description of these motors has been given in section 4.10. The theoretical characteristics curves of the motors are shown in Figure 4-315 in arbitrary units.

If ΔP is the differential pressure across the motor, due to leakage between the rotor and stator, the RPM decreases as follows:

New Motors

$$RPM = 4.0 - 1.0 \times \Delta P \tag{4-228}$$

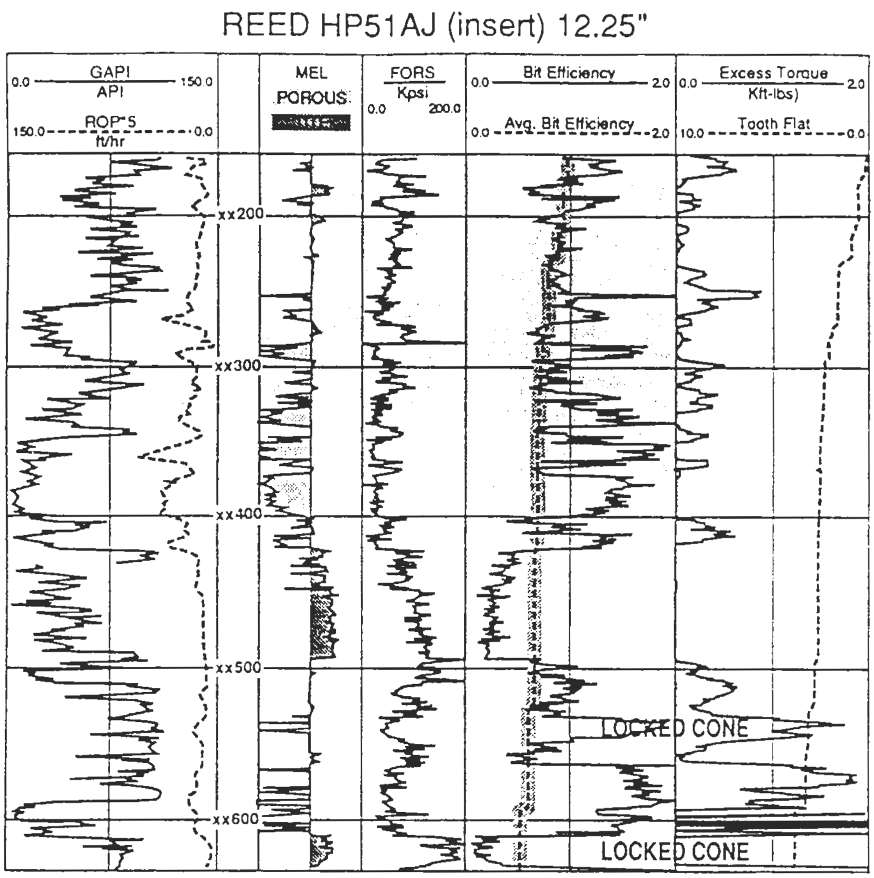


Figure 4-314. Bit efficiency curve. *(Courtesy Anadrill.)*

Worn out motors

$$RPM = 4.0 - 2.0 \times \Delta P \tag{4-229}$$

The torque (T) increases with ΔP as follows:

$$T = 2.0 \times \Delta P \tag{4-230}$$

The horse power (MHP) goes through a maximum according to the formula:

New motors

$$MHP = 8.0 \times \Delta P - 2.0 \times \Delta P^2 \tag{4-231}$$

Worn out motors

$$MHP = 8.0 \times \Delta P - 4.0 \times \Delta P^2 \tag{4-232}$$

Figure 4-316 shows the characteristic curves for a new 8-in. OD, 7:8 lobe, and three stages.

The recommended full load ΔP is 490 psi. When the pressure is measured inside the drill collars and below the motor, ΔP can be determined. The flowrate is also known and the hydraulic power can be calculated in real time:

$$HHP = \frac{\Delta P \bullet Q}{1714} \tag{4-233}$$

where HHP = motor hydraulic power in hp
ΔP = motor pressure drop psi
Q = flowrate in gal/min

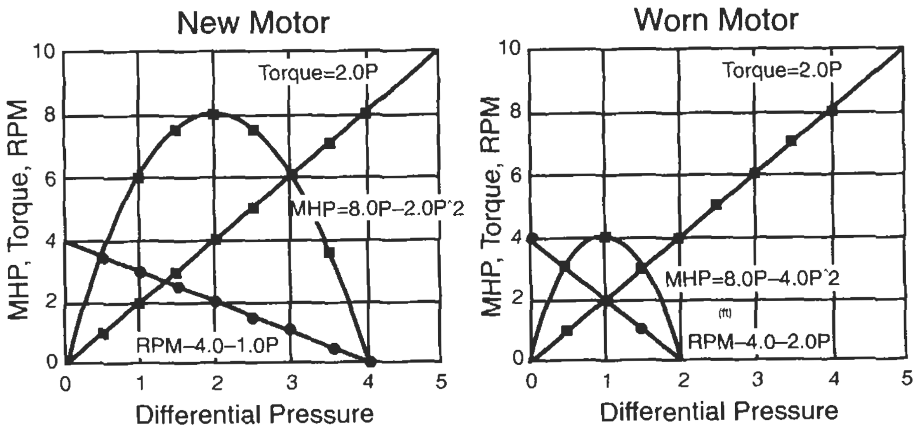


Figure 4-315. Characteristics of new and worn out positive displacement mud motors. *(Courtesy Anadrill.)*

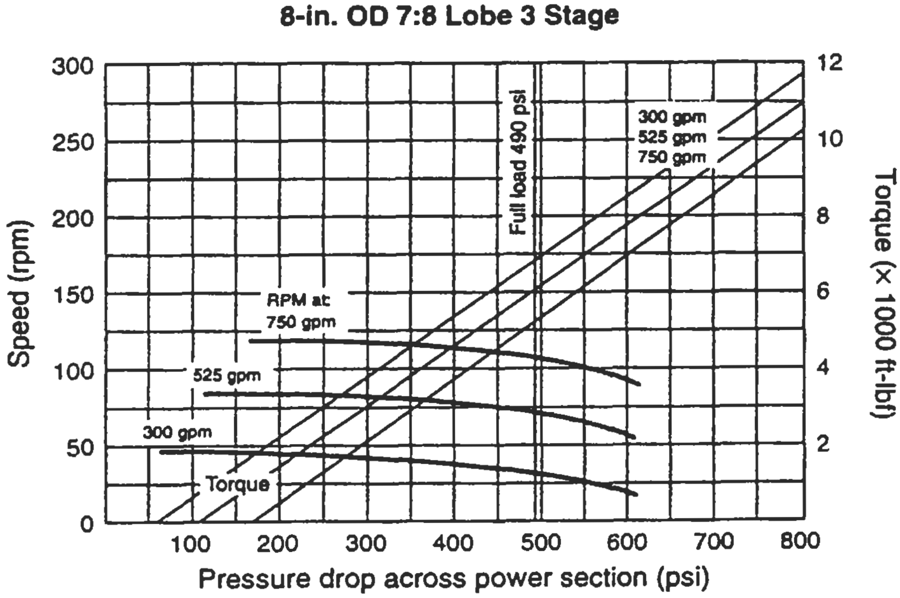


Figure 4-316. Characteristic curves for a positive displacement mud motor with 8-in. OD, 7:8 lobe, and three stages. *(Courtesy Anadrill.)*

For the motor of Figure 4-316 at ΔP = 490 psi and Q = 525 gal/min, we have

HHP = (490 × 525)/1714 = 150 hp

HHP = 3,378,060 × 0.03312 = 111,877 W = 150 hp (metric units)

The mechanical power can be calculated with the formula when the RPM and torque are known. Both are measured downhole in Anadrill geosteering system:

$$MHP = \frac{T \bullet RPM}{5263} \tag{4-234}$$

$$MHP = \frac{2\pi}{60} \times T \times RPM \tag{4-235}$$

where MHP = motor mechanical power in HP
T = motor torque in ft-lb
RPM = motor rotation speed in rpm

For the motor of Figure 4-316 at ΔP = 490 psi, we read 6,000 ft-lb and 74 rpm:

MHP = (6000 × 74)/5,247 = 85 HP

= (2π/60) × 8,134.8 × 74 = 63,067 W = 85 HP (metric units)

The efficiency of the motor of Figure 4-316 is

$$\eta = \frac{MHP}{HHP} = \frac{85}{150} = 56\%$$

To monitor the motors performance, a plot can be made in real time of the mechanical horsepower versus the hydraulic horsepower. The curve should be similar to the MHP curve of Figure 4-315 since, according to equation 4-233, at constant Q, the hydraulic power is proportional to ΔP. An example is shown in Figure 4-317 where measurements have been made at various RPM.

Motorwise, the optimal setting is marked on the screen with a large X.

Rock Mechanical Properties. In the previous section (Figure 4-313), the wear of the bit teeth can be determined in shales by plotting the dimensionless bit torque (T_D) versus the dimensionless ROP (R_D). By introducing a new parameter, namely the apparent formation strength, the bit effects can be separated from the lithology effects.

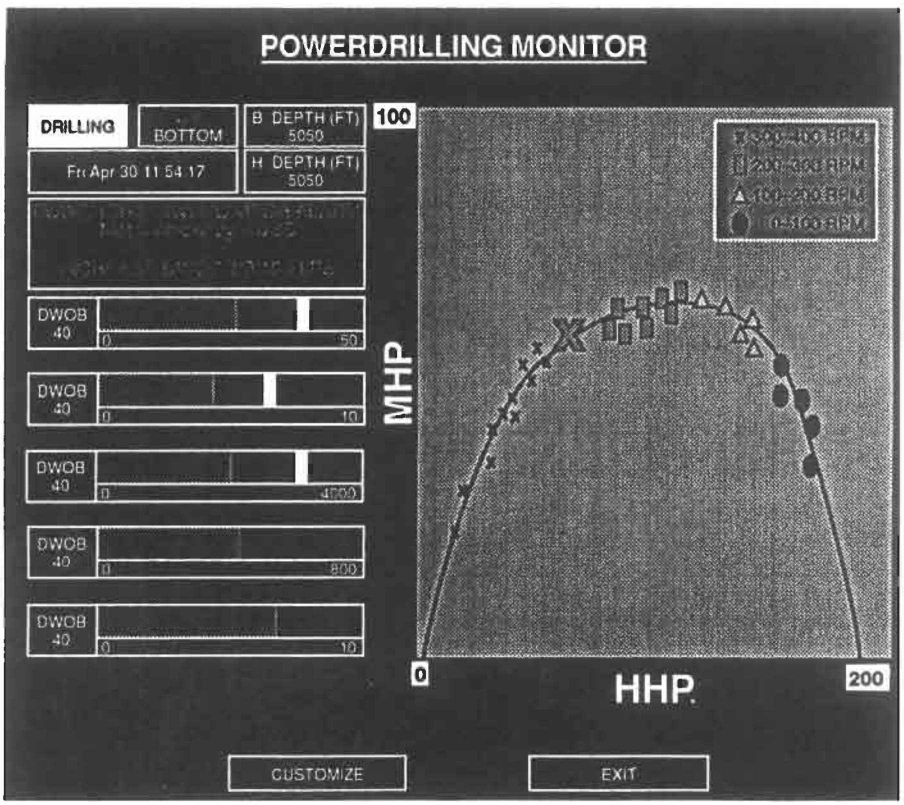


Figure 4-317. Power drilling monitor example. *(Courtesy Anadrill* [113].*)*

The apparent formation strength (FORS) is calculated with the following equation:

$$FORS = 40 \times \frac{DWOB \times RPM \times A_1 \times E_d}{ROP \times BD} \tag{4-236}$$

where FORS = apparent formation strength in psi
DWOB = downhole weight-on-bit in lb
RPM = bit rotation per minute
A_1 = dimensionless gouging coefficient (A_1 = 0.07 to 0.12 for milled tooth bit, A_1 = 0.2 to 0.35 for PDC bits)
E_d = bit efficiency (one for a sharp bit, zero for a worn bit)
ROP = rate of penetration in ft/hr
BD = bit diameter in in.

This equation produces a formation strength that is independent of the wear state of the bit and other measurable drilling variables. Figure 4-318 is a schematic plot of the dimensionless torque (T_D) versus the apparent formation strength (FORS).

Once the shale line has been established for a certain drill bit, the porous and tight zones can be determined regardless of bit condition.

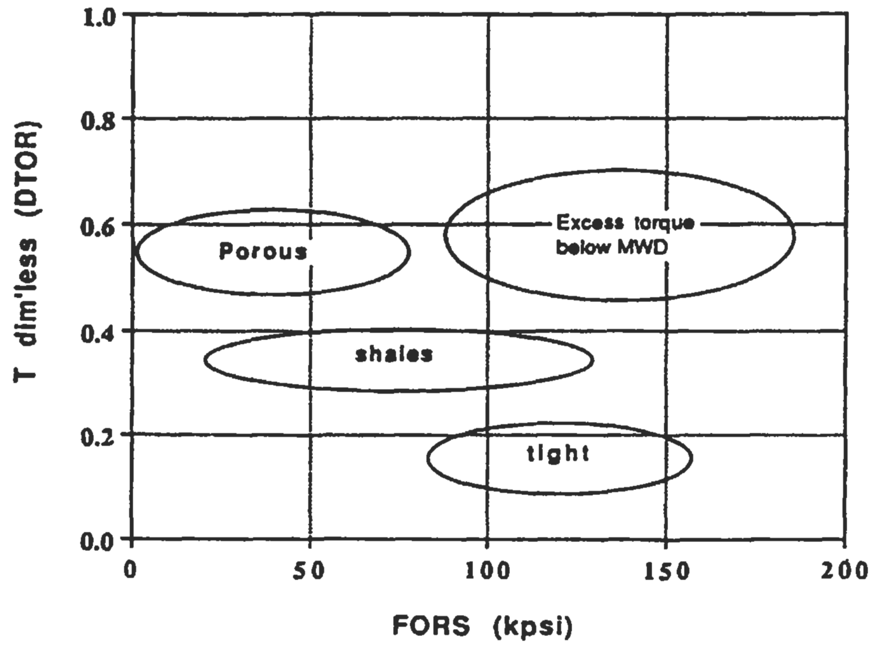


Figure 4-318. Schematic plot of the dimensionless torque T_D versus the apparent formation strength.

Demonstration. If DWOB = 14,000 lb, RPM = 100, A_1 = 0.1, E_d = 1 (new bit), ROP = 80 ft/hr, BD = 8.5 in.

FORS = 8,235 psi. Typical shallow Gulf Coast formations.

The techniques described provide the following answers at the rig site, in real-time:

- rock strength
- wear state of the bit teeth in shales
- lithological correlations
- excess torque or cone locking

This last point is illustrated in Figure 4-314 where FORS = 190 kpsi, when the cones lock.

Example 20: Drag and Friction Coefficients

Figure 4-319 shows the friction (due to rotation) and the drag recorded in a slanted borehole.

Borehole data:

- Average inclination: 24°
- KOP at 3,000 ft
- Mud weight: 16 lb/gal

Drillstring data:

• Drillpipe OD:	$4\frac{1}{2}$ in.
weight:	22 lb/ft
metal displacement:	0.2855 gal/ft
length:	1,000 ft
• Drill collars OD:	8 in.
weight:	223 lb/ft
metal displacement:	2.2882 gal/ft

1. Compute the drag coefficient at 16,200 ft.
2. Compute the friction coefficient at 16,200 ft.
3. Compare with the data computed and recorded at 16,200 ft.

Solution

1. Length of inclined drillpipes: 16,200 – 3,000 – 1,000 = 12,200 ft.
   Weight of drillpipes in air: 12,200 × 22 = 268.4 klb.
   Weight of displaced mud: 0.2855 × 12,200 × 16 = 55.73 klb.
   Weight of drillpipes in mud: 268.4 – 55.73 = 212.67 klb.
   Weight of drill collars in air: 223 × 1,000 = 223 klb.
   Weight of displaced mud: 2.2882 × 1,000 × 16 = 36 klb.
   Weight of drill collars in mud: 223 – 36 = 187 klb.
   Total weight of inclined pipe and drill collars: 212.67 + 187 = 399.67 klb (~400 klb).
   Surface weight-on-bit: 23 klb
   Downhole weight-on-bit: 5 klb
   Using Equation 4-218

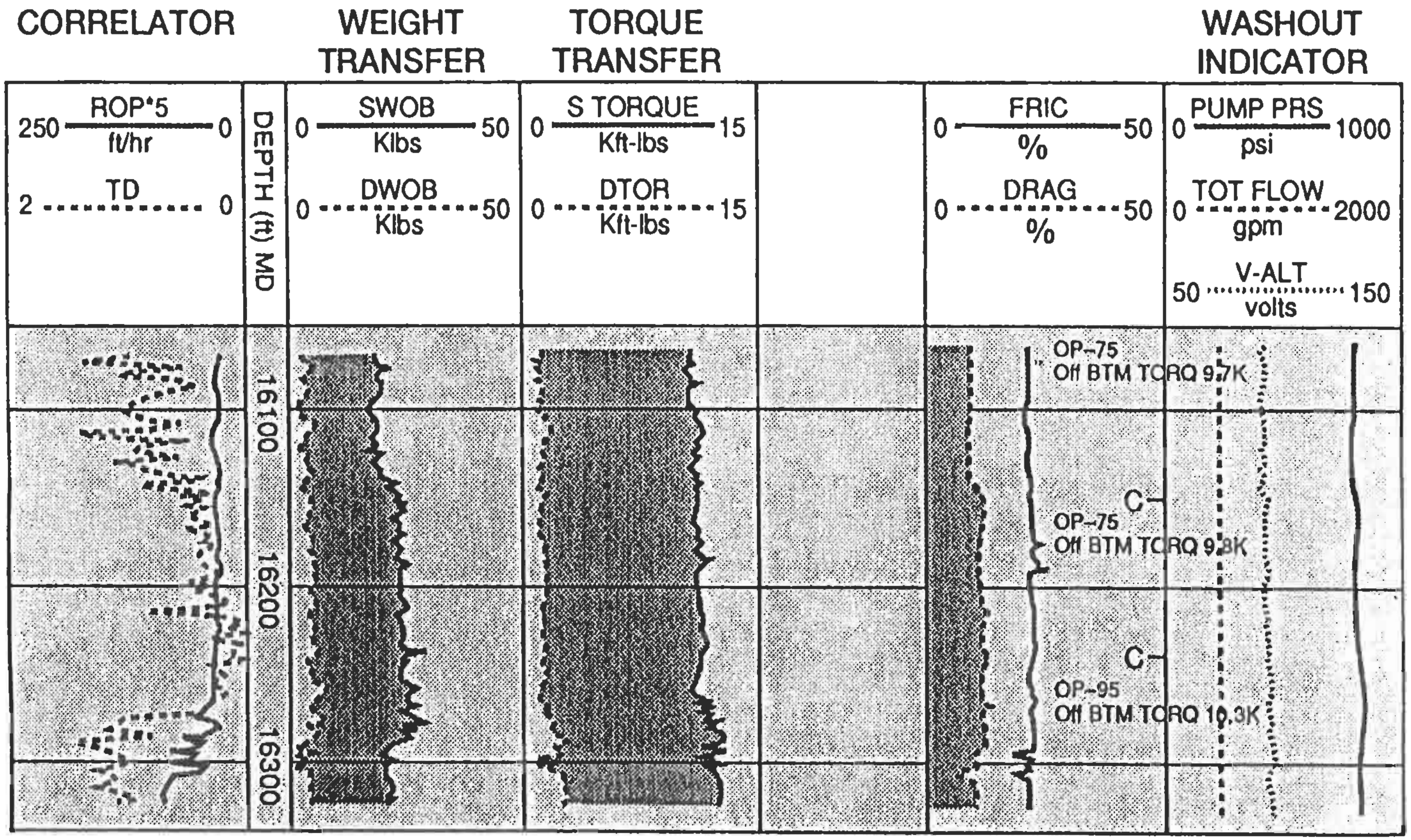


Figure 4-319. Friction (due to rotation) and drag recorded in a slanted borehole. *(Courtesy Anadrill.)*

$$\text{DRAG} = \frac{23 - 5}{400 \times \sin 24} = 0.11 = 11\%$$

2. Surface torque: 11.1 kft • lb
   Downhole torque: 1.5 kft • lb
   Using Equation 4-219

$$\text{FRIC} = \frac{(11.1 - 1.5) \times 12}{\sin 24 \times (212.67 \times 2.25 + 187 \times 4)} = 0.23 = 23\%$$

3. The values read on Figure 4-319

   DRAG = 11%

   FRIC = 22%

The algorithms used for computing the values of Figure 4-319 are identical to those used in questions 1 and 2.

Example 21: Drag and Friction Coefficients

Figure 4-320 shows the data recorded in a highly deviated well.
Borehole data:

- Mud weight: 12 lb/gal (water-base)
- KOP: 2,000 ft
- Average inclination: 46°.

Drillstring data:

- Drillpipe
  - OD: $4\frac{1}{2}$ in.
  - weight: 22 lb/ft
  - metal displacement: 0.2855 gal/ft
- Drill collar
  - OD: 8 in.
  - length: 1,000 ft
  - weight: 223 lb/ft
  - metal displacement: 2.28 gal/ft

1. Compute the drag and friction coefficients at 11,000 ft.
2. What was the effect of the "pump pill" and short trip at 11,020 ft?
3. Was the short trip at 11,350 ft necessary?
4. If the pump pills decrease the drag, what is the probable reason for drag increase?
5. If the short trips decrease the drag, what is the probable reason for drag increase?

Solution

1. Drag: 0.05 (5%)
   Friction: 0.18 (18%)
   Same values computed on the log.

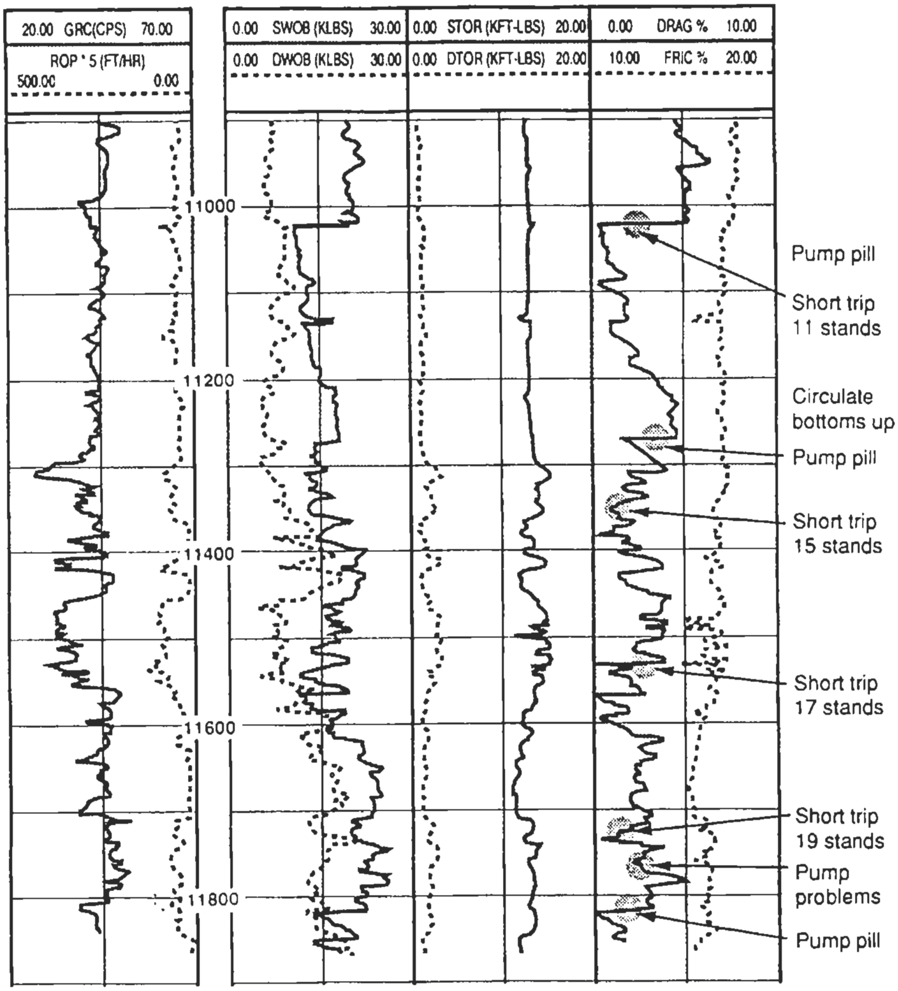


Figure 4-320. Example of drilling data recorded in a highly slanted borehole. *(Courtesy Anadrill.)*

2. It decreased the drag from 5 to 0.5% for the next 200 ft of drilling. Cutting buildup may be the problem.
3. Probably not.
4. Probably cutting buildup.
5. Probably stabilizers hanging.

Example 22: Shocks Recording

Figure 4-321 shows the data recorded in a borehole. Shocks have been monitored. The curve at the right shows the number of shocks per second with an amplitude greater than 25 g.

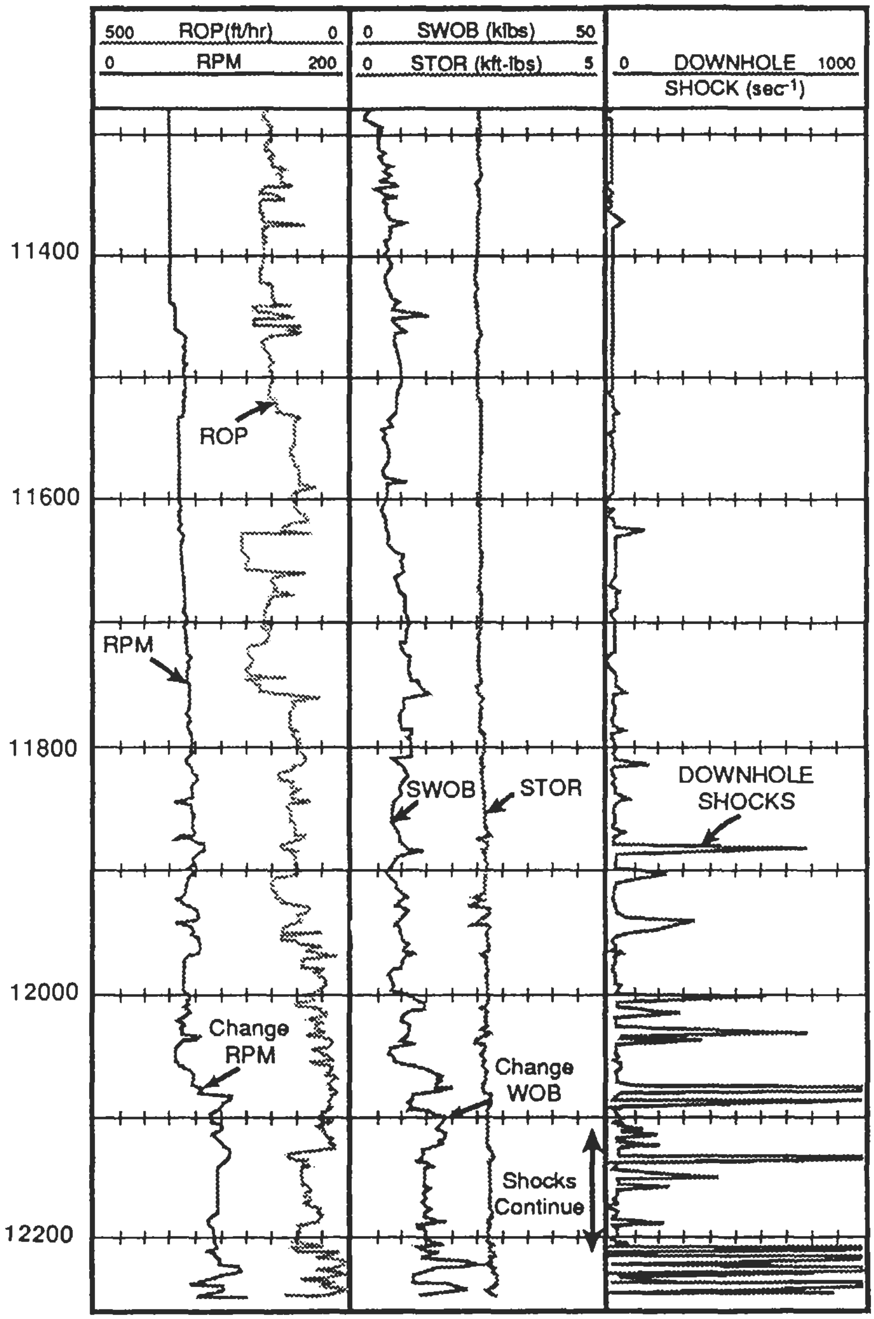


Figure 4-321. Example of drilling data including shocks recorded in a slanted borehole. *(Courtesy Anadrill.)*

Only surface WOB and torque are available. Borehole inclination is 40°.

1. Is there a correlation between the shock number, surface weight-on-bit, rate of penetration, and RPM? Explain.
2. What is the probable reason for the shocks?
3. At 12,040 ft, would you have increased or decreased RPM for reducing the shock number?
4. How can the weight-on-bit increase/decrease the shock number?
5. Could something else have been tried?

Solution

1. Yes; with the shock number, RPM increases, ROP decreases and SWOB increases.
2. Probably stabilizer digging in a stick-slip fashion.
3. In theory, an increase in RPM decreases stabilizer digging due to increase in the BHA momentum. However, here it did not help. We may have tried the RPM value used around 11,400 ft where no shocks were registered.
4. Again in theory, a decrease in the weight-on-bit improves the stabilizer digging by reducing the side forces. Here weight-on-bit was increased.
5. A short trip of 20 stands may have helped.

Example 23: Weight-On-Bit and Torque Interpretation

The data given in the first four columns of Table 4-132 were recorded during drilling with an MWD downhole WOB/TOR sub.

1. Compute the normalized torque (TOR), the normalized ROP, and plot TOR versus ROP.
2. Compute the drilling formation strength (FORS) and plot versus depth.
3. Plot TOR versus FORS.
4. Comment on the results.

Solution

1. Table 4-133b gives calculated values of normalized TOR, normalized ROP and FORS.
2. Figure 4-322 gives normalized TOR versus normalized ROP.
3. Figures 4-323 gives FORS versus depth.
4. Figure 4-324 gives normalized TOR versus FORS.
5. Some tight formations can be seen at 9,250 and 9,400 in. The representative point at 8,900-ft plots in the porous area of the graph. The other points seem to be mostly shale points. Confirmation should be sought with logging data.

Abnormal Pressure Detection

Definition of Concepts. The *hydrostatic pressure* in a borehole is the pressure exerted by a column of fluid that height is the true vertical depth. This is

$$P_H = 0.052 \bullet W_M \bullet Z_V \tag{4-237}$$

Table 4-132
Interpreting Downhole Drilling Data Recorded in a Vertical Well

FORS Calculation						
DIA =	12.25	inches				
RPM =	160	rpm				
Norm. TOR =	12*TOR/(WOB*DIA)					
Norm. ROP =	(12*ROP/(RPM*DIA))^0.5					
FORS= 40*0.0563*WOB*RPM/(60*ROP*DIA)						
Depth ft	WOB Klbs	ROP ft/min	TOR kft.lbs	Norm TOR	Norm. ROP	FORS (kpsi)
8900	39.29	0.83	3.57	0.089	0.071	23.21
8950	39.29	0.71	3.33	0.083	0.066	27.13
9000	39.29	0.59	3.33	0.083	0.060	32.65
9050	46.43	0.71	3.8	0.080	0.066	32.06
9100	44.64	1	3.8	0.083	0.078	21.88
9150	44.64	0.59	3.57	0.078	0.060	37.09
9200	46.43	0.77	3.81	0.080	0.069	29.56
9250	46.43	0.59	3.33	0.070	0.060	38.58
9300	53.57	1.07	4.52	0.083	0.081	24.54
9350	50	0.47	4.05	0.079	0.054	52.15
9400	53.57	0.59	3.81	0.070	0.060	44.51

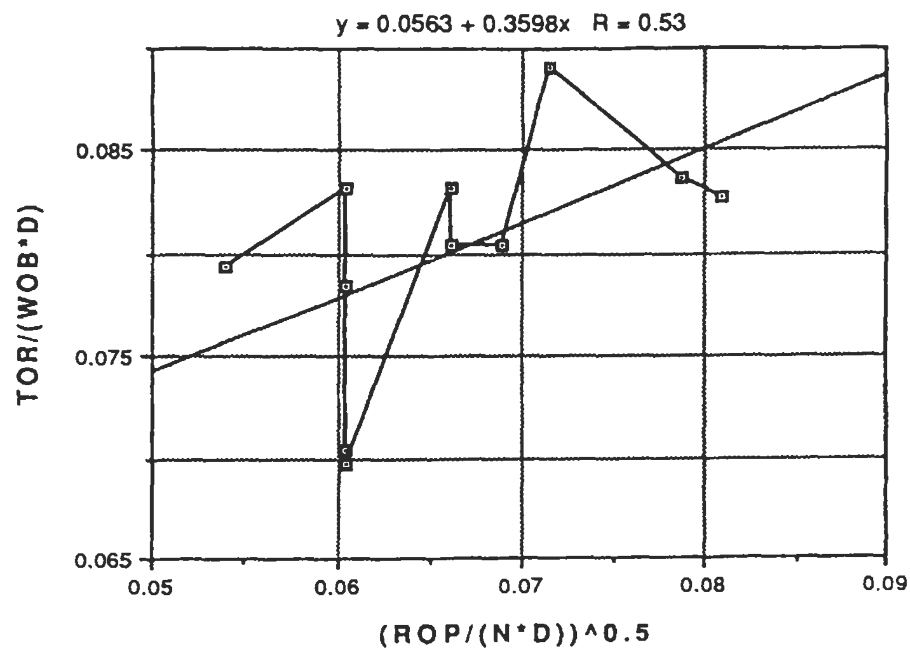


Figure 4-322. Dimensionless torque versus dimensionless ROP: (a) grid; (b) plot of calculated values.

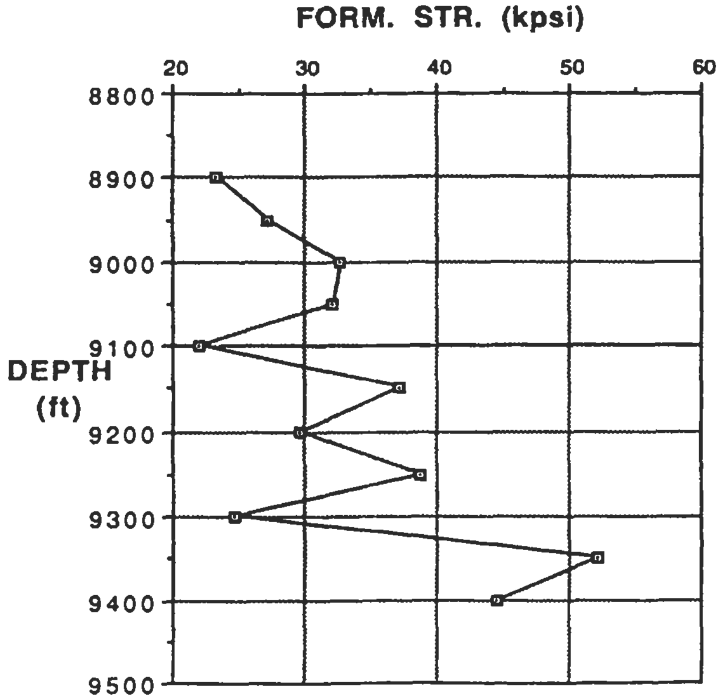


Figure 4-323. Formation strength versus depth: (a) grid; (b) plot of calculated values.

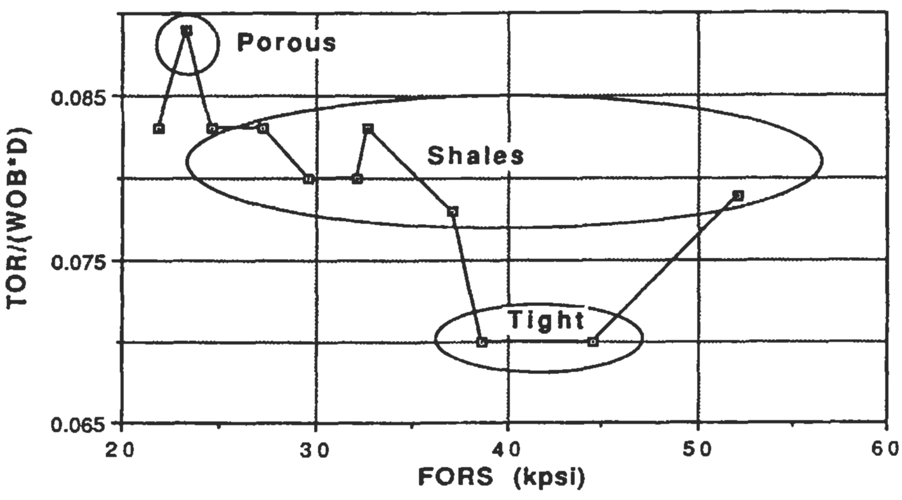


Figure 4-324. Dimensionless torque versus formation strength: (a) grid; (b) plot of calculated values.

where P_H = hydrostatic pressure in psi
W_M = fluid density in lb/gal
Z_V = true vertical depth in ft

The *mud equivalent circulation specific weight* includes the pressure drop in the annulus. Thus,

$$ECW = \frac{P_H + P_D}{0.052 \bullet Z_V} \tag{4-238}$$

where ECW = mud equivalent circulation density in lb/gal

Note that P_H must be calculated using the true mud specific weight loaded with cuttings.

The *gradient* (G) is the specific weight of the fluid in a column of height Z_V. Thus,

$$G = \frac{P_H}{0.052 \bullet Z_V} \tag{4-239}$$

The gradient concept here is different from the mathematical gradient. This concept of gradient is also applicable to a column of a solid.

Demonstration. If

W_M = 12 lb/gal

Z_V = 10,000 ft

P_D = 200 psi

P_H = 6,240 psi

ECW = 12.38 lb/gal

We shall say that the gradient is 12 lb/gal without circulation and 12.38 lb/gal with circulation (no cuttings).

The *geostatic* or *overburden gradient* is the average density of a column of sediments of height Z_V. The overburden pressure is calculated with the Equation 4-237 by replacing W_M with the average sediment density. Since the density of sediments generally increases with depth, the overburden gradient and Poisson's ratio increase with depth, but not linearly (see Figure 4-325). The following equation may be used to compute the overburden gradient G_{OV}:

$$G_{OV} = a(\ln Z_V)^2 + b(\ln Z_V) + c \tag{4-240}$$

$$G_{OV} = d(\ln Z_V)^2 + e(\ln Z_V) + f \tag{4-241}$$

The coefficients a, b, c, d, e, f are given in Table 4-133.

Figure 4-325a shows a plot of overburden pressure gradient versus depth for typical soft (1) and hard (2) provinces. Figure 4-325b gives similar data for Poisson's ratio.

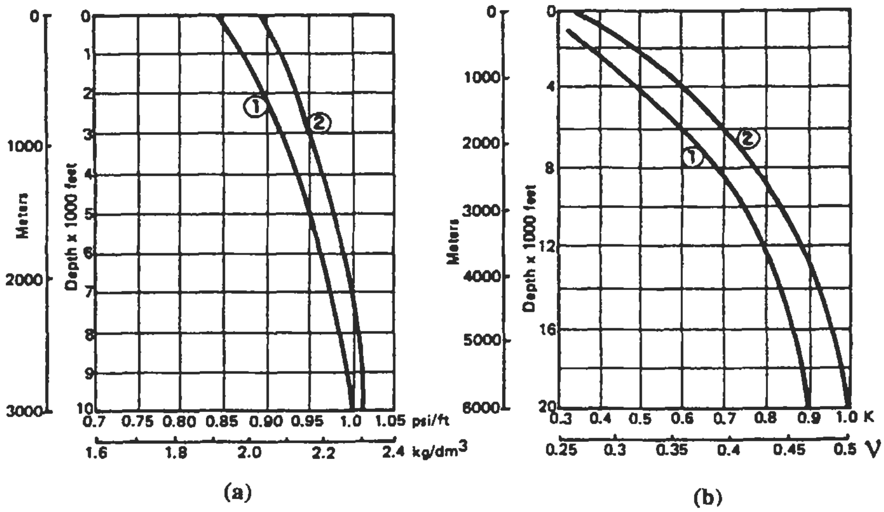


Figure 4-325. Overburden gradient and Poisson ratio: (a) variation of the overburden gradient with depth; (b) variation of the Poisson ratio with depth. *(Courtesy Editions Technip.)*

Table 4-133
Values of Coefficients a, b, c, d, e, f in Equations 4-241 and 4-242

	Gulf Coast (Southern Texas)	Santa Barbara (California)
Formation	**Soft (1)**	**Hard (2)**
a	−0.00312	0.02176
b	0.11715	−0.31307
c	0.18241	2.06162
d	0.01264	−0.04998
e	−0.04670	0.85514
f	1.66734	−1.34235

The *fracture gradient* is an important parameter that must be taken into account while increasing the mud weight and during well control operations. Several equations have been proposed for calculating the fracture gradient. The best results seem to be obtained with the one suggested by Mathews and Kelly [101]:

$$G_{FRAC} = (G_{OB} - G_p) \bullet K_p + G_p \qquad (4\text{-}242)$$

where G_{FRAC} = fracture gradient in psi/ft
G_{OB} = overburden gradient in psi/ft
G_p = pore pressure gradient in psi/ft
K_p = coefficient related to Poisson ratio

with

$$K_p = \frac{\nu}{1 - \nu}$$

where ν = static Poisson ratio

The static Poisson ratio is determined in a triaxial cell. The dynamic Poisson ratio is calculated with the sonic compressional and shear wave velocities. They usually are different.

When controlling a well, be careful not to exceed the fracture pressure anywhere in the open-hole section. An intermediate casing may have to be set to avoid the fracture condition.

The *normal formation pressure gradient* is the density of a column of saltwater of length Z_V is expressed in psi-ft in customary units. Table 4-134 gives normal gradient values for areas around the world. Note that for freshwater or quasi-freshwater G_F = 0.433 psi/ft = 8.345 lb/gal.

The *formation pressure* is said "normal" when it corresponds to the hydrostatic pressure of a column of water of length Z_v; the water having the densities stated in Table 4-134.

An *abnormal formation pressure* is a formation pressure greater than the normal formation pressure. Subnormal formation pressures are also encountered in drilling; they are generally due to depleted reservoirs.

Abnormal Formation Pressures Origin. Four main causes are attributed to abnormal formation pressures: compaction effects, diagenetic effects, differential density effects and fluid migration.

The compaction effects are the most common. According to Terzaghi's work in soil mechanics, the overburden pressure mentioned earlier, is equal to [119]

$$P_{OB} = \sigma_v + \beta P_F \tag{4-243}$$

where P_{OB} = overburden pressure in psi
σ_v = vertical stress in psi
P_F = formation pressure in psi
β = poroelasticity constant (0.75 to 1.0)

Table 4-134
Normal Formation Pressure Gradients in Various Areas [101]

Area	Pressure Gradient (psi/ft)	Equivalent Water Specific Weight (lbs/gal)
West Texas	0.433	8.345
Gulf of Mexico coastline	0.465	8.963
North Sea	0.452	8.712
Malaysia	0.442	8.520
Mackenzie Delta	0.442	8.520
West Africa	0.442	8.520
Anadarko Basin	0.433	8.345
Rocky Mountains	0.436	8.403
California	0.439	8.462

During compaction, if the fluid can escape, the formation pressure stays equal to the normal formation pressure.

If the fluid cannot escape due to permeability barriers, for example, then the fluid supports part or most of the overburden load. Under these conditions the formation pressure can be up to twice the normal formation pressure.

The *diagenetic effects* are related to the alteration of rock mineral, shales in particular. Under certain conditions, montmorillonite clays change to illites, chlorites and kaolinites. The water of hydration that desorbs in the form of free water occupies a larger volume. This volume increase will cause abnormal pressures if the water cannot escape.

The *differential density effects* are especially related to thick gas reservoirs or highly dipping reservoirs. If we assume that at the gas/water contact a normal pressure exists, as we come up the reservoir or updip, the normal pressure due to the water column decreases more rapidly than the gas pressure.

Demonstration. For example, assume a gas reservoir with a gas/water contact located at 10,000 ft. The reservoir thickness, or change in depth due to dip, is 2,000 ft. The normal gradient is 0.433 psi/ft. The normal pressure at 10,000 ft is

$$P_H(10{,}000\ \text{ft}) = 4{,}330\ \text{psi}$$

The normal pressure at 8,000 ft is

$$P_H(8{,}000\ \text{ft}) = 3{,}464\ \text{psi}$$

Assuming the gas causes a local gradient of 0.0866 psi/ft, the pressure due to a 2,000-ft gas column is

$$\Delta P_G = 173.2\ \text{psi}$$

Consequently, the gas pressure at 8,000 ft will be

$$P_G(8{,}000\ \text{ft}) = 4{,}330 - 173.2 = 4{,}155\ \text{psi}$$

The overpressure will then be

$$P_{op} = 4{,}155 - 3{,}464 = 691\ \text{psi}$$

The increase in mud specific weight to balance this overpressure at 8,000 ft should be

$$\Delta W_M = 1.66\ \text{lb/gal}$$

The pressure variation with depth is shown on the lower part of Figure 4-326.

The *fluid migration effects* occur when communication through a cement channel along the casing lets the fluids migrate from one zone to another. The upper zone is being "charged" by the lower zone. The overpressure in the upper zone may be very large.

Demonstration. A gas sand with normal pressure at 6,000 is charging a sand at 3,000 ft. The normal pressure at 5,000 ft is 2,603 psi. Assuming that the gas has caused an overburden gradient of 0.0433 psi/ft, then the pressure due to 3,000 ft of gas column is

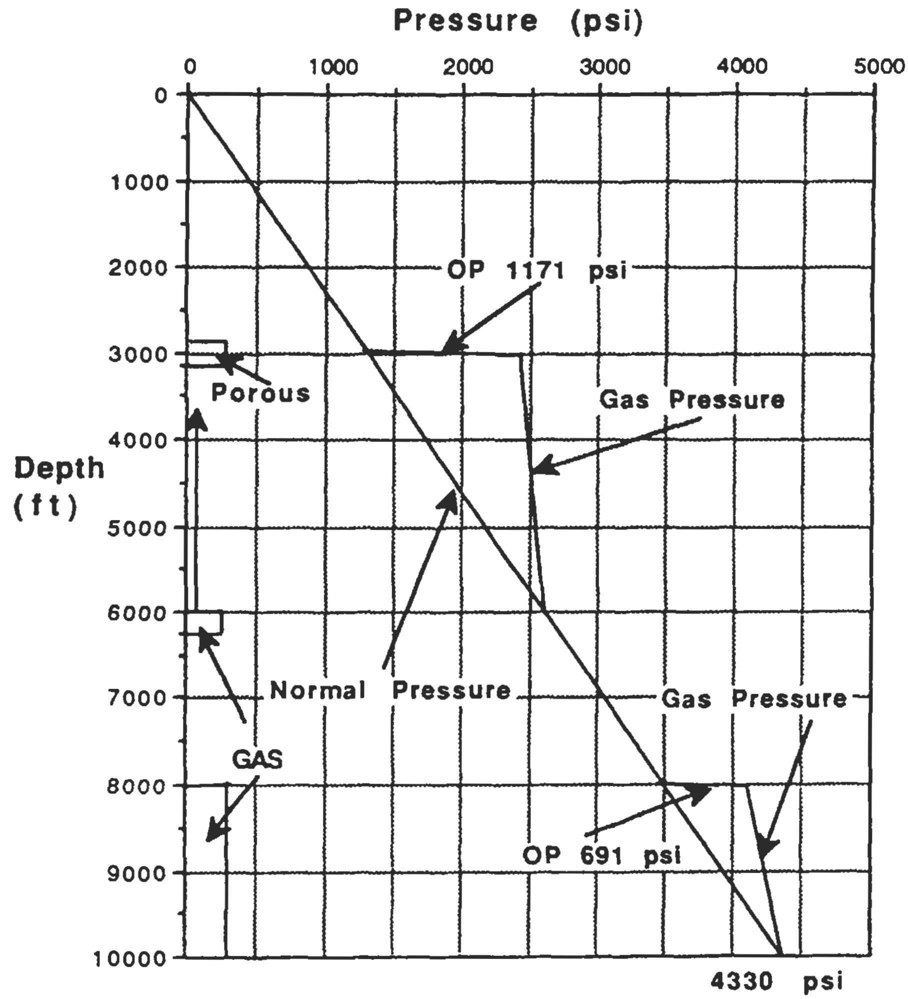


Figure 4-326. Schematic of pressure versus depth for a thick gas formation and for a channel communication behind casing.

$\Delta P_G = 130$ psi

The pressure at 3,000 ft will be

$P_G(3{,}000 \text{ ft}) = 2{,}603 - 130 = 2{,}473$ psi

The normal pressure at 3,000 ft:

$P_H = 1{,}302$ psi

The overpressure at 3,000 ft will then be

$$P_{op} = 2{,}473 - 1{,}302 = 1{,}171 \text{ psi}$$

The increase in mud weight to balance this overpressure at 3,000 ft should be

$$\Delta W_M = 7.51 \text{ lb/gal}$$

This large increase at a shallow depth would probably fracture the formation—such communication charging would pose very grave problems when drilling new wells.

Abnormal Formation Pressure Detection with Drilling Parameters. There are four groups of parameters that can be used for abnormal formation pressure detection:

- Drilling parameters for detection "at the bit"
- Logging parameters for detection 30 to 50 ft later
- Mud logging parameters for delayed detection
- Kicks

In 1966, Jorden and Shirley [120] introduced the *d-exponent method* designed to allow real-time pressure estimation while drilling by analyzing the drilling data, mainly: ROP, RPM and WOB. The equation is as follows:

$$ROP = K \times RPM \times \left(\frac{WOB}{D}\right)^d \tag{4-244}$$

where ROP = rate of penetration in ft/hr
RPM = rotary speed in rpm
WOB = weight-on-bit in klb
D = bit diameter in in.
K = rock drillability factor
d = weight-on-bit exponent

Assuming a drillability of one,

$$d = \frac{\ln\left(\dfrac{ROP}{60RPM}\right)}{\ln\left(\dfrac{12WOB}{1000D}\right)} \tag{4-245}$$

where ROP = rate of penetration in ft/hr
RPM = rotary rotation per minute
WOB = weight-on-bit in klb
D = bit diameter in in.

The d-exponent method provided a qualitative overpressure detection, because other parameters did not appear in the equation.

In 1971, Rehm and McClendon [121] introduced the *corrected d-exponent*, d_{cs}, for mud weight.

$$d_{cs} = \frac{G}{ECW} \times d \tag{4-246}$$

where G = normal gradient
ECW = equivalent weight with the mud used
d = d-exponent

The d_{cs} exponent is mostly used in Eaton's equation for a quantitative estimation of the pore pressure using the following equation [122]:

$$\frac{P_p}{Z_v} = \frac{P_{ob}}{Z_v} - \left[\frac{P_{ob}}{Z_v} - \left(\frac{P_p}{Z_v}\right)_n\right] \times \left(\frac{d_{cs}}{d_{csn}}\right)^{1.2} \tag{4-247}$$

where P_p = formation pore pressure in psi
Z_v = true vertical depth in ft
P_{ob} = overburden pressure in psi
$(P_p/Z_v)_n$ = normal gradient in psi/ft
d_{csn} = extrapolated normal corrected d-exponent, in normally compacted shales, not overpressured
d_{cs} = observed corrected d-exponent

Eaton's equation is the most commonly used relationship in the industry. It has provided good results in many areas of the world, even though the basic theory remains questionable. Bit wear corrections were attempted, but since bit wear could not be quantified prior to Anadrill T_D/R_D technique (see Equations 4-225 and 4-226), the bit wear correction in the d_{cs} technique is rarely used.

Another technique used for interpreting the d_{cs} curve is the *equivalent depth method.* The concept is as follows. The porosity, thus d_{cs}, observed in the overpressure section at a given depth, Z_2, is the same as would be expected at a shallower depth, Z_1, in a normal pressure environment. Terzaghi's relation (Equation 4-243) can be written at both depths with $\beta = 1$.

$$P_{ob1} = \sigma_{v1} + P_{F1} \text{ at depth } Z_1$$

$$P_{ob2} = \sigma_{v2} + P_{F2} \text{ at depth } Z_2$$

Since an equal d_{cs} implies equal σ_v we have

$$P_{F2} = P_{F1} + (P_{ob2} - P_{ob1}) \tag{4-248}$$

An example will be worked out later.

Other attempts to develop a pore pressure evaluation method from different drilling equations were made by Combs, by Bourgoyne and Young and by Bellotti and Giacca [101]. The three models follow the same general approach: a drilling equation is developed assuming that all variables are independent. The ROP is then normalized to eliminate the effect of each variable but the pore pressure. These equations attempt to take more variables into account.

Demonstration. Using the example in Figure 4-327, we will plot the corrected d-exponent data, expressed in d-units, in a linear scale, and a logarithmic scale [101].

Both scales give the same results; however, the correlations in normally pressure shales may be easier in the logarithmic scale. Applying Eaton's equation (Equation 4-247) at 13,000 ft, with a normal gradient of 0.465 psi/ft, an overburden gradient of 1 psi/ft, d_{csn} = 1.64 and d_{cs} = 1.17, we find

$$\frac{P_p}{Z_v} = 0.643 \text{ psi/ft}$$

$$P_p = 8{,}362 \text{ psi } (P_{pn} = 6{,}045 \text{ psi})$$

At 16,000 ft, where d_{csn} = 1.8 and d_{cs} = 0.8,

$$\frac{P_p}{Z_v} = 0.797 \text{ psi/ft}$$

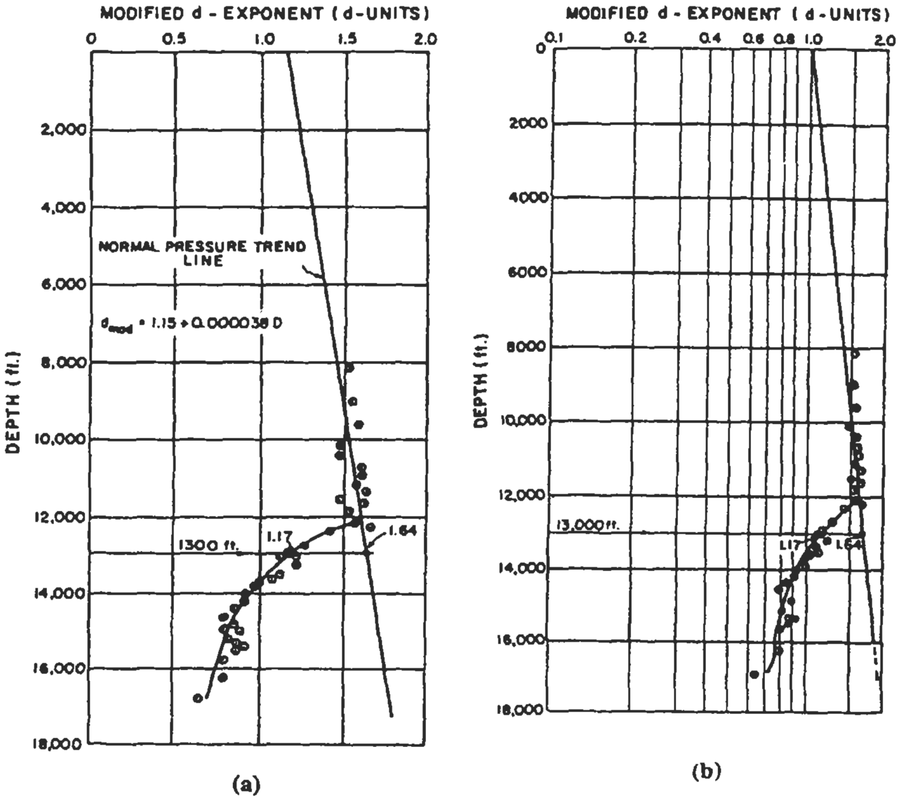


Figure 4-327. d_{cs} plots for a U.S. Gulf Coast well in shales: (a) linear scale; (b) logarithmic scale. *(Courtesy SPE* [101].*)*

P_p = 10,371 psi (P_{pn} = 7,440 psi)

Applying the equivalent depth technique at 13,000 ft, the same d_{cs} is read on the extrapolated normal trend at 4,000 ft. This gives

P_{F1} = 1,860 psi

P_{ob2} = 13,000 psi

P_{ob1} = 4,000 psi

P_{F2} = 10,860 psi

This technique is probably less accurate than the technique using Eaton's equation. This equation has been established statistically in the Gulf Coast.

No independent pressure measurements are available for this demonstration calculation.

When *downhole drilling measurements* are available, the techniques based on weight-on-bit, can be implemented with a better accuracy since the drag is eliminated. However, they are still based on a trend line determined in non-overpressured shales. Sometimes, such shales do not exist. Changes in lithology may affect the trend line determination. Early or real-time pressure detection can be made using the formation strength parameter FORS discussed previously (Equation 4-236). FORS is first normalized to a 9-lb/gal mud and 0 psi pressure using

$$FORS_{0,9} = FORS \times \frac{1}{\frac{2W_M}{9} - 1.0} \times \frac{1}{0.002P_M + 1.0} \tag{4-249}$$

where $FORS_{0,9}$ = FORS with 9 lb/gal and 0 psi
FORS = measured formation strength
W_M = mud weight in lb/gal
P_M = mud pressure in psi

$FORS_{0,9}$ is used to determine the excess effective porosity in the formation using an equation greatly simplified here:

$$\Phi_{op} = A - (B \bullet FORS_{0,9})$$

A and B are functions of many parameters such as non-shale-rock strength, volume of shale, shale rock strength, effective porosity, and overpressured shale rock strength. These parameters are first initialized with local knowledge, for example, the volume of shale, V_{cl}, of 65% sandstone FORS, etc. The calculated pore pressure is referred to as the "bit pore pressure." Petrophysical data coming later during drilling (gamma ray, resistivity, neutron density) are used to refine the parameter values for a more accurate pressure evaluation. The technique will be described in more detail later on. The total pore pressure is derived with the following equation:

$$P_p = P_H + k \bullet \sigma_v \bullet \log\left(1 + \frac{\Phi_{op}}{\Phi_n}\right) \tag{4-250}$$

where P_p = pore pressure in the overpressured zone
P_H = normal formation pressure
k = constant
σ_v = vertical stress
Φ_{op} = excess effective porosity
Φ_n = normal shale porosity at the depth of burial (well depth)

No trend line is needed to determine the formation pressure. However, petrophysical data are required to refine the results. A computer is mandatory to implement the technique.

Abnormal Formation Pressure Detection with Logging Parameters. The first logging parameter used for detecting overpressure zones is the *shale resistivity*. The first observation of the shale resistivity decrease in overpressured zone was made by Hottman and Johnson [123]. A normal resistivity increase trend exists in normally pressured shales. In overpressured shales the resistivity decreases sharply as shown in Figure 4-328. The shale resistivity, as seen in the amplified short normal, decreases from approximately 1 Ω • m at 9,000 ft to about 0.5 Ω • m at 10,000 ft. After setting casing at 10,150 ft, the mud weight had to be raised progressively to 18 lb/gal to keep the well under control.

Hottman and Johnson developed an empirical correlation to relate the ratio of resisitivities to the pore pressure gradient. In 1972, Eaton developed an empirical relationship that he modified in 1975 to the following [122]:

$$\frac{P_p}{Z_v} = \frac{P_{ob}}{Z_v} - \left[\frac{P_{ob}}{Z_v} - \left(\frac{P_p}{Z_v}\right)_n\right] \times \left(\frac{R_{sh}}{R_{shn}}\right)^{1.2} \tag{4-251}$$

where P_p = formation pressure in psi
Z_v = true vertical depth in ft
P_{ob} = overburden pressure in psi
$(P_p/Z_v)_n$ = normal gradient in psi/ft
R_{sh} = observed shale resistivity in Ω • m
R_{shn} = extrapolated shale resistivity in normally compacted shales in Ω • m

In another example from Bourgoyne, the shale conductivity (inverse of resistivity) was plotted as shown in Figure 4-329 [101].

Demonstration. Both resistivity and conductivity scales have been used with approximately the same success. At 13,000 ft, C_{sh} = 1,700 mS/m and C_{shn} = 369 mS/m. Assuming a normal gradient of 0.465 psi/ft and an overburden gradient of 1 psi/ft, we get

$$\frac{P_p}{Z_v} = 0.92 \text{ psi/ft}$$

$$P_p = 11{,}960 \text{ psi}$$

Using the Matthews and Kelly (1967) correlation [101] (shown in Figure 4-330) for South Texas, we get P_p = 10,660 psi using the Frio correlation, and P_p = 11,960 psi using the Vicksburg correlation.

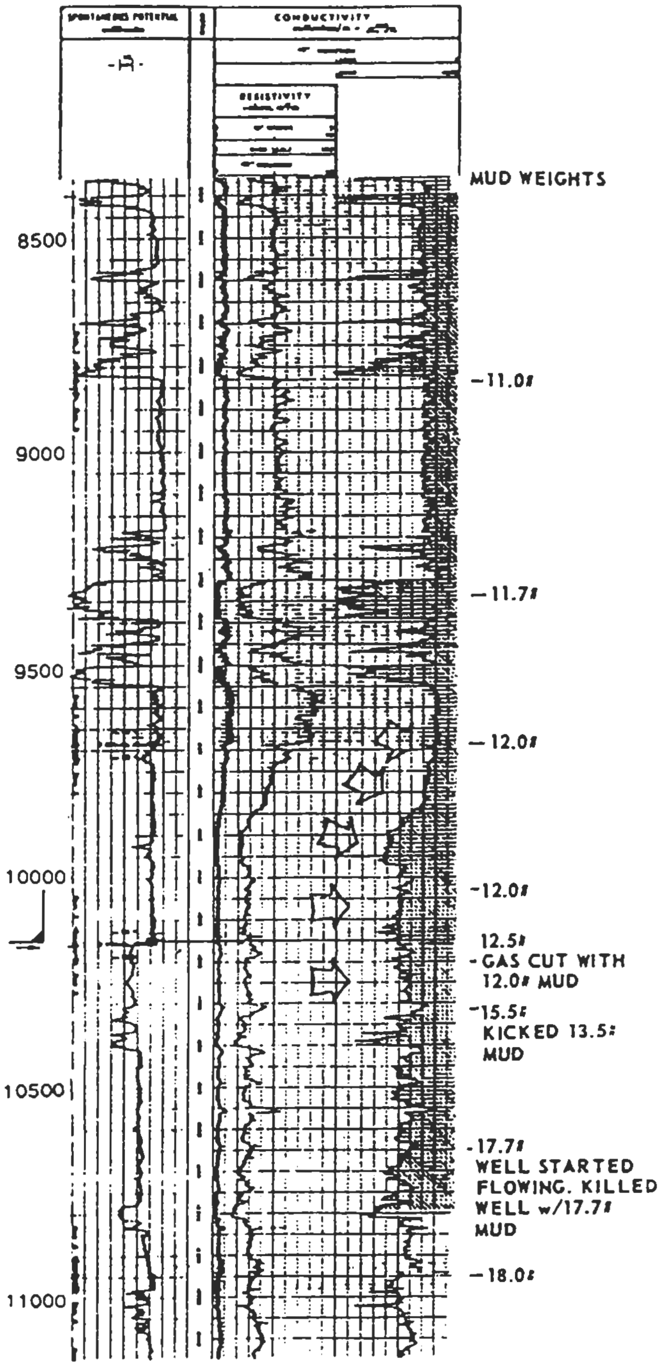


Figure 4-328. Example of resistivity decrease in overpressured shales.

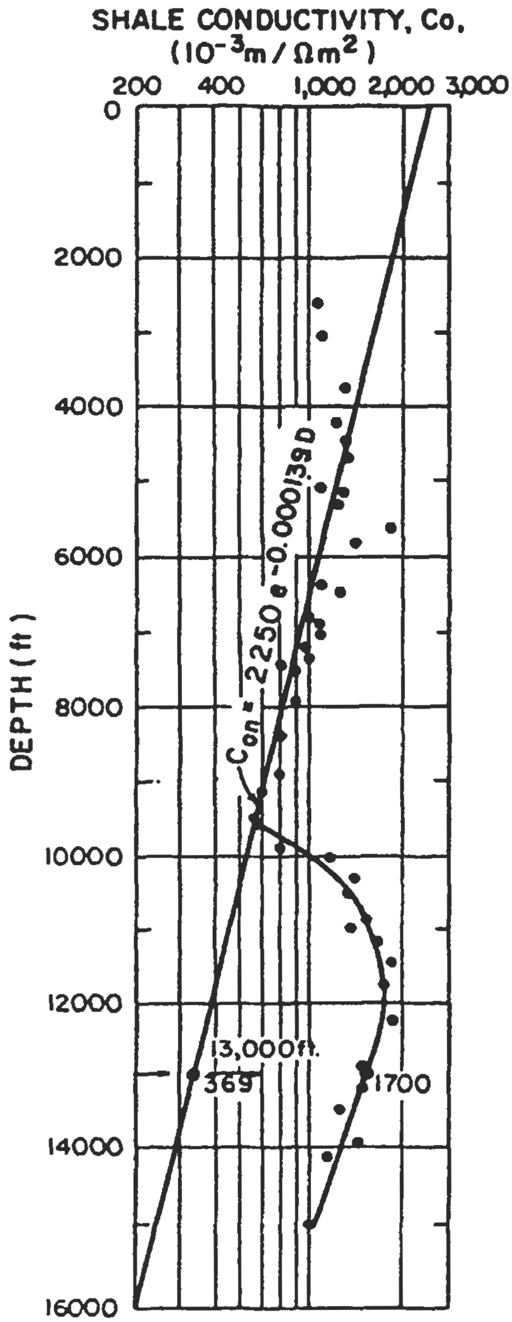


Figure 4-329. Example of shale conductivity plot in the Frio sand in South Texas. *(Courtesy SPE* [101].*)*

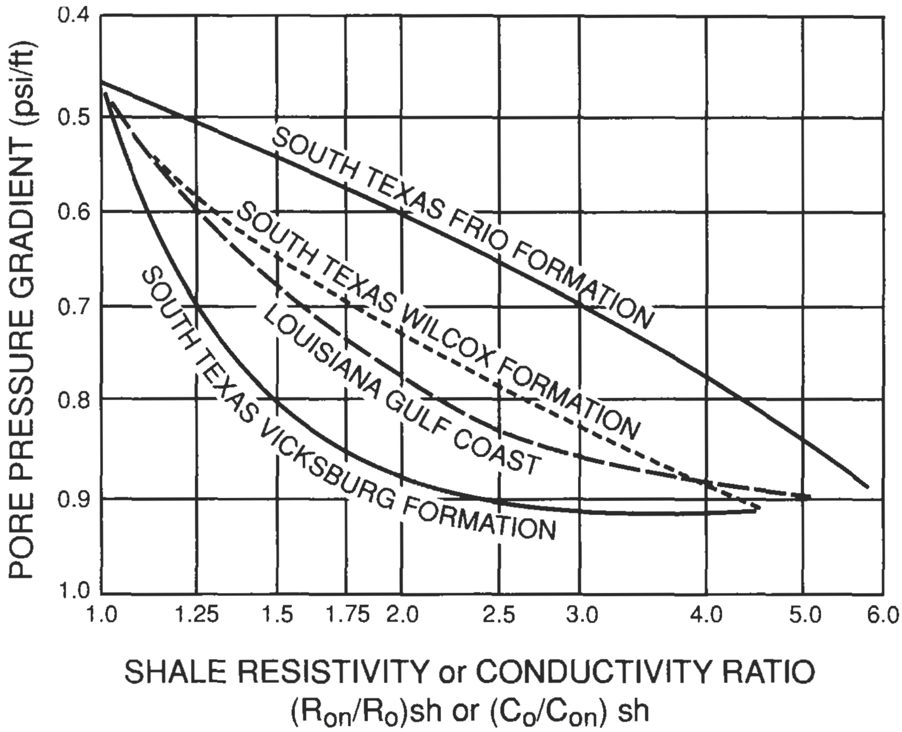


Figure 4-330. Matthews and Kelly relationship between formation pore pressure and shale resistivity for the South Texas Gulf Coast. *(Courtesy SPE* [101].*)*

A large difference in pressure prediction will occur if the formation is not well known. Several exponents might have to be used in Eaton's equation. Exponent 1.2 seems to match the Vicksburg formation as was seen in the preceding example.

Another technique based on shale resistivity was proposed by Alixant [124]. This technique does not use a normal trend line or empirical correlations and is based solely on the shale resistivity. The procedure is as follows:

- Select a shale interval. Using MWD data, gamma ray, and resistivity, determine R_{sh}.
- Estimate the formation temperature at the corresponding depth and determine bound water resistivity using the equation [124]

$$R_{wb} = \beta \bullet T^{-\gamma} \tag{4-252}$$

where R_{wb} = bound water resistivity in $\Omega \bullet m$
β = coefficient equal to 297.6
T = formation temperature in °F
γ = coefficient equal to −1.76

- Calculate the formation factor using

$$F = \frac{R_{sh}}{R_{wb}} \tag{4-253}$$

- Calculate the shale porosity using the Perez-Rosales equation [124]

$$F = 1 + M \bullet \frac{1 - \Phi}{\Phi - \Phi_r} \tag{4-254}$$

where F = formation factor
M = geometrical factor, usually 1.85
Φ = porosity in fraction
Φ_r = residual porosity in fraction, usually 0.1

- Determine the vertical effective stress, using the void ratio [$e = \Phi/(1 - \Phi)$] with the equation

$$\sigma_v = 10^{(e-e_i)/c_v} \tag{4-255}$$

where σ_v = vertical stress
e = void ratio
e_i = void ratio for σ_v = 1 psi
c_c = average constant compression index

Numerically:

$$\sigma_v = 10^{(e-3.84)/-1.1} \tag{4-256}$$

- Apply the Terzaghi relationship (Equation 4-243) to determine the formation pressure.

The procedure is summarized in Figure 4-331.

The technique has been tried in various areas of the world with good results provided that the shales are very clayey. Silt, sand or carbonates will lead to erroneous results. Figure 4-332 is an example of calculation in a North Sea well.

A good formation pressure agreement is seen at 9,500 ft with the wireline formation tester data.

The *sonic log* can also be used to detect overpressured zones. The sonic measurements until recently were available only on wireline. Now, MWD sonic tools have been developed adding one more parameter for overpressure detection while drilling. Two equations relating the formation porosity to the transit time are used:

The Wyllie equation (simplified form) [125]

$$\Delta t = V_{sh} \bullet \Delta t_{sh} \bullet V_{ma} \bullet \Delta t_{ma} + \Phi \bullet \Delta t_{mf} \tag{4-257}$$

The Hunt-Raymer-Gardner equation [126]

$$\frac{1}{\Delta t} = \frac{1}{\Delta t_{ma}} - \frac{\Phi}{0.625 \bullet \Delta t_{ma}} \tag{4-258}$$

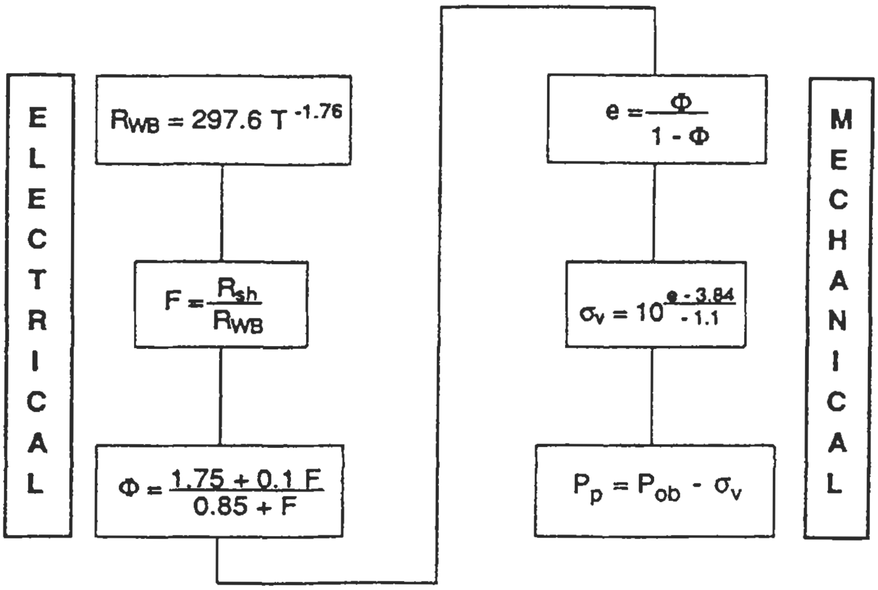


Figure 4-331. Summary of useful numerical equations in the Alixant's method. *(Courtesy Louisiana State University* [124].*)*

$$\Delta t_{ma} = \frac{V_{sh} \bullet \Delta t_{sh} + V_{ma} \bullet \Delta t_{ma}}{V_{sh} + V_{ma}} \qquad (4\text{-}259)$$

where Δt = transit time of the sonic wave in µs/ft
V_{sh} = volume of shale in fraction
Δt_{sh} = transit time in the shale in ms/ft
V_{ma} = volume of rock matrix in fraction
Δt_{ma} = transit time in the rock matrix µs/ft
F = porosity in fraction
Δt_{mf} = transit time in the mud filtrate in µs/ft

Equation 4-258 gives better results in most formations, particularly in unconsolidated formations. Both of these equations show a variation of the transit time with the porosity. Since the porosity increases in overpressured zones, the transit time increases. Figure 4-333 shows a typical plot for the shale formations of a well in Jefferson County, Texas.

Demonstration. A normal compaction trend can be seen down to 9,000 ft, then the overpressured zone is clearly outlined. At 12,000 ft the difference between the extrapolated normal trend transit time Δt_{shn} and the measured transit time Δt_{sh} is

$$\Delta t_{sh} - \Delta t_{shn} = 39\ \mu s/ft$$

The empirical correlation of Hottman and Johnson of Figure 4-334 can be used [101].

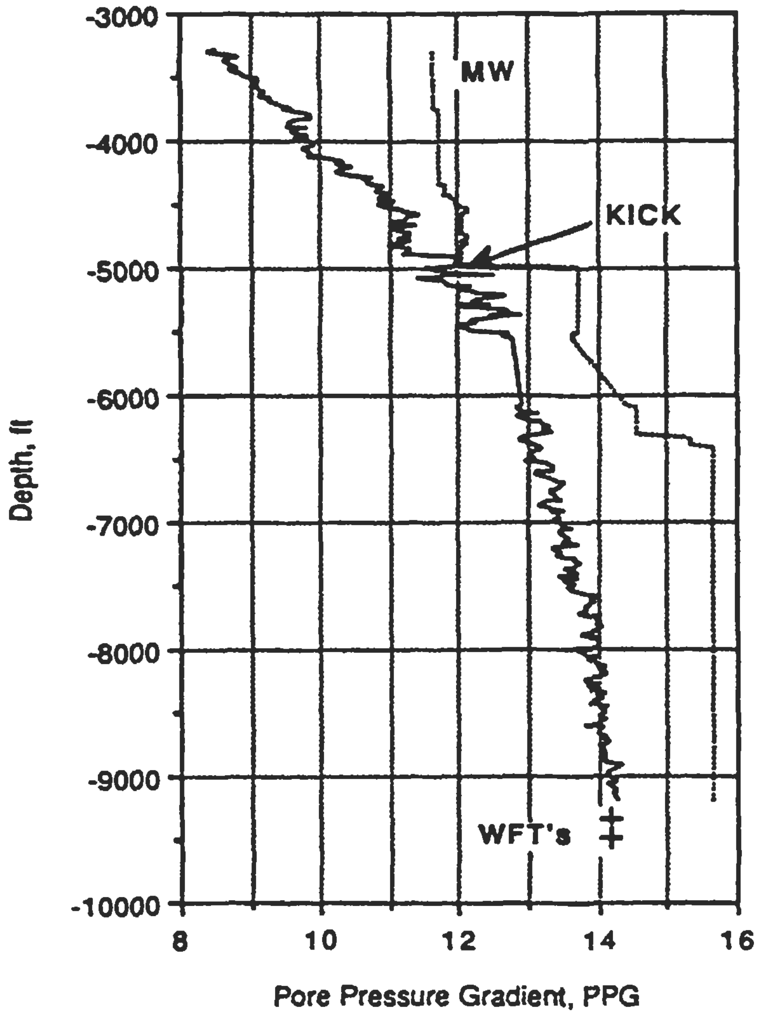


Figure 4-332. Pore pressure evaluation in a North Sea well. *(Courtesy Louisiana State University* [124].*)*

The formation pressure gradient corresponding to the difference of transit times is 0.93 psi/ft. The formation pressure is

$$0.93 \times 12{,}000 = 11{,}160 \text{ psi}$$

With the availability of the MWD/LWD drilling parameters, gamma ray resistivity, neutron-density Pe, a global approach of interpretation has been implemented by Anadrill when all measurements have been made in a given strata. A particular strata is first analyzed with the drilling data for a "pressure at the bit" estimate. Then, it is reanalyzed later when gamma ray resistivity data

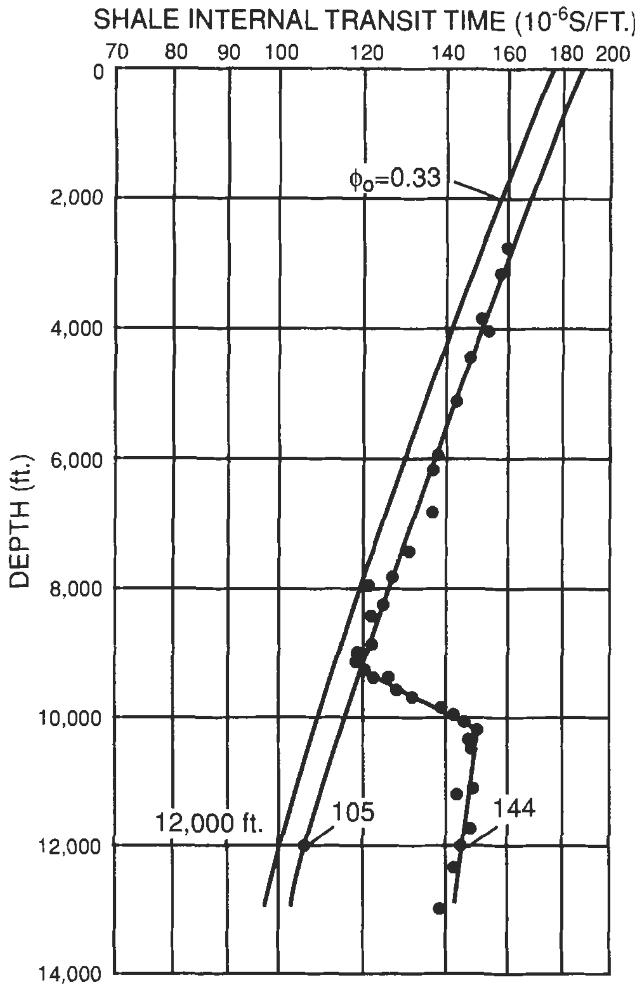


Figure 4-333. Example of sonic log data collected in the shale formations of a well in Jefferson County, Texas. *(Courtesy SPE* [101].*)*

become available. The full global program is implemented still later when the neutron-density Pe data also become available. The global technique used for pressure determination is similar to that used for wireline log interpretation.

The various volumes of rock matrix, shale, porosity, overpressure porosity and hydrocarbons are used to compute the various tool responses according to a model. The responses are compared to the measured values and a volume optimization is made to minimize the errors grouped in an incoherence function. The value of the incoherence function for the best fit determines the quality of the answer. Figure 4-335 is an example of formation pressure calculation as well as formation evaluation for lithology and fluid content.

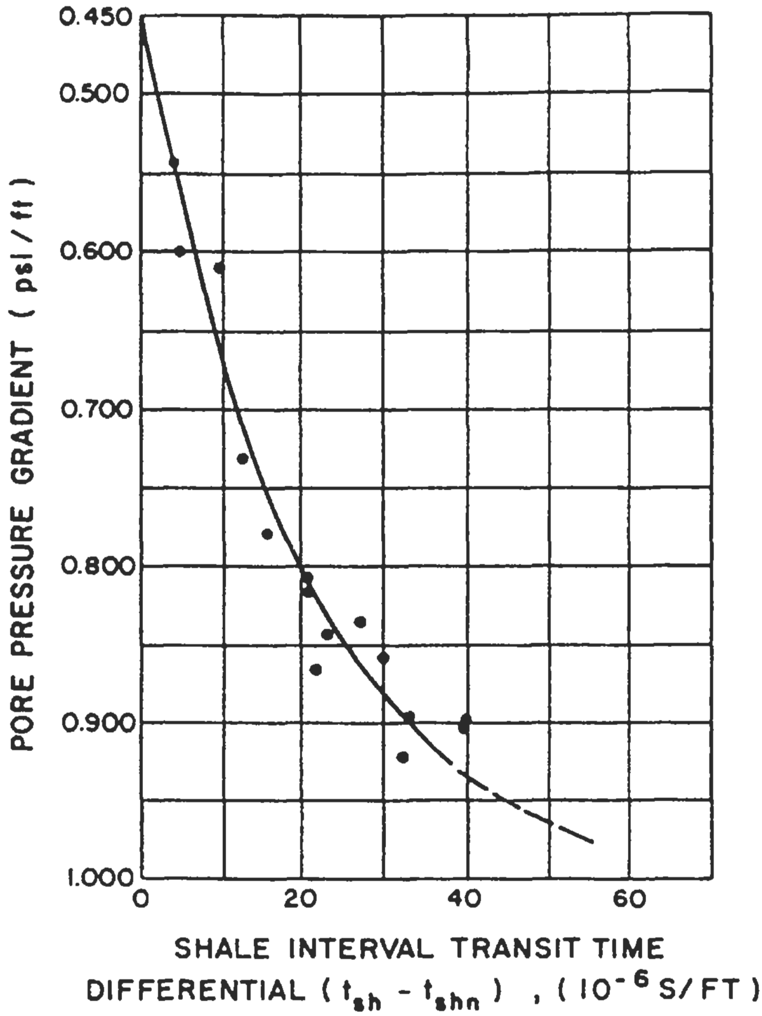


Figure 4-334. Empirical correlations between shale transit time and the pore pressure in the Gulf Coast. *(Courtesy SPE* [101].*)*

A standard error on the formation pressure of ±1.8 lb/gal (±216 kg/m³) can be seen clearly on the figure for the lower 55 ft when only formation strength data are available. For the next 38 ft, when resistivity and gamma ray become available, the standard error is ±0.9 lb/gal (±108 kg/m³), to reach ±0.35 lb/gal (±43 kg/m³) when all measurements are available.

Abnormal Formation Pressure Detection with Mud Logging Parameters. The main parameter used in mud logging for abnormal pressure detection is the *shale cutting bulk density.* Four techniques are commonly used:

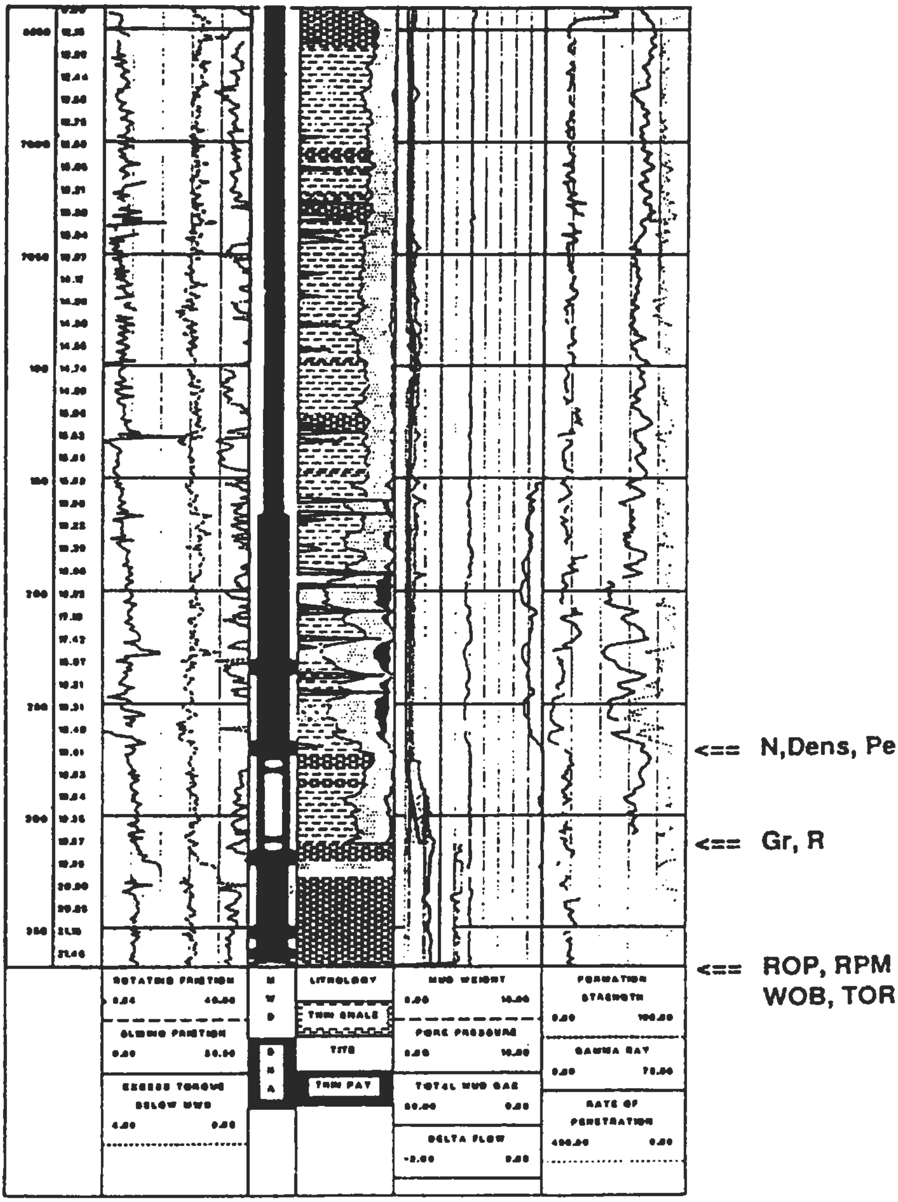


Figure 4-335. Computations of the formation pressure, lithology, and fluid content made with MWD formation-strength, LWD resistivity, gamma ray, neutron, and density measurements. *(Courtesy Anadrill* [113].*)*

- mercury pump technique
- mud balance
- variable density column
- microsol scale

In the *mercury pump technique,* the volume of a 25-g sample of shale is obtained by pressuring a chamber to 24 psig with and without the shale sample. In the *mud balance technique,* shale cuttings are added in the mud balance cup until they weigh the equivalent of a cup of water. The volume of shale can be expressed as

$$V_{sh} = \frac{\rho_w}{\rho_{sh}} \bullet V_c \tag{4-260}$$

where V_{sh} = volume of shale
ρ_w = density of water
ρ_{sh} = density of shale
V_c = cup volume

Adding water to fill the cup and measuring the density of the mixture gives

$$\rho_m = \rho_{sh} \bullet \frac{V_{sh}}{V_c} + \rho_w \bullet \frac{(V_c - V_{sh})}{V_c} \tag{4-261}$$

Substituting Equation 4-260 into 4-261 gives

$$\rho_{sh} = \frac{\rho_w^2}{2\rho_w - \rho_m} \tag{4-262}$$

A third technique used a *variable density column.* The variable density column is obtained by mixing gently a dense liquid, generally bromoform (density = 2.85 g/cm^3) with a light solvent, for example trichlorethane in a graduated cylinder. Calibration density beads are placed in the column for calibration. Shale cuttings are introduced carefully. They float at a level corresponding to their density.

The *microsol scale technique* is used to weigh a shale sample out of and in water, thus giving weight and volume. The results are plotted versus depth. Low shale densities (high porosity filled with water) indicate overpressured zone. A demonstration example is shown in Figure 4-336.

Demonstration. The transition zone is clearly indicated at 12,000 ft. The trend line is determined as an exponential function. The difference between the trend line value (2.48 g/cm^3) and the measured value (2.28 g/cm^3) at 13,000 ft can be correlated to the formation pressure gradient using the empirical correlation curve established by Boatman and shown in Figure 4-337 [101].

The formation pressure gradient corresponding to the density difference of 0.2 g/cm^3 is 0.86. The formation pressure is

$$P_p = 0.86 \times 13{,}000 = 11{,}180 \text{ psi}$$

The *shape of the shale cuttings* also gives a qualitative evaluation of the formation pressure. The cuttings are larger and more angular than normal. They

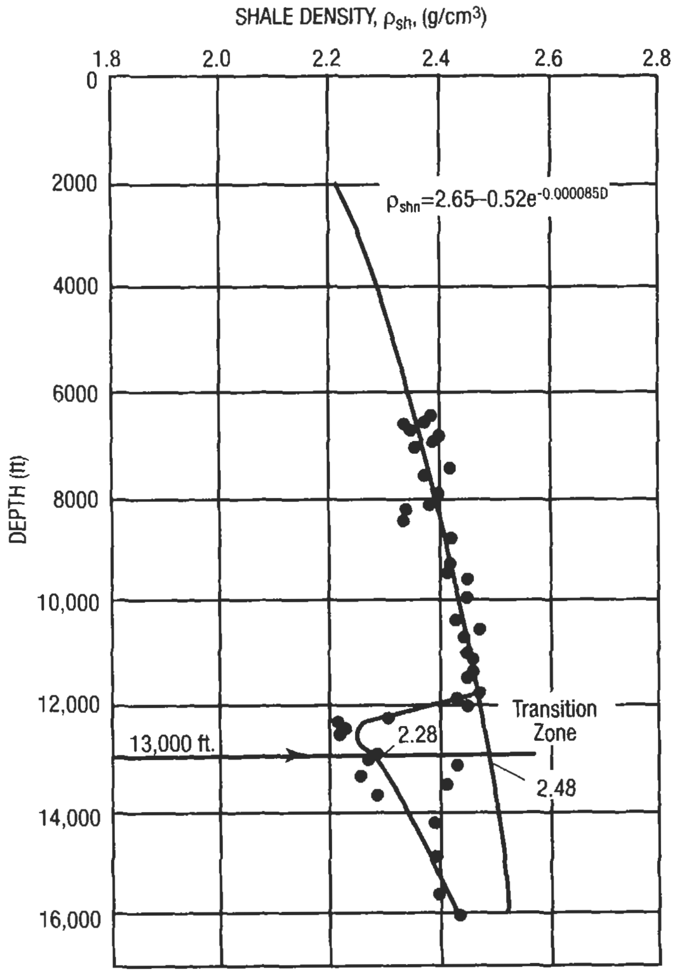


Figure 4-336. Example of shale cutting density plot. *(Courtesy SPE [101].)*

appear to be "exploded" into the borehole. Less crushing by the bit occurs as they rapidly leave the bit region.

The *gas content of the mud* is also an overpressure indicator. The background gas level (liberated and recycled gas) increases when drilling shales in underbalanced conditions. The connection gas (produced during connection when the bottomhole pressure decreases due to stopping the mud circulation, and working the drillstring) increases in underbalanced conditions. Finally, the trip gas (gas entering the mud column when the drillstring is out of the hole) also increases in underbalanced conditions. Bit swabbing may accentuate the increase.

Background, connection, and trip gas increase are indications that previously drilled zones are underbalanced and absolute control should be established before a trip is attempted.

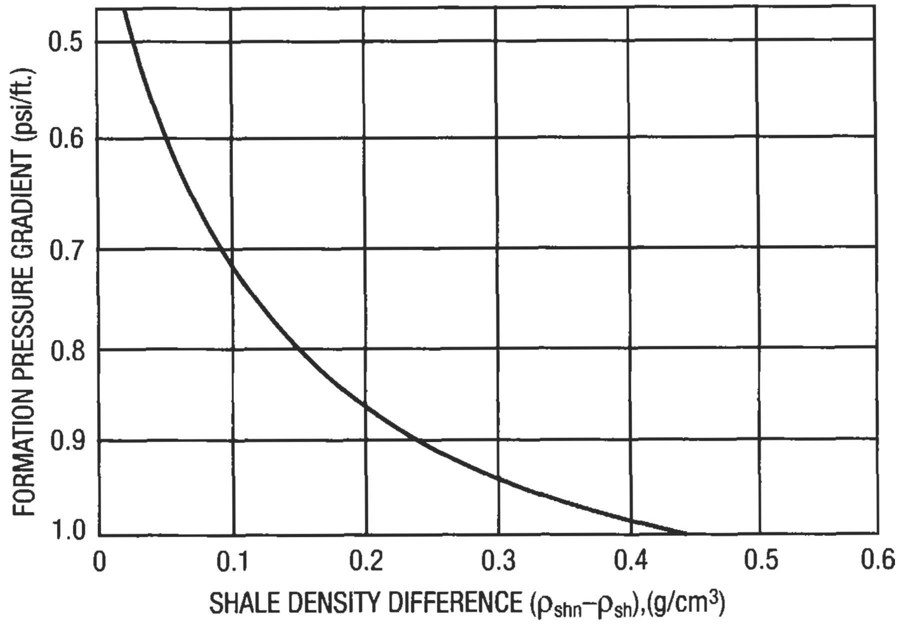


Figure 4-337. Boatman correlation between formation pressure gradient and bulk density of shale cuttings. *(Courtesy SPE* [101].*)*

Flowline temperature is also an overpressure indicator. In overpressured zones the formation temperature increases. The flow line temperature gradient (increase in temperature per 100 ft) can increase by 2° to 10° over the normal gradient. However, other effects (salt domes, lithology changes) may also cause gradient changes.

Salinity reversals or decrease of the formation brine salinity with depth are generally associated with overpressured formations. The mud salinity or chloride content reflects the formation water salinity if there is a close control over the mud properties and analyses.

Abnormal Formation Pressure Detection from Kicks. The kicks, or flow of formation fluids into the borehole, are the ultimate indication that thc well has encountered an overpressured zone. Kick detection during drilling usually is achieved by use of a pit-volume indicator and/or a flow indicator. The usual pit-volume alert is 10 barrels drilling fluid volume increase. A differential mud flow indicator can also be used to detect kicks more quickly.

Kick detection while tripping is done by monitoring the volume of mud added to keep the hole full. If the volume required to fill the hole is less than the volume of pipe removed, a kick may be in progress.

When a kick is detected while drilling or while tripping, well control operations should be started immediately.

Example 24: Overpressure Detection with Rate of Penetration

The mud logging log recorded in a well of the Gulf Coast is shown in Figure 4-338.

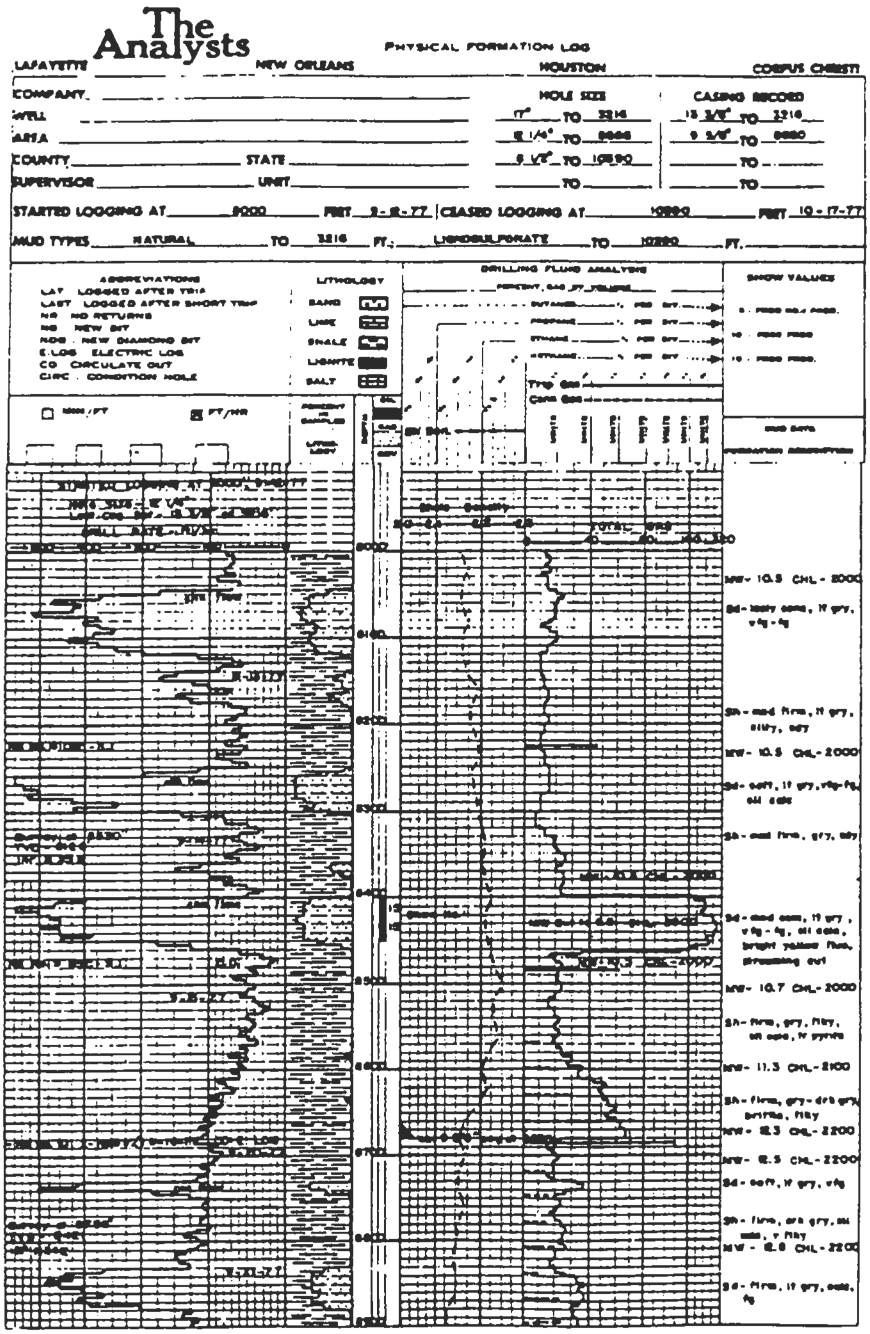


Figure 4-338. Example of mud log for interpreting ROP in terms of formation pressure. *(Courtesy The Analysts, now Anadrill.)*

The hole has been drilled with a $12\frac{1}{4}$-in. bit and approximately 25,000 lb weight down to 8,685 ft. Then a $8\frac{1}{2}$-in. bit was used with 17,000 lb weight to TD. Bit rpm was approximately constant and equal to 100 rpm for the whole section. The normal hydrostatic gradient for the area, G_H, is 8.4 lb/gal. The mud weight MW is shown on the log. The overburden gradient for the area G_{ob} is 1 psi/ft. [Note: 1 g/cm^3 = 62.4 lb/ft^3 = 8.345 lb/gal = 0.433 psi/ft]

1. Compute the d-exponent corrected for mud weight assuming that

   $$ECD = MW + 0.5 \text{ lb/gal}$$

   for the depths listed in the first column of Table 4-135 (average ROP for ± 10 ft at each depth).
2. Plot the d_c values versus depth on the graph as in Figure 4-339. Mark the slow shale points with a dot (•) and the fast sand points with a cross (×).
3. Plot the normal shale trend and determine the top of the high pressure zone if any. Is the sand at 8850 ft high pressured? Will it produce hydrocarbons? Why?
4. Compute the formation pressure gradient and formation pressure at 8,460 ft using the d_c-exponent technique.
5. Compute the fracturation pressure gradient and fracturation pressure at 8,460 ft assuming a Poisson ratio of 0.4.
6. Compute the formation pressure gradient and fracturation pressure at 8,800 ft using the d_c-exponent technique. Why was a casing set at 8,680 ft?
7. Compute the formation pressure of the formation at 8,400 to 8,450 ft. Will it be a hydrocarbon producer? Do we have enough information to know if it will produce oil or gas?

Solution

1. See Table 4-135:

$$dc = \frac{\ln\left(\dfrac{ROP}{60 \bullet RPM}\right)}{\ln\left(\dfrac{12 \bullet WOB}{1{,}000{,}000 \bullet D}\right)} \bullet \frac{G_H}{ECD}$$

2. See graph in Figure 4-339.
3. Top of high pressure zone at 8,500 ft; dc decreases, yes sand is high pressured; no, because no gas and no fluorescence.
4. $d_c = d_{cn}$; G_{fp} = 8.4 lb/gal; FP = 3,695 psig.
5. K_p = 0.667; GOB = 19.27 lb/gal; G FRAC = 15.65 lb/gal; P_{frac} = 6,885 psig.
6. d_c = 0.673; d_{cn} = 1.00; G_{fp} = 11.9 lb/gal; FP = 5,445 psig. To avoid fracturing openhole above when increasing mud weight in high pressure zone.
7. dc = dcn; above and below zone, no overpressure. FP = 3,680 psig at 8,425 ft. Yes, it will produce HC because gas show in mud. Oil because oil in cuttings, fluorescence.

Example 25: Overpressure Detection with Rate of Penetration

The data in the first six columns of Table 4-136 were taken in a mud logging log of a South Louisiana well.

Table 4-135
d_c Exponent Calculation Table

Z ft	ROP	rpm	WOB	D	GH	ECD	dc
8020	70	100	25000	12.25	8.4	11	0.9163
8060	600	100	25000	12.25	8.4	11	0.474
8220	65	100	25000	12.25	8.4	11	0.9315
8260	900	100	25000	12.25	8.4	11	0.3905
8330	60	100	25000	12.25	8.4	11	0.948
8420	800	100	25000	12.25	8.4	11	0.4148
8500	50	100	25000	12.25	8.4	11.2	0.968
8600	70	100	25000	12.25	8.4	11.8	0.8542
8680	120	100	25000	12.25	8.4	12.8	0.6921
8720	120	100	17000	8.5	8.4	13	0.6777
8800	110	100	17000	8.5	8.4	13.3	0.6772
8850	500	100	17000	8.5	8.4	13.3	0.4208

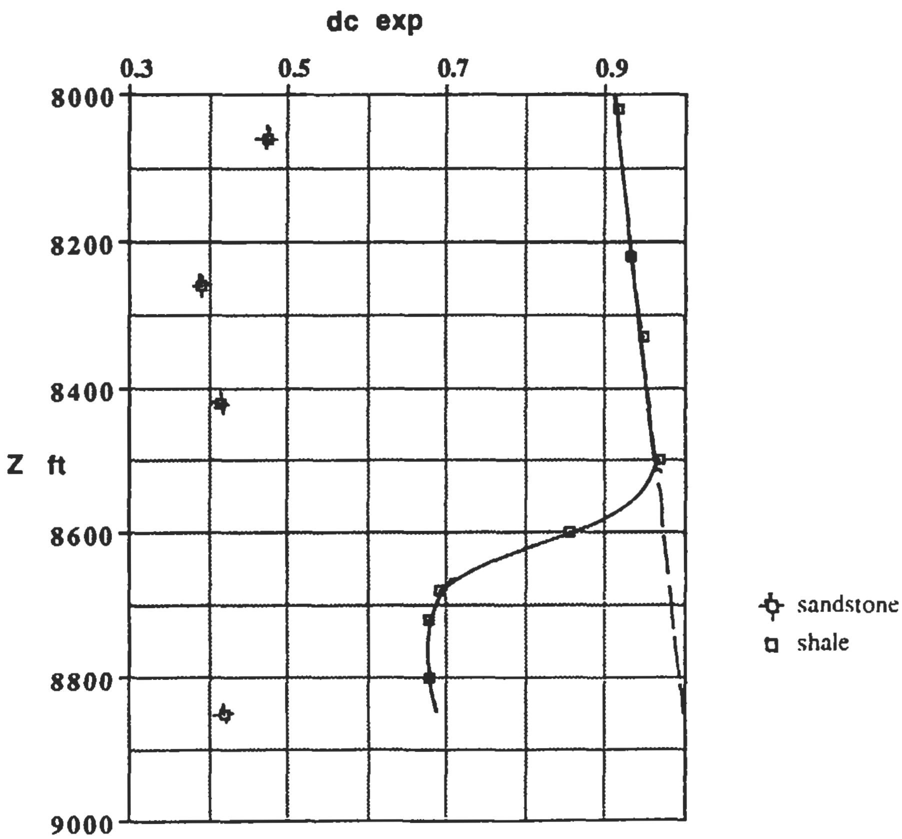


Figure 4-339. D-exponent graph: (a) grid; (b) plot of the results.

Table 4-136
d-Exponent and d_c-Exponent Corrected Calculation Table

Depth (ft)	Hole size (in)	ROP (ft/hr)	WOB (1000 lbs)	RPM (rpm)	Mud W. (lb/gal)	d-exp	d-exp.cor
4000	12	126.1	50	120	9.5	1.35	1.18
5000	9.875	49.4	40	110	9.5	1.62	1.41
6000	9.875	53.7	40	100	9.5	1.56	1.36
7000	9.875	16.7	30	70	9.5	1.67	1.46
8000	9.875	60.1	55	140	9.5	1.83	1.59
9000	9.875	77.6	60	150	9.7	1.82	1.55
10000	9.875	15.7	50	100	9.7	2.12	1.82
11000	9.875	27.6	50	80	10	1.84	1.53
12000	6.5	15.9	30	80	12	1.97	1.38
13000	6.5	33.1	35	100	12.2	1.90	1.30
14000	6.5	54.6	30	100	12.3	1.82	1.11
15000	6.5	5.3	15	50	12.3	1.77	1.21

1. Compute the d-exponent and d-exponent corrected in the last two columns of Table 4-136.
2. Plot the penetration rate, d-exponent, and d-exponent corrected versus depth.
3. Compute the pore pressure at 15,000 ft using Eaton's equation, a normal gradient of 0.453 psi/ft, and an overburden gradient of 1 psi/ft assuming the well vertical.
4. Should the mud weight be increased?

Solution

1. The d-exponent and d-exponent corrected have been computed in Table 4-136.
2. The plot of the results are shown in Figure 4-340.
3. At 15,000 ft: d_{cn} = 2.08; d_c = 1.2; P_p/Z_v = 0.729 psi/ft; P_p = 10,937 psi.
4. Equivalent mud weight: 14 lb/gal. The mud specific weight should be increased to 14.5 lb/gal to avoid a kick in the next permeable zone.

Example 26: Overpressure Detection with Shale Resistivity

The data of Figure 4-341 was taken from the log of a Humble Oil and Refining Co. well in Pecan Island, Louisiana.

1. Compute the formation pressure at 15,000 and 18,000 ft assuming a normal gradient of 0.465 psi/ft, an overburden gradient of 1 psi/ft and a vertical well. (Use Eaton's equation).
2. What mud weight should be used for drilling at 15,000 and 18,000 ft?
3. What will be the danger uphole if casing is not set around 12,000 ft?

Solution

1. The top of the overpressure zone is around 13,000 ft.
   15,000 ft:

   R_{shn} = 1.35; R_{sh} = 0.6 $\Omega \bullet$ m

   P_p/Z_v = 0.797 psi/ft; P_p = 11,955 psi

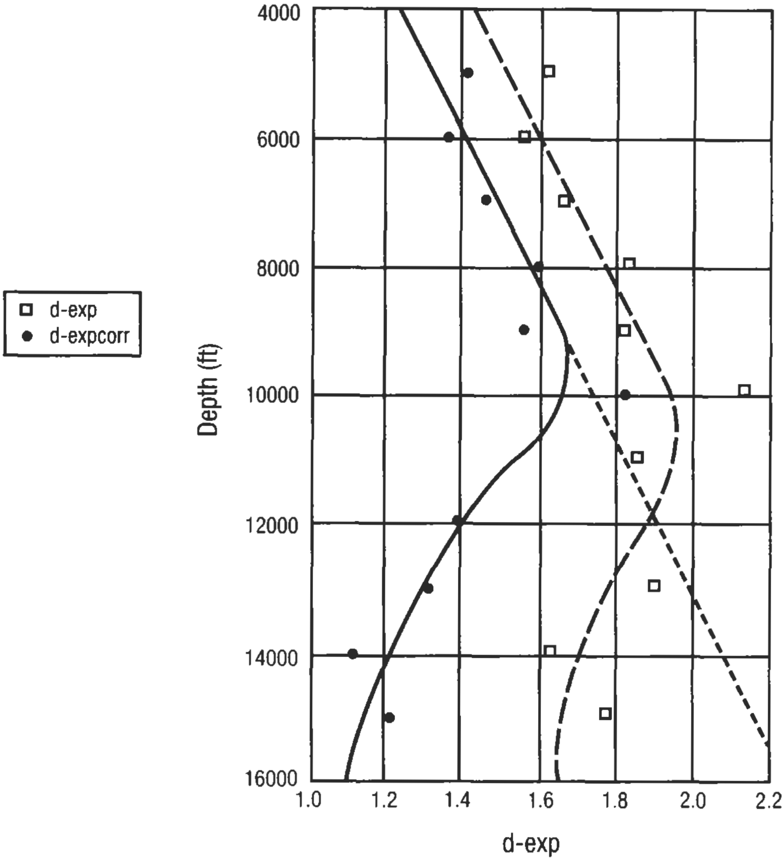


Figure 4-340. D-exponent graph: (a) grid; (b) plot of the d-exponent and of the d-exponent corrected.

18,000 ft:

R_{shn} = 1.67; R_{sh} = 0.6 $\Omega \bullet$ m

P_p/Z_v = 0.842 psi/ft; P_p = 15,180 psi

2. 15,000 ft:

0.797 psi/ft = 15.36 lb/gal. Recommended mud weight = 15.86 lb/gal.

18,000 ft:

0.842 psi/ft = 16.23 lb/gal. Recommended mud weight = 17.73 lb/gal.

3. The formation may fracture uphole.

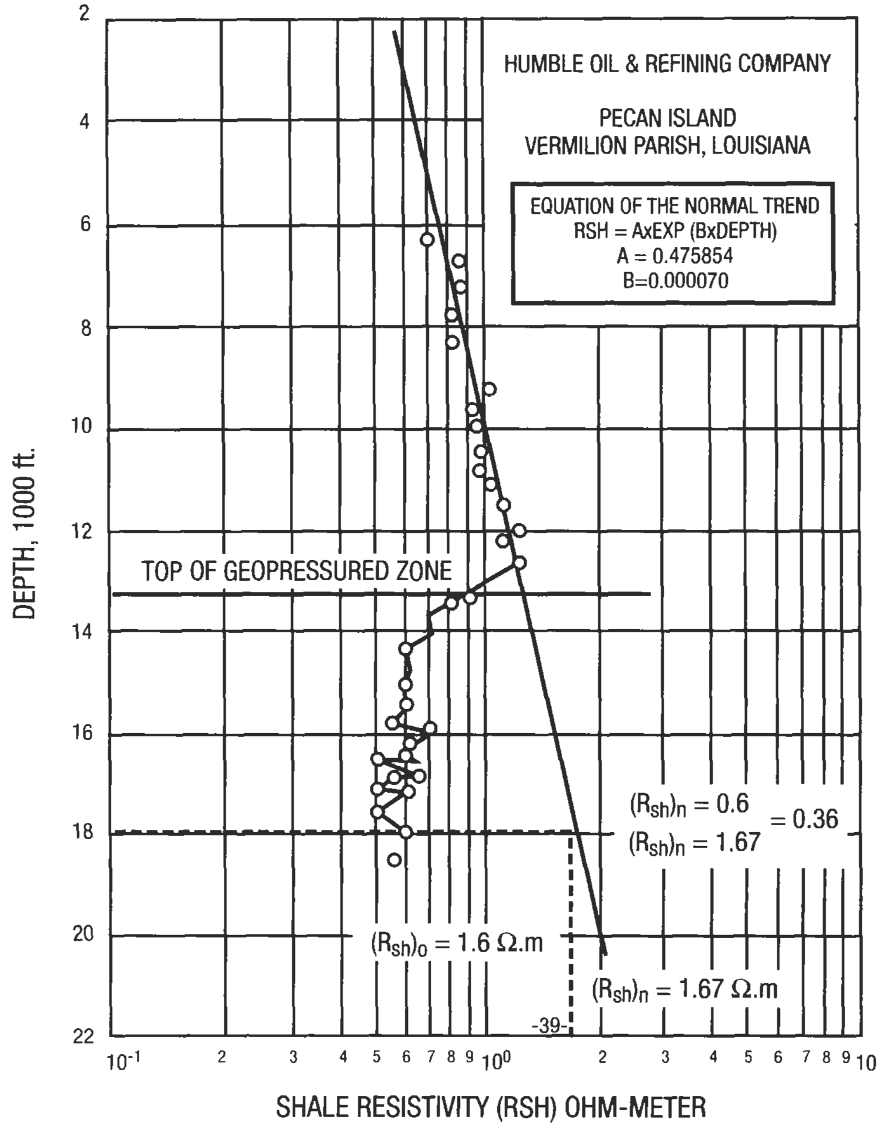


Figure 4-341. Plot of the resistivity of shales versus depth in a Gulf Coast land well (Louisiana).

Example 27: Overpressure Detection with Shale Resistivity

A shale resistivity plot versus depth is shown in Figure 4-342 has been prepared with electric log data for a well drilled in south Louisiana.

1. Estimate the depth of the top of the overpressured zone.
2. Compute the pore pressure using Eaton's equations, assuming a normal gradient of 0.465 psi/ft, an overburden gradient of 1 psi/ft at 14,000 ft and a vertical well.

Solution

1. Probable top of overpressure: 12,800 ft.
2. $R_{shn} = 1$; $R_{sh} = 0.7$; $P_p/Z_v = 0.651$ psi/ft; $P_p = 9.114$ psi.

Drilling Safety, Kick Alert

The safety priority at the well site must be given to personnel safety. The utmost care must be given to avoid casualties and injuries. The second priority is equipment safety. Breakdowns, fishing, and lost hole must be avoided or at least minimized.

Personnel Safety

The greatest danger is uncontrolled kick or blowout. When mud logging, MWD and LWD can play a major role to improve safety by preventing or minimizing the danger of kicks and blowouts. The conventional kick detection is done with the *mud pit level* measurement. The alert is generally given for a pit gain of 5 or 10 barrels. Another system is the *differential mud flowrate* measurement. If the difference outflow rate minus inflow rate increases, fluid is entering the borehole.

These techniques will work for gas, oil, or water inflow with any types of mud, water-base or oil-base. For gas inflow in oil-base mud, large amounts of gas go into solution at high pressures and reduce the flowrate increase. The change is less important, if detectable, with oil or water entry. The gas in solution in oil-base mud cannot be detected easily. The latest and most promising equipment is the *kick alert* (*Anadrill*). Figure 4-343 shows a schematic of "Kick Alert."

The kick alert detects the free gas over the whole annulus length by comparing the pump noise in the standpipe to the pump noise picked up a few feet below the flowline. The phase shift is correlated to the annulus sonic velocity, itself a function of the free gas.

The operational constraints are

- pumps must be operating;
- mud level must cover the annulus transducer;
- only gas influxes are detected;
- it works in water-base or oil-base muds;
- gas may be entering anywhere.

Another important feature of mud logging and MWD/LWD, is the capability of *overpressure zone detection*. As was discussed previously in this section, overpressure estimates can be done during drilling with many techniques and accuracy increases with the number of parameters available.

Drilling operations will never be 100% safe for the personnel but the various techniques available in mud logging, MWD and LWD greatly improve the safety on the drill site.

(text continued on page 1070)

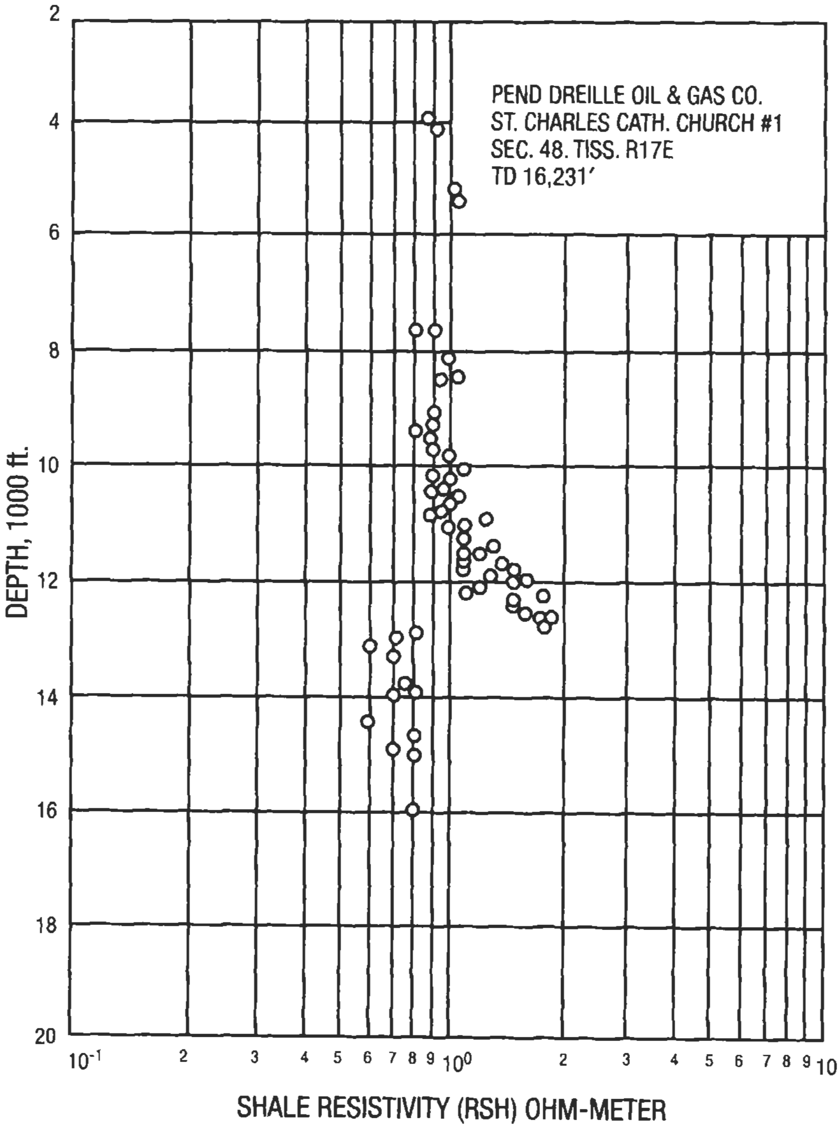


Figure 4-342. Plot of the resistivity of shales versus depth in a Gulf Coast land well (Louisiana).

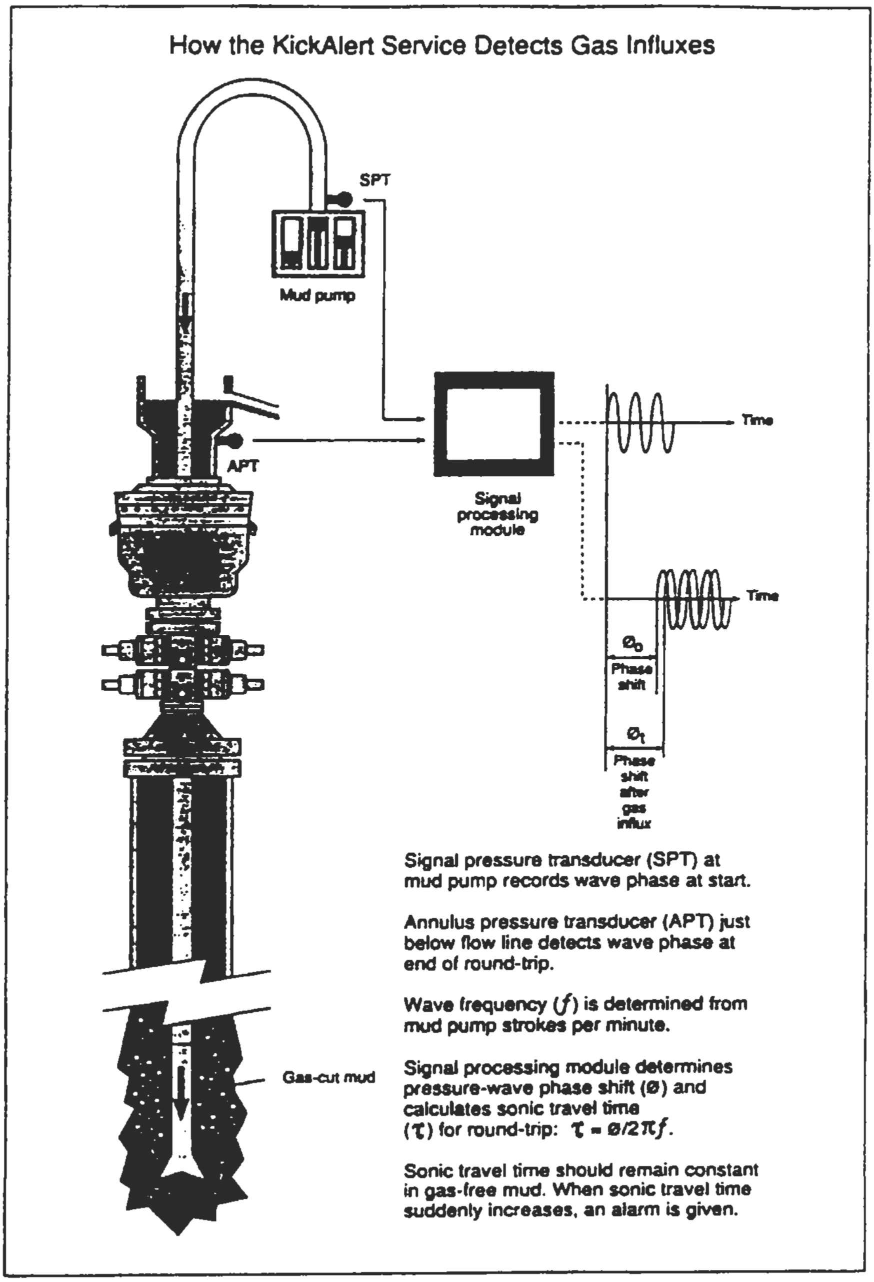


Figure 4-343. Schematic of the "Kick Alert" system of Anadrill. *(Courtesy Anadrill* [113].*)*

(text continued from page 1067)

Equipment Safety. The greatest risk for the equipment is sticking the drillstring. Before MWD, the drilling personnel had to rely on surface measurements only. The drilling parameters can now be measured practically "at the bit." Calculations of drag and friction can be carried out in real-time alerting the driller of any danger of pipe sticking.

Downhole flowrate measurements indicate drillstring washouts before the drillstring breaks and, consequently, avoiding expensive and dangerous fishing.

Bit efficiency calculated in real time gives an indication that the drill bit is damaged and that cones may be lost. Downhole vibration and shock measurements are also valuable indicators used to avoid damaging the drillstring.

All of these MWD measurements are definitively improving safety, minimizing breakdowns and making drilling more efficient.

Horizontal Drilling, Geosteering

To attain higher performances, the oil and gas companies are demanding greater drilling efficiency in conditions such as extended reach and horizontal drilling. The improved production and return on investment can be achieved from fewer wells, but better quality wells. Figures 4-344 and 4-345 show the theoretical vertical profile for a buildup to horizontal with respectively 1°/10 m (3°/100 ft) and 2°/10 m (6°/100 ft). In the first case, the distance below kick-off point (KOP) to reach horizontal is 570 m (2,870 ft) TVD, with a measured depth of 900 m (2,952 ft). In the second case, the corresponding lengths are 290 m (951 ft) and 450 m (1,476 ft).

To follow accurately the theoretical trajectory, MWD techniques must be used.

When the borehole is near horizontal, logging or surveying tools cannot be lowered by gravity anymore. They must be pumped down the drillpipes for directional measurements. Conventional logging has to be carried out by conveying the logging sondes downhole at the tip of the drillstring. The logging operation becomes long, expensive and dangerous. A much more efficient way is to survey the trajectory and record the logs while drilling. The logging data can be used to ascertain that the borehole is being drilled in the anticipated pay zone. If not, immediate remedial action is taken to "steer" the well towards the pay zone. The most advanced technique in use today is the *"geosteering" technique.*

Geosteering is usually done with a mud motor. A mud motor with bent sub allows changing of orientation and inclination without pulling the drillstring out. Steering is done by rotating it a small angle.

In *classical geosteering* the sensors for inclination, azimuth, drilling parameters, and logging are located above the mud motor and the distances may be in the order of those shown in Figure 4-296 that is 30 ft or more above the drill bit. Although radial measurements can be performed to verify that the borehole is being drilled in the pay zone, it is often too late to make a correction and the borehole leaves the pay zone.

The new geosteering system offers measurements "at the bit" (below the mud motor) of inclination, rpm, azimuthal gamma ray, azimuthal resistivity, and bit resistivity as seen in Figure 4-298. The signals are transmitted electromagnetically to the MWD sub located above the mud motor, then relayed to surface with the standard mud pressure transmission system. To summarize, the following is recorded just above the drill bit:

- inclination
- revolution per minute

- azimuthal gamma ray
- azimuthal resistivity
- bit resistivity

Above the mud motor, the following is recorded:

- weight-on-bit
- torque
- inclination
- azimuth
- tool face
- neutron
- density
- Pe

Other parameters, such as alternator voltage (for flowrate), temperature and pressure, can also be monitored.

A frame of 16 words of information can be updated in 27 s at a rate of 6 bits/s. Even at 100 ft/hr of drilling rate (never reached in horizontal drilling), updating would occur every 0.75 ft.

An example of three horizontal wells drilled in a 2 m (6 ft) in the North Sea is shown in Figure 4-346.

Well No. 1 was drilled with inclination and azimuth data only. The sensors were located above the mud motor. Only a short section (63 m; 207 ft) was drilled in the reservoir.

Well No. 2 was "geologically" steered by adding gamma ray and resistivity capability. Only a short section is out of the reservoir, making a total of 168 m (552 ft) in the reservoir.

Well No. 3 was steered with the new geosteering system. A smooth trajectory was obtained with the whole interval in the reservoir. The last section was dipped intentionally to investigate the lower reservoir.

We see that horizontal drilling can be carried out satisfactorily if the following is available:

- sophisticated MWD/LWD technology
- computer capability
- positive displacement motor

Suggestions have been made recently to drill *horizontal branches* in the horizontal portion of the horizontal wells. If this technique is developed, the drainage capacity of horizontal wells will be even better.

Types of Directional Drilling

Horizontal drilling is performed using "long radius" of curvature to reach horizontal: 1 to 2°/10 m (3 to 6°/100 ft). Attempts have been made for years to improve production with medium, short and ultrashort radius of curvature. The sketch of Figure 4-347 shows the four types of curvatures [127].

Table 4-137 gives the range of values of radius, angle change with depth, and usual horizontal lengths drilled in each case.

Ultrashort radius wells are usually called "drainholes." They are drilled with special equipment and completed with $1\frac{1}{4}$ to $2\frac{1}{2}$-in. tubing. The tubing is

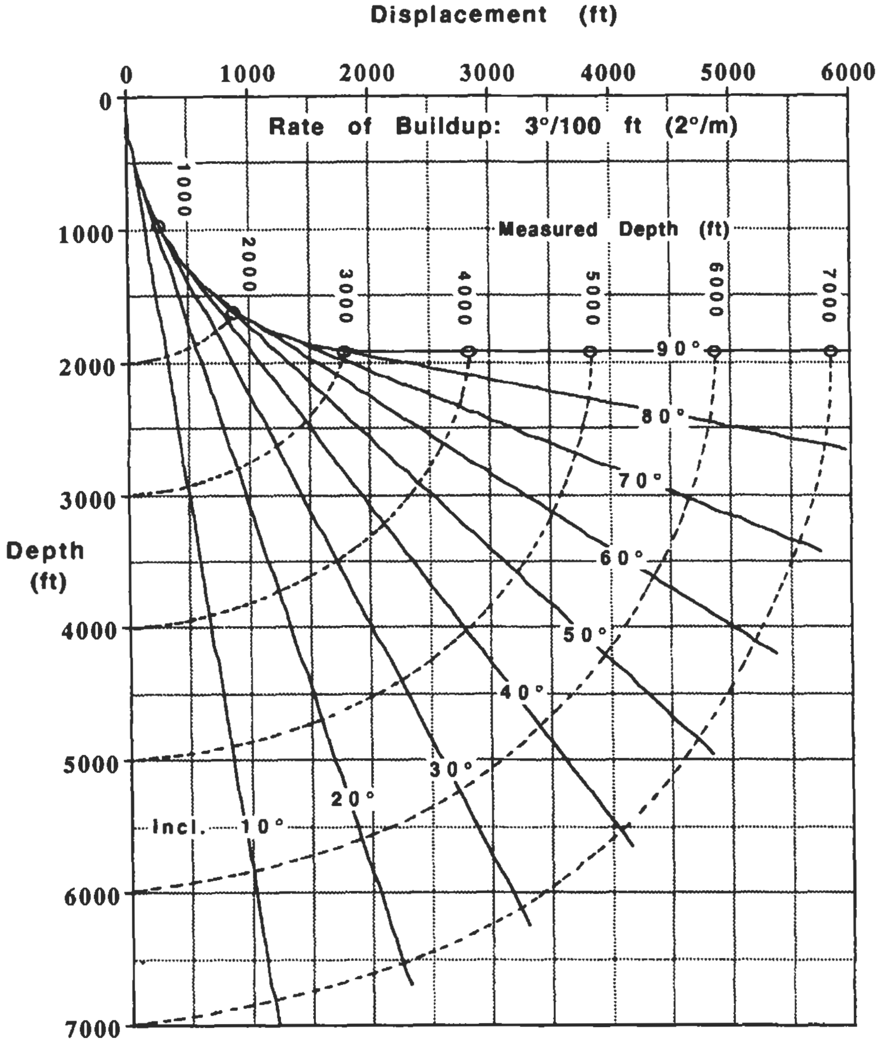


Figure 4-344. Theoretical vertical profile for a buildup rate of 1°/10 m (3°/100 ft) for a well reaching horizontal. *(Courtesy Inst. Fr. du Petr.)*

perforated and severed where it reached the main vertical hole. No MWD or LWD operations are carried out in these drainholes.

Short radius wells are usually drilled from a cased or uncased vertical well. Articulated drill collars are used to drill to 90° or beyond. A second stabilized assembly is used to drill the rest of the hole, usually in $4\frac{3}{4}$ or $6\frac{3}{4}$-in. diameter.

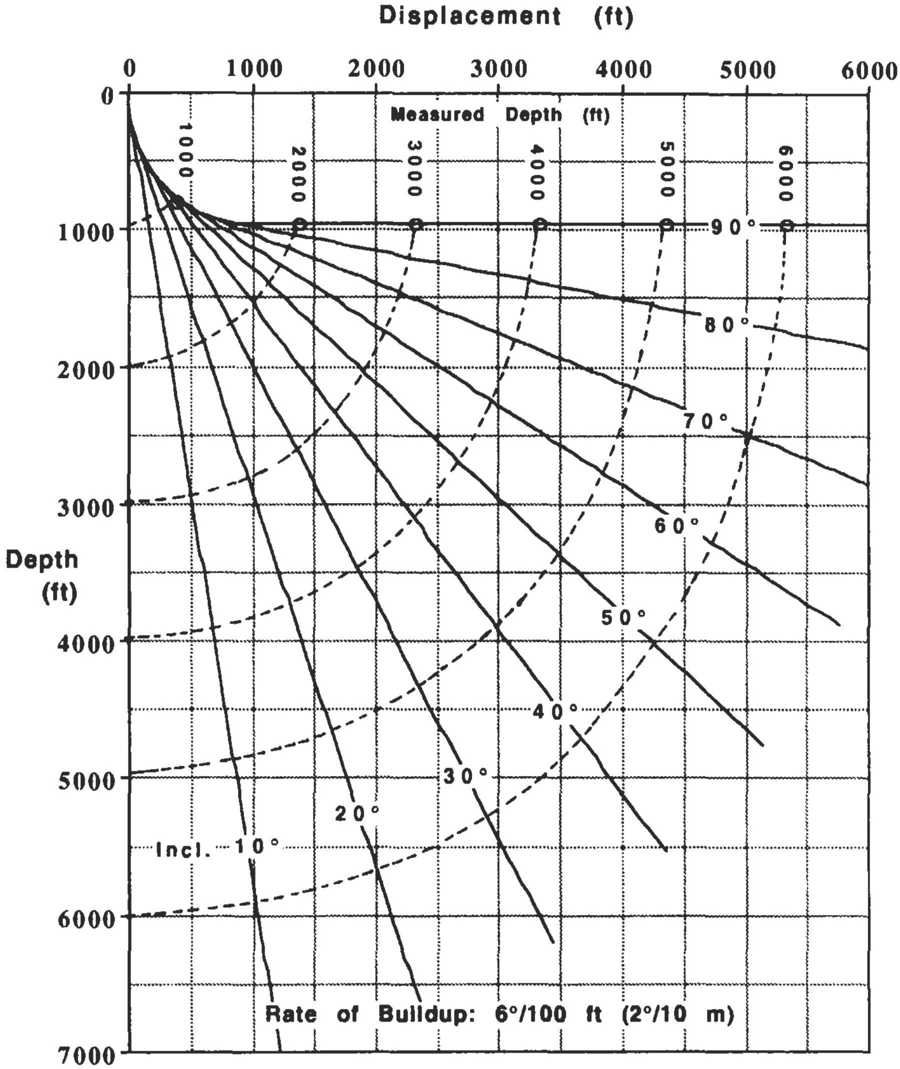


Figure 4-345. Theoretical vertical profile for a buildup rate of 2°/10 m (6°/100 ft) for a well reaching horizontal. *(Courtesy Inst. Fr. du Petr.)*

No MWD or LWD equipment exist to date to log these wells. However, service companies are developing equipment for MWD purpose in 40-ft (12-m) curvature radius or 1.5°/ft (4.6°/m). The equipment consists of articulated mud motors and inclination and azimuth sensors.

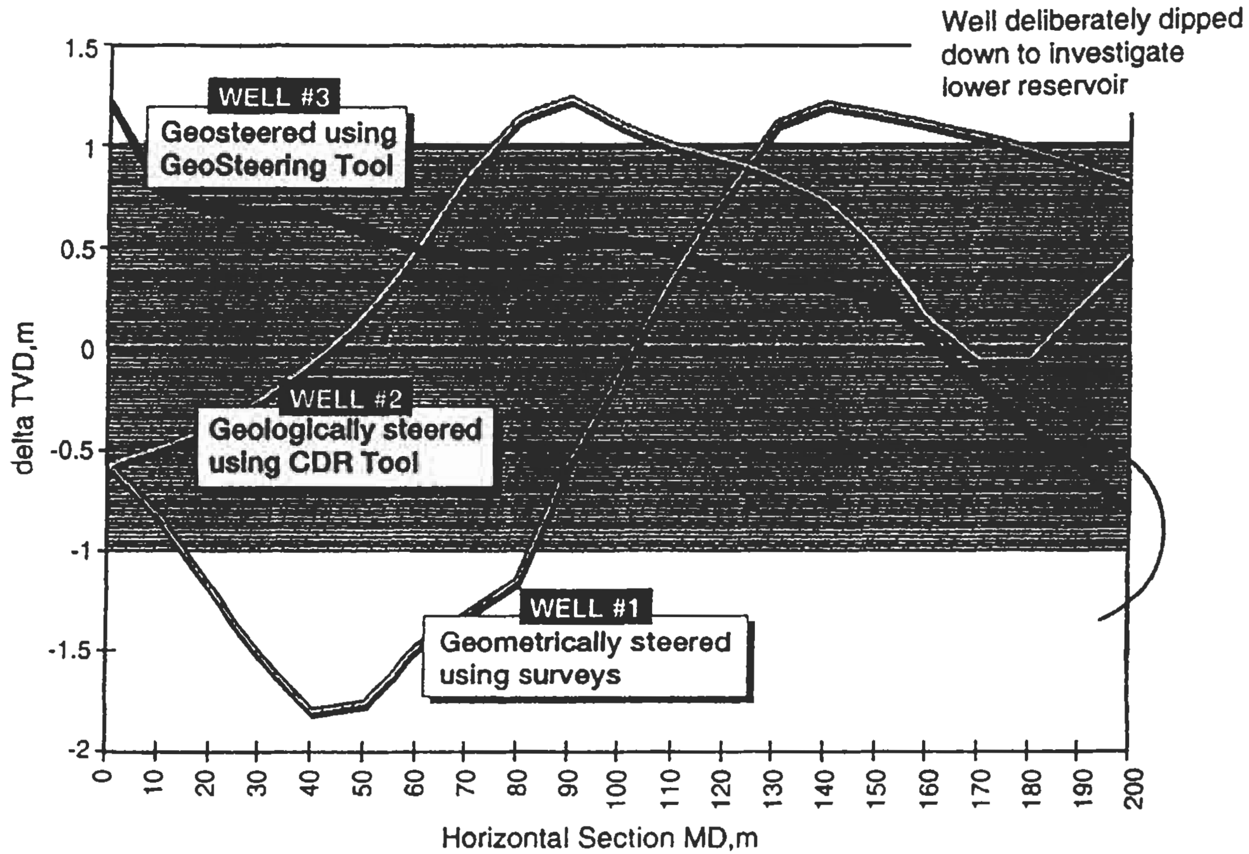


Figure 4-346. North Sea geosteering example. *(Courtesy Anadrill* [113].*)*

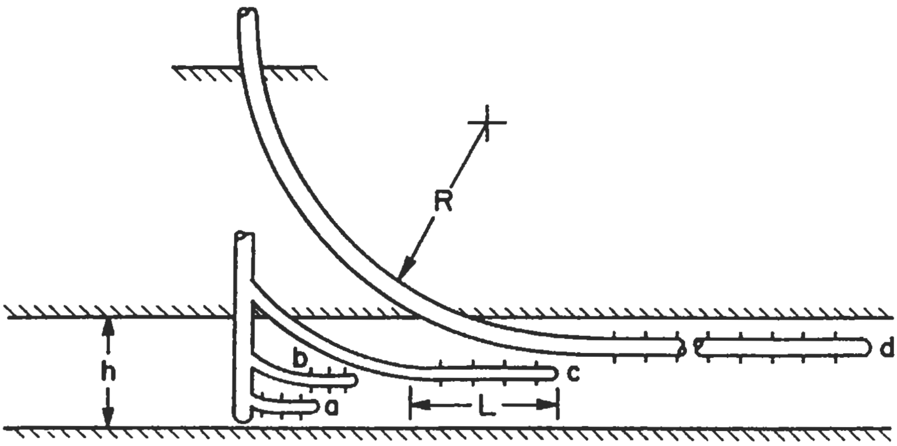


Figure 4-347. Schematics of different types of wells or drains: (a) ultrashort radius; (b) short radius; (c) medium radius; (d) long radius. (From Ref. [127].)

Table 4-137
Characteristics of Horizontal Wells

(a) Metric Units

Radius	R (m)	°/10 m	L (m)
Ultrashort	0.3–0.6	—	30–60
Short	6–12	91–46	30–240
Medium	90–240	6–2.3	300–1,200
Long	≥300	≤2	300–1,200

(b) English Units

Radius	R (ft)	°/10 ft	L (ft)
Ultrashort	1–2	—	100–200
Short	20–40	280–140	100–800
Medium	300–800	19–7	1,000–4,000
Long	≥1,000	≤6	1,000–4,000

Medium radius and long radius wells are drilled with conventional oilfield tools. Both MWD and LWD are usable in these wells. Downhole motors are mostly used in medium radius wells to avoid fatigue of the BHA. Long radius wells have been drilled with both mud motors and rotary techniques.

Air drilling operations can also be utilized for directional drilling and MWD/LWD technologies. Due to the absence of reliable downhole pneumatic motors, this technology is not yet fully developed.

Example 28: Overpull Estimation

Figure 4-348 shows the proposed trajectory of a horizontal well. The planned mud weight is 12 lb/gal water base. The drillstring will be as follows:

- 200 ft of drill collars:
  - $6\frac{3}{4}$-in OD
  - 2.81-in. ID
  - weight 149 lb/ft
  - displacement 1.537 gal/ft
- 4,000 ft of heavy drillpipes:
  - 3.5-in. OD
  - weight 37.7 lb/ft
  - displacement 0.387 gal/ft
- 4,904 ft of regular drillpipes:
  - 3.5-in. OD
  - weight 16.57 lb/ft
  - displacement 0.2236 gal/ft

The drag coefficient is 0.3.

1. Compute the overpull with a straight trajectory in the horizontal plan at TD.
2. Compute the overpull with the horizontal trajectory shown in Figure 4-348 at TD. Assuming that the direction change of 40° is at the junction of the regular drill pipes and heavy drill pipes. (The capstan-effect will be neglected.)

Solution

1. Drill collar drag:
   Weight in air: 200 × 149 = 29.8 klb
   Displaced mud weight: 200 × 1.537 × 12 = 3.689 klb
   Weight in mud: 29.8 − 3.689 = 26.11 klb
   Drag: 26.11 × 0.3 × sin 90° = 7 klb
   Heavy drillpipe drag:
   1,804 ft horizontal
   Weight in air: 1,804 × 37.7 = 68 klb
   Displaced mud weight: 1,804 × 0.387 × 12 = 8.378 klb
   Weight in mud: 59.622 klb
   Drag: 59.622 × 0.3 × sin 90° = 17.887 klb
   2,196 ft at 62° inclination
   Weight in air: 2196 × 37.7 = 82.789 klb
   Displaced mud weight: 2196 × 0.387 × 12 = 10.198 klb
   Weight in mud: 72.591 klb
   Drag: 72.591 × 0.3 × sin 62° = 19.228 klb
   Drillpipe drag:
   3,904 ft at 62° (includes buildup)
   Weight in air: 3904 × 16.37 = 63.908 klb
   Displaced mud weight: 3904 × 0.2236 × 12 = 10.475 klb
   Weight in mud: 53.433 klb
   Drag: 53.433 × 0.3 × sin 62° = 14.153 klb
   Total drag (overpull) = 7 + 17.887 + 19.228 + 14.153 = 58.268 klb
2. Forces due to drag below the direction change: 7 + 17.887 + 19.228 = 44.115 klb

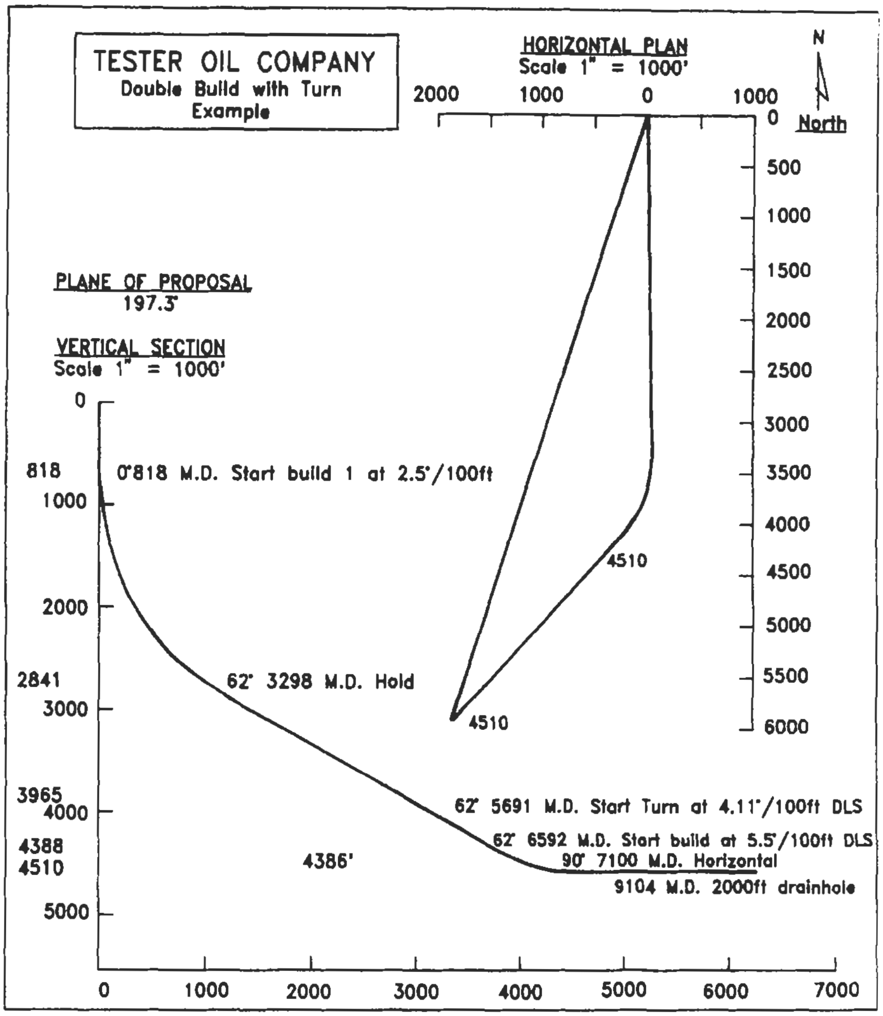


Figure 4-348. Proposed trajectory of a horizontal well.

Force acting on the dog leg (approx.) = 2 × 44.115 × cos ((180 − 40)/2) = 30.176 klb
Drag due to dog leg force: 30.176 × 0.3 = 9.052 klb
Total drag (overpull): 58.268 + 9.052 = 67.32 klb

Comparison of LWD Logs with Wireline Logs

The best scientific study comparing the LWD logs with the wireline logs has been done in the framework of a project sponsored by the oil and service companies in Conoco test facility, Kay County, Oklahoma in 1990.

Five service companies have field tested both their commercial and prototype MWD/LWD petrophysical sensors in oil-base mud and freshwater-base mud test wells. The data recorded were compared to equivalent wireline measurements. The effect of the rate of penetration (ROP) on some devices was investigated. The water-base mud well was drilled to 3.035 ft with $7\frac{7}{8}$-in. bit. The oil-base mud well was drilled to 2,400 ft with an $8\frac{3}{4}$-in. bit.

Each MWD company ran a full suite of MWD/LWD logs that were compared to "benchmark" wireline logs obtained by averaging the wireline logs with the least standard deviation errors. The data have been analyzed in the oil and service companies [128] and at Louisiana State University (LSU) where two master of science theses were completed utilizing this data [129,130]. The general conclusions of the LSU studies are as follows:

Gamma Ray. A linear relationship is generally established when comparing LWD and wireline gamma ray logs. Therefore, the LWD gamma ray data can be used with confidence as a replacement of wireline gamma ray for formation evaluation. Furthermore, LWD gamma ray logs generally have a better bed resolution than their wireline equivalent.

Resistivity. The 2-MHz LWD amplitude and phase-shift resistivity logs match the wireline deep and medium induction very well. Excellent results are obtained when the invasion is not severe (less than 40 in. in diameter) and in formations 20 $\Omega \cdot$m or less.

The focused resistivity log offered by one of the service companies is very sensitive to borehole diameter and can be used only in a qualitative manner in its present form.

Neutron. The neutron porosity values recorded with most tools match closely the wireline thermal neutron logs in the lower porosity ranges (under 25 porosity units). In high porosity zones, the LWD neutron porosities lie between thermal and epithermal wireline values.

In all zones the discrepancies between LWD and wireline porosity data are in the range of one to five porosity units. Thus, the LWD neutron data are suitable for formation evaluation.

Density. The best data are obtained with stabilizer-type tools. In good borehole sections, a close match between the LWD data and the wireline data was found. Discrepancies of less than 0.05 g/cm^3 were generally noticed.

Washouts, rugosity, and drillstring wobbling (or vibration) will affect the LWD density readings. However, the LWD density data are generally suitable for formation evaluation.

Photoelectric Effect (Pe). Only one service company was offering a commercial Pe log. The readings of the LWD tool were very sensitive to washouts. For a qualitative lithology identification of the strata, the LWD Pe curve is satisfactory.

In conclusion, the logs available now with LWD are perfectly suitable for a good basic formation evaluation in all types of formations. It should be possible to complete a well successfully with LWD data alone in most instances.

Comparison of MWD Data with Other Drilling Data

Before MWD, most drilling data were recorded at the surface. Mud logging data, the only downhole data, were available with a time delay corresponding to the time required by the mud to reach the surface.

Drilling surface data such as weight-on-bit and torque were difficult to interpret because they were loosely related to downhole values. MWD for the first time in the history of drilling gives values of parameters measured at the bit or close by. Rock strength, bit wear, drag and friction can be calculated in real time. Shocks, temperature and pressure can also be measured.

Sophisticated programs and codes are now available to interpret all these data in real time. The result is a more efficient and safer drilling process.

DIRECTIONAL DRILLING

Glossary of Terms Used in Directional Drilling

The glossary of terms used in directional drilling [131] has been developed by the API Subcommittee on Controlled Deviation Drilling under the jurisdiction of the American Petroleum Institute Production Department's Executive Committee on Drilling and Production Practice.

The most frequently used terms are listed below.

Angle of buildup Rate of change (°/100 ft) of the inclination angle in the section of the hole where the inclination from the vertical is increasing.

Angle drop-off Rate of change (°/100 ft) of the inclination angle in the section of the wellbore that is decreasing toward vertical.

Angle of inclination (angle of drift) The angle, in degrees, taken at one or at several points of variation from the vertical as revealed by a deviation survey; sometimes called the inclination or angle of deviation.

Angle of twist The azimuth change through which the drillstring must be turned to offset the twist caused by the reactive torque of the downhole motor.

Anisotropic formation theory Stratified or anisotropic formations are assumed to possess different drillabilities parallel and normal to the bedding planes with the result that the bit does not drill in the direction of the resultant force.

Azimuth Direction of a course measured in a clockwise direction from 0° to 360° as North; also bearing.

Back-torque Torque on a drillstring causing a twisting of the string.

Bent sub Sub used on top of a downhole motor to give a nonstraight bottom assembly. One of the connecting threads is machined at an angle to the axis of the body of the sub.

Big-eyed bit Drill bit with one large-sized jet nozzle, used for jet deflection.

Bit stabilization Refers to stabilization of the downhole assembly near the bit; a stabilized bit is forced to rotate around its own axis.

Borehole direction Refers to the azimuth in which the borehole is heading.

Borehole directional survey Refers to the measurements of the inclinations, azimuths and specified depths of the stations through a section of borehole.

Bottom-hole assembly (BHA) Assembly composed of the rock bit, stabilizers, reamers, drill collars, subs, etc., used at the bottom of the drillstring.

Bottomhole location Position of the bottom of the hole with respect to some known surface location.

Bottomhole orientation sub (BHO) A sub in which a free-floating ball rolls to the low side and opens a port indicating an orientation position.

Build-and-hold wellbore A wellbore configuration where the inclination is increased to some terminal angle of inclination and maintained at that angle to the specified target.

Buildup That portion of the hole in which the inclination angle is increased.

Clearance Space between the outer diameter of the tool in question and the side of the drilled hole; the difference in the diameter of the hole and the tool.

Clinograph An instrument to measure and record inclination.

Closed traverse Term used to indicate the closeness of two surveys; one survey going in the hole and the second survey coming out of the hole.

Corrective jetting runs Action taken with a directional jet bit to change the direction or inclination of the borehole.

Course The axis of the borehole over an interval length.

Course bearing The azimuth of the course.

Crooked hole Wellbore that has been inadvertently deviated from a straight hole.

Crooked-hole area An area where subsurface formations are so composed or arranged that it is difficult to drill a straight hole.

Cumulative fatigue damage The total of fatigue damage caused by repeated cyclic stresses.

Deflection tools Drilling tools and equipment used to change the inclination and direction of the drilled wellbore.

Departure Horizontal displacement of one station from another in an east or west direction.

Deviation angle See "Angle of inclination."

Deviation Control Techniques

- **Fulcrum technique** Utilizes a bending movement principle to create a force on the bit to counteract reaction forces that are tending to push the bit in a given direction.
- **Mechanical technique** Utilizes bottomhole equipment which is not normally a part of the conventional drillstring to aid deviation control. This equipment acts to force the bit to turn the hole in direction and inclination.
- **Packed-hole technique** Utilizes the hole wall to minimize bending of the bottomhole assembly.
- **Pendulum technique** The basic principle involved is gravity or the "plumb-bob effect."

Directional drilling contractor A service company that supplies the special deflecting tools, BHA, survey instruments and a technical representative to perform the directional drilling aspects of the operation.

Direction of inclination Direction of the course.

Dogleg Total curvature in the wellbore consisting of a change of inclination and/or direction between two points.

Dogleg severity A measure of the amount of change in the inclination and/or direction of a borehole; usually expressed in degrees per 100 ft of course length.

Drift angle The angle between the axis of the wellbore and the vertical (see "Inclination").

Drainholes Several high-angle holes drilled laterally from a single wellbore into the producing zone.

Drag The extra force needed to move the drillstring resulting from the drillstring being in contact with the wall of the hole.

Drop off The portion of the hole in which the inclination is reduced.

Goniometer An instrument for measuring angles, as in surveying.

Gyroscopic survey A directional survey conducted using a gyroscope for directional control, usually used where magnetic directional control cannot be obtained.

Hole curvature Refers to changes in inclination and direction of the borehole.

Hydraulic orienting sub Used in directional holes, with inclination greater than 6°, to find the low side of the hole. A ball falls to the low side of the sub and restricts an orifice, causing an increase in the circulating pressure. The position of the tool is then known with relation to the low side of the hole.

Hydraulically operated bent sub A deflection sub that is activated by hydraulic pressure of the drilling fluid.

Inclination angle The angle of the wellbore from the vertical.

Inclinometer An instrument that measures a position angle of deviation from the vertical.

Jet bit deflection A method of changing the inclination angle and direction of the wellbore by using the washing action of a jet nozzle at one side of the bit.

Keyseat A condition wherein the borehole is abraded and extended sideways, and with a diameter smaller than the drill collars and the bit; usually caused by the tool joints on the drillpipe.

Kickoff point (kickoff depth) The position in the wellbore where the inclination of the hole is first purposely increased (KOP).

Lead angle A method of setting the direction of the wellbore in anticipation of the bit walking.

Magnetic declination Angular difference, east or west, at any geographical location, between true north or grid north and magnetic north.

Magnetic survey A directional survey in which the direction is determined by a magnetic compass deflecting the earth's magnetic field.

Measured depth Actual length of the wellbore from its surface location to any specified station.

Mechanical orienting tool A device to orient deflecting tools without the use of subsurface surveying instruments.

Methods of orientation

- **Direct method** Magnets embedded in the nonmagnetic drill collar are used to indicate the position of the tool face with respect to magnetic north. A picture of a needle compass pointing to the magnets is superimposed on the picture of a compass pointing to magnetic north. By knowing the position of the magnets in the tool, the tool can be positioned with respect to north.
- **Indirect method** A method of orienting deflecting tools in which two survey runs are needed, one showing the direction of the hole and the other showing the position of the tool.
- **Surface readout** A device on the rig floor to indicate the subsurface position of the tool.
- **Stoking** Method of orienting a tool using two pipe clamps, a telescope with a hair line, and an aligning bar to determine the orientation at each section of pipe run in the hole.

Monel (K monel) A permanently nonmagnetic alloy used in making portions of downhole tools in the bottomhole assembly (BHA) where the magnetic survey tools are placed for obtaining magnetic direction information. Monel refers to a family of nickel–copper alloys.

Mud motor Usually a positive displacement or turbine-type motor.

Mule shoe A shaped form used on the bottom of orienting tools to position the tool. The shape resembles a mule shoe or the end of a pipe that has been cut both diagonally and concave. The shaped end forms a wedge to rotate the tool when lowered into a mating seat for the mule shoe.

Multishot survey A directional survey in which multiple data points are recorded with one trip into the wellbore. Data are usually recorded on rolls of film.

Near-bit stabilizer A stabilizer placed in the bottomhole assembly just above the bit.

Ouija board® (registered trademark of Eastern Whipstock) An instrument composed of two protractors and a straight scale that is used to determine the positioning for a deflecting tool in an inclined wellbore.

Permissible dogleg A dogleg through which equipment and/or tubulars can be operated without failure.

Pendulum effect Refers to the pull of gravity on a body; tendency of a pendulum to return to a vertical position.

Pendulum hookup A bit and drill collar with a stabilizer to attain the maximum effect of the pendulum.

Rat hole A hole that is drilled ahead of the main wellbore and which is of a smaller diameter than the bit in the main borehole.

Reamer A tool employed to smooth the wall of a wellbore, enlarge the hole, stabilize the bit and straighten the wellbore where kinks and abrupt doglegs are encountered.

Rebel tool® (registered trademark of Eastman Whipstock) A tool designed to prevent and correct lateral drift (walk) of the bit tool. It consists of two paddles on a common shaft that are designed to push the bit in the desired direction.

Roll off A correction in the facing of the deflection tool, usually determined by experience, and which must be taken into consideration to give the proper facing to the tool.

Setting off course A method of setting the direction of the wellbore in anticipation of the bit walking.

Side track An operation performed to redirect the wellbore by starting a new hole at a position above the bottom of the original hole.

Slant hole A nonvertical hole; usually refers to a wellbore purposely inclined in a specific direction; also used to define a wellbore that is nonvertical at the surface.

Slant rig Drilling rig specifically designed to drill a wellbore that is nonvertical at the surface. The mast is slanted and special pipe-handling equipment is needed.

Spiraled wellbore A wellbore that has attained a changing configuration such as a spiral or helical form.

Spud bit In directional drilling, a special bit used to change the direction and inclination of the wellbore.

Stabilizer A tool placed in the drilling assembly to:

(1) change or maintain the inclination angle in a wellbore by controlling the location of the contact point between the hole and drill collars;

(2) center the drill collars near the bit to improve drilling performance; and/or

(3) prevent wear and differential sticking of the drill collars.

Surveying frequency Refers to the number of feet between survey records.

Target area A defined area, at a prescribed vertical depth, that is planned to be intersected by the wellbore.

Tool azimuth angle The angle between north and the projection of the tool reference axis onto a horizontal plane.

Tool high-side angle The angle between the tool reference axis and a line perpendicular to the hole axis and lying in the vertical plane.

Total curvature Implies three-dimensional curvature.

True north The direction from any geographical location on the earth's surface to the north geometric pole.
True vertical depth (TVD) The actual vertical depth of an inclined wellbore.
Turbodrill A downhole motor that utilizes a turbine for power to rotate the bit.
Turn A change in bearing of the hole; usually spoken of as the right or left turn with the orientation that of an observer who views the well course from the surface site.
Walk (of hole) The tendency of a wellbore to deviate in the horizontal plane.
Wellbore survey calculation methods Refers to the mathematical methods and assumptions used in reconstructing the path of the wellbore and in generating the space curve path of the wellbore from inclination and direction angle measurements taken along the wellbore. These measurements are obtained from gyroscopic or magnetic instruments of either the single-shot or multi-shot type.
Whipstock A long wedge and channel-shaped piece of steel with a collar at its top through which the subs and drillstring may pass. The face of the whipstock sets an angle to deflect the bit.
Woodpecker drill collar (indented drill collar) Round drill collar with a series of indentations on one side to form an eccentrically weighted collar.

Dogleg Severity (Hole Curvature) Calculations

Currently there are several analytical methods available for calculating dogleg severity.

These methods include:

- tangential
- radius of curvature
- average angle
- trapezoidal (average tangential)
- minimum curvature

Here the tangential and radius of curvature methods are outlined.

Tangential Method [171]

The overall angle change is calculated from

$$\Delta a = 2 \text{ arc sin}\left(\sin \frac{2\Delta v}{2} + \sin \frac{2\Delta h}{2} \sin V_0 \sin V_1\right)^{0.5} \quad (4\text{-}263)$$

where Δa = overall angle change
Δh = change of horizontal angle (in horizontal plane)
Δv = change of vertical angle (in vertical plane)
V_0, V_1 = hole inclination angle in two successive surveying stations

The hole curvature is

(4-264)

where L = course length between the surveying stations

Example 1

Two surveying measurements were taken 30 ft apart. The readings are as below:
Station 1:
Hole inclination, 3°30′
Hole direction, N11°E
Station 2:
Hole inclination, 4°30′
Hole direction, N23°E
Find the dogleg severity.

Solution

Change in horizontal angle is

$$\Delta h = 23 - 11 = 12°$$

Change in vertical angle is

$$\Delta v = 4.5 - 3.5 = 1°$$

The overall angle change is

$$\Delta a = 2 \text{ arc } \sin[\sin^2(0.5) = \sin^2(6) \sin(3.5) \sin(4.5)]^{0.5} = 1.3°$$

Hole curvature is

$$C = \frac{1.3}{30} 100 = 4.33°/100\,\text{ft} \cong 4°20'/100\,\text{ft}$$

Radius of Curvature Method [133]

The dogleg severity is calculated from:

$$C = 100[a^2 \sin^4\phi + b^2]^{0.5} \tag{4-265}$$

where a = rate of change in direction angle in °/ft
b = rate of change in inclination angle in °/ft
ϕ = inclination angle in °

The sequence of computations involved is explained in the following example.

Example 2

From two successive directional survey stations is obtained:

	Station 1	Station 2
Hole inclination angle:	30° (Φ_1)	40° (Φ_2)
Hole direction angle:	N11°E (θ_1)	N18°E (θ_2)

The distance between the stations is 60 ft. Determine the dogleg severity.

Solution

Rate of change in inclination angle is

$$b = \frac{40 - 30}{60} = 0.1667°/\text{ft}$$

Radius of curvature in vertical plane is

$$R_v = \frac{180}{b\pi} = \frac{180}{0.1667\pi} = 363\,\text{ft}$$

Horizontal departure (arc length of projection of wellbore in horizontal plane) is

$$Hd = R_v (\cos \phi_1 - \cos \phi_2) = 363 (\cos 30° - \cos 40°) = 34.28 \text{ ft}$$

Rate of change in hole direction is

$$a = \frac{\theta_2 - \theta_1}{Hd} = \frac{18 - 11}{34.28} = 0.2042°/\text{ft}$$

Hole curvature at the first station is

$$c_1 = 100 \left[(0.2042)^2 \sin^4 30° + (0.1667)^2\right]^{0.5} = 7.43°/100 \text{ ft}$$

Hole curvature at the second station is

$$c_2 = 100 \left[(0.2042)^2 \sin^4 40° + (0.1667)^2\right]^{0.5} = 18.68°/100 \text{ ft}$$

The average value is

$$c = \frac{17.43 + 18.68}{2} = 18.05°/100\text{ft}$$

Deflection Tool Orientation

Application of a deflecting tool (e.g., downhole motor with a bent sub) requires determining the orientation of the tool so that the hole takes the desired course. There are three effects to consider when setting a deflection tool:

1. the existing borehole inclination angle
2. the existing borehole direction angle
3. the bent sub angle of the deflection tool itself

These three effects in combination will result in a new dogleg of a wellbore.

The deflection tool orientation parameters can be obtained using the vectorial method of D. Ragland, the Ouija Board®, or the three-dimensional mathematical deflecting model.

Vectorial Method of D. Ragland [134]

This method is explained by solving two example problems.

Example 3

Determine the deflection tool-face orientation, tool deflection angle and tool facing change from original course line angle if the data are as follows:

Existing hole inclination angle: 10°
Existing hole direction: N30°W
Desired change of azimuth: 25° (to the left)
Desired hole inclination: 8°

Solution

The solution to this problem is shown in Figure 4-349. The following steps are involved in preparing a Ragland diagram.

1. Lay a quadrant N-S-W-E.
2. Select a scale for the angles.
3. With a protractor lay off an angle 30° from N.
4. Using the selected angle scale find p.B (10° from p.A).
5. With a protractor lay off an angle 25° to the left of line AB.
6. Find p.C using the angle scale (8° from p.A).
7. Describe a circle about p.B with a radius of CB = 4°6' (read off from the diagram). This is the desired tool deflection angle.
8. Read off required tool-face orientation: S22°W
9. Read off required tool facing change from original course line: 131°.

Example 4

The original hole direction and inclination were measured to be S60°W and 6°, respectively. To obtain a new hole direction of S75°W with an inclination angle of 7°, a whipstock with the deflection angle of 2° was used and oriented correctly.

After drilling 90 ft, a checkup measurement was performed that revealed that the hole direction is S72°W and the inclination is 6°30'. Determine the magnitude of roll-off for this system and the real deflection angle of the whipstock. How should the whipstock be oriented to drill a hole with a direction of S37°W and inclination of 5.5°?

Solution

The solution to this problem is shown in Figure 4-350. Steps 1 and 2 are the same as in Example 3.

3. Lay off an angle 60° from S.
4. Draw a line 00'.
5. Draw a line OA that will represent the desired new hole direction (S72°W). Point A is found at a distance of 7° from point O.
6. Read off the original whipstock orientation N64°W.
7. Draw a line OB (direction, S72°W; inclination, 6°30').

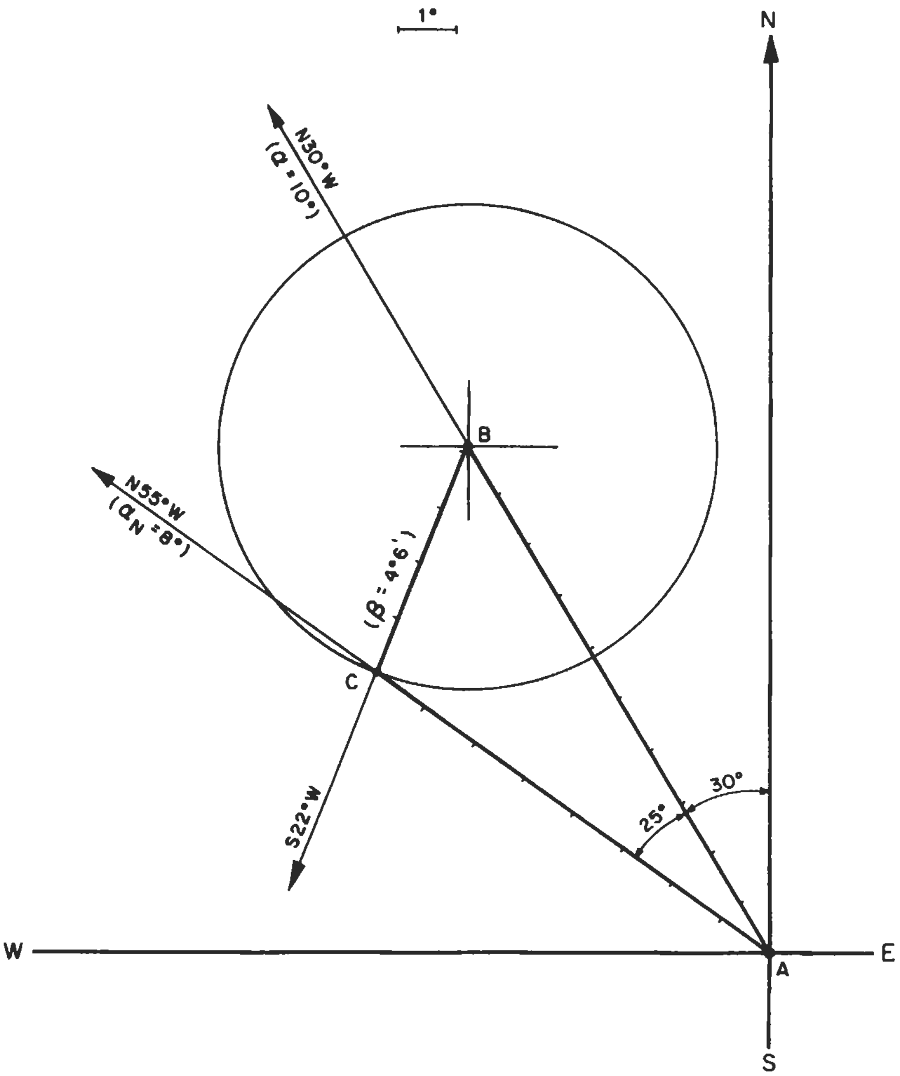


Figure 4-349. Graphic solution of Example 3.

8. Describe a circle about p.A′ with a radius of OB = 1°36' which is the whipstock true deflection angle.
9. The angle AO′B is the roll-off angle. Read off the roll-off angle = 9° to the right.
10. Draw a line OC (S37°W and inclination 5.5°).
11. If the roll-off effect is not considered, the whipstock direction is S71°E. Including the roll-off effect, the whipstock direction is S80°E.

Note: With a Dyna-Drill downhole motor there will be a left-hand reaction torque; therefore, the tool should be turned counterclockwise from the ideal position.

Three-Dimensional Deflecting Model

Recently a mathematical model has been presented [135] that enables one to analyze and plan deflection tool runs. The model is a set of equations that relate the original hole inclination angle (α), new hole inclination angle (α_N), overall angle change (β), change of direction ($\Delta\varepsilon$), and tool-face rotation from original course direction (γ).

These equations are given below:

$$\beta = \text{arc } \cos(\cos \alpha \cos \alpha_N + \sin \alpha \sin \alpha_N \cos \Delta\varepsilon) \quad (4\text{-}266)$$

$$\alpha_N = \text{arc } \cos(\cos \alpha \cos \beta - \sin \beta \sin \alpha \cos \gamma) \quad (4\text{-}267)$$

$$\Delta\varepsilon = \text{arc } \tan g\left(\frac{\tan\beta \sin\gamma}{\sin\alpha + \tan\beta\cos\alpha\cos\gamma}\right) \quad (4\text{-}268)$$

The above equations can be rearranged to the form suitable for a solution of a particular problem. A hand-held calculator may be used to perform required calculations [136].

Practical usefulness of this model is presented below.

Example 5

Original hole direction and inclination is N20°E and 10°, respectively. It is desired to deviate the borehole so that the new hole inclination is 13° and direction N30°E.

What is the tool orientation and deflecting angle (dogleg) necessary to achieve this turn and build?

Solution

From Equation 4-266,

$$\beta = \text{arc } \cos(\cos 10 (\cos \beta) + \sin 10 (\sin \beta) \cos 10) = 3.59°$$

The desired tool deflection angle (e.g., bent sub) is 3.59°.

Solving Equation 4-267 for γ yields

$$\gamma = \text{arc } \cos\frac{\cos\alpha\cos\beta - \cos\alpha_N}{\sin\beta\sin\alpha} = \text{arc } \cos\left(\frac{\cos 10(\cos 3.59) - \cos 13}{\sin 3.59 \sin 10}\right)$$

$$= 38.53°$$

Consequently the tool-face orientation is

20 + 38.53 = N58.53°N

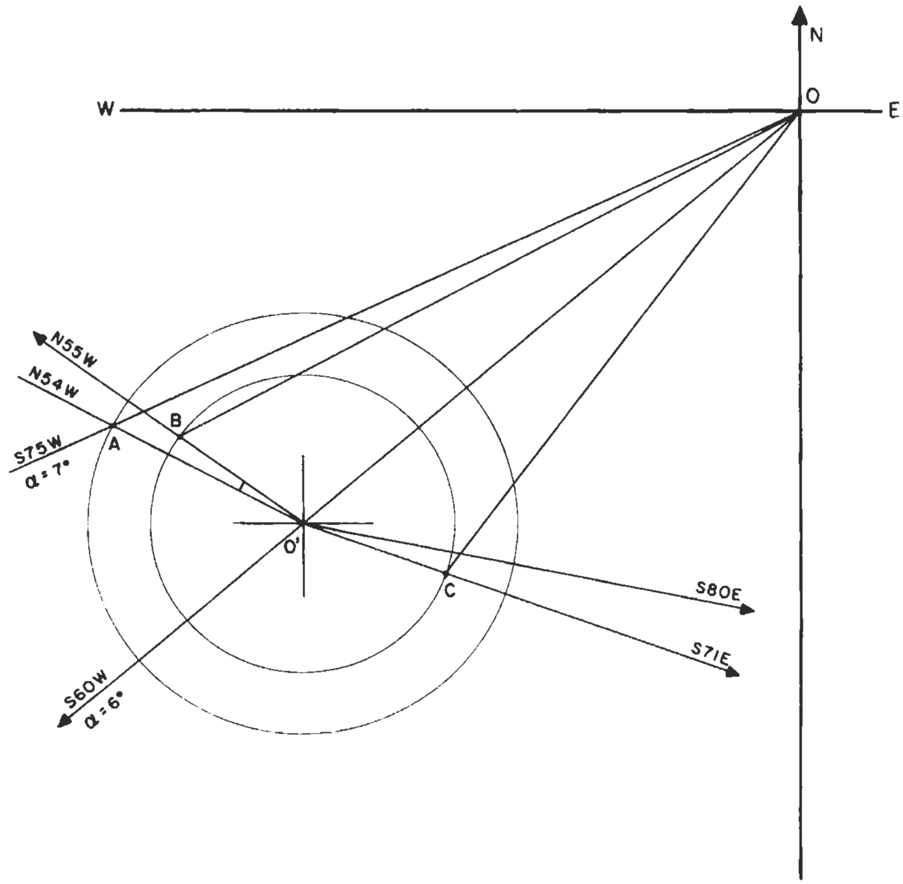


Figure 4-350. Graphical solution of Example 4.

Example 6

Surveying shows that the hole drift angle is 22° and direction S36°W. It is desired to turn the hole of 6° to the right and build angle. For this purpose a deflection tool with a dogleg of 3° is used. What is the expected new hole inclination angle? What is the required tool direction?

Solution

Equation 4-266 can be solved for α_N by applying the following rearrangements:

$$\cos\beta = \cos\alpha \cos\alpha_N + \sin\alpha \cos\Delta\varepsilon \sin\alpha_N$$

Denote

$$a = \sin\alpha \cos\Delta\varepsilon$$

$$b = \cos\alpha$$

Then

$$\cos\beta = (a^2 + b^2)^{0.5}\cos(\alpha_N - \phi)$$

where

$$\phi = \arc\tan\frac{a}{b}$$

so

$$\alpha_N = \arc\cos\left(\frac{\cos\beta}{(a^2 + b^2)^{0.5}}\right) + \phi$$

$$\arc\cos\left(\frac{\cos 3}{(\cos^2 22 + \sin^2 22\cos^2 6)^{0.5}}\right) = 1.99°$$

$$\phi = \arc\tan\left(\frac{\sin 22\cos 6}{\cos 22}\right) = 21.89°$$

and

$$\alpha_N = 23.88°$$

Using Equation 4-267, the tool-face rotation is found to be 54.02°. Consequently the required tool direction is N90°W.

SELECTION OF DRILLING PRACTICES

Factors Affecting Drilling Rates

The factors that influence the rate of penetration during a conventional rotary drilling process are numerous. In general these factors can be classified into five groups.

1. Formation mechanical and physical properties (strength, hardness, abrasiveness, porosity, permeability, etc.)
2. Rock bit type (tooth shape, journal angle, cone offset, fluid circulation system)
3. Drilling fluid type and properties (density, viscosity, fluid loss, etc., solids content, differential pressure, etc.)
4. Drilling parameters (weight on bit, rotary speed, nozzle size, flowrate)
5. Rig type and performance of drilling personnel (rig size, degree of automation, power equipment, etc.)

The detailed and rigorous analysis of the relationship between the drilling rate and the aforementioned factors is not available as of today.

Selection of Weight on Bit, Rotary Speed and Drilling Time (Bit Rotating Time)

To select properly the weight on bit W (lb), rotary speed N (rpm) and bit rotating time T_b (hr), a drilling engineer should choose a criterion for selection and a drilling model. As criteria for the drilling process evaluation and the drilling parameters selection the following functions are generally used.

Drilling cost per foot C_f ($/ft) is

$$C_f = \frac{(T_b + T_t)C_r + C_b}{F} \tag{4-269}$$

Run cycle speed RCS (ft/hr) is

$$RCS = \frac{F}{T_b + T_t} \tag{4-270}$$

Footage F (ft) is

$$F = ROP \times T_b \tag{4-271}$$

where C_r = hourly rig cost in $/hr
C_b = bit cost in $
T_t = trip time (nonrotating time) in hr
ROP = an average rate of penetration in ft/hr

The drilling model presented here is a simplified and modified model developed by Bourgoyne and Young [137]. The model includes three equations.

Instantaneous drilling rate equation

$$\frac{dF}{dt} = K\, I_1 \frac{1}{1 + Ch} \tag{4-272}$$

where

$$I_1 = \left(\frac{W}{4D_b}\right)^{a_1} \left(\frac{N}{100}\right)^{a_2}$$

Rate of bit tooth wear equation

$$\frac{dh}{dt} = \frac{1}{A_f} I_2 \frac{1}{1 + H_2 h}$$

where

$$I_2 = H_3\left(\frac{N}{100}\right)^{H_1}\left[\frac{\left(\frac{W}{D_b}\right)_m - 4.0}{\left(\frac{W}{D_b}\right)_m - \frac{W}{D_b}}\right]\left(1 + \frac{H_2}{2}\right) \tag{4-273}$$

Rate of bit bearing wear equation is

$$\frac{dB}{dt} = \frac{1}{S} I_3 \tag{4-274}$$

where

$$I_3 = \left(\frac{N}{100}\right)\left(\frac{W}{4D_b}\right)^b$$

where

F = footage in ft
t = bit rotating time in hr
W = weight on bit in 10^3 lb
N = bit rotary speed in rpm
D_b = bit diameter in in.
h = rock bit tooth dullness (fraction of original height worn away), $0 \leq h \leq 1$
B = bearing wear (fraction of a bearing life expected), $0 \leq B \leq 1$
C = constant dependent upon bit and formation type
A_f = formation abrasiveness parameter in hr
K = formation drillability parameter in ft/hr
S = bit bearing parameter in hr
a_1, a_2 = bit weight and rotary speed exponents
H_1, H_2, H_3 or $(W/D_b)_m$ = constants which depend upon bit type (see Table 4-138)
b = constant depending upon bearing and drilling fluid type (see Table 4-139)
4 = normalized weight on bit in 4×10^3 lb
100 = normalized rotary speed in rpm

Although the presented model is not a perfect simulation of the real system, it provides a possibility of determining the optimal WOB, RPM and T_b.

The formation abrasiveness, drillability and bit-bearing parameters are calculated from the following formulas based on data available from previous drilling experience.

$$A_f = \frac{I_2 T_b}{h_f + \frac{H_2}{2} h_f^2} \tag{4-275}$$

$$K = \frac{F I_2}{I_1 A_f \left[\frac{H_2}{C} h_f + \frac{C - H_2}{C^2} \ln\left(C h_f + 1\right)\right]} \tag{4-276}$$

Table 4-138
Recommended Tooth Wear Parameters
(After Bourgoyne and Young [137])

Bit Class	H_1	H_2	H_3	$(W/D_b)max$
1-1 to 1-2	1.9	7	1.0	7.0
1-3 to 1-4	1.84	6	0.8	8.0
2-1 to 2-2	1.80	5	0.6	8.5
2-3	1.76	4	0.48	9.0
3-1	1.70	3	0.36	10.0
3-2	1.65	2	0.26	10.0
3-3	1.60	2	0.20	10.0
4-1	1.50	2	0.18	10.0

Table 4-139
Recommended Value of Exponent "b" (After Maratier [137])

Bearing type	Drilling fluid	b
Nonsealed	Barite mud	1.00
	Sulfide mud	1.25
	Water	1.30
	Clay mud	2.04
	Oil base mud	2.55
Sealed		2.8

$$S = \frac{I_3 T_b}{B} \tag{4-277}$$

Taking the minimum drilling cost as a criterion for determining the optimum magnitude of WOB, RPM and T_b, the following equations are applicable:

$$W_{(opt)} = \frac{a_1 H_1 \left(\frac{W}{D_b}\right)_m}{a_1 H_1 + a_2} D_b \tag{4-278}$$

$$T_{b(opt)} = \left(\frac{C_b}{C_r} + T_t\right)\left(\frac{H_1}{a_2} - 1\right) \tag{4-279}$$

$$N_{(opt)} = 100\left(\frac{A_f\left(h_f + \frac{H_2}{2}h_f^2\right)\left[\left(\frac{W}{D_b}\right)_m - \left(\frac{W}{D_b}\right)\right]}{T_{b(opt)}H_3\left[\left(\frac{W}{D_b}\right)_m - 4.0\right]\left(1+\frac{H_2}{2}\right)}\right)^{1/H_1} \quad (4\text{-}280)$$

Equations 4-278, 4-279 and 4-280 are valid only if the bit life is limited by tooth wear or economical drilling rate.

Example 1

The following data are available from an offset well:

Weight on bit, $W = 45 \times 10^3$ lb;
Rotary speed, $N = 130$ rpm;
Footage, $F = 580$ ft;
Bit rotating time, $T_b = 16$ hr;
Trip time, $T_t = 9$ hr;
Bit diameter, $D_b = 12\frac{1}{4}$ in.;
Bit type: SDS;
From Table 4-138: $H_1 = 1.9$, $H_2 = 7$, $H_3 = 1.0$, $(W/D_b)_m = 7$;
Bearing type: nonsealed, clay mud;
From Table 4-139: $b = 2.06$;
Tooth wear, $h_f = \frac{7}{8}$;
Bearing wear, $B = \frac{6}{8}$;
Exponents: $a_1 = 1.0$, $a_2 = 0.8$;
Constant: $C = 6.5$.

Find the formation abrasiveness and drillability parameters and the bit-bearing constant.

Solution

Calculate the dimensionless functions:

$$I_1 = \left(\frac{W}{4D_b}\right)^{a_1}\left(\frac{N}{100}\right)^{a_2} = \left(\frac{45}{4 \times 12.25}\right)^{1.0}\left(\frac{130}{100}\right)^{0.8} = 1.323$$

$$I_2 = H_3\left(\frac{N}{100}\right)^{H_1}\left(\frac{\left(\frac{W}{D_b}\right)_m - 4.0}{\left(\frac{W}{D}\right)_m - \frac{W}{D_b}}\right)\left(1+\frac{H_2}{2}\right)$$

$$= \left(\frac{130}{100}\right)^{1.3}\left(\frac{7-4}{7-\frac{45}{12.25}}\right)\left(1+\frac{7}{2}\right) = 6.68$$

and

$$I_3 = \left(\frac{N}{100}\right)\left(\frac{W}{4D_b}\right)^b = \left(\frac{130}{100}\right)\left(\frac{45}{4 \times 12.25}\right)^{2.04} = 1.09$$

Applying Equations 4-275, 4-276 and 4-277 yields the following:

Formation abrasiveness parameter is

$$A_f = \frac{6.68 \times 16}{0.875 + \frac{7}{2}0.875^2} = 30.07 \text{ hr}$$

Formation drillability parameter is

$$K = \frac{580 \times 6.68}{1.323 \times 30.07\left(\frac{7}{6.5}0.785 + \frac{6.5 - 7}{6.5^2}(6.5 \times 0.785 + 1)\right)} = 133.66 \text{ ft/hr}$$

and the bearing constant is

$$S = \frac{1.09 \times 16}{0.75} = 23.31 \text{ hr}$$

Note: Using the above obtained values for K, A_f and S, one may attempt to optimize drilling parameters from Equations 4-278, 4-279 and 4-280; however, in the case considered, the bit life is limited by bearings wear. Consequently Equations 4-278, 4-279 and 4-280 are not applicable. Nevertheless a simple trial-and-error calculation can be used to find the desired parameters.

Example 2

Find the optimal weight on bit and rotary speed for minimum drilling cost per foot if the data are as follows:

Formation drillability parameter, K = 28.5 ft/hr;
Formation abrasiveness parameter, A_f = 24.7 hr;
Bit bearings constant, S = 45 hr;
Bit diameter, D_b = 9.875 in.;
Bit constants: H_1 = 1.84, H_2 = 6, H_3 = 0.8, $(W/d_b)_m$ = 8;
Exponents: a_1 = 1.2; a_2 = 0.6, b = 2.04;
Trip time, T_t = 7 hr;
Bit cost, C_b = \$900;
Rig cost, C_r = 600 \$/hr;
Bit formation constant, C = 2.5.

Solution

Optimal weight on bit is

$$W_{opt} = 9.875\frac{1.2 \times 1.84 \times 8}{1.2 \times 1.84 + 0.6} = 62.1 \times 10^3 \text{ lb}$$

Optimal bit rotating time is

$$T_{b,opt} = \left(\frac{900}{600} + 7\right)\left(\frac{1.84}{0.6} - 1\right) = 17.5 \text{ hr}$$

Optimal rotary speed is

$$N_{opt} = 100\left(\frac{24.7\left(h_f + \frac{6}{2}h_f^2\right)\left(8 - \frac{62.1}{9.875}\right)}{17.5 \times 0.8 \times (8-4)(1+3)}\right)^{1/1.86}$$

$$= 40.4\left(h_f + 3 \times h_f^2\right)^{0.54347}$$

Take $h_f = 1.0$:

$$N_{(hf = 1.0)} = 85.8 \cong 86 \text{ rpm}$$

Expected bearings wear is

$$B = I_3 \frac{T_b}{S} = \left(\frac{86}{100}\right)\left(\frac{62.1}{4 \times 9.875}\right)^{2.06} \frac{17.5}{45} = 0.84$$

Conclusion: Bit life is limited by tooth wear or economical drilling rate.

Expected magnitude of footage at $W = 62.1 \times 10^3$ lb, $N = 86$ rpm, and $h_f = 1.0$. Calculate I_1 and I_2:

$$I_1 = \left(\frac{62.1}{4 \times 9.875}\right)^{1.2}\left(\frac{86}{100}\right)^{0.6} = 1.572$$

$$I_2 = 0.8\left(\frac{86}{100}\right)^{1.84}\left(\frac{8.0 - 4.0}{8.0 - 6.288}\right)(1+3) = 5.665$$

The expected footage is

$$F = 28.5 \times 24.7 \frac{1.572}{5.665}\left(\frac{6}{2.5}1 + \frac{2.5 - 6}{2.5^2}(2.5 \times 1 + 1)\right) = 334 \text{ ft}$$

and the expected cost per foot is

$$C_f = \frac{(7 + 17.5)900 + 600}{334} = 67.8 \text{ \$/ft}$$

Further calculations can be performed for $h_f = 0.9$; however, in the case considered, the expected cost per foot is higher. Consequently, the bit life is limited by tooth wear, and the computations are completed.

Selection of Optimal Nozzle Size and Mud Flowrate

Optimal hydraulics is the proper balance of hydraulic parameters (flowrate and equivalent nozzle size) that satisfy chosen criteria of optimization. Hydraulic quantities used to characterize jet bit performance include hydraulic horsepower, jet impact force, jet velocity, Reynolds number at the nozzle, generalized drilling rate or cost per foot drilled. While designing the hydraulic program the limitations due to *cuttings* transport in the annulus and pump performance characteristics must be included.

An estimate of the minimum required mud flow velocity in annular space to assure adequate cuttings transport can be estimated from

$$V_{an} = \frac{196}{D_h \bar{\gamma}_m} \tag{4-281}$$

where V_{an} = annular velocity in ft/s
D_h = bore-hole diameter in in.
$\bar{\gamma}_m$ = mud weight in lb/gal

The mud pump performance characteristic can be thought of as being composed of two operating ranges.

Range 1 is

$$p_p = p_b + p_d = \text{const} \tag{4-282}$$

Range 2 is

$$HP_p = HP_b + HP_d = \text{const} \tag{4-283}$$

where p_p = pump pressure in psi
p_b = bit pressure drop in psi
p_d = pressure drop through remaining circulating system in psi
HP_p = pump hydraulic horsepower
HP_b = hydraulic horsepower developed at the bit
HP_d = hydraulic horsepower dissipated through the remaining circulating system

Range 1 of the mud pump performance characteristic is defined by the performance of the smallest liner, and range 2 is defined by the remaining liners. The pressure loss in a circulating system, except for bit (p_d), can be estimated from numerous theoretical formulas or from a flowrate test. Data obtained from a flowrate test can be approximated using a curve-fitting technique by the following function:

$$p_d = Kq^m \tag{4-284}$$

The quantities K and m may also be found by plotting p_d vs. Q (flowrate) on log-log graph paper. Here, an outline of a hydraulic program design for the maximum jet impact force is presented. The jet impact force F_j (lb) is calculated from

$$F_j = 1.73 \times 10^{-2} Q (P_b)^{0.5} \tag{4-285}$$

where q is the flowrate (gal/min).

For the first pump operating range (p_p = const = p_{pm}) ($0 < Q \leq Q_m$), where p_{pm} is maximum desirable pump pressure. The optimal flow rate q_{opt} is

$$q_{opt(1)} = \left(\frac{2 p_{pm}}{K(m+2)} \right)^{1/m} \tag{4-286}$$

The pressure drop across the bit is

$$p_b = \frac{m}{m+2} p_{pm} \tag{4-287}$$

For the second pump operating range (HP_p = const = HP_m) ($q_m < q < q_m{}^1$), where HP_m is maximum desirable pump hydraulic horsepower. The optimal flowrate is

$$q_{opt(2)} = \left(\frac{1{,}714 \times HP_{pm}}{K \times (m+2)} \right)^{1/m+1} \tag{4-288}$$

and the corresponding pressure change across the bit is

$$p_b = \frac{m+1}{m+2} p_{p(opt)} \tag{4-289}$$

where

$$p_{p(opt)} = \frac{1{,}714 HP_{pm}}{q_{opt(2)}} \tag{4-290}$$

On determining the optimal flowrate q_{opt} (gal/min) and pressure drop across the bit nozzles p_b, the total flow area is calculated from

$$A_n = \left(\frac{\bar{\gamma}_m q^2_{(opt)}}{10{,}858 p_b} \right)^{0.5} \tag{4-291}$$

Example 3

Calculate the optimal hydraulic parameters for the hole depth of 8,555 ft if the available data are as given below: Type of pump: Continental Emsco F-1600 with 97% pump volumetric efficiency. To prevent rapid pump wear, the pump pressure should not exceed about 80% of the maximum pump pressure at any liner size. The hole size is $12\frac{1}{4}$ in. and the mud weight is 9.7 lb/gal.

Upon completing a drilling run at the depth of 8,555 ft, the flowrate test was performed and the following measurements were obtained:

q (gal/min)	p_p (psi)
375	1,950
400	2,125
425	2,400
450	2,725

Nozzle size at flowrate test is $3 \times \frac{12}{32}$. The minimum required flowrate is 350 gal/min.

Solution

Applying the least-squares technique, the equation for pressure loss P_d is

$p_d = 0.0522q^{1.623}$

Based on a nominal pump performance characteristic, the following table is prepared.

Liner size (in.)	Pump output at 130 strokes/min gal/min	Max pump pressure (psi)	Pump hydr. hrspwr.
5½	481	5,558	1,560
5¾	526	5,078	1,558
6	573	4,665	1,560
6¼	621	4,299	1,558
6½	671	4,012	1,570
6¾	724	3,688	1,557
7	778	3,423	1,554
7¼	836	3,197	1,560
7½	894	2,988	1,558

Introducing the volumetric efficiency (97%) and the limitations for pump pressure (80%) we obtain

Liner size (in.)	Pump output at 120 strokes/min gal/min	Max pump pressure (psi)	Pump hydr. hrspwr.
5½	430	4,446	1,115
5¾	470	4,062	1,114
6	513	3,732	1,117
6¼	556	3,439	1,112
6½	600	3,209	1,123
6¾	648	2,950	1,115
7	696	2,738	1,112
7¼	720	2,557	1,074
7½	800	2,390	1,115

For further calculations the maximum pump pressure of 4,446 psi and maximum hydraulic horsepower of 1,110 hp is accepted. Also, q_m = 430 gal/min and q'_m = 800 gal/min. Since $q_m > q_r$, we calculate $q_{opt(2)}$ from Equation 4-288:

$$q_{opt(2)} = \left(\frac{1,716 \times 1,100}{(1.623 + 2)0.0522}\right)^{1/1.623+1} = 466 \text{ gal/min}$$

then, applying Equation 4-290,

$$p_{popt} = \frac{1,716 \times 1,100}{466} = 4,045 \text{ psi}$$

The liner size is $5\frac{3}{4}$ in. The number of strokes per minute is 119. Pressure drop across the bit from Equation 4-289 is

$$p_b = \frac{2.623}{3.623} 4,045 = 2,928 \text{ psi}$$

Calculate the total nozzle area from Equation 4-291:

$$A_n = \left(\frac{9.7 \times 466^2}{10,858 \times 2,928}\right)^{0.5} = 0.2574 \text{ in.}^2$$

The nozzles sizes are $\frac{8}{32}$, $\frac{10}{32}$ and $\frac{13}{32}$ in. For the selected set of nozzles, the total flow area is 0.2554 in.^{2}, which will result in slightly higher pump pressure than 4,446 psi; however, this seems to be acceptable.

WELL PRESSURE CONTROL

Introduction

Basically all formations penetrated during drilling are porous and permeable to some degree. Fluids contained in pore spaces are under pressure that is overbalanced by the drilling fluid pressure in the well bore. The bore-hole pressure is equal to the hydrostatic pressure plus the friction pressure loss in the annulus. If for some reason the borehole pressure falls below the formation fluid pressure, the formation fluids can enter the well. Such an event is known as a *kick*. This name is associated with a rather sudden flowrate increase observed at the surface.

A formation fluid influx (a kick) may result from one of the following reasons:

- abnormally high formation pressure is encountered
- lost circulation
- mud weight too low
- swabbing in during tripping operations
- not filling up the hole while pulling out the drillstring
- recirculating gas or oil cut mud.

If a kick is not controlled properly, a blowout will occur. A blowout may develop for one or more of the following causes:

- lack of analysis of data obtained from offset wells
- lack or misunderstanding of data during drilling
- malfunction or even lack of adequate well control equipment

Surface Equipment

A formation fluid kick can be efficiently and safely controlled if the proper equipment is installed at the surface. One of several possible arrangements of pressure control equipment is shown in Figure 4-351. The blowout preventer (BOP) consists of a spherical preventer (Hydril), blind and pipe rams, and the drilling spool.

A spherical preventer contains a packing element that seals the space around a drillpipe. This preventer is not designed to shut off the well when the drillpipe is out of the hole, although it allows stripping operations and some pipe rotation. Hydril Corporation, Shaffer and other manufacturers provide several models with different packing system designs for specific types of service. The ram-type preventer seals the annulus around the drillpipe; however, each size of ram is designated for only one size of drillpipe. In other words, the preventer with 5-in. pipe rams can provide a seal only on 5-in. drillpipe. The preventer with blind rams is used to shut in the well if the pipe is not in the hole. If this type of preventer is activated with the pipe in the hole, the pipe can be cut.

There are also specially designed preventers with shear-blind rams. This type of ram will certainly cut the pipe and seal the open hole. Special precautions should be taken to ensure that the blind rams cannot be mistakenly closed.

A drilling spool is the element of the BOP stack to which mud and kill lines are attached. The pressure rating of the drilling spool and its outlets must be consistent with the BOP stack. The kill line serves to pump drilling fluid directly into the annulus in case of need.

The choke lines are attached to the BOP to enable controlled circulation of the drilling and formation fluids out of the hole.

A degasser is installed on the mud line to remove gas from drilling fluid while penetrating gas bearing formations. Samples of gas are analyzed using the gas chromatograph.

If the well cannot be shut in (for some reason) to implement regular kick killing procedures, a diverter stack is used rather than a regular BOP stack. A diverter is furnished with a blow-down line to allow the well to blow out away from the rig.

When and How to Close the Well

There are certain warning signals while drilling which, if analyzed properly, can lead to early detection of formation fluid entry into the wellbore.

1. *Drilling break.* A relatively sudden increase in the instantaneous drilling rate is called the *drilling break.* The drilling break may occur due to a decrease in the difference between the borehole pressure and formation pore pressure. When a drilling break is observed, the pumps should be stopped and the well watched for flow at the mud line. If the well does not flow, it means that the overbalance is not lost or simply that a softer formation has been encountered.
2. *Decrease in pump pressure.* When less dense formation fluid enters the borehole, the hydrostatic head in the annulus is decreased; thus, the pressure supplied by mud pumps is decreased. Although reduction in pump pressure

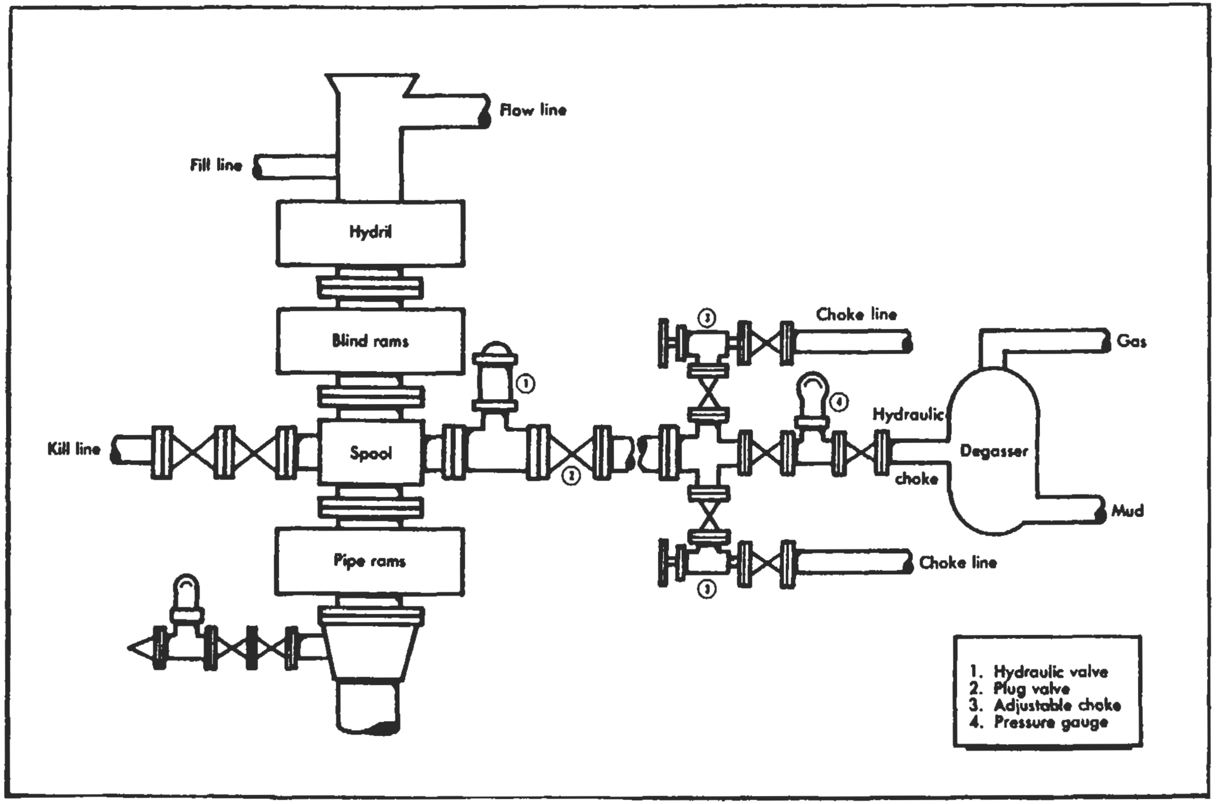


Figure 4-351. Pressure control equipment. *(From Ref. [144].)*
Source: Preston Moore's "Drilling Practices Manual, 2nd Edition" Copyright PennWell Books, 1986.

may be caused by several other factors, drilling personnel should consider formation fluid influx in the wellbore as one possible cause. Again, the pumps should be stopped and the flow line watched carefully.
3. *Increase in pit level.* This is a definite signal of formation fluid invasion into the hole. The well must be closed as soon as possible.
4. *Gas-cut mud.* When drilling through gas-bearing formations, small quantities of gas occur in the cuttings. As these cuttings are circulated up the annulus, the gas expands. Consequently a reduction in mud weight is observed at the surface. Here again stopping the pumps and watching the flow line return helps to find out whether or not the overbalance is lost.

If the kick is gained while tripping, the only warning signal we have is an increase in fluid volume at the surface (pit gain). Once it is determined that the pressure overbalance is lost, the well must be closed as quickly as possible. The sequence of operations involved in closing the well is as follows:

1. Shut off the mud pumps.
2. Raise kelly above the BOP stack.
3. Open the choke line.
4. Close the spherical preventer.
5. Close choke slowly.
6. Record the pit level increase.
7. Record the stabilized pressure on drillpipe and annulus pressure gages.
8. Notify the company personnel.
9. Prepare the kill procedure.

If the well kicks while tripping, the sequence of necessary steps can be as given below:

1. Close the safety valve (kelly cock) on the drillpipe.
2. Pick up and install the kelly.
3. Open safety valve (kelly cock).
4. Open the choke line.
5. Close the annular preventer.
6. Record the pit gain and the shut-in drillpipe and casing pressure.
7. Notify the company personnel.
8. Prepare the kill procedure.

Depending upon the drilling rig type and company policy, the sequence of operations described above may be changed.

Gas-Cut Mud

A gas-cut mud is a warning signal of possible formation fluid influx, although it is not necessarily a serious problem. Due to gas expandibility, it usually gives the appearance of being a more serious problem than it actually is.

The bottomhole hydrostatic head of gas-cut mud and the expected pit gain can be calculated from the following equations:

$$P_h = P_s + \frac{\gamma_s H}{(1 - a_s)} - \frac{P_s a_s}{(1 - a_s)} \ln\left(\frac{P_h}{P_s}\right) \tag{4-292}$$

$$V_p = \frac{A_n P_s a_s}{\bar{\gamma}_s} \ln\left(\frac{P_h}{P_s}\right) \tag{4-293}$$

where

$$\gamma_s = \frac{P_s M a_s}{zRT_{av}} + \gamma_m (1 - a_s) \tag{4-294}$$

where P_h = hydrostatic pressure of gas-cut mud in lb/ft², abs
P_s = casing pressure at the surface in lb/ft², abs
H = vertical hole depth with the gas-cut mud in ft
a_s = gas concentration in mud (ratio of gas to total fluid volume)
$\bar{\gamma}_s$ = gas-cut mud specific weight at surface in lb/ft³
A_n = cross-sectional area of annulus in ft²
γ_m = original mud specific weight in lb/ft³
z = the average gas compressibility factor
M = gas molecular mass
T_{av} = the average fluid temperature in °R
R = universal gas constant in lb • ft

Example 1

Calculate the expected reduction in bottomhole pressure and pit gain for the data as given below:

Hole size = $9\frac{5}{8}$ in.
Hole depth = 10,000 ft
Drilling rate = 60 ft/hr
Original mud density = 10 lb/gal
Mud flowrate = 350 gal/min
Formation pore pressure gradient = 0.47 psi/ft
Porosity = 30%
Water saturation = 25%
Gas saturation = 75%
Gas specific gravity = 0.6 (air specific gravity is 1.0)
Surface temperature = 540°R
Temperature gradient = 0.01°F/ft

Solution

Gas volumetric rate entering the annulus

$$\dot{V} = \frac{\pi(9.625)^2}{4 \times 144} \times 60 \times \frac{1}{60} \times 0.3 \times 0.75 = 0.1136 \text{ ft}^3/\text{min} = 0.85 \text{ gal/min}$$

Gas volumetric flowrate at surface (at constant temperature) is

$$\dot{V} = \frac{0.85 \times 4700}{14.7} = 272.6 \text{ gal/min}$$

The surface pressure is assumed to be 14.7 psia. Gas concentration at surface is

$$a_s = \frac{272.6}{272.6 + 350} = 0.4378$$

Mud weight at surface (assume z = 1.0) is

$$\gamma_s = \frac{14.7 \times 144 \times 0.6 \times 29 \times 0.4378}{1{,}544 \times 540} + 10 \times 7.48(1 - 0.4378)$$

$$= 42.07 \text{ lb/ft}^3, \text{ or } 5.62 \text{ lb/gal}$$

Hydrostatic pressure of gas-cut mud is

$$p_h = 14.7 + \frac{0.052 \times 5.62 \times 9{,}038}{(1 - 0.4378)} - \frac{14.7 \times 0.4378}{1 - 0.4378} \ln\left(\frac{p_h}{14.7}\right)$$

Solving the above yields

$$p_h = 4646 \text{ psia}$$

Hydrostatic head at the bottom of the hole is

$$P_{bh} = 0.052 \times 10(10{,}000 - 9{,}038) + 4{,}646 = 5{,}146 \text{ psia}$$

Since the hydrostatic pressure of the original mud is 5,214.7 psia, the reduction in the hydrostatic pressure is about 69 psi. Because the pore pressure at the vertical depth of 10,000 ft is 4,700 psi, the hydrostatic pressure of the gas-cut mud is sufficient to prevent any formation fluid kick into the hole.

The Closed Well

Upon shutting in the well, the pressure builds up both on the drillpipe and casing sides. The rate of pressure buildup and time required for stabilization depend upon formation fluid type, formation properties, initial differential pressure and drilling fluid properties. In Ref. [143] technique is provided for determining the shut-in pressures if the drillpipe pressure is recorded as a function of time. Here we assume that after a relatively short time the conditions are stabilized. At this time we record the shut-in drillpipe pressure (SIDPP) and the shut-in casing pressure (SICP). A small difference between their pressures indicates liquid kick (oil, saltwater) while a large difference is evidence of gas influx. This is true for the same kick size (pit gain).

Assuming the formation fluid does not enter the drillpipe, we know that the SIDPP plus the hydrostatic head of the drilling fluid inside the pipe equals the pressure of the kick fluid (formation pressure). The formation pressure is also equal to the SICP plus the hydrostatic head of the original mud, plus the hydrostatic head of the kick fluid in the annulus.

Thus

$$p_p = 0.052\bar{\gamma}_m H + SIDPP \tag{4-295}$$

$$= 0.052\bar{\gamma}_m (H - L) + 0.052\bar{\gamma}_f L + SICP \tag{4-296}$$

where H = vertical hole depth in ft
$\bar{\gamma}_m$ = original mud specific weight in lb/gal
$\bar{\gamma}_f$ = formation fluid specific weight in lb/gal
L = vertical length of the kick in ft
p_p = formation pore pressure in psi

If the hole is vertical the kick length can be calculated as

$$L = \frac{V}{VC_{dc}} \tag{4-297}$$

or

$$L = L_{dc} + \frac{V - L_{dc}VC_{dc}}{VC_{dp}} \tag{4-298}$$

where V = pit gain in bbl
VC_{dc} = annulus volume capacity opposite the drill collars in bbl/ft
L_{dc} = length of drill collar in ft
VC_{dp} = annulus volume capacity opposite the drillpipe in bbl/ft

Equation 4-297 is applicable if the kick length is shorter than the drill collars while Equation 4-298 is used if the kick fluid column is longer than the drill collars.

From Equations 4-295 and 4-296 it is found that the required mud weight increase can be calculated from

$$\Delta\gamma_m = \frac{19.23 \times SIDPP}{H} \tag{4-299}$$

The formation fluid density can be obtained from

$$\bar{\gamma}_f = \bar{\gamma}_m - \frac{SICP - SIDPP}{0.052H} \tag{4-300}$$

Kick Control Procedures

There are several techniques available for kick control (kick-killing procedures). In this section only three methods will be addressed.

1. *Driller's method.* First the kick fluid is circulated out of the hole and then the drilling fluid density is raised up to the proper density (kill mud

density) to replace the original mud. An alternate name for this procedure is the two circulation method.

2. *Engineer's method.* The drilling fluid is weighted up to kill density while the formation fluid is being circulated out of the hole. Sometimes this technique is known as the *one circulation method.*
3. *Volumetric method.* This method is applied if the drillstring is off the bottom.

The guiding principle of all these techniques is that bottomhole pressure is held constant and slightly above the formation pressure at any stage of the process. To choose the most suitable technique one ought to consider (a) complexity of the method, (b) drilling crew experience and training, (c) maximum expected surface and borehole pressures and (d) time needed to reestablish pressure overbalance and resume normal drilling operations.

Driller's Method

The driller's method of controlling a kick is accomplished in two main steps:

Step 1. The well is circulated at half the normal pump speed while keeping the drillpipe pressure constant (see Figure 4-352a). This is accomplished by adjusting the choke on the mud line so that the bottomhole pressure is constant and above the formation fluid pressure. To maintain a constant bottomhole pressure the formation fluid is allowed to expand, which usually results in a noticeable increase in casing pressure. This step is completed when the formation fluid is out of the hole. At this time casing pressure should be equal to the initial SIDPP if the well could be shut in.

Step 2. When the formation fluid is out of the hole, a kill mud is circulated down the drillpipe. To obtain constant bottomhole pressure, the casing pressure is kept constant (see Figure 4-352b) while the drillpipe pressure drops. Once the kill mud reaches the bottom of the hole the control moves back to the drillpipe side. The drillpipe pressure is maintained constant (almost constant) while the new mud fills the annulus.

Example 2

Consider the following data:

Vertical hole depth = 10,000 ft
Hole diameter = $8\frac{3}{4}$ in.
Drillpipe diameter = $4\frac{1}{2}$ in.
Mud density = 12 lb/gal
Yield point = 12 lb/100 ft^2
Circulating pressure at reduced speed = 800 psi
Shut-in drillpipe pressure = 300 psi

Calculate the following:

1. required mud weight to restore the safe overbalance
2. initial circulating pressure (ICP)
3. final circulating pressure (FCP)
4. specific weight of formation fluid

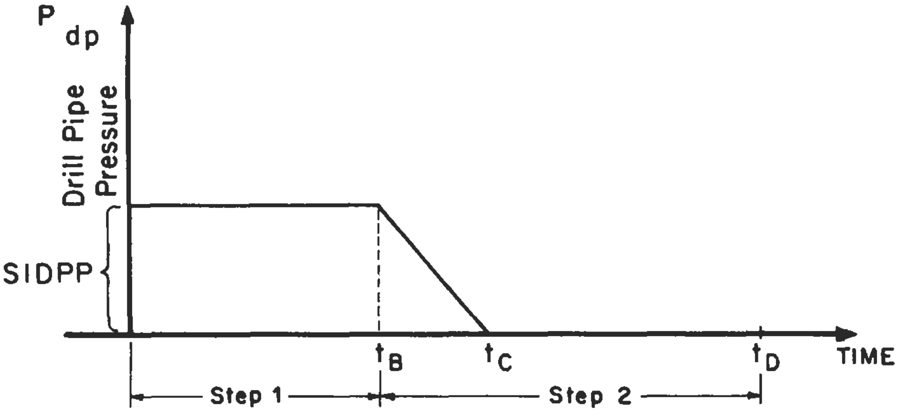


Figure 4-352a. Driller's method—Schematic diagram of drillpipe pressure vs. time.

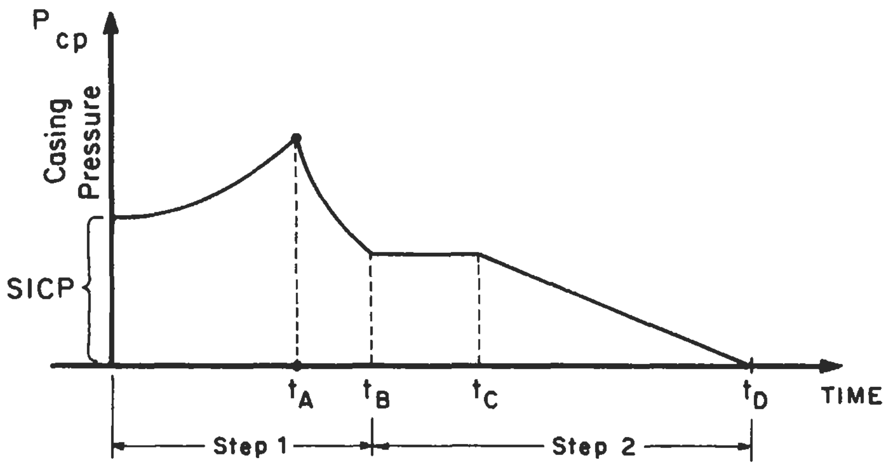


Figure 4-352b. Driller's method—Schematic diagram of casing pressure vs. time. t_A = kick fluid out the top of the hole; t_B = kick fluid out of the hole; t_C = kill mud at the bottom of the hole; t_D = killing procedure completed.

Solution

Mud weight increase to balance formation pressure is

$$\Delta\bar{\gamma}_m \cong \frac{20 \times 300}{10{,}000} = 0.6 \text{ lb/gal}$$

Trip margin $\Delta\bar{\gamma}_t$ is

$$\Delta\bar{\gamma}_t = \frac{yp}{6(D_h - D_p)} = \frac{12}{6(8.75 - 4.5)} = 0.47 \text{ lb/gal}$$

Consequently the required mud weight is

$$\bar{\gamma}_m = 12 + 0.6 + 0.47 \cong 13.1 \text{ lb/gal}$$

Initial circulating pressure is

$$\text{ICP} = 800 + 300 = 1{,}100 \text{ psi}$$

Final circulating pressure is the system pressure loss corrected for new mud weight. This is

$$\text{FCP} \cong 800\left(\frac{13.1}{12}\right) = 873 \text{ psi}$$

Engineer's Method

This method consists of four phases.

Phase 1. During this phase the drilling fluid, weighted to the desired density, is placed in the drillpipe. When the drillpipe is filled with heavier mud, the standpipe pressure is gradually reduced. The expected drillpipe pressure versus the number of pump strokes (or time) must be prepared in advance. Only by pumping with a constant number of strokes and simultaneously maintaining the standpipe pressure in accordance with the schedule can one keep the bottomhole pressure constant and above the formation pressure. The annulus pressure at the surface generally rises due to formation fluid expansion, although for some formation fluid the casing pressure may decrease. This depends on phase behavior of the formation fluid and irregularities in the hole geometry.

Phase 2. This phase is initiated when the kill mud begins filling the annulus and is finished when the formation fluid reaches the choke. The standpipe pressure remains essentially constant by proper adjustment of the choke.

Phase 3. The formation fluid is circulated out of the hole while heavier mud fills the annulus. Again the choke operator maintains the drillpipe pressure constant and constant pumping speed.

Phase 4. During this phase the original mud that follows the kick fluid is circulated out of the hole and a kill mud fills up the annulus. The choke is opened more and more to keep the drillpipe pressure constant. At the end of this phase the safe pressure overbalance is restored.

A qualitative relationship between the drillpipe pressure, casing pressure and circulating time is shown in Figures 4-353a and 4-353b, respectively.

Volumetric Method

This method can be used if the kick is taken during tripping up the hole with the bit far from the bottom of the hole. Again the constant bottomhole pressure principle is used to control the situation.

The fundamental principle of this method is equating the pit volume change with the corresponding change in annulus pressure. We write

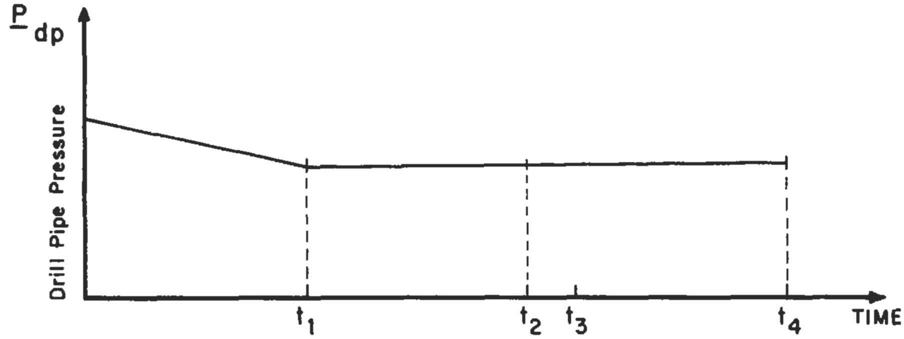


Figure 4-353a. Engineer's method—Drillpipe pressure v. time.

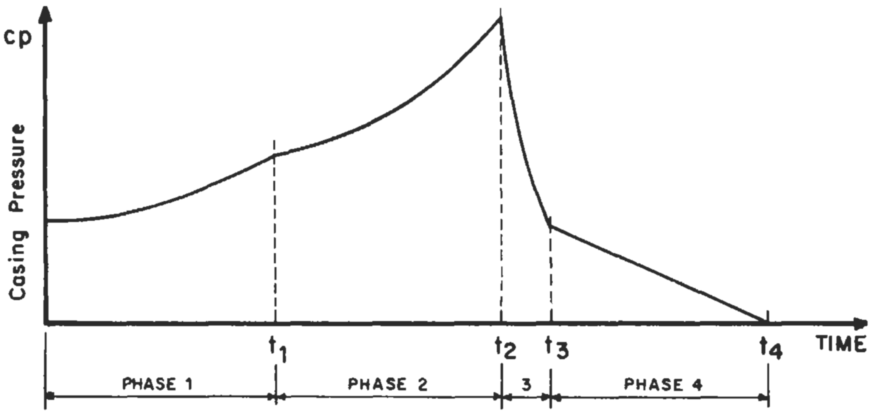


Figure 4-353b. Engineer's method—Casing pressure vs. time. t_1 = kill mud at the bottom of the hole; t_2 = formation fluid reaches the choke; t_3 = kick fluid out of the hole; t_4 = pressure overbalance is restored.

$$\Delta p_s = \frac{\Delta V_p}{AV} 0.052\bar{\gamma}_m \tag{4-301}$$

where Δp_s = change in surface pressure at casing side in psi
ΔV_p = change in pit volume (volume gained) in bbl
$\bar{\gamma}_m$ = mud specific weight in lb/gal
AV = volume capacity of annulus in bbl/ft

Note that the term ($0.052\ \gamma_m/AV$) expresses the expected increase of casing pressure when 1 bbl of pit gain is recorded.

The magnitude of the casing pressure during kick control is

$$P = SICP + \frac{\Delta V_p}{AV} 0.052\bar{\gamma}_m \tag{4-302}$$

When the pit volume stabilizes, there is equilibrium in the annulus and the kick fluid is out of the hole.

Maximum Casing Pressure

Determination of the maximum expected casing pressure is required for selection of the kick control technique. If the driller's method is used for kick control, the maximum casing pressure $P_{c\,max}$ (psi) is calculated assuming gas influx into the hole. This is

$$p_{c\,max} = \frac{SIDPP}{2} + \left[\left(\frac{SIDPP}{2}\right)^2 + 0.052\bar{\gamma}_m \frac{p_p \Delta V_p T_2 z_2}{z_1 T_1 AV}\right]^{0.5} \tag{4-303}$$

where SIDPP = shut-in drillpipe pressure in psi
$\bar{\gamma}_m$ = original mud density specific weight in lb/gal
P_p = formation pressure
ΔV_p = pit gain in bbl
T_2 = gas temperature at surface in °R
T_1 = gas temperature at the bottom of the hole in °R
z_1 = gas compressibility factor at surface conditions
z_2 = gas compressibility factor at the bottom of the hole
AV = volume capacity of annulus in bbl/ft

For the engineer's method the maximum expected casing pressure can be calculated from

$$p_{c\,max} = 0.026(\bar{\gamma}_k - \bar{\gamma}_m)L_m + \left(\left[0.026(\bar{\gamma}_k - \bar{\gamma}_m)L_m\right]^2 + 0.052\bar{\gamma}_k \frac{p_p \Delta V_p z_2 T_2}{z_1 T_1 AV}\right)^{0.5} \tag{4-304}$$

where $\bar{\gamma}_k$ = kill mud density
L_m = length of original mud in the annulus at the moment the bubble reaches the top of the hole

Example 3

Calculate the maximum expected casing pressure for the driller's and engineer's techniques of kick control for the data as below:

Vertical hole depth = 12,500 ft
Original mud weight = 12.0 lb/gal
Shut-in drillpipe pressure = 260 psi
Pit gain = 30 bbl
Volume of mud inside the drillstring = 175 bbl
Annular volume capacity = 0.13 bbl/ft
Gas compressibility ratio z_1/z_2 = 1.35
Gas temperature at the surface = 120°F

Well-bore temperature gradient = 1.2°F/100 ft
Surface temperature = 80°F

Solution

Required mud weight for killing the well is

$$\bar{\gamma}_k = 12 + \frac{260}{0.052 \times 12{,}500} + 0.5 = 12.9 \text{ lb/gal}$$

For trip margin, a 0.5 lb/gal was added arbitrarily.
Bottomhole temperature is

$$T_1 = 80 + 1.2 \times 125 = 230°F$$

Formation pore pressure is

$$p_p = 260 + 0.052 \times 12 \times 12{,}500 = 8{,}060 \text{ psi}$$

For the driller's method

$$p_{c\,max} = \frac{260}{2} + \left[\left(\frac{260}{2}\right)^2 + 0.052 \times 12 \frac{8{,}060 \times 30 \times 1.35 \times 580}{690 \times 0.13}\right]^{0.5} = 1{,}285 \text{ psi}$$

For the engineer's method

$$p_{c\,max} = 0.026(12.9 - 12.0)\frac{175}{0.13}$$

$$+ \left[(0.026 \times 0.9 \times 1{,}346.1)^2 + 0.052 \times 12.9 \frac{8{,}060 \times 30 \times 1.35 \times 580}{690 \times 0.13}\right]^{0.5}$$

$$= 1{,}222 \text{ psi}$$

The engineer's method normally results in lower maximum surface pressure than the driller's method. For fast field-type calculations, the following equation for the engineer's method can be used:

$$p_{c\,max} = 200\left(\frac{p_p \Delta V_p \bar{\gamma}_k}{AV}\right)^{0.5} \tag{4-305}$$

All symbols in Equation 4-305 are the same as those used above.

Maximum Borehole Pressure

When the top of the gas bubble just reaches the casing setting depth, the open part of the hole is exposed to the highest pressure. If this pressure is less than the formation fraction pressure, the kick can be circulated out of the hole safely without danger of an underground blowout.

If the driller's method is used, this maximum pressure p_{bm} can be obtained from

$$p_{bm}^2 - (p_p - 0.052 \times \bar{\gamma}_m \times H + 0.052\bar{\gamma}_m D_1)p_{bm} - 0.052\bar{\gamma}_m \frac{\Delta V_p p_p}{AV} = 0 \qquad (4\text{-}306)$$

Example 4

The data for this example are the same as the data used in Example 3.

Casing setting depth = 9,200 ft
Formation fracturing pressure at casing shoe = 7,550 psi

It is required to determine whether or not the formation at the casing shoe will break down while circulating the kick out of the hole. The annulus volume capacity in an open hole is 0.046 bbl/ft.

Solution

Using Equation 4-306 we write

$$p_{bm}^2 - (8{,}060 - 0.052 \times 12 \times 12{,}500 + 0.052 \times 12 \times 7{,}550)p_{bm}$$

$$- 0.052 \times 12 \times \frac{30 \times 8{,}060}{0.046} = 0$$

Solving the above equation yields p_{bm} = 5,561 psi. Since the maximum expected pressure in an open hole is less than the formation fracture pressure, the kick can be safely circulated out of the hole.

FISHING OPERATIONS AND EQUIPMENT

Fishing Operations

A fish is a part of the drillstring that separates from the upper remaining portion of the drillstring. This can result from the drillstring failing mechanically, or from the lower portion of the drillstring becoming stuck and having to be disconnected from the lower portion to instigate an operation to free and retrieve the lower portion with a strengthened, specialized string. Junk are small items that fall into the borehole, or are left behind in the borehole during drilling operations. These are nondrillable items that must be retrieved before drilling operations can continue [146–148].

Whenever there is a fish or junk in the hole, it must be removed from the borehole. The technique of removing the fish or junk from the borehole is called *fishing.*

It is important to remove the fish or junk from the borehole as quickly as possible. The longer these items remain in a borehole, the more difficult they will be to retrieve. Further, if the fish or junk is in an open-hole section of a borehole, the longer such a borehole remains open, the more likely borehole stability problems are to occur.

There is an important tradeoff that must be made during any fishing operation. Although the actual cost of a fishing operation is normally small compared to the cost of the drilling rig and other investments in the borehole, if a fish or junk cannot be removed from the borehole in a timely fashion, it may be necessary to sidetrack (directionally drill) around the fish or junk, or drill another borehole. Thus, the economies of the fishing operation and the other incurred costs at the well site must be carefully and continuously considered while the fishing operation is under way. It is very important to know when to terminate the fishing operation and get on with the primary objective of drilling the borehole.

Causes and Prevention

There are a number of causes for fishing operations. Many of the causes are preventable by careful planning of the drilling operation and being very watchful for the indication of possible future trouble [149].

The major causes are:

1. Mechanical fatigue and overstress of drillstring components probably accounts for a large portion of the fish and junk left in a borehole. The most common location of a drillstring failure is in the drillpipe just above the drill collars, usually in a tool joint at the base of the threaded pin. Also, drill collar tool joints are notorious failure locations. Again, the base of the threaded pin is the most likely location. Such possible failures can be prevented by conducting nondestructive testing on these drillstring components prior to placing them back in the borehole. Such nondestructive testing programs have been responsible for reducing fishing operations over the past two decades.
2. Stuck drillstring is responsible for many fishing operations. A drillstring can become stuck because of a number of problems. Pressure differential sticking, caving of the borehole wall, cuttings accumulations and key-seating of drill collars are a few of these problems. Often when the drillstring becomes stuck it is necessary to unscrew the unstuck portions of the drillstring, remove this portion, and return to the fish with a strengthened, specialized string for removing the fish. Usually there are signs that the drillstring is in danger of being stuck prior to the actual sticking of the string. The drilling crew must be constantly alert for these signs and react to them quickly. If these signs are not ignored, fishing operations can be avoided.
3. Broken bit components left behind in a borehole when the drillstring is removed and hand tools and other foreign objects falling in the borehole constitute junk that must be retrieved. These components cannot be drilled up during normal drilling operations. They may be milled with metal drilling bits and other special apparatus that can eventually remove these items in pieces. Such junk items can be very difficult to remove.
4. Logging cable and wireline can part due to the logging tool becoming stuck. Such cable and wireline can be removed by special fishing tools.
5. Production tubulars after long periods of service in a borehole can corrode and become weakened. When such tubulars are removed during well workovers, these tubulars may fail mechanically. Programs that have minimized

corrosion in production wells have decreased the necessity for fishing operations in production wells.

Fishing Tools

There are basically three categories of fishing tools. These are (1) those used to define the geometry and orientations of the fish or junk in the borehole, (2) those used to recover tubular items (fish), and (3) those used to recover miscellaneous items (junk).

Geometry and Orientation Tools

The tools used to define the geometry and orientation of the fish or junk in the borehole vary from sophisticated downhole televiewers to very simple impression blocks such as that shown in Figure 4-354. The impression block has a soft lead insert in the lower end of its steel housing. The impression block is made up to the end of the fishing drillstring and is lowered to the top of the fish (or junk). A small amount of weight is placed on the impression block, which

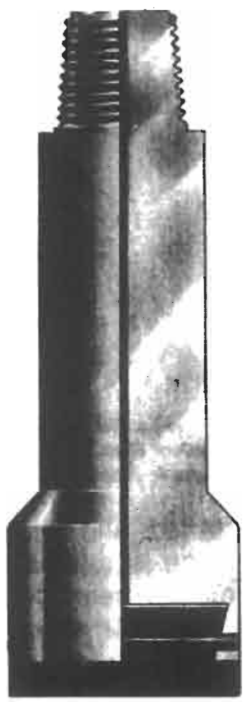


Figure 4-354. Impression block. *(Courtesy Bowen Tools, Inc.)*

places an imprint of the top of the fish on the lead insert. This imprint when recovered enables the operator to assess the configuration of the fish and its location in the borehole. The operator is then in a better position to select the proper tool needed to successfully complete the fishing operation. Sophisticated televiewers are run in the borehole much like any other logging tool. These services are obtained from companies that specialize in such equipment.

Tubular Fishing Tools

There are two subcategories of tubular fishing tools: those that can attach to the inside of the tubular fish and those that can attach to the outside of the tubular fish.

Inside Fishing Tools. Most of the fishing tools designed to catch tubular items from the inside are variations on the tap or spear concept. Figure 4-355 shows an example of a simple tap. A tapered tap is a tool with case-hardened lead. The tap when lowered on a drillpipe string can engage a fish when the upper part of the fish consists of an inside open element (box). The taper of the fishing tool permits easy entry into the upper part of the fish. The coarse lead allows for positive engagement with the box end of the upper part of the fish. A box tap uses a specific threaded spear to screw into a specific open box element.

Figure 4-356 shows an example of a complex spear fishing tool. Once the spear is inside the fish, rotation to the left will place the grapple in the engaging position. Upward pull will wedge the grapple in the fish. The fishing tool can be released if it becomes necessary to come out of the hole. Bump downward

Figure 4-355. Inside fishing tool, tap. *(Courtesy Bowen Tools, Inc.)*

to break the hold of the grapple and turn to the right a few turns and the fish should release.

Outside Fishing Tools. Outside fishing tools must pass over the outside of the fish before attaching themselves. The simplest of the outside fishing tools is the die collar, which is shown in Figure 4-357 with lipped guide. This is a short tubular that has case-hardened coarse threads cut in the inner surface of the tool. Such a tool is used to recover tubular items.

The overshot is another outside fishing tool. This tool consists of a tapered bowl in which slips are free to move up and down. Figure 4-358 shows a release type of overshot. The entire tool is designed to fit over the upper part of a fish. In the example the overshot is lowered on a drillpipe string and rotated slowly to the right until the fish is inside the overshot. Lifting the string allows the slips (or basket grapple) to wedge the upper part of the fish in the overshot. If it is necessary to release the grip of the overshot, the weight of the fish is dropped and simultaneously rotates to the right. This should release the fish when the string is slowly lifted.

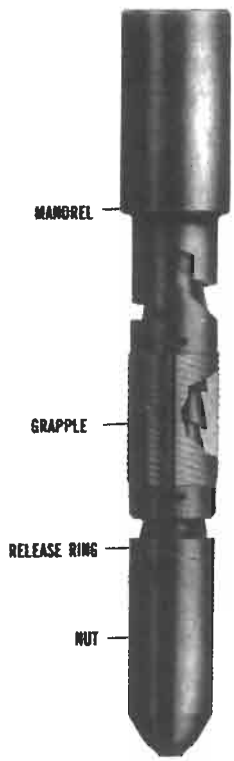


Figure 4-356. Inside fishing tool, spear. *(Courtesy Bowen Tools, Inc.)*

Figure 4-357. Outside fishing tool, die collar with lipped guide. *(Courtesy Bowen Tools, Inc.)*

Auxiliary Equipment. There are a number of special tools that are used that allow the fishing tools to attach themselves to the fish. An important auxiliary tool is the knuckle joint. Figure 4-359 shows the typical knuckle joint in the knuckled position and in the straight position. Such a device used in conjunction with an overshot (see Figure 4-358) and a wall-hook guide (see Figure 4-360) can be used effectively to engage a fish in a washed-out portion of the borehole. Figure 4-361 shows the useful action of the knuckle joint and the wall-hook guide in retrieving the fish from its position in the washout.

In some cases a stuck drillpipe or drill collar cannot be recovered by simple grappling of the pipe and pulling out of the hole. Often if circulation has ceased, incomplete formations will cave around the fish. In such situations, wash-over equipment must be used to expose the upper part of the fish so that the fishing tool can effectively engage it. In such situations a wash-over shoe is placed on the bottom of the fishing string. Above the wash-over shoe is about 200 to 500 ft of wash-over pipe and above the wash-over pipe is usually a wash-over safety joint. The entire wash-over string is placed on the bottom of a string of drillpipe (entire assembly constitutes the fishing string). The wash-over string must have an inside diameter that will allow the fish to pass into its interior once the cavings and cuttings are cut and washed away from the sides of the fish. Figure 4-362 shows a typical wash-over shoe. Figure 4-363 shows the wash-over safety joint that allows the wash-over string to be released if the string becomes stuck and provides the cross-over between the wash-over pipe and the drillpipe. Once the wash-over operation is completed and the upper end of the fish exposed, normal fishing tools can be used to retrieve the fish.

There are a number of other devices that can be used to aid in the retrieval of a fish. These vary from mechanical tubular cutting equipment to explosive devices. This type of equipment can be very useful in separating the fish into smaller parts that can be retrieved more easily than the whole fish.

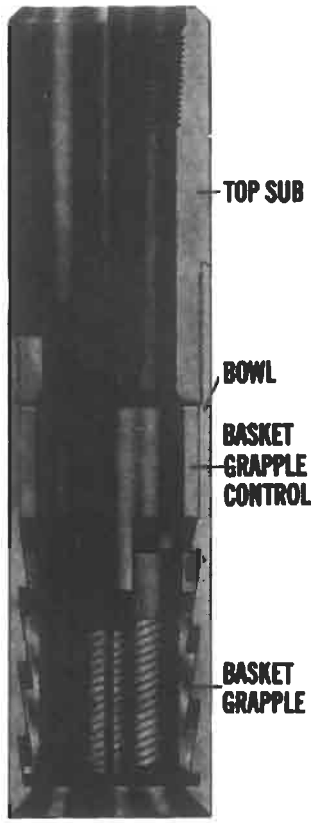
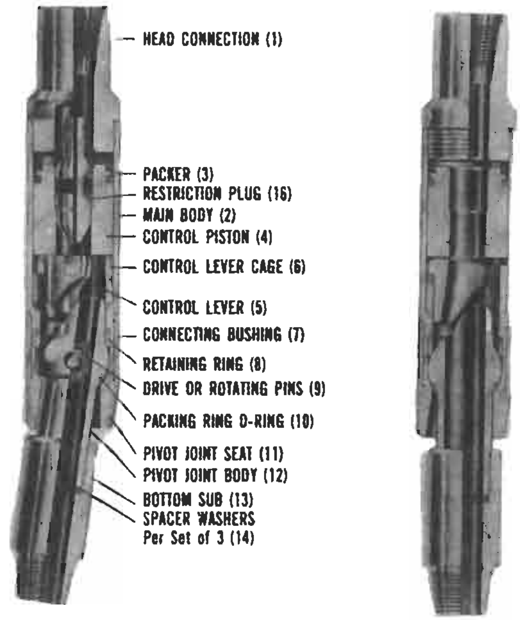


Figure 4-358. Outside fishing tool, overshot. *(Courtesy Bowen Tools, Inc.)*

Hydraulic and Impact Tools. When additional force is needed to remove a fish from the borehole, a hydraulic pulling tool can be used. The hydraulic pulling tool is essentially a downhole hydraulic jack. This tool attaches to the top of the fish and to slips that engage the casing. Using a series of pistons, a mechanical advantage as high as 100 to 1 can be obtained. This force is transmitted to the casing instead of to the derrick. In general, this tool can apply much greater and more direct force to a fish than can be brought to bear through the derrick and the drilling line. The hydraulic pulling tool is used in conjunction with normal fishing tools and is used above the fishing tools in the string. This special tool must be used in the cased portion of the borehole.

A jar is a device for providing an impact load to the fish when the fish cannot be retrieved by normal string and derrick forces. There are purely mechanical jars and hydraulic jars (see the section titled "Drilling Bits and Downhole Tools" for details on drilling jars). In a fishing operation the jar is usually placed

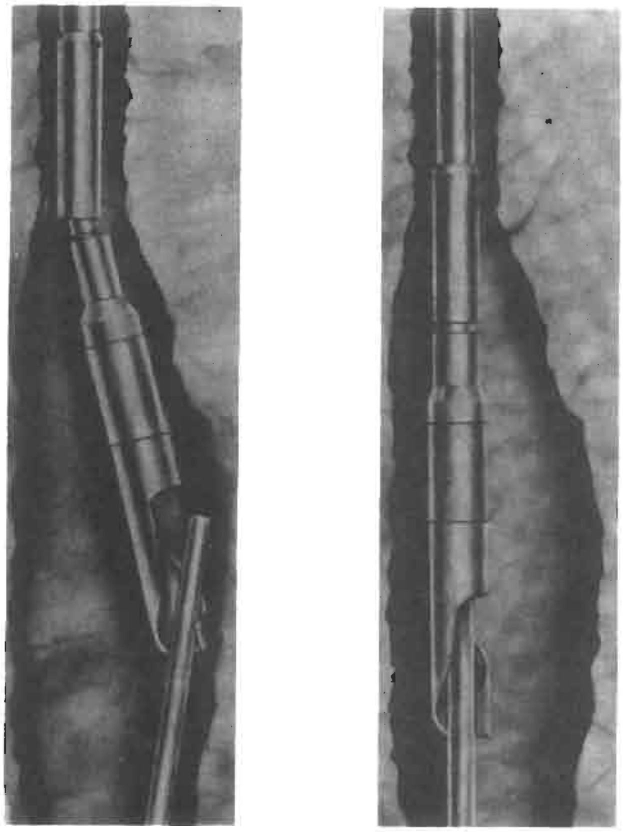


Knuckle Joint
in Knuckled Position

Knuckle Joint
in Straight Position

Figure 4-359. Knuckle joint auxiliary tool. *(Courtesy Bowen Tools, Inc.)*

Figure 4-360. Well-hook guide auxiliary tool. *(Courtesy Bowen Tools, Inc.)*



Knuckle Joint with Overshot and Wallhook Guide contacting Fish in a cavity

After contacting the Fish, ready for retrieval

Figure 4-361. Knuckle joint action to retrieve a fish in a washout. *(Courtesy Bowen Tools, Inc.)*

immediately above the fishing tool in the string. In this location the shock, or impulse, that the jar provides to the fish is attenuated the least.

The mechanical jar provides for a direct mechanical impact load from a metal-to-metal contact that is a multiple of the tension (or compression) attainable by the drillstring.

The hydraulic jar again uses a direct mechanical impact blow. The hydraulic fluid in this tool acts mainly to provide a delay while the desired derrick pull is achieved prior to actuation of the tool. Such tools may also be operated by compressed gas in a closed chamber. The compressed gas can be used to drive a hammer within the jar that strikes the top of a tool anvil.

Figure 4-362. Washover shoe. *(Courtesy Bowen Tools, Inc.)*

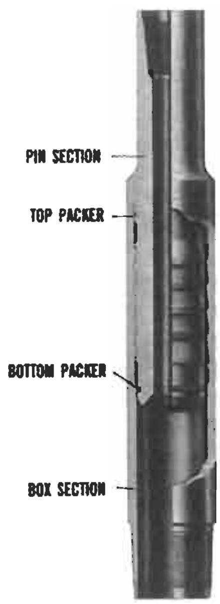


Figure 4-363. Washover safety joint. *(Courtesy Bowen Tools, Inc.)*

Junk Fishing Tools

There are items other than tubulars that must be fished from boreholes. These are drill bit parts, pieces of downhole tools left in the borehole and hand tools accidentally dropped in the borehole. Nearly all of these items cannot be drilled by a normal rock bit. Such items are called junk.

Milling Tool. The milling tool is a common device used for fishing junk. The milling tool is used to grind the junk into small pieces so that the pieces can be circulated to the surface or removed in a junk basket. Figure 4-364 shows the various milling tools that can be used to drill up junk in the bottom of the

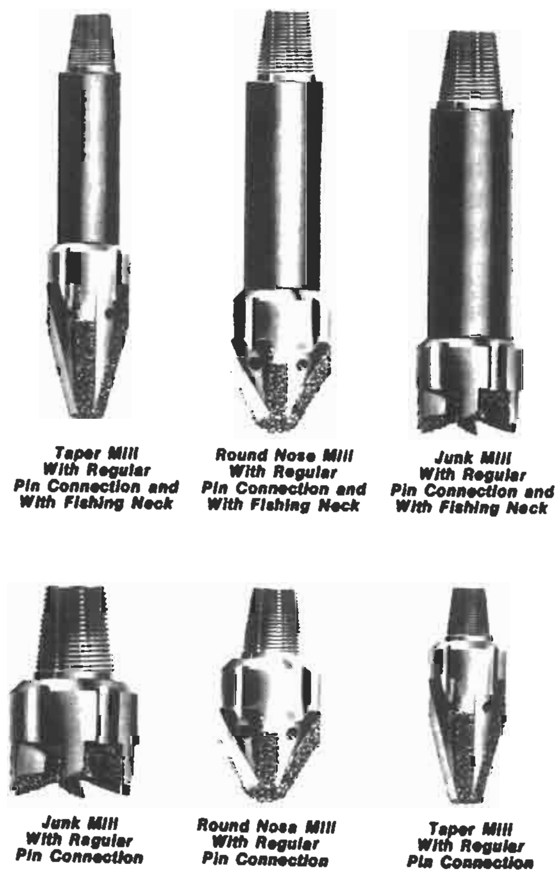


Figure 4-364. Milling tools. *(Courtesy Bowen Tools, Inc.)*

borehole. These milling tools are especially designed to drill steel. The milling tool is placed on the bottom of a drillstring designed to drill up junk. Such milling tools are designed with carbide-cutting surfaces. A properly designed milling tool will drill up the steel junk rapidly and not be effective in drilling rock.

Junk Basket. The junk basket is a tool that can be run in conjunction with a milling tool or separately. It is designed to recover the smaller pieces that have been milled or pieces too large to be circulated to the surface. Figure 4-365 shows a typical junk basket. The junk basket is placed directly above the milling tool in the string.

Magnetic Fishing Tools. Since many of the items that are lost in the borehole are steel, the use of a magnet to recover small junk has been very successful. Magnetic fishing tools may be either a permanent magnet type, or an electromagnetic type. Most in use are of the permanent magnet type. In the permanent magnet type, the permanent magnet is located inside a nonmagnetic section. Figure 4-366 shows a fishing magnet with ports to allow circulation through the tool body. The permanent magnet is a separate section of the tool. The fishing magnet is placed at the bottom of a string of drillpipe and is lowered to the bottom of the hole. There the magnet attracts and holds the junk in place while the string is retrieved.

Free Point

When a drillstring or other tubular becomes stuck in a borehole it is very important that the depth where the pipe is stuck be determined. In most cases this can be accomplished rather simply. The depth to where the drillstring is free and where sticking of the pipe commences is called the *free point* [148,150]. This free point depth can be calculated using measurements taken on the rig floor.

Figure 4-367 shows a pipe stuck at some depth L_f (ft) less than the total measure depth D (ft). The length L_f is the distance to the free point, or the length from the surface to where the pipe stuck. To obtain L_f, the following procedure is carried out:

1. An upward force F_1 is applied to the pipe via the drawworks. This force is slightly greater than the total weight of the drillstring. This ensures that the entire drillstring is in tension.
2. With this tension on the drillstring, a reference mark is made on the drillstring exposed at the top of the rotary table.
3. A larger upward force F_2 is then applied to the drillstring. This causes the free portion of the drillstring to elastically stretch by an amount L (ft). The stretch (or elastic displacement) is measured by the movement of the original reference mark. The magnitude of F_2 is limited by the yield, or elastic limit of the pipe steel.

Knowing the stretch L and the forces applied F_1 and F_2 and using Hooke's law (see the section titled *Strength of Materials* in Chapter 2), the length to the free point is

$$L_f = EA \frac{L}{(F_2 - F_1)} \tag{4-307}$$

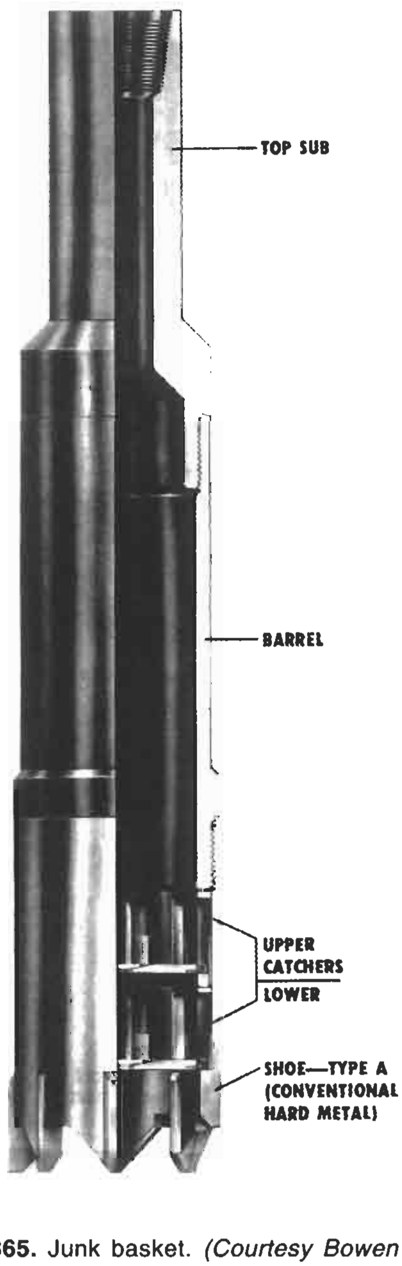


Figure 4-365. Junk basket. *(Courtesy Bowen Tools, Inc.)*

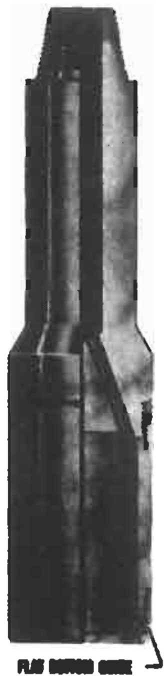


Figure 4-366. Fishing magnet. *(Courtesy Bowen Tools, Inc.)*

where E = Young's modulus (30×10^6 psi)
A = cross-sectional area of the pipe in ft^2
L = stretch distance in ft
F_1 and F_2 = forces placed on the drillstring in lb

Example 1

A string of $3\frac{1}{2}$ in. diameter, 13.3 lb/ft, Grade E drillpipe is stuck in a 12,000-ft borehole. The driller places 150,000 lb of tension on the drillstring above the normal weight of the drillstring and makes a mark on the drillstring at the top of the rotary table. The driller then increases the load to 210,000 lb beyond the weight of the drillstring (which is less than the yield of the pipe). The original mark on the drillstring shows that the free portion of the drillstring has stretched 4 ft due to the additional load placed on the drillstring by the drawworks. Where is the drillstring stuck?

The cross-sectioned area of the drillpipe is approximately

$$A = \frac{13.3}{490}$$

$$= 0.0271 \text{ ft}^2$$

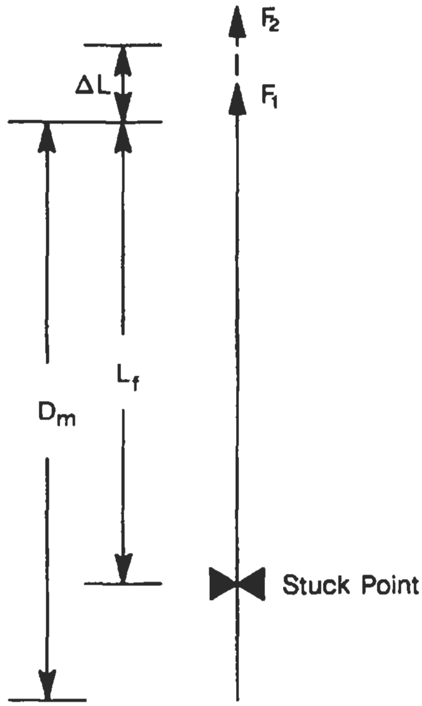


Figure 4-367. Free point.

where the specific weight of steel is 490 lb/ft³.

Equation 4-308 becomes

$$L_f = \frac{(30 \times 10^6)(144)(0.0271)(4)}{(210{,}000 - 150{,}000)}$$

$$= 7805 \text{ ft}$$

CASING AND CASING STRING DESIGN

Types of Casing

Based upon the primary function of the casing string, there are five types of casing to be distinguished.

Stove or Surface Casing

The stove pipe is usually driven to sufficient depth (15–60 ft) to protect loose surface formation and to enable circulation of the drilling fluid. This pipe is sometimes cemented in predrilled holes.

Conductor String

This string acts as a guide for the remaining casing strings into the hole. The purpose of conductor string is also to cover unconsolidated formations and to seal off shallow overpressured formations. The conductor string is the first string that is always cemented to the top and equipped with casing head and blowout prevention (BOP) equipment.

Surface Casing

This is set deeply enough to protect the borehole from caving-in in loose formations frequently encountered at shallow depths, and protects the freshwater sands from contamination while subsequently drilling a deeper hole. In case the conductor string has not been set, the surface casing is fitted with casing head and BOP.

Intermediate Casing

Also called *protection string,* this is usually set in the transition zone before abnormally high formation pressure is encountered, to protect weak formations or to case off loss-of-circulation zones. Depending upon geological conditions, the well may contain two or even three intermediate strings. Production string (oil string) is the string through which the well is produced.

Intermediate or production string can be set as a liner string. The liner string extends from the bottom of the hole upward to a point about 150–250 ft above the lower end of the upper string.

Casing Program Design

Casing program design is accomplished by two steps. In the first step, the casing sizes and corresponding bit sizes should be determined. In the second step, the setting depth of individual casing strings ought to be evaluated. Before starting the casing program design, the designer ought to know the following basic information:

- the purpose of the well (exploratory or development drilling);
- geological cross-sections that should consist of type of formations, expected hole problems, pore and formation's fracture pressure, number and depth of water, oil, gas horizons;
- available rock bits and casing sizes;
- load capacity of a derrick and mast if the type of rig has already been selected.

Before starting the design, it must be assumed that the production casing size and depth of the well have been established by the petroleum engineer in cooperation with a geologist, so that the hole size (rock bit diameter) for the casing may be selected. Considering the diameter of the hole, a sufficient clearance beyond the coupling outside diameter must be provided to allow for mud cake and also for a good cementing job. Field experience shows that the casing clearance should range from about 1.0 in to 3.5 in. Larger casing sizes require greater value of casing clearance. Once the hole size for production string has been selected, the smallest casing through which a given bit will pass is next determined. The bit diameter should be a little less (0.05 in.) than casing drift diameter. After choosing the casing with appropriate drift diameter, the outside coupling diameter of this casing may be found. Next, the appropriate size of the bit should be determined and the procedure repeated.

Example 1

The production casing string for a certain well is to consist of 5-in. casing. Determine casing and corresponding bit sizes for the intermediate, surface and conductor string. Take casing data and bit sizes from Table 4-140.

Solution

For production hole, select a $6\frac{3}{4}$-in. rock bit. Therefore, the casing clearance = 6.75 – 5.563 = 1.187 in.

For intermediate string, select a $7\frac{5}{8}$-in. casing, assuming that wall thicknesses that correspond to drift diameter of 6.640 and 6.5 in. will not be used. For the $7\frac{5}{8}$-in. intermediate string, use a $9\frac{7}{8}$-in. bit. The casing clearance = 9.875 – 8.5 = 1.375 in.

For surface string, select $10\frac{3}{4}$-in. casing. Note that only unit weights corresponding to drift diameters of 10.036 and 9.896 in. can be used. For the $10\frac{3}{4}$-in. casing, use a $13\frac{3}{4}$-in. bit, so the casing clearance = 13.75 – 11.75 = 2.0 in.

For conductor string, select 16-in. casing; the bit size will then be 20 in. and the casing clearance = 20 – 17 = 3 in.

Having defined bit and casing string sizes, the setting depth of the individual strings should be determined.

The operation of setting is governed by the principle according to which casing should be placed as deep as possible. However, the designer must remember to ensure the safety of the drilling crew from possible blowout, and to maintain the hole stability, well completion aspects (formation damage) and state regulations.

In general, casing should be set:

- where drilling fluid could contaminate freshwater that might be used for drinking or other household purposes;
- where unstable formations are likely to cave or slough into the borehole;
- where loss of circulation may result in blowout;
- where drilling fluid may severely damage producing horizon.

Currently, a graphical method of casing setting depth determination is used. The method is based on the principle according to which the borehole pressure should always be greater than pore pressure and less than fracture pressure.

For practical purposes, a safety margin for reasonable kick conditions should be imposed (see Figure 4-368). Even when the borehole pressure is adjusted correctly, problems may arise from the contact between the drilling fluid and the formation. It depends upon the type of drilling fluid and formation, but, in general, the more time spent drilling in an open hole, the greater the possibility of formation caving or sloughing into the borehole. Formation instability may lead to expensive work in the borehole, which influences the time and cost of the drilling operation. To arrest or reduce this problem, special treatment drilling fluids might be used, but these special drilling fluids are

Table 4-140
Casing and Rock Bit Sizes

Outside Diameter of Casing (inches)	Outside Diameter of Coupling (inches)	Drift Diameter (inches)
4½	5.000	3.965
		3.875
		3.795
5	5.563	4.435
		4.369
		4.283
		4.151
5½	6.050	4.919
		4.887
		4.825
		4.767
		4.653
		4.545
6⅝	7.390	6.010
		5.924
		5.796
		5.666
		5.550
7	7.656	6.413
		6.331
		6.241
		6.151
		6.059
		5.969
		5.879
		5.795
7⅝	8.500	7.000
		6.900
		6.844
		6.750
		6.640
		6.500
8⅝	9.625	7.972
		7.892
		7.796
		7.700
		7.600
		7.500
		7.386
9⅝	10.625	8.907
		8.845
		8.765
		8.679
		8.599
		8.525
		8.379
10¾	11.750	10.036
		9.894
		9.794
		9.694
		9.604
		9.504
		9.404
11¾	12.750	10.994
		10.928
		10.844
		10.724
		10.616
13⅜	14.375	12.559
		12.459
		12.359
		12.259
		12.191
16	17.000	15.188
		15.062
		14.938
18⅝	20.000	17.567
20	21.000	18.936

These data have been extracted from API Bulletin 5C2, API Specifications 5A and Hughes Tool Company Bit Handbook. Source: From Ref. [155].

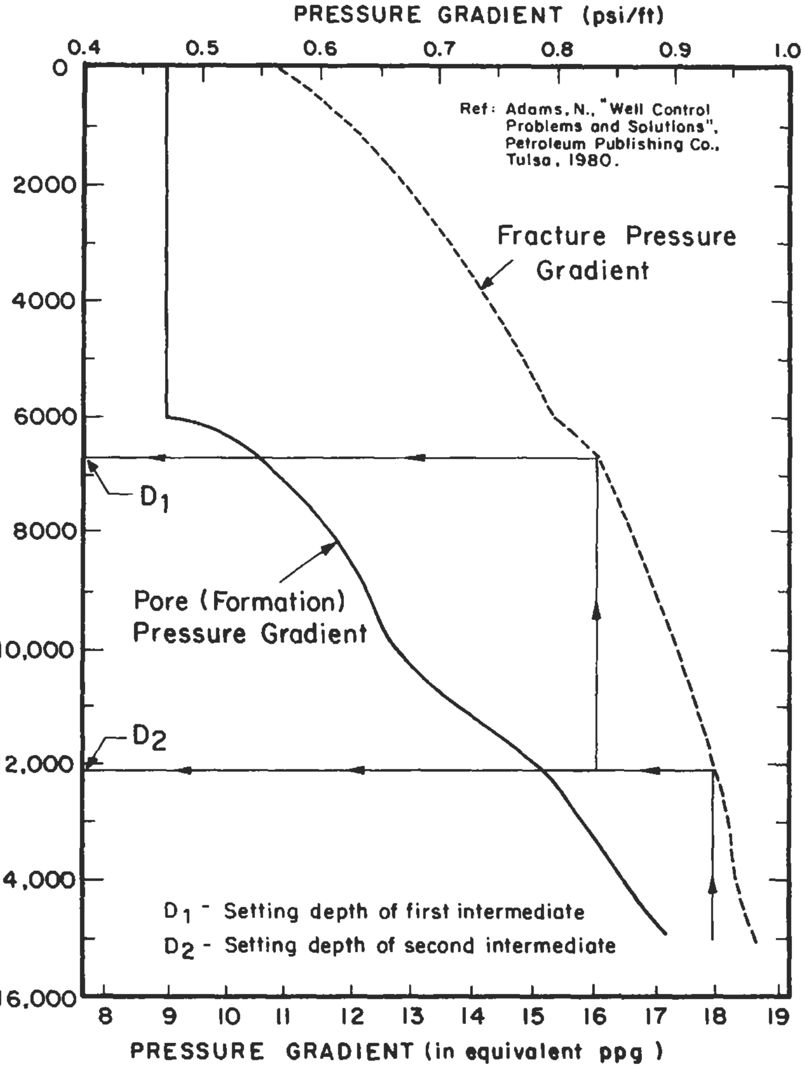


Figure 4-368. Well planning. *(From Ref. [157].)*

expensive. Therefore, the casing and drilling fluid programs depend on each other, and solving the issue of the correct casing setting depth evaluation is a rather complicated, optimizing problem.

Example 2

Suppose that in some area the expected formation pressure gradient is 0.65 psi/ft and formation fracture pressure gradient is 0.85 psi/ft. A gas-bearing formation is expected at depth of 15,000 ft. Assume that a gas kick occurs that, to be removed from the hole, induces a surface pressure of 2,000 psi. The first intermediate casing is set at a depth of 7,200 ft. Determine the setting depth for the second intermediate casing string if required in given conditions. Assume drilling fluid pressure gradient = 0.65 psi/ft.

Solution

The formation fracture pressure line is

$$P_f = (0.85)(D)$$

The borehole pressure line is

$$P_{bh} = 2000 + (0.65)(D)$$

so

$$(0.85)(D) = 2{,}000 + (0.65)(D)$$

$$D = \frac{2000}{0.2} = 10{,}000\,\text{ft}$$

The second intermediate casing string is required and must be set at a depth of 10,000 ft.

Casing Data

Special Note: Nothing in this specification should be interpreted as indicating a preference by the committee for any material or process, nor as indicating equality between the various materials or processes. In the selection of materials and processes, the purchaser must be guided by his or her experience and by the service for which the pipe is intended.

Casing is classified according to its manner of manufacture, steel grade, dimensions and weights, and the type of coupling.

The following are excerpts from API Spec. 5A, 35th Edition, March 1981.

Process of Manufacture

Casing and liners shall be seamless or electric welded as defined below and as specified on the purchase.

a. Seamless pipe is defined as a wrought steel tubular product made without a welded seam. It is manufactured by hot working steel or, if necessary, by subsequently cold finishing the hot-worked tubular product to produce the desired shape, dimensions and properties.
b. Electric-welded pipe is defined as pipe having one longitudinal seam formed by electric-flash welding or electric-resistance welding, without the addition of extraneous metal. The weld seam of electric-welded pipe shall be heat treated after welding to a minimum temperature of 1,000°F (538°C), or processed in such a manner that no untempered martensite remains.

Physical Properties (Section 4, API Specification 5A)

Tensile Properties. The tensile properties, except elongation, of the upset ends of the casing and tubing shall comply with the requirements given for the pipe body (see Table 4-141). In case of dispute, the tensile properties (except elongation) of the upset shall be determined from a tensile test cut from the upset.

Where elongation is recorded or reported, the record or report shall show the nominal width of the test specimen when strip specimens are used, or state when full section specimens are used.

Yield Strength. The yield strength (except of steel grades P-105, P-110) shall be the tensile strength required to produce a total elongation of 0.5% of the gage length as determined by an extensometer or by multiplying dividers. In case of steel grade P-105 and P-110, the yield strength is defined as the tensile stress required to produce a total elongation of 0.6% of the gage length.

Tensile Tests. Tensile properties shall be determined by tests on longitudinal specimens conforming to the requirements of Paragraph 4.4 (see API Specification 5A) and ASTM A370: Mechanical Testing of Steel Products, Supplement II, Steel Tabular Products.

Dimensions, Weights and Lengths (Section 6, API Specification 5A)

Dimensions and Weights

Pipes shall be furnished in the sizes, wall thicknesses and weights as specified on the purchase order. (See Tables 4-142 to 4-145 for more detailed information on dimensions and weights of casing.)

Table 4-141
Casing Properties

Steel Grade	Yield Strength (psi) min	Yield Strength (psi) max	Tensile Strength (psi) min	Min. elongation %, 2" specimens
H–40	4,000	—	60,000	The minimum elongation in 2" specimens is determined by formula:
J–55	55,000	80,000	75,000	$e = 62500(A^{0.2}/U^{0.9})$
K–55	55,000	80,000	95,000	where:
C–75	75,000	90,000	95,000	A = cross-sectional area of tensile test specimen, in.^{2};
N–80	80,000	110,000	100,000	U = specified tensile strength, psi.
L–80	80,000	95,000	95,000	
C–95	95,000	110,000	105,000	
P–105 (tubing)	105,000	135,000	120,000	
P-110	110,000	135,000	125,000	

Note: There are also other grades of casing available from various manufacturers. For example, Lone Star offers steel grades SS-95 for sour service, S-80 and S-95 for nominal service. Mannesmann offers a C-90 for sour service and S00-95, 124, 140 for high-strength service. There is also available a tentative API grade V-150 (Y_{min} = 150,000 psi, Y_{max} = 180,000 psi), which has not yet been adopted by the API.

Table 4-142
Short Round-Thread Casing Dimensions and Weights

1	2	3	4	5	6
				Calculated Weight	
Size: Outside Diameter, in. D	Nominal Weight:[1] Threads and Coupling, lb per ft	Wall Thickness, in. t	Inside Diameter, in. d	Plain End, lb/ft w_{pe}	Threads[2] and Coupling lb e_w
4½	9.50	0.205	4.090	9.40	4.20
4½	10.50	0.224	4.052	10.23	3.80
4½	11.60	0.250	4.000	11.35	3.40
5	11.50	0.220	4.560	11.23	5.40
5	13.00	0.253	4.494	12.83	4.80
5	15.00	0.296	4.408	14.87	4.20
5½	14.00	0.244	5.012	13.70	5.40
5½	15.50	0.275	4.950	15.35	4.80
5½	17.00	0.304	4.892	16.87	4.40
6⅝	20.00	0.288	6.049	19.49	11.00
6⅝	24.00	0.352	5.921	23.58	9.60
7	17.00	0.231	6.538	16.70	10.00
7	20.00	0.272	6.456	19.54	9.40
7	23.00	0.317	6.366	22.63	8.00
7	26.00	0.362	6.276	25.66	7.20
7⅝	24.00	0.300	7.025	23.47	15.80
7⅝	26.40	0.328	6.969	25.56	15.20
8⅝	24.00	0.264	8.097	23.57	23.60
8⅝	28.00	0.304	8.017	27.02	22.20
8⅝	32.00	0.352	7.921	31.10	20.80
8⅝	36.00	0.400	7.825	35.14	19.40
9⅝	32.30	0.312	9.001	31.03	24.40
9⅝	36.00	0.352	8.921	34.86	23.00
9⅝	40.00	0.395	8.835	38.94	21.40
10¾	32.75	0.279	10.192	31.20	29.00
10¾	40.50	0.350	10.050	38.88	26.40
10¾	45.50	0.400	9.950	44.22	24.40
10¾	51.00	0.450	9.850	49.50	22.60
10¾	55.50	0.495	9.760	54.21	20.80
11¾	42.00	0.333	11.084	40.60	29.60
11¾	47.00	0.375	11.000	45.56	27.60
11¾	54.00	0.435	10.880	52.57	25.00
11¾	60.00	0.489	10.772	58.81	22.60
13⅜	48.00	0.330	12.715	45.98	33.20
13⅜	54.50	0.380	12.615	52.74	30.80
13⅜	61.00	0.430	12.515	59.45	28.40
13⅜	68.00	0.480	12.415	66.11	25.80
13⅜	72.00	0.514	12.347	70.60	24.20
16	65.00	0.375	15.250	62.58	42.60
16	75.00	0.438	15.124	72.72	38.20
16	84.00	0.495	15.010	81.97	34.20
18⅝	87.50	0.435	17.755	84.51	73.60
20	94.00	0.438	19.124	91.51	47.00
20	106.50	0.500	19.000	104.13	41.60
20	133.00	0.635	18.730	131.33	30.00

[1]Nominal weights, threads and coupling (Col. 2), are shown for the purpose of identification in ordering.

[2]Weight gain due to end finishing.

Source: From Ref. [151].

Table 4-143
Long Round-Thread Casing Dimensions and Weights

1	2	3	4	5	6
				Calculated Weight	
Size: Outside Diameter, in. D	Nominal Weight:[1] Threads and Coupling, lb. per ft.	Wall Thickness, in. t	Inside Diameter in. d	Plain End, lb/ft w_{pe}	Threads[2] and Coupling, lb e_w
4½	11.60	0.250	4.000	11.85	3.80
4½	13.50	0.290	3.920	13.04	3.20
5	13.00	0.253	4.494	12.83	5.80
5	15.00	0.296	4.408	14.87	5.20
5	18.00	0.362	4.276	17.93	4.20
5	21.40	0.437	4.126	21.30	2.95
5	24.10	0.500	4.000	24.03	1.95
5½	15.50	0.275	4.950	15.85	5.80
5½	17.00	0.304	4.892	16.87	5.40
5½	20.00	0.361	4.778	19.81	4.40
5½	23.00	0.415	4.670	22.54	3.20
6⅝	20.00	0.288	6.049	19.49	13.60
6⅝	24.00	0.352	5.921	23.58	12.00
6⅝	28.00	0.417	5.791	27.65	10.20
6⅝	32.00	0.475	5.675	31.20	8.80
7	23.00	0.317	6.366	22.63	10.40
7	26.00	0.362	6.276	25.66	9.40
7	29.00	0.408	6.184	28.72	8.00
7	32.00	0.453	6.094	31.68	6.60
7	35.00	0.498	6.004	34.58	5.60
7	38.00	0.540	5.920	37.26	4.40
7⅝	26.40	0.328	6.969	25.56	19.00
7⅝	29.70	0.375	6.875	29.04	17.40
7⅝	33.70	0.430	6.765	33.04	15.80
7⅝	39.00	0.500	6.625	38.05	13.60
7⅝	42.80	0.562	6.501	42.39	12.01
7⅝	47.10	0.625	6.375	46.72	10.16
8⅝	32.00	0.352	7.921	31.10	27.60
8⅝	36.00	0.400	7.825	35.14	25.60
8⅝	40.00	0.450	7.725	39.29	23.80
8⅝	44.00	0.500	7.625	43.39	21.80
8⅝	49.00	0.557	7.511	48.00	19.60
9⅝	36.00	0.352	8.921	34.86	32.00
9⅝	40.00	0.395	8.835	38.94	30.00
9⅝	43.50	0.435	8.755	42.70	28.20
9⅝	47.00	0.472	8.681	46.14	26.60
9⅝	53.50	0.545	8.535	52.85	23.40
20	94.00	0.438	19.124	91.51	61.20
20	106.50	0.500	19.000	104.13	54.80
20	133.00	0.635	18.730	131.33	40.60

[1]Nominal weights, threads and coupling (Col. 2), are shown for the purpose of identification in ordering.

[2]Weight gain due to end finishing.

Source: API Spec 5A, 35th Edition, March 1981, p. 23. From Ref. [151].

Diameter

The outside diameter shall be within the tolerance specified in Table 4-146. For threaded pipe, the outside diameter at the threaded ends shall be such that the length L_4 and the full-crest thread length L_c are within the dimensions and tolerances in API Standard 5B. (Inside diameters are governed by the outside diameters and weight tolerances.)

Wall Thickness

Each length of pipe shall be measured for conformance to wall thickness requirements. The wall thickness at any place shall not be less than the tabulated thickness minus the permissible undertolerance specified in Table 4-146. Wall thickness measurements shall be made with a mechanical caliper or with a properly calibrated nondestructive testing device of appropriate accuracy. In

Table 4-144
Buttress Thread Casing Dimensions and Weights

1	2	3	4	5	6	7
				Calculated Weight		
					Threads and Coupling[2]	
Size: Outside Diameter, in. D	Nominal Weight:[1] Threads and Coupling, lb per ft	Wall Thickness, in. t	Inside Diameter, in. d	Plain End, lb/ft w_{pe}	Regular lb e_w	Special Clearance lb e_w
4½	10.50	0.224	4.052	10.23	5.00	2.56
4½	11.60	0.250	4.000	11.35	4.60	2.16
4½	13.50	0.290	3.920	13.04	4.00	1.56
5	13.00	0.253	4.494	12.83	6.60	2.42
5	15.00	0.296	4.408	14.87	5.80	1.62
5	18.00	0.362	4.276	17.93	4.40	0.22
5	21.40	0.437	4.126	21.30	2.46	−1.72
5	24.10	0.500	4.000	24.03	1.24	−2.94
5½	15.50	0.275	4.950	15.35	6.40	2.10
5½	17.00	0.304	4.892	16.87	5.80	1.50
5½	20.00	0.361	4.778	19.81	4.60	0.30
5½	23.00	0.415	4.670	22.54	3.40	−0.90
6⅝	20.00	0.288	6.049	19.49	14.40	2.38
6⅝	24.00	0.352	5.921	23.58	12.60	0.58
6⅝	28.00	0.417	5.791	27.65	10.60	−1.42
6⅝	32.00	0.475	5.675	31.20	9.00	−3.02
7	23.00	0.317	6.366	22.63	11.00	1.60
7	26.00	0.362	6.276	25.66	9.60	0.20
7	29.00	.0.408	6.184	28.72	8.20	−1.20
7	32.00	0.453	6.094	31.68	6.80	−2.60
7	35.00	0.498	6.004	34.58	5.60	−3.80
7	38.00	0.540	5.920	37.26	4.20	−5.20
7⅝	26.40	0.328	6.969	25.56	20.60	6.21
7⅝	29.70	0.375	6.875	29.04	18.80	4.41
7⅝	33.70	0.430	6.765	33.04	17.00	2.61
7⅝	39.00	0.500	6.625	38.05	14.60	0.21
7⅝	42.80	0.562	6.501	42.39	11.39	−3.01
7⅝	47.10	0.625	6.375	46.72	9.23	−5.17
8⅝	32.00	0.352	7.921	31.10	28.20	6.03
8⅝	36.00	0.400	7.825	35.14	26.20	4.03
8⅝	40.00	0.450	7.725	39.29	24.20	2.03
8⅝	44.00	0.500	7.625	43.39	22.20	0.03
8⅝	49.00	0.557	7.511	48.00	19.80	−2.37
9⅝	36.00	0.352	8.921	34.86	31.00	6.48
9⅝	40.00	0.395	8.835	38.94	29.00	4.48
9⅝	43.50	0.435	8.755	42.70	27.20	2.68
9⅝	47.00	0.472	8.681	46.14	25.60	1.08
9⅝	53.50	0.545	8.535	52.85	22.40	−2.12
10¾	40.50	0.350	10.050	38.88	34.40	7.21
10¾	45.50	0.400	9.950	44.22	31.80	4.61
10¾	51.00	0.450	9.850	49.50	29.40	2.21
10¾	55.50	0.495	9.760	54.21	27.00	−0.19
11¾	47.00	0.375	11.000	45.56	35.80	....
11¾	54.00	0.435	10.880	52.57	32.40	....
11¾	60.00	0.489	10.772	58.81	29.60	....
13⅜	54.50	0.380	12.615	52.74	40.20	....
13⅜	61.00	0.430	12.515	59.45	36.80	....
13⅜	68.00	0.480	12.415	66.11	33.60	....
13⅜	72.00	0.514	12.347	70.60	31.60	....
16	75.00	0.438	15.124	72.72	45.60	....
16	84.00	0.495	15.010	81.97	39.60	....
18⅝	87.50	0.435	17.755	84.51	86.40	....
20	94.00	0.438	19.124	91.51	54.80	....
20	106.50	0.500	19.000	104.13	48.40	....
20	133.00	0.635	18.730	131.33	35.20	....

[1]Nominal weights, threads and coupling (Col. 2), are shown for the purpose of identification in ordering.
[2]Weight gain due to end finishing.

Source: From API Spec 5A, 35th Edition, March 1981, p. 24, 25. From Ref. [151].

case of dispute, the measurements determined by use of the mechanical caliper shall govern.

Weight

Each length of casing shall be weighed separately. Threaded-and-coupled pipe shall be weighed with the couplings screwed on or without couplings, provided proper allowance is made for the weight of the couplings. Threaded and coupled pipe, integral joint pipe and pipe shipped without coupling shall be weighed without thread protectors except for carload weighings for which proper allowances shall be made for the weight of thread protectors. The weights determined as described above shall conform to the specified calculated weights (or adjusted calculated weight) for the end finish specified on the purchase order, within the tolerances stipulated in Table 4-146. Calculated weights shall be determined in accordance with the following formula:

$$W_L = (W_{pe} \times L) + e_w$$

Table 4-145
Extreme-Line Casing Dimensions and Weights

1	2	3	4	5	6	7
	Nominal Weight:			Calculated Weight		
					Upset and Threads	
Size: Outside Diameter, in. OD	Upset and Threaded lb per ft	Wall Thickness, in. WALL	Inside Dia., in. ID	Plain End, lb/ft w_{pe}	Std. lb e_w	Opt. lb e_w
5	15.00	0.296	4.408	14.87	4.60	....
5	18.00	0.362	4.276	17.93	1.40	....
5½	15.50	0.275	4.950	15.35	5.80	4.20
5½	17.00	0.304	4.892	16.87	4.80	3.20
5½	20.00	0.361	4.778	19.81	1.40	— 0.20
5½	23.00	0.415	4.670	22.54	0.00	— 1.60
6⅝	24.00	0.352	5.921	23.58	3.40	1.80
6⅝	28.00	0.417	5.791	27.65	0.20	— 1.40
6⅝	32.00	0.475	5.675	31.20	— 1.40	— 3.00
7	23.00	0.317	6.366	22.63	6.00	4.20
7	26.00	0.362	6.276	25.66	2.80	1.00
7	29.00	0.408	6.184	28.72	0.60	— 1.20
7	32.00	0.453	6.094	31.68	— 0.60	— 2.40
7	35.00	0.498	6.004	34.58	1.00	— 1.80
7	38.00	0.540	5.920	37.26	— 0.20	— 3.00
7⅝	26.40	0.328	6.969	25.56	6.40	4.00
7⅝	29.70	0.375	6.875	29.04	2.60	0.20
7⅝	33.70	0.430	6.765	33.04	0.00	— 2.40
7⅝	39.00	0.500	6.625	38.05	— 2.20	— 4.60
8⅝	32.00	0.352	7.921	31.10	13.20	9.80
8⅝	36.00	0.400	7.825	35.14	7.60	4.20
8⅝	40.00	0.450	7.725	39.29	4.00	0.60
8⅝	44.00	0.500	7.625	43.39	1.60	— 1.80
8⅝	49.00	0.557	7.511	48.00	— 0.80	— 4.20
9⅝	40.00	0.395	8.835	38.94	10.60	7.20
9⅝	43.50	0.435	8.755	42.70	5.40	2.00
9⅝	47.00	0.472	8.681	46.14	2.20	— 1.20
9⅝	53.50	0.545	8.535	52.85	— 1.20	— 4.60
10¾	45.50	0.400	9.950	44.22	21.20	....
10¾	51.00	0.450	9.850	49.50	18.40	....
10¾	55.50	0.495	9.760	54.21	15.80	....

Table 4-145 (continued)

1	2	8	9	10	11	12	13	14	15	16
		Pin and Box Dimensions—Finished								
						Pin and Box Made-up[2] (Power-Tight)				
Size: Outside Diameter, in.	Nominal Weight: Upset and Threaded lb. per ft.	Pin and Box Outside Dia. (Turned) in. +.020 −.010		Pin Inside Dia. (Bored) in. +.015 −.015	Box Inside Dia. (Bored) in. +.030 −.000	Outside Dia. in. +.020 −.010		Inside Dia. in. +.010 −.005	Drift Dia. for Finish Bored Upset Members in.	Drift Dia.[4] for Full Length Drifting, in. min.
OD		Std. M	Opt. M	Std. & Opt. B	Std & Opt. D	Std. 2	Opt. 2	Std. & Opt.	Std. & Opt.	Std. & Opt.
5	15.00	5.860	....	4.208	4.235	5.360	....	4.198	4.183	4.151
5	18.00	5.360	....	4.208	4.235	5.360	....	4.198	4.183	4.151
5½	15.50	5.860	5.780	4.746	4.773	5.860	5.780	4.736	4.721	4.653
5½	17.00	5.860	5.780	4.711	4.738	5.860	5.780	4.701	4.686	4.653
5½	20.00	5.860	5.780	4.711	4.738	5.860	5.780	4.701	4.686	4.653
5½	23.00	5.860	5.780	4.619	4.647	5.860	5.780	4.610	4.595	4.545
6⅝	24.00	7.000	6.930	5.792	5.818	7.000	6.930	5.781	5.766	5.730
6⅝	28.00	7.000	6.930	5.741	5.768	7.000	6.930	5.731	5.716	5.666
6⅝	32.00	7.000	6.930	5.624	5.652	7.000	6.930	5.615	5.600	5.550
7	23.00	7.390	7.310	6.182	6.208	7.390	7.310	6.171	6.156	6.151
7	26.00	7.390	7.310	6.182	6.208	7.390	7.310	6.171	6.156	6.151
7	29.00	7.390	7.310	6.134	6.160	7.390	7.310	6.123	6.108	6.059
7	32.00	7.390	7.310	6.042	6.069	7.390	7.810	6.032	6.017	5.969
7	35.00	7.530	7.390	5.949	5.977	7.530	7.390	5.940	5.925	5.879
7	38.00	7.530	7.390	5.869	5.897	7.580	7.390	5.860	5.845	5.795
7⅝	26.40	8.010	7.920	6.782	6.807	8.010	7.920	6.770	6.755	6.750
7⅝	29.70	8.010	7.920	6.782	6.807	8.010	7.920	6.770	6.755	6.750
7⅝	33.70	8.010	7.920	6.716	6.742	8.010	7.920	6.705	6.690	6.640
7⅝	39.00	8.010	7.920	6.575	6.602	8.010	7.920	6.565	6.550	6.500
8⅝	32.00	9.120	9.030	7.737	7.762	9.120	9.030	7.725	7.710	7.700
8⅝	36.00	9.120	9.030	7.737	7.762	9.120	9.030	7.725	7.710	7.700
8⅝	40.00	9.120	9.030	7.674	7.700	9.120	9.030	7.663	7.648	7.600
8⅝	44.00	9.120	9.030	7.575	7.602	9.120	9.030	7.565	7.550	7.500
8⅝	49.00	9.120	9.030	7.460	7.488	9.120	9.030	7.451	7.436	7.386
9⅝	40.00	10.100	10.020	8.677	8.702	10.100	10.020	8.665	8.650	8.599
9⅝	43.50	10.100	10.020	8.677	8.702	10.100	10.020	8.665	8.650	8.599
9⅝	47.00	10.100	10.020	8.633	8.658	10.100	10.020	8.621	8.606	8.525
9⅝	53.50	10.100	10.020	8.485	8.512	10.100	10.020	8.475	8.460	8.379
10¾	45.50	11.460	....	9.829	9.854	11.460	....	9.819	9.804	9.794
10¾	51.00	11.460	....	9.729	9.754	11.460	....	9.719	9.704	9.694
10¾	55.50	11.460	....	9.639	9.664	11.460	....	9.629	9.614	9.604

[1] Due to the nature of extreme-line casing, certain dimensional symbols and nomenclature differ from those for similar details for other pipe covered by this specification.
[2] Made-up joint OD is same as outside diameter dimension M.
[3] Shown for reference.
[4] See Table 6.11, Ref. 4.15.1.
[5] Weight gain or loss due to end finishing. See Par. 6.5, Ref. 4.15.1.
Source: From API Spec 5A, 35th Edition, March 1981, p. 26, 27. From Ref. [151].

where W_L = calculated weight of a piece of pipe of length L in lb
W_{pe} = plain-end weight
L = length of pipe, including end finish (see note on length determination) in ft
e_w = weight gain or loss due to end finish in lb

Length

Pipe shall be furnished in range lengths conforming to Table 4-147, as specified on the purchase order. When pipe is furnished with threads and couplings, the length shall be measured to the outer face of the coupling, or if

Table 4-146
Casing Weight Tolerance

Quantity	Tolerance
Outside Diameter, D	
4 in and smaller (tubing)	±0.031 in
4½ in and larger (casing)	+0.75 percent
Wall thickness, t	−12.75 percent
Weight	
Single Lengths	+6.5 percent −3.5 percent
Carload Lots	−1.75 percent
A carload is considered to be a minimum of 40,000 lbs.	

Table 4-147
Casing Length Range

Range Total range length, ft	1 16–25	2 25–34	3 34–48
Range length for 95 percent or more of carload			
Permissible variation max. ft	6	5	6
Permissible length min. ft	18	28	36

measured without couplings proper allowance shall be made to include the length of coupling. The extreme-line casing and integral joint tubing lengths shall be measured to the outer face of the box end. For pup joints and connectors, the length shall be measured from end to end.

Casing Jointers

If so specified on the purchase order for round-thread casing only, jointers (two pieces coupled to make a standard length) may be furnished to a maximum of 5% of the order, but no length used in making a jointer shall be less than 5 ft.

Coupling

API standards established three types of threaded joints:

1. coupling joints with rounded thread (Figures 4-369 and 4-370) (long or short)
2. coupling joints with asymmetrical trapezoidal thread buttress (Figure 4-371)
3. extreme-line casing with trapezoidal thread without coupling (Figure 4-372)

There are also many non-API joints, like Hydril "CTS," Hydril "Super FJ-P," Armco SEAL-LOCK, Mannesmann metal-to-metal seal casing and others.

The following are excerpts both from API Specification 5A, 35th Edition, March 1981 and API RP 5B1, 1st Edition, April 1983.

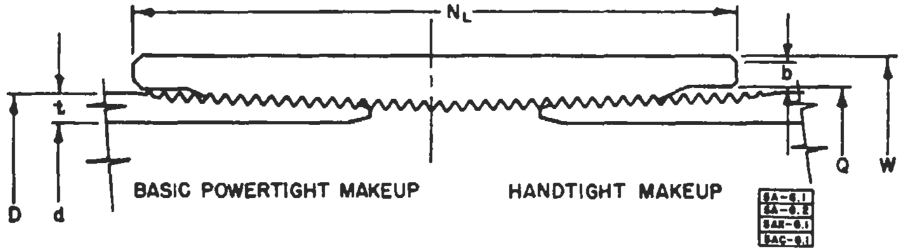


Figure 4-369. Short round-thread casing and coupling. *(From Ref. [51].)* Taken from API Spec 5A 35th Edition, Ref. [151], p. 22.
See Table 4.15.3 for pipe dimensions. See Table 4.15.9 for coupling dimensions. See API Std 5B for thread details.

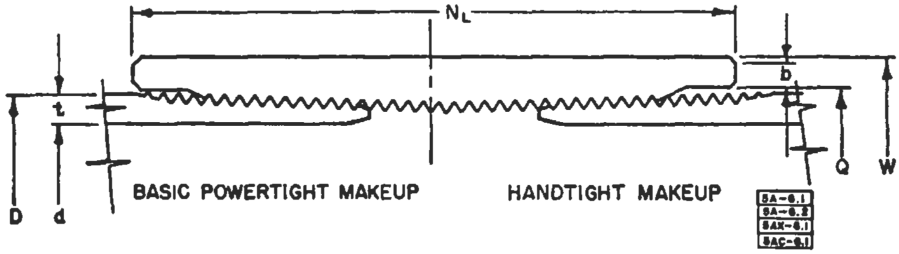


Figure 4-370. Long round-thread casing and coupling. *(From Ref. [51].)* From API Spec 5A, 35th Edition, March 1981, Ref. [151], p. 23.
See Table 4.15.4 for pipe dimensions. See Table 4.15.9 for coupling dimensions. See API Std 5B for thread details.

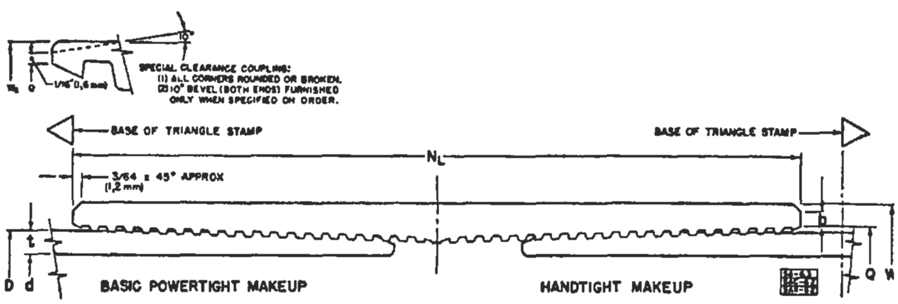
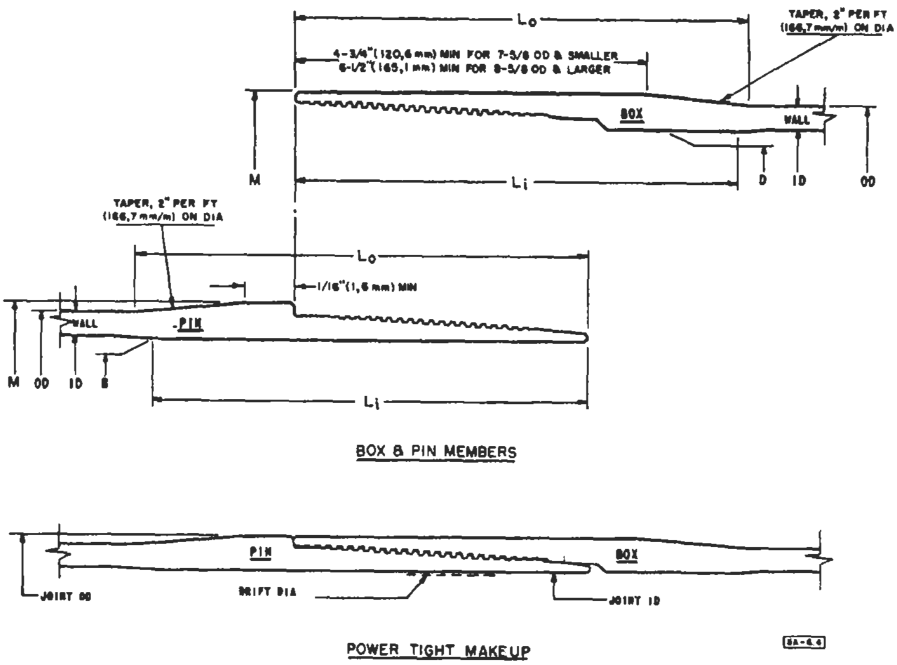


Figure 4-371. Buttress thread casing and coupling. *(From Ref. [51].)* From API Spec 5A, 35th Edition, March 1981, Ref. [151], p. 25.
See Table 4.15.5 for pipe dimensions. See Table 4.15.10 for coupling dimensions. See API Std 5B for thread details.



Size,	Length of Upset					
	Pin,† Min.		Box,† Min.		Pin or Box, Max	
in. OD	in.	mm (L_i)	in.	mm (L_i)	in.	mm (L_o)
5	6⅝	168,3	7	177,8	8	203,2
5½	6⅝	168,3	7	177,8	8	203,2
6⅝	6⅝	168,3	7	177,8	8	203,2
7	6⅝	168,3	7	177,8	8*	203,2*
7⅝	6⅝	168,3	7	177,8	8	203,2
8⅝	8	203,2	8¾	222,2	11	279,4
9⅝	8	203,2	8¾	222,2	11	279,4
10¾	8	203,2	8¾	222,2	12¾	323,8

†L_i is the minimum length from end of pipe of the machined diameter B on pin, or machined diameter D plus length of thread on box, to the beginning of the internal upset runout.

*L_o shall be 9 in. (228,6 mm) max. for 7 in. — 35 lb/ft and 7 in. — 38 lb/ft casing.

Figure 4-372. Extreme-line casing. *(From Ref. [14].)*
From API Spec 5A, 35th Edition, March 1981, p. 28.
See Table 4.15.6 for pipe dimensions. See API Std 5B for thread details.

Elements of Threads

Threaded connections are complicated mechanisms consisting of many elements that must interact in prescribed fashion to perform a useful function. Each of these elements of a thread may be gauged individually as described in API RP 5B1, 1st Edition, April 1983. The thread elements are defined as:

1. *Thread height or depth.* The thread height or depth is the distance between the threaded crest and the thread root normal to the axis of the thread (Figure 4-373).
2. *Lead.* For pipe thread inspection purposes, lead is defined as the distance from a point on a thread to a corresponding point on the adjacent thread measured parallel to the thread axis (Figure 4-373).
3. *Taper.* Taper is the change in diameter of a thread expressed in in./ft of thread length.

Round Threads (Figure 4-373)

The purpose of round top (crest) and round bottom (roots) is:

a. to improve the resistance of the threads from galling in make-up;
b. to provide a controlled clearance between make-up thread crest and root for foreign particles or contaminants; and
c. to make the crests less susceptible to harmful damage from minor scratches or dents. If insufficient interference is applied during makeup, the leak path through the connection could be through the annular clearance between mated crest and roots. Proper thread compound is necessary to ensure leak resistance.

Buttress Thread (Figure 4-374 and 4-375)

Buttress threads are designed to resist high axial tension or compression loading in addition to offering resistance to leakage.

The 3° load flank offers resistance to disengagement under high axial tension loading, while the 10° stub flank offers resistance to high axial compressive loading. In any event, leak resistance is again accomplished with use of proper thread compound and/or thread coating agents. Leak resistance is controlled by proper assembly (interference) within the perfect thread only.

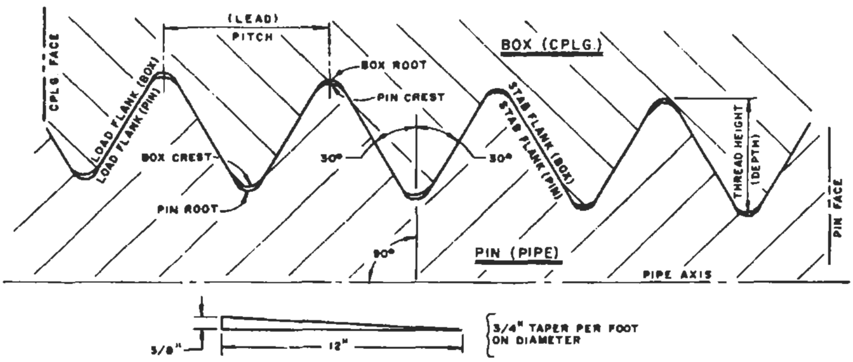


Figure 4-373. Round-thread casing and tubing thread configuration. *(From Ref. [154].)*

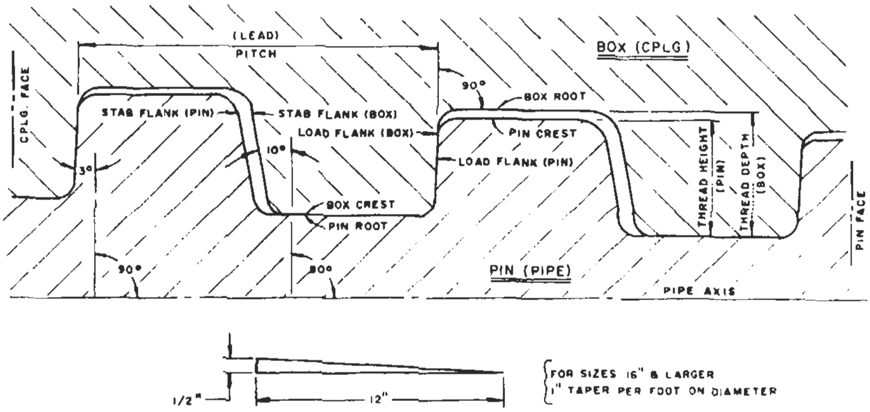


Figure 4-374. Buttress thread configuration for 16-in. outside diameter and larger casing. *(From Ref. [54].)*

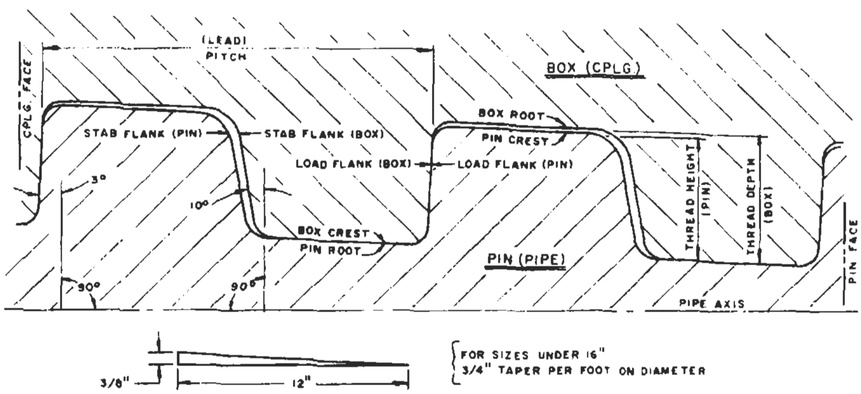


Figure 4-375. Buttress thread configuration for 13⅜-in. outside diameter and smaller casing. *(From Ref. [154].)*

Extreme-Line Casing (Integral Connection) (Figure 4-376)

Extreme-line casing in all sizes uses a modified acme type thread having a 12° included angle between stub and load flanks, and all threads have crests and roots flat and parallel to the axis (Figure 4-376). For all sizes, the threads are not intended to be leak resistant when made up. Threads are used purely as a mechanical means to hold the joint members together during axial tension loading. The connection uses upset pipe ends for pin and box members that are an integral part of the pipe body. Axial compressive load resistance is primarily offered by external shouldering to the connection or makeup.

Leak resistance is obtained on makeup by interference of metal-to-metal seal between a long radius curved seal surface on the pin member engaging a conical metal seal surface of the box member (Figures 4-372 and 4-376).

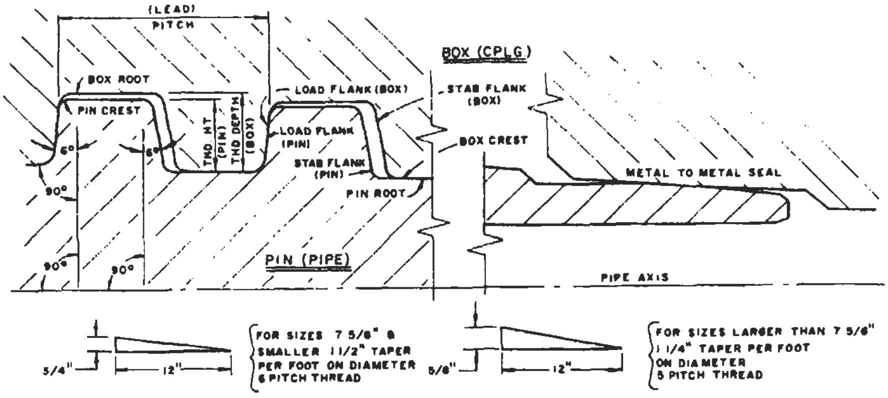


Figure 4-376. Extreme-line casing thread configuration. *(From Ref. [154].)*

Thread compound is not necessarily a critical agent to ensure leak resistance, but instead is used primarily as an antigalling or antiseizure agent.

Material

Couplings for pipe (both casing and tubing) shall be seamless and, unless otherwise specified on the purchase order, shall be of the same grade as the pipe, except grade H-40 and J-55 pipe which may be furnished with grade J-55 or K-55 couplings.

Note: Most buttress thread couplings will not develop the highest minimum joint strength unless couplings of the next higher order are specified. (See API Specification 5A for more detailed information.)

Physical Properties

Couplings for grades J-55, K-55 and N-80 shall conform to the tensile requirements specified in Table 4-142. A tensile test shall be made on each heat of steel from which couplings are produced, and the coupling manufacturer shall maintain a record of such tests. This record shall be open to inspection by the purchaser. Either round specimens proportioned as specified in ASTME 8: Tension Testing of Metallic Materials, or strip specimens shall be used, at the option of the manufacturer.

Dimensions and Tolerances

Couplings shall conform to the dimensions and tolerances shown in Tables 4-148 and 4-149. Unless otherwise specified, threaded and coupled casing and tubing shall be furnished with regular couplings.

Note: Couplings inspection procedures are described by API RP 5B1, First Edition, April 1983.

Thread Protectors

Design. The manufacturer shall apply external and internal thread protectors of such design, material and mechanical strength to protect the threaded end

Table 4-148
Round-Thread Casing Coupling Dimensions, Weights, and Tolerances

1	2	3	4	5	6	7	8
Size[1]	Outside Diameter W	Minimum Length		Diameter of Recess Q	Width of Bearing Face b	Weight, lb	
		Short N_L	Long N_L			Short	Long
4½	5.000	6¼	7	4 19/32	5/32	8.05	9.07
5	5.563	6½	7¾	5 3/32	3/16	10.18	12.56
5½	6.050	6¾	8	5 13/32	1/8	11.44	14.03
6⅝	7.390	7¼	8¾	6 23/32	¼	19.97	24.82
7	7.656	7¼	9	7 3/32	3/16	18.34	23.67
7⅝	8.500	7½	9¼	7 23/32	¼	26.93	34.23
8⅝	9.625	7¾	10	8 23/32	9/32	35.58	47.48
9⅝	10.625	7¾	10½	9 23/32	9/32	39.51	55.77
10¾	11.750	8	....	10 27/32	9/32	45.53	....
11¾	12.750	8	....	11 27/32	9/32	49.61	....
13⅜	14.375	8	....	13 15/32	5/16	56.23	....
16	17.000	9	....	16 3/32	5/16	78.98	....
18⅝	20.000	9	....	18 23/32	5/16	118.94	....
20	21.000	9	11½	20 3/32	5/16	98.25	126.74

Tolerance on outside diameter W, ±1 per cent, but not greater than ±⅛ in.

[1]The size of the coupling is the same as the corresponding pipe size.

All dimensions in inches. See Figures 4.15.2 and 4.15.3. Source: From Ref. [151], p. 38.

Table 4-149
Buttress Thread Casing Coupling Dimensions, Weights, and Tolerances

1	2	3	4	5	6	7	8
	Outside Diameter					Weight, lb	
Size[1]	Regular W	Special Clearance W_c	Minimum Length N_L	Diameter of Chamfer Q	Width of Bearing Face b	Regular	Special Clearance
4½	5.000	4.875	8⅞	4.640	⅛	10.11	7.67
5	5.563	5.375	9⅛	5.140	5/32	12.99	8.81
5½	6.050	5.875	9¼	5.640	5/32	14.14	9.84
6⅝	7.390	7.000	9⅝	6.765	¼	24.46	12.44
7	7.656	7.375	10	7.140	7/32	23.22	13.82
7⅝	8.500	8.125	10⅝	7.765	5/16	34.84	20.45
8⅝	9.625	9.125	10⅝	8.765	⅜	45.94	23.77
9⅝	10.625	10.125	10⅝	9.765	⅜	50.99	26.47
10¾	11.750	11.250	10⅝	10.890	⅜	56.68	29.49
11¾	12.750	.....	10⅝	11.890	⅜	61.74	....
13⅜	14.375	.....	10⅝	13.515	⅜	69.95	....
16	17.000	......	10⅝	16.154	⅜	87.56	....
18⅝	20.000	....	10⅝	18.779	⅜	138.03	....
20	21.000	....	10⅝	20.154	⅜	110.33	....

Tolerance on outside diameter W, ±1 per cent, but not greater than ±⅛ in.
Tolerance on outside diameter W_c $+\frac{1}{32}$, $-\frac{1}{32}$ in.

[1]The size of the coupling is the same as the corresponding pipe size.
All dimensions in inches. See Figure 14.5.4. Source: From Ref. [151], p. 38.

of the pipe from damage under normal handling and transportation. External thread protectors shall cover the full length of the thread on the pipe and internal thread protectors shall cover the equivalent total pipe thread length of internal thread. Thread protectors shall exclude water and dirt from the thread during transportation and normal storage period. Normal storage period shall be considered as approximately 1 year. The thread forms in protectors shall be such that the product threads are not damaged by the protectors.

Material. Protector material shall contain no compounds capable of causing corrosion or promoting adherence of the protectors of the threads and shall be suitable for service temperature (-50 to +150°F) (-46 to +66°C).

Performance Properties. Results of years of field experience have revealed that to reduce the risk of failure, the minimum yield strength should be used instead of average yield strength to determine the performance properties of casing.

Values for collapse resistance, internal yield pressure, pipe body, and joint strength for steel grades as in Table 4-141 are given in Table 4-150. Table 4-150 is directly taken from API Bulletin 5C2, 17th Edition, 1980. Formulas and procedures for calculating the values in Table 4-150 are given in Bulletin 5C3 [152] and are as follows:

Collapse Pressure (Section 1 of API 5C3)

Yield Strength Collapse Pressure Formula. The yield strength collapse pressure is not a true collapse pressure but rather the external pressure p_{yp} that generates minimum yield stress Y_p on the inside wall of a tube as calculated by

$$p_{yp} = 2Y_p\left|\frac{(D/t)-1}{(D/t)^2}\right| \tag{4-308}$$

The applicable D/t ratios for yield strength collapse are shown in Table 4-151.

Plastic Collapse Pressure Formula. The minimum collapse pressure for the plastic range of collapse is calculated by

$$p_P = Y_P\left|\frac{A}{D/t} - B\right| - C \tag{4-309}$$

The formula for minimum plastic collapse pressure is applicable for D/t values as shown in Table 4-151. The factors A, B, and C are given in Table 4-152.

Transition Collapse Pressure Formula. The minimum collapse pressure for the plastic to elastic transition zone P_T is calculated by

$$p_T = Y_P\left|\frac{F}{D/t} - G\right| \tag{4-310}$$

(text continued on page 1154)

Table 4-150
Minimum Performance of Casing

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
					Threaded and Coupled			Extreme Line				Internal Yield Pressure at Minimum Yield, psi							★ Joint Strength — 1,000 lbs: Threaded and Coupled						Extreme Line	
													Round Thread		Buttress Thread				Round Thread		Buttress Thread					
															Regular Coupling		Special Clearance Coupling									
Size: Outside Diameter in. *D*	Nominal Weight, Threads and Coupling lb per ft	Grade	Wall Thickness in. *t*	Inside Diameter in. *d*	Drift Diameter in.	Outside Diameter of Coupling in. *W*	Outside Diameter Special Clearance Coupling in. W_c	Drift Diameter in.	Outside Diameter of Box Powertight in. *M*	Collapse Resistance psi	Pipe Body Yield Strength 1000 lbs	Plain End or Extreme Line	Short	Long	Same Grade	Higher Grade*	Same Grade	Higher Grade	Short	Long	Regular Coupling	Regular Coupling Higher Grade:	Special Clearance Coupling	Special Clearance Coupling Higher Grade‡	Standard Joint	Optional Joint
4½	9.50	H-40	0.205	4.090	3.965	5.000	—	—	—	2,700	111	3,190	3,190	—	—	—	—	—	77	—	—	—	—	—	—	—
	9.50	J-55	0.205	4.090	3.965	5.000	—	—	—	3,310	152	4,380	4,380	—	—	—	—	—	101	—	—	—	—	—	—	—
	10.50	J-55	0.224	4.052	3.927	5.000	4.875	—	—	4,010	165	4,790	4,790	—	4,790	4,790	4,790	4,790	132	—	203	203	203	203	—	—
	11.60	J-55	0.250	4.000	3.875	5.000	4.875	—	—	4,960	184	5,350	5,350	5,350	5,350	5,350	5,350	5,350	154	162	225	225	225	225	—	—
	9.50	K-55	0.205	4.090	3.965	5.000	—	—	—	3,310	152	4,380	4,380	—	—	—	—	—	112	—	—	—	—	—	—	—
	10.50	K-55	0.224	4.052	3.927	5.000	4.875	—	—	4,010	165	4,790	4,790	—	4,790	4,790	4,790	4,790	146	—	249	249	249	249	—	—
	11.60	K-55	0.250	4.000	3.875	5.000	4.875	—	—	4,960	184	5,350	5,350	5,350	5,350	5,350	5,350	5,350	170	180	277	277	277	277	—	—
	11.60	C-75	0.250	4.000	3.875	5.000	4.875	—	—	6,150	250	7,290	—	7,290	7,290	—	7,290	—	—	212	288	—	288	—	—	—
	13.50	C-75	0.290	3.920	3.795	5.000	4.875	—	—	8,170	288	8,460	—	8,460	8,460	—	7,490	—	—	257	331	—	320	—	—	—
	11.60	L-80	0.250	4.000	3.875	5.000	4.875	—	—	6,350	267	7,780	—	7,780	7,780	7,780	7,780	7,780	—	212	291	—	291	—	—	—
	13.50	L-80	0.290	3.920	3.795	5.000	4.875	—	—	8,540	307	9,020	—	9,020	9,020	9,020	7,990	9,020	—	257	334	—	320	—	—	—
	11.60	N-80	0.250	4.000	3.875	5.000	4.875	—	—	6,350	267	7,780	—	7,780	7,780	7,780	7,780	7,780	—	223	304	304	304	304	—	—
	13.50	N-80	0.290	3.920	3.795	5.000	4.875	—	—	8,540	307	9,020	—	9,020	9,020	9,020	7,990	9,020	—	270	349	349	337	349	—	—
	11.60	C-95	0.250	4.000	3.875	5.000	4.875	—	—	7,010	317	9,240	—	9,240	9,240	—	9,240	—	—	234	325	325	325	—	—	—
	13.50	C-95	0.290	3.920	3.796	5.000	4.875	—	—	9,630	364	10,710	—	10,710	10,710	—	9,490	—	—	284	374	374	353	—	—	—
	11.60	P-110	0.250	4.000	3.875	5.000	4.875	—	—	7,560	367	10,690	—	10,690	10,690	10,690	10,690	10,690	—	279	385	385	385	385	—	—
	13.50	P-110	0.290	3.920	3.795	5.000	4.875	—	—	10.670	422	12,410	—	12,410	12,410	12,410	10,990	12,410	—	338	443	443	421	443	—	—
	15.10	P-110	0.337	3.826	3.701	5.000	4.875	—	—	14,320	485	14,420	—	14,420	13,460	14,420	10,990	14,310	—	406	509	509	421	509	—	—
5	11.50	J-55	0.220	4.560	4.435	5.563	—	—	—	3,060	182	4,240	4,240	—	—	—	—	—	133	—	—	—	—	—	—	—
	13.00	J-55	0.253	4.494	4.369	5.563	5.375	—	—	4,110	208	4,870	4,870	4,870	4,870	4,870	4,870	4,870	169	182	252	252	252	252	—	—
	15.00	J-55	0.296	4.408	4.283	5.563	5.375	4.151	5.360	5,550	241	5,700	5,700	5,700	5,700	5,700	5,150	5,700	207	223	293	293	287	293	328	—
	11.50	K-55	0.220	4.560	4.435	5.563	—	—	—	3,060	182	4,240	4,240	—	—	—	—	—	147	—	—	—	—	—	—	—
	13.00	K-55	0.253	4.494	4.360	5.563	5.375	—	—	4,140	208	4,870	4,870	4,870	4,870	4,870	4,870	4,870	186	201	309	309	309	309	—	—
	15.00	K-55	0.296	4.408	4.283	5.563	5.375	4.151	5.360	5,550	241	5,700	5,700	5,700	5,700	5,700	5,150	5,700	228	246	359	359	359	359	416	—
	15.00	C-75	0.296	4.408	4.283	5.563	5.375	4.151	5.360	6,970	328	7,770	—	7,770	7,770	—	6,990	—	—	295	375	—	364	—	416	—
	18.00	C-75	0.362	4.276	4.151	5.563	5.375	4.151	5.360	10,000	396	9,500	—	9,500	9,290	—	6,990	—	—	376	452	—	364	—	446	—
	21.40	C-75	0.437	4.126	4.001	5.563	5.375	—	—	11,060	470	11,470	—	10,140	9,290	—	6,990	—	—	466	510	—	364	—	—	—
	24.10	C-75	0.500	4.000	3.875	5.563	5.375	—	—	13,500	530	13,130	—	10,140	9,920	—	6,990	—	—	538	510	—	364	—	—	—
	15.00	L-80	0.296	4.408	4.283	5.563	5.375	4.151	5.360	7,250	350	8,290	—	8,290	8,290	8,290	7,460	8,290	—	295	379	—	364	—	416	—
	18.00	L-80	0.362	4.276	4.151	5.563	5.375	4.151	5.360	10,490	422	10,140	—	10,140	9,910	10,140	7,460	10,140	—	376	457	—	364	—	446	—
	21.40	L-80	0.437	4.126	4.001	5.563	5.375	—	—	12,760	501	12,240	—	10,810	9,910	—	7,460	—	—	466	510	—	364	—	—	—
	24.10	L-80	0.500	4.000	3.875	5.563	5.375	—	—	14,400	566	14,000	—	10,810	9,910	—	7,460	—	—	538	510	—	364	—	—	—
	15.00	N-80	0.296	4.408	4.283	5.563	5.375	4.151	5.360	7,250	350	8,290	—	8,290	8,290	8,290	7,460	8,290	—	311	396	396	383	396	437	—
	18.00	N-80	0.362	4.276	4.151	5.563	5.575	4.151	5.360	10,400	422	10,140	—	10,140	9,910	10,140	7,460	10,140	—	396	477	477	383	477	469	—
	21.40	N-80	0.437	4.126	4.001	5.563	5.375	—	—	12,760	501	12,240	—	10,810	9,910	12,240	7,460	10,250	—	490	537	566	383	479	—	—
	24.10	N-80	0.500	4.000	3.875	5.563	5.375	—	—	14,400	566	14,000	—	10,810	9,910	13,620	7,460	10,250	—	567	537	639	383	479	—	—
	15.00	C-95	0.296	4.408	4.283	5.563	5.375	4.151	5.360	8,090	415	9,840	—	9,840	9,840	—	8,850	—	—	326	424	—	402	—	459	—
	18.00	C-95	0.362	4.276	4.151	5.563	5.375	4.151	5.360	12,010	501	12,040	—	12,040	11,770	—	8,850	—	—	416	512	—	402	—	493	—
	21.40	C-95	0.437	4.126	4.001	5.563	5.375	—	—	15,150	595	14,550	—	12,840	11,770	—	8,850	—	—	515	563	—	402	—	—	—
	24.10	C-95	0.500	4.000	3.875	5.563	5.375	—	—	17,100	672	16,630	—	12,850	11,770	—	8,850	—	—	505	563	—	402	—	—	—
	15.00	P-110	0.206	4.408	5.283	5.563	5.375	4.151	5.360	8,830	481	11,400	—	11,400	11,400	11,400	10,250	11,400	—	388	503	503	479	503	547	—
	18.00	P-110	0.362	4.276	4.151	5.563	5.375	4.151	5.360	13,450	580	13,940	—	13,940	13,620	13,910	10,250	13,940	—	495	606	606	479	606	587	—
	21.40	P-110	0.437	4.126	4.001	5.563	5.375	—	—	17,550	689	16,820	—	14,870	13,620	16,820	10,250	13,980	—	613	671	720	479	613	—	—
	24.10	P-110	0.500	4.000	3.875	5.563	5.375	—	—	10,800	778	19,250	—	14,870	13,620	18,580	10,250	13,080	—	708	671	812	479	613	—	—

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
					Threaded and Coupled			Extreme Line				Internal Yield Pressure at Minimum Yield, psi							★ Joint Strength — 1,000 lbs: Threaded and Coupled						Extreme Line	
													Round Thread		Buttress Thread				Round Thread		Buttress Thread					
															Regular Coupling		Special Clearance Coupling									
Size: Outside Diameter in. D	Nominal Weight, Threads and Coupling lb per ft	Grade	Wall Thickness in. t	Inside Diameter in. d	Drift Diameter in.	Outside Diameter of Coupling in. W	Outside Diameter Special Clearance Coupling in. W_c	Drift Diameter in.	Outside Diameter of Box Power-tight in. M	Collapse Resistance psi	Pipe Body Yield Strength 1000 lbs	Plain End or Extreme Line	Short	Long	Same Grade	Higher Grade	Same Grade	Higher Grade	Short	Long	Regular Coupling	Regular Coupling Higher Grade	Special Clearance Coupling	Special Clearance Coupling Higher Grade‡	Standard Joint	Optional Joint
5½	14.00	H-40	0.244	5.012	4.887	6.050	—	—	—	2,630	161	3,110	3,110	—	—	—	—	—	130	—	—	—	—	—	—	—
	14.00	J-55	0.244	5.012	4.887	6.050	—	—	—	3,120	222	4,270	4,270	—	—	—	—	—	172	—	—	—	—	—	—	—
	15.50	J-55	0.275	4.950	4.825	6.050	5.875	4.653	5.860	4,040	248	4,810	4,810	4,810	4,810	4,810	4,730	4,810	202	217	300	300	300	300	339	339
	17.00	J-55	0.304	4.892	4.767	6.050	5.875	4.653	5.860	4,910	273	5,320	5,320	5,320	5,320	5,320	4,730	5,320	229	247	329	329	318	329	372	372
	14.00	K-55	0.244	5.012	4.887	6.050	—	—	—	3,120	222	4,270	4,270	—	—	—	—	—	189	—	—	—	—	—	—	—
	15.50	K-55	0.275	4.050	4.825	6.050	5.875	4.653	5.860	4,040	248	4,810	4,810	4,810	4,810	4,810	4,730	4,810	222	239	366	366	366	366	429	429
	17.00	K-55	0.304	4.892	4.767	6.050	5.875	4.653	5.860	4,910	273	5,320	5,320	5,320	5,320	5,320	4,730	5,320	252	272	402	402	402	402	471	471
	17.00	C-75	0.304	4.892	4.767	6.050	5.875	4.653	5.860	6,070	372	7,250	—	7,250	7,250	—	6,450	—	—	327	423	—	403	—	471	471
	20.00	C-75	0.361	4.778	4.653	6.050	5.875	4.653	5.860	8,410	457	8,610	—	8,610	8,430	—	6,450	—	—	403	497	—	403	—	497	479
	23.00	C-75	0.415	4.670	4.545	6.050	5.875	4.545	5.860	10,460	497	9,900	—	9,260	8,430	—	6,450	—	—	473	550	—	403	—	549	479
	17.00	L-80	0.304	4.892	4.767	6.050	5.875	4.653	5.860	6,280	367	7,740	—	7,740	7,740	7,740	6,880	7,740	—	338	428	—	403	—	471	471
	20.00	L-80	0.361	4.778	4.653	6.050	5.875	4.653	5.860	8,830	466	9,190	—	9,190	8,990	9,190	6,880	9,190	—	416	503	—	403	—	497	479
	23.00	L-80	0.415	4.670	4.545	6.050	5.875	4.545	5.860	11,160	530	10,560	—	9,880	8,990	10,560	6,880	9,460	—	489	550	—	403	—	549	479
	17.00	N-80	0.304	4.892	4.767	6.050	5.875	4.653	5.860	6,280	397	7,740	—	7,740	7,740	7,740	6,880	7,740	—	348	446	446	424	446	496	496
	20.00	N-80	0.361	4.778	4.653	6.050	5.875	4.653	5.860	8,830	466	9,190	—	9,190	8,990	9,190	6,880	9,190	—	428	524	524	424	524	523	504
	23.00	N-80	0.415	4.670	4.545	6.050	5.875	4.545	5.860	11,160	530	10,560	—	9,880	8,990	10,560	6,880	9,460	—	502	579	596	424	530	577	504
	17.00	C-95	0.304	4.892	4.767	6.050	5.875	4.653	5.860	8,930	471	9,190	—	9,190	9,190	—	8,170	—	—	374	480	—	445	—	521	521
	20.00	C-95	0.361	4.778	4.653	6.050	5.875	4.653	5.860	10,000	554	10,910	—	10,910	10,680	—	8,170	—	—	460	563	—	445	—	549	530
	23.00	C-95	0.415	4.070	4.515	6.050	5.875	4.545	5.860	12,920	630	12,540	—	11,730	10,680	—	8,170	—	—	540	608	—	445	—	606	530
	17.00	P-110	0.304	4.892	4.767	6.050	5.875	4.653	5.860	7,460	546	10,640	—	10,640	10,640	10,640	9,460	10,640	—	445	568	568	530	568	620	620
	20.00	P-110	0.361	4.778	4.653	6.050	5.875	4.653	5.860	11,080	641	12,640	—	12,640	12,360	12,640	9,460	12,640	—	548	667	667	530	667	654	630
	23.00	P-110	0.415	4.070	4.515	6.050	5.875	4.545	5.860	14,520	729	14,520	—	13,580	12,360	14,520	9,460	12,890	—	645	724	759	530	668	732	630
6	20.00	H-40	0.288	6.019	5.924	7.390	—	—	—	2,520	229	3,040	3,040	—	—	—	—	—	184	—	—	—	—	—	—	—
	20.00	J-55	0.288	6.019	5.924	7.234	7.000	—	—	2,970	315	4,180	4,180	4,180	4,180	4,180	4,060	4,180	245	266	374	374	374	374	—	—
	28.00	J-55	0.352	5.921	5.756	7.390	7.000	5.730	7.000	4,560	382	5,110	5,110	5,110	5,110	5,110	4,060	5,110	314	340	453	453	390	453	477	477
	20.00	K-55	0.288	6.019	5.924	7.234	—	—	—	2,970	315	4,180	4,180	4,180	4,180	4,180	4,060	4,180	267	290	453	453	453	453	—	—
	28.00	K-55	0.352	5.921	5.756	7.390	7.000	5.730	7.000	4,560	382	5,110	5,110	5,110	5,110	5,110	4,060	5,110	342	372	548	548	494	520	605	605
	24.00	C-75	0.352	5.921	5.796	7.390	7.000	5.730	7.000	5,570	520	6,970	—	6,970	6,970	—	5,540	—	—	453	583	—	494	—	605	605
	28.00	C-75	0.417	5.741	5.666	7.390	7.000	5.666	7.000	7,830	610	8,260	—	8,260	8,260	—	5,540	—	—	552	683	—	494	—	648	644
	32.00	C-75	0.475	5.675	5.550	7.390	7.000	5.550	7.000	9,830	688	9,410	—	9,410	9,200	—	5,540	—	—	638	771	—	494	—	717	644
	24.00	L-80	0.352	5.921	5.796	7.390	7.000	5.730	7.000	5,760	555	7,440	—	7,440	7,440	—	5,910	—	—	473	592	—	494	—	605	605
	28.00	L-80	0.417	5.741	5.666	7.390	7.000	5.666	7.000	8,170	651	8,810	—	8,810	8,810	—	5,910	—	—	576	693	—	494	—	648	644
	32.00	L-80	0.475	5.675	5.550	7.390	7.000	5.550	7.000	10,320	734	10,040	—	10,040	9,820	—	5,910	—	—	666	783	—	494	—	717	644
	24.00	N-80	0.352	5.921	5.796	7.390	7.000	5.730	7.000	5,760	555	7,440	—	7,440	7,440	7,440	5,910	7,440	—	481	615	615	520	615	637	637
	28.00	N-80	0.417	5.741	5.666	7.390	7.000	5.666	7.000	8,170	651	8,810	—	8,810	8,810	8,810	5,910	8,120	—	586	721	721	520	650	682	678
	32.00	N-80	0.475	5.675	5.550	7.390	7.000	5.550	7.000	10,320	734	10,040	—	10,040	9,820	10,040	5,910	8,120	—	677	814	814	520	650	756	678
	24.00	C-95	0.352	5.921	5.796	7.390	7.000	5.730	7.000	6,290	659	8,830	—	8,830	8,830	—	7,020	—	—	545	605	—	546	—	608	608
	28.00	C-95	0.417	5.741	5.666	7.390	7.000	5.666	7.000	9,200	773	10,460	—	10,460	10,460	—	7,020	—	—	605	780	—	546	—	716	712
	32.00	C-95	0.475	5.675	5.550	7.390	7.000	5.550	7.000	11,800	872	11,920	—	11,920	11,660	—	7,020	—	—	769	880	—	546	—	793	712
	24.00	P-110	0.352	5.921	5.796	7.390	7.000	5.730	7.000	6,710	763	10,230	—	10,230	10,230	10,230	8,120	10,230	—	641	786	786	650	786	796	796
	28.00	P-110	0.417	5.741	5.666	7.390	7.000	5.666	7.000	10,140	895	12,120	—	12,120	12,120	12,120	8,120	11,080	—	781	922	922	650	832	852	848
	32.00	P-110	0.475	5.675	5.550	7.390	7.000	5.550	7.000	13,200	1009	13,800	—	13,800	13,500	13,800	8,120	11,080	—	904	1,040	1,040	650	832	944	848

Table 4-150 (continued)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
					Threaded and Coupled			Extreme Line				Internal Yield Pressure at Minimum Yield, psi							★ Joint Strength — 1,000 lbs: Threaded and Coupled						Extreme Line	
													Round Thread		Buttress Thread				Round Thread		Buttress Thread					
															Regular Coupling		Special Clearance Coupling									
Size: Outside Diameter in. D	Nominal Weight, Threads and Coupling lb per ft	Grade	Wall Thickness in. t	Inside Diameter in. d	Drift Diameter in.	Outside Diameter of Coupling in. W	Outside Diameter Special Clearance Coupling in. W_c	Drift Diameter in.	Outside Diameter of Box Powertight in. M	Collapse Resistance psi	Pipe Body Yield Strength 1000 lbs	Plain End or Extreme Line	Short	Long	Same Grade	Higher Grade	Same Grade	Higher Grade	Short	Long	Regular Coupling	Regular Coupling Higher Grade*	Special Clearance Coupling	Special Clearance Coupling Higher Grade†	Standard Joint	Optional Joint
7	17.00	H-40	0.231	6.578	6.413	7.656	—	—	—	1,450	196	2,310	2,310	—	—	—	—	—	122	—	—	—	—	—	—	—
	20.00	H-40	0.272	6.456	6.331	7.656	—	—	—	1,980	200	2,720	2,720	—	—	—	—	—	176	—	—	—	—	—	—	—
	20.00	J-55	0.272	6.456	6.331	7.656	—	—	—	2,270	316	3,740	3,740	—	—	—	—	—	234	—	—	—	—	—	—	—
	23.00	J-55	0.317	6.346	6.241	7.656	7.375	6.151	7.390	3,270	366	4,360	4,360	4,360	4,360	4,360	3,950	4,360	284	313	432	432	421	432	499	499
	26.00	J-55	0.362	6.276	6.131	7.656	7.375	6.131	7.390	4,320	415	4,980	4,980	4,980	4,980	4,980	3,950	4,980	334	367	490	490	421	490	506	506
	20.00	K-55	0.272	6.456	6.331	7.656	—	—	—	2,270	316	3,740	3,740	—	—	—	—	—	254	—	—	—	—	—	—	—
	23.00	K-55	0.317	6.368	6.241	7.656	7.375	6.151	7.390	3,270	364	4,360	4,360	4,360	4,360	4,350	3,950	4,360	309	341	522	522	522	522	632	632
	26.00	K-55	0.362	6.276	6.135	7.656	7.375	6.151	7.390	4,320	415	4,980	4,980	4,980	4,980	4,980	3,950	4,980	364	401	592	592	533	561	641	641
	23.00	C-75	0.317	6.366	6.211	7.656	7.375	6.151	7.390	3,770	499	5,940	—	5,940	5,940	—	5,380	—	—	416	557	—	533	—	685	632
	26.00	C-75	0.362	6.276	6.151	7.656	7.375	6.151	7.390	5,250	566	6,790	—	6,790	6,790	—	8,380	—	—	489	631	—	533	—	611	641
	29.00	C-75	0.408	6.184	6.059	7.656	7.375	6.059	7.390	6,760	634	7,650	—	7,650	7,650	—	8,380	—	—	562	707	—	533	—	695	674
	32.00	C-75	0.458	6.004	5.069	7.656	7.375	5.960	7.390	8,230	699	8,490	—	8,490	7,930	—	5,380	—	—	633	779	—	533	—	681	674
	35.00	C-75	0.498	6.004	5.879	7.656	7.375	5.879	7.530	9,710	763	9,340	—	8,660	7,930	—	8,380	—	—	701	833	—	533	—	856	781
	38.00	C-75	0.540	5.920	5.795	7.656	7.375	5.795	7.530	10,680	822	10,120	—	8,660	7,930	—	8,380	—	—	767	833	—	533	—	917	781
	23.00	L-80	0.317	6.366	6.241	7.656	7.375	6.151	7.390	3,830	532	6,340	—	6,340	6,340	8,346	5,740	6,340	—	435	565	—	533	—	632	632
	26.00	L-80	0.362	6.276	6.151	7.656	7.375	6.151	7.390	5,410	604	7,240	—	7,240	7,210	7,210	5,740	7,210	—	511	641	—	533	—	641	641
	29.00	L-80	0.408	6.184	6.059	7.656	7.375	6.059	7.390	7,020	676	8,160	—	8,160	8,160	8,160	5,740	7,890	—	587	718	—	533	—	685	674
	32.00	L-80	0.483	6.094	5.368	7.656	7.375	5.969	7.390	8,600	745	9,060	—	9,060	8,160	9,060	5,740	7,890	—	661	791	—	533	—	761	674
	35.00	L-80	0.498	6.004	5.879	7.656	7.375	5.879	7.530	10,180	814	9,900	—	9,240	8,460	9,960	5,740	7,890	—	734	833	—	533	—	856	761
	38.00	L-80	0.590	5.920	5.793	7.656	7.375	6.795	7.530	11,390	877	10,800	—	9,240	8,460	10,800	5,740	7,890	—	801	853	—	533	—	917	761
	23.00	N-80	0.317	6.366	6.241	7.656	7.375	6.151	7.390	3,830	532	6,340	—	6,340	6,340	6,340	5,740	6,340	—	442	588	588	561	588	666	666
	26.00	N-80	0.362	6.276	6.151	7.666	7.375	6.151	7.390	5,410	604	7,240	—	7,240	7,240	7,240	5,740	7,240	—	519	667	667	561	667	675	675
	29.00	N-80	0.408	6.184	6.059	7.656	7.375	6.059	7.390	7,020	676	8,160	—	8,160	8,160	8,160	5,740	7,890	—	507	745	746	561	702	721	709
	32.00	N-80	0.453	6,036	5.069	7.656	7.375	5.969	7.390	8,600	743	9,060	—	9,060	8,460	9,060	5,740	7,890	—	672	823	823	561	702	801	709
	35.00	N-80	0.498	6,006	5.879	7.656	7.375	5.879	7.530	10,180	814	9,960	—	9,240	8,460	9,960	5,740	7,890	—	746	876	898	561	702	805	801
	38.00	N-80	0.540	5.920	5.795	7.656	7.375	5.795	7.530	11,330	877	10,800	—	9,240	8,460	10,800	5,740	7,890	—	814	876	968	561	702	895	801
	23.00	C-95	0.317	6.366	6.711	7.656	7.375	6.151	7.390	4,150	632	7,530	—	7,530	7,530	—	6,810	—	—	605	656	—	589	—	699	699
	26.00	C-95	0.362	6.274	6.151	7.656	7.375	6.151	7.390	5,870	717	8,600	—	8,600	8,600	—	6,810	—	—	593	722	—	589	—	709	709
	29.00	C-95	0.408	6.184	6.059	7.656	7.375	6.059	7.390	7,820	803	9,690	—	9,690	9,690	—	6,810	—	—	683	808	—	889	—	757	744
	32.00	C-95	0.453	6.094	5.969	7.656	7.375	5.959	7.390	9,730	885	10,760	—	10,760	10,050	—	6,810	—	—	768	891	—	589	—	841	744
	35.00	C-95	0.498	6.001	5.879	7.656	7.375	5.879	7.330	11,640	966	11,830	—	10,970	10,050	—	6,810	—	—	853	920	—	589	—	940	841
	38.00	C-95	0.540	5.820	5.795	7.656	7.375	5.793	7.530	13,429	1041	12,820	—	10,970	10,050	—	6,810	—	—	931	920	—	589	—	1,013	841
	26.00	P-110	0.562	6.276	6.151	7.656	7.375	6.151	7.390	6,210	850	9,960	—	9,960	9,960	9,960	7,890	9,960	—	693	853	853	702	853	844	844
	29.00	P-110	0.408	4.184	6.059	7.656	7.375	6.059	7.390	8,310	929	11,220	—	11,220	11,220	11,220	7,890	10,760	—	797	955	955	702	898	902	884
	32.00	P-110	0.453	6.091	5.969	7.656	7.375	5.959	7.390	10,760	1025	12,460	—	12,460	11,640	12,460	7,890	10,760	—	897	1,053	1,053	702	898	1,002	886
	35.00	P-110	0.498	6.004	5.879	7.656	7.375	5.879	7.530	13,010	1119	13,700	—	12,700	11,640	13,700	7,890	10,760	—	996	1,096	1,150	702	898	1,118	1,002
	38.00	P-110	0.540	8.920	5.795	7.656	7.375	5.795	7.530	15,110	1205	14,850	—	12,700	11,640	14,850	7,890	10,760	—	1,087	1,096	1,239	702	898	1,207	1,002
7⅝	24.00	H-40	0.300	7.025	6.900	8.500	—	—	—	2,040	276	2,750	2,750	—	—	—	—	—	212	—	—	—	—	—	—	—
	26.40	J-55	0.328	6.969	6.844	8.500	8.125	6.750	8.010	2,890	414	4,140	4,140	4,140	4,140	4,140	4,140	4,140	316	346	483	483	483	483	558	553
	26.40	K-55	0.328	6.969	6.844	8.500	8.125	6.750	8.010	2,890	414	4,140	4,140	4,140	4,140	4,140	4,140	4,140	342	377	581	581	581	581	700	700
	26.40	C-75	0.328	6.969	6.814	8.500	8.125	6.750	8.010	3,280	564	5,650	—	5,650	5,650	—	5,650	—	—	461	624	—	624	—	700	700
	29.70	C-75	0.375	6.875	6.750	8.500	8.125	6.750	8.010	4,670	611	6,450	—	6,450	6,450	—	6,140	—	—	542	709	—	709	—	700	700
	33.70	C-75	0.430	6.755	6.610	8.500	8.125	6.640	8.010	6,320	725	7,400	—	7,400	7,400	—	6,140	—	—	635	806	—	735	—	766	744
	39.00	C-75	0.500	6.625	6.500	8.500	8.125	6.500	8.010	8,430	839	8,610	—	8,610	8,610	—	6,140	—	—	751	929	—	735	—	851	744
	42.80	C-75	0.562	6.501	6.374	8.500	8.125	—	—	10,240	231	9,670	—	9,670	9,190	—	6,140	—	—	852	1,035	—	735	—	—	—
	47.10	C-75	0.625	6.375	6.250	8.500	8.125	—	—	11,200	1031	10,760	—	9,840	9,190	—	6,140	—	—	953	1,140	—	735	—	—	—
	26.40	L-80	0.328	6.969	6.844	8.500	8.125	6.750	8.010	3,400	602	6,020	—	6,020	6,020	6,020	6,020	6,020	—	482	635	—	635	—	700	700
	29.70	L-80	0.375	6.875	6.750	8.500	8.125	6.750	8.010	4,700	683	6,890	—	6,890	6,890	6,890	6,550	6,890	—	566	721	—	721	—	700	700
	33.70	L-80	0.400	6.765	6.620	8.500	8.125	6.610	8.010	6,560	778	7,900	—	7,900	7,900	7,900	6,550	7,900	—	664	820	—	735	—	766	744
	39.00	L-80	0.500	6.625	6.500	8.500	8.125	6.500	8.010	8,810	895	9,180	—	9,180	9,180	9,180	6,550	9,000	—	786	945	—	735	—	851	744
	42.00	L-80	0.562	6.301	6.374	8.500	8.125	—	—	10,810	928	10,320	—	10,320	9,790	—	6,550	—	—	892	1,053	—	735	—	—	—
	47.10	L-80	0.625	6.375	6.250	8.500	8.125	—	—	12,040	1100	11,480	—	10,490	9,970	—	6,550	—	—	997	1,160	—	735	—	—	—
	26.40	N-80	0.328	6.969	6.844	8.500	8.125	6.750	8.010	3,400	602	6,020	—	6,020	6,020	6,020	6,020	6,020	—	490	659	659	659	659	737	737
	29.70	N-80	0.375	6.875	6.750	8.500	8.125	6.750	8.010	4,700	683	6,890	—	6,890	6,890	6,890	6,550	6,890	—	575	749	749	749	749	737	737
	33.70	N-80	0.430	6.765	4.810	8.500	8.125	6.640	8.010	6,560	778	7,900	—	7,900	7,900	7,900	6,550	7,000	—	674	852	852	773	852	806	784
	39.00	N-80	0.500	6.625	6.500	8.500	8.125	6.500	8.010	5,810	695	9,180	—	9,180	9,180	9,180	6,550	9,000	—	798	981	981	773	967	896	784
	42.80	N-80	0.562	6.301	6.776	8.500	8.125	—	—	10,810	998	10,320	—	10,320	9,700	10,320	6,550	9,000	—	905	1,003	1,093	773	967	—	—
	47.10	N-80	0.625	6.375	6.230	8.500	8.125	—	—	12,010	1100	11,480	—	10,490	9,790	11,480	6,550	9,000	—	1,013	1,205	1,204	773	967	—	—
	26.40	C-95	0.328	6.969	6.814	8.500	8.125	6.750	8.010	3,710	714	7,150	—	7,150	7,150	—	7,150	—	—	560	716	—	716	—	774	774
	29.70	C-95	0.375	6.875	6.750	8.500	8.125	6.720	8.010	5,120	811	8,180	—	8,180	8,180	—	7,780	—	—	659	813	—	812	—	774	774
	33.70	C-95	0.400	6.765	6.610	8.500	8.125	6.640	8.010	7,260	923	9,380	—	9,380	9,380	—	7,780	—	—	772	925	—	812	—	846	823
	38.90	C-95	0.500	6.625	6.500	8.500	8.125	6.500	8.010	9,389	1061	10,900	—	10,900	10,900	—	7,780	—	—	914	1,065	—	812	—	941	823
	42.80	C-95	0.562	6.501	6.725	8.500	8.125	—	—	12,400	1185	12,250	—	12,250	11,620	—	7,780	—	—	1,037	1,187	—	812	—	—	—
	47.10	C-95	0.625	6.375	6.250	8.500	8.125	—	—	14,300	1396	13,630	—	12,460	11,620	—	7,780	—	—	1,159	1,300	—	812	—	—	—
	29.70	P-110	0.375	6.875	6.750	8.500	8.125	6.750	8.010	5,340	940	9,470	—	9,470	9,470	9,470	9,000	9,470	—	769	960	960	960	960	922	922
	33.70	P-110	0.430	6.755	6.410	8.500	8.125	6.600	8.010	7,950	1069	10,860	—	10,860	10,860	10,860	9,000	10,860	—	901	1,093	1,093	967	1,093	1,008	979
	39.00	P-110	0.500	6.425	6.500	8.500	8.125	6.500	8.010	11,140	1231	12,620	—	12,620	12,620	12,620	9,000	12,280	—	1,066	1,258	1,258	967	1,237	1,120	979
	42.80	P-110	0.542	6.501	6.376	8.500	8.125	—	—	13,910	1372	14,190	—	14,190	13,460	14,100	9,000	12,280	—	1,210	1,402	1,402	967	1,237	—	—
	47.10	P-110	0.625	6.375	6.250	8.500	8.124	—	—	16,330	1512	15,780	—	14,430	13,460	15,780	9,000	12,280	—	1,353	1,545	1,545	967	1,237	—	—

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
					Threaded and Coupled			Extreme Line				Internal Yield Pressure at Minimum Yield, psi							★ Joint Strength — 1,000 lbs: Threaded and Coupled						Extreme Line	
													Round Thread		Buttress Thread: Regular Coupling		Buttress Thread: Special Clearance Coupling		Round Thread		Buttress Thread					
Size: Outside Diameter in. D	Nominal Weight, Threads and Coupling lb per ft	Grade	Wall Thickness in. t	Inside Diameter in. d	Drift Diameter in.	Outside Diameter of Coupling in. W	Outside Diameter Special Clearance Coupling in. W_c	Drift Diameter in.	Outside Diameter of Box Powertight in. M	Collapse Resistance psi	Pipe Body Yield Strength 1000 lbs	Plain End or Extreme Line	Short	Long	Same Grade	Higher Grade	Same Grade	Higher Grade	Short	Long	Regular Coupling	Regular Coupling Higher Grade†	Special Clearance Coupling	Special Clearance Coupling Higher Grade‡	Standard Joint	Optional Joint
8⅝	28.00	H-40	0.304	8.017	7.892	9.625	—	—	—	1,640	318	2,470	2,470	—	—	—	—	—	233	—	—	—	—	—	—	—
	32.00	H-40	0.352	7.921	7.796	9.625	—	—	—	2,210	366	2,860	2,860	—	—	—	—	—	279	—	—	—	—	—	—	—
	24.00	J-55	0.264	8.097	7.972	9.625	—	—	—	1,370	381	2,950	2,950	—	—	—	—	—	244	—	—	—	—	—	—	—
	32.00	J-55	0.352	7.921	7.796	9.625	9.125	7.700	9.120	2,530	503	3,930	3,930	3,930	3,930	3,930	3,930	3,930	372	417	579	579	579	579	686	686
	36.00	J-55	0.400	7.825	7.700	9.625	9.125	7.700	9.120	3,450	568	4,460	4,460	4,460	4,460	4,460	4,060	4,460	434	486	654	654	654	654	688	688
	24.00	K-55	0.264	8.097	7.972	9.625	—	—	—	1,370	381	2,950	2,950	—	—	—	—	—	263	—	—	—	—	—	—	—
	32.00	K-55	0.352	7.921	7.796	9.625	9.125	7.700	9.120	2,530	503	3,930	3,930	3,930	3,930	3,930	3,930	3,930	402	452	690	690	690	690	869	869
	36.00	K-55	0.400	7.825	7.700	9.625	9.125	7.700	9.120	3,450	568	4,460	4,460	4,460	4,460	4,460	4,060	4,460	468	526	780	780	780	780	871	871
	36.00	C-75	0.400	7.825	7.700	9.625	9.125	7.700	9.120	4,020	775	6,090	—	6,090	6,090	—	5,530	—	—	648	847	—	839	—	871	871
	40.00	C-75	0.450	7.725	7.600	9.625	9.125	7.600	9.120	5,350	867	6,850	—	6,850	6,850	—	5,530	—	—	742	947	—	839	—	942	886
	44.00	C-75	0.500	7.625	7.300	9.625	9.125	7.500	9.120	6,680	957	7,610	—	7,610	7,610	—	5,530	—	—	834	1,046	—	839	—	1,007	886
	49.00	C-75	0.557	7.511	7.386	9.625	9.125	7.386	9.120	8,200	1059	8,480	—	8,480	8,480	—	5,530	—	—	939	1,157	—	839	—	1,007	886
	36.00	L-80	0.400	7.825	7.700	9.625	9.125	7.700	9.120	4,100	827	6,490	—	6,490	6,490	—	5,900	—	—	678	864	—	839	—	871	871
	40.00	L-80	0.450	7.725	7.600	9.625	9.125	7.600	9.120	5,520	925	7,300	—	7,300	7,300	—	5,900	—	—	776	966	—	839	—	942	886
	44.00	L-80	0.500	7.625	7.500	9.625	9.125	7.500	9.120	6,950	1021	8,120	—	8,120	8,120	—	5,900	—	—	874	1,066	—	839	—	1,007	886
	49.00	L-80	0.557	7.511	7.386	9.625	9.125	7.386	9.120	8,570	1129	9,040	—	9,040	9,040	—	5,900	—	—	983	1,180	—	839	—	1,007	886
	36.00	N-80	0.400	7.825	7.700	9.625	9.125	7.700	9.120	4,100	827	6,490	—	6,490	6,490	6,490	5,900	6,490	—	688	895	895	883	895	917	917
	40.00	N-80	0.450	7.725	7.600	9.625	9.125	7.600	9.120	5,520	925	7,300	—	7,300	7,300	7,300	5,900	7,300	—	788	1,001	1,001	883	1,001	992	932
	44.00	N-80	0.500	7.625	7.500	9.625	9.125	7.500	9.120	6,950	1021	8,120	—	8,120	8,120	8,120	5,900	8,110	—	887	1,105	1,105	883	1,103	1,060	932
	49.00	N-80	0.557	7.511	7.386	9.626	9.125	7.386	9.120	8,570	1129	9.040	—	9,040	9,040	9,040	5,900	8,110	—	997	1,222	1,222	883	1,103	1,060	932
	36.00	C-95	0.400	7.825	7.706	9.625	9.125	7.700	9.120	4,360	982	7,710	—	7,710	7,710	—	7,010	—	—	789	976	—	927	—	963	963
	40.00	C-95	0.450	7.725	7.600	9.625	9.125	7.600	9.120	6,010	1098	8,670	—	8,670	8,670	—	7,010	—	—	904	1,092	—	927	—	1,042	979
	44.00	C-95	0.500	7.625	7.500	9.625	9.125	7.500	9.120	7,730	1212	9,640	—	9,640	9,640	—	7,010	—	—	1,017	1,206	—	927	—	1,113	979
	49.00	C-95	0.557	7.511	7.386	9.625	9.125	7.386	9.120	9,690	1341	10,740	—	10,740	10,740	—	7,010	—	—	1,144	1,334	—	927	—	1,113	979
	40.00	P-110	0.450	7.725	7.600	9.625	9.125	7.600	9.120	6,380	1271	10,040	—	10,040	10,040	10,040	8,110	10,040	—	1,055	1,288	1,288	1,103	1,288	1,240	1,165
	44.00	P-110	0.500	7.625	7.500	9.625	9.125	7.500	9.120	8,400	1404	11,160	—	11,160	11,160	11,160	8,110	11,060	—	1,186	1,423	1,423	1,103	1,412	1,326	1,165
	49.00	P-110	0.557	7.511	7.386	9.625	9.125	7.386	9.120	10,720	1553	12,430	—	12,430	12,430	12,430	8,110	11,060	—	1,335	1,574	1,574	1,103	1,412	1,326	1,165
9⅝	32.30	H-40	0.312	9.001	8.845	10.625	—	—	—	1,400	365	2,270	2,270	—	—	—	—	—	254	—	—	—	—	—	—	—
	36.00	H-40	0.352	8.921	8.765	10.625	—	—	—	1,740	410	2,560	2,560	—	—	—	—	—	294	—	—	—	—	—	—	—
	36.00	J-55	0.352	8.921	8.765	10.625	10.125	—	—	2,020	569	3,520	3,520	3,520	3,520	3,520	3,520	3,520	394	453	639	639	639	639	—	—
	40.00	J-55	0.395	8.835	8.679	10.625	10.125	8.599	10.100	2,570	630	3,950	3,950	3,950	3,950	3,950	3,660	3,950	455	520	714	714	714	714	770	770
	36.00	K-55	0.352	8.921	8.765	10.625	10.125	—	—	2,020	564	3,520	3,520	3,520	3,520	3,520	3,520	3,520	425	489	755	755	755	755	—	—
	40.00	K-55	0.395	8.835	8.679	10.625	10.125	8.599	10.100	2,570	630	3,950	3,950	3,950	3,950	3,950	3,660	3,950	486	561	843	843	843	843	975	975
	40.00	C-75	0.395	8.835	8.679	10.625	10.125	8.599	10.100	2,980	859	5,390	—	5,390	5,390	—	4,990	—	—	694	926	—	926	—	975	975
	43.50	C-75	0.435	8.755	8.599	10.625	10.125	8.599	10.100	3,750	942	5,930	—	5,930	5,930	—	4,990	—	—	776	1,016	—	934	—	975	975
	47.00	C-75	0.472	8.681	8.525	10.625	10.125	8.525	10.100	4,630	1018	6,440	—	6,440	6,440	—	4,990	—	—	852	1,098	—	934	—	1,032	1,032
	53.50	C-75	0.545	8.535	8.579	10.625	10.125	8.379	10.100	6,380	1166	7,430	—	7,430	7,430	—	4,990	—	—	999	1,257	—	934	—	1,175	1,053
	40.00	L-80	0.395	8.835	8.679	10.625	10.125	8.599	10.100	3,090	916	5,750	—	5,750	5,750	—	5,320	—	—	727	947	—	934	—	975	975
	43.50	L-80	0.435	8.755	8.599	10.625	10.125	8.599	10.100	3,810	1005	6,330	—	6,330	6,330	—	5,320	—	—	813	1,038	—	934	—	975	975
	47.00	L-80	0.472	8.681	8.525	10.625	10.125	8.525	10.100	4,750	1086	6,870	—	6,870	6,870	—	5,320	—	—	893	1,122	—	934	—	1,032	1,032
	53.50	L-80	0.545	8.535	8.379	10.625	10.125	8.379	10.100	6,620	1244	7,930	—	7,930	7,930	—	5,320	—	—	1,047	1,286	—	934	—	1,173	1,053
	40.00	N-80	0.395	8.835	8.679	10.625	10.125	8.539	10.100	3,090	916	5,750	—	5,750	5,750	5,750	5,320	5,750	—	797	979	979	979	979	1,027	1,027
	43.50	N-80	0.435	8.755	8.599	10.625	10.125	8.590	10.100	3,610	1005	6,330	—	6,330	6,330	6,330	5,320	6,330	—	825	1,074	1,074	983	1,074	1,027	1,027
	47.00	N-80	0.472	8.681	8.525	10.625	10.125	8.325	10.100	4,750	1086	6,870	—	6,870	6,870	6,870	5,320	6,870	—	905	1,161	1,161	983	1,161	1,086	1,086
	53.50	N-80	0.545	8.535	8.379	10.625	10.125	8.379	10.100	6,620	1244	7,930	—	7,930	7,930	7,930	5,320	7,310	—	1,062	1,329	1,329	983	1,229	1,235	1,109
	40.00	C-95	0.395	8.835	8.679	10.625	10.125	8.599	10.100	3,330	1088	6,820	—	6,820	6,820	—	6,310	—	—	847	1,074	—	1,032	—	1,078	1,078
	43.50	C-95	0.435	8.755	8.599	10.625	10.125	8.599	10.100	4,130	1193	7,510	—	7,510	7,510	—	6,310	—	—	948	1,178	—	1,032	—	1,078	1,078
	47.00	C-95	0.472	8.681	8.525	10.625	10.125	8.525	10.100	5,080	1239	8,150	—	8,150	8,150	—	6,310	—	—	1,010	1,273	—	1,032	—	1,141	1,141
	53.50	C-95	0.545	8.535	8.379	10.625	10.125	8.379	10.100	7,230	1477	9,410	—	9,410	9,410	—	6,310	—	—	1,220	1,458	—	1,032	—	1,297	1,164
	43.50	P-110	0.435	8.755	8.599	10.625	10.125	8.599	10.100	4,430	1381	8,700	—	8,700	8,700	8,700	7,310	8,700	—	1,106	1,388	1,388	1,229	1,388	1,283	1,283
	47.00	P-110	0.472	8.681	8.525	10.625	10.125	8.525	10.100	5,310	1403	9,440	—	9,440	9,440	9,440	7,310	9,440	—	1,213	1,500	1,500	1,229	1,500	1,358	1,358
	53.50	P-110	0.545	8.535	8.379	10.625	10.125	8.379	10.100	7,930	1710	10,900	—	10,900	10,900	10,900	7,310	9,970	—	1,422	1,718	1,718	1,229	1,573	1,544	1,386

Table 4-150 (continued)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
					Threaded and Coupled			Extreme Line				Internal Yield Pressure at Minimum Yield, psi							* Joint Strength — 1,000 lbs: Threaded and Coupled						Extreme Line	
													Round Thread		Buttress Thread				Round Thread		Buttress Thread					
															Regular Coupling		Special Clearance Coupling									
Size: Outside Diameter in. D	Nominal Weight, Threads and Coupling lb per ft	Grade	Wall Thickness in. t	Inside Diameter in. d	Drift Diameter in.	Outside Diameter of Coupling in. W	Outside Diameter Special Clearance Coupling in. W_c	Drift Diameter in.	Outside Diameter of Box Power-tight in. M	Collapse Resistance psi	Pipe Body Yield Strength 1000 lbs	Plain End or Extreme Line	Short	Long	Same Grade	Higher Grade	Same Grade	Higher Grade	Short	Long	Regular Coupling	Regular Coupling Higher Grade†	Special Clearance Coupling	Special Clearance Coupling Higher Grade‡	Standard Joint	Optional Joint
10¾	32.75	H-40	0.279	10.192	10.036	11.750	—	—	—	880	367	1,820	1,820	—	—	—	—	—	205	—	—	—	—	—	—	—
	40.50	H-40	0.350	10.050	9.894	11.750	—	—	—	1,420	457	2,280	2,280	—	—	—	—	—	314	—	—	—	—	—	—	—
	40.50	J-55	0.350	10.050	9.894	11.750	11.250	—	—	1,580	629	3,130	3,130	—	3,130	3,130	3,130	3,130	420	—	700	700	700	700	—	—
	45.50	J-55	0.400	9.950	9.794	11.750	11.250	9.794	11.460	2,090	715	3,580	3,580	—	3,580	3,580	3,290	3,580	493	—	796	796	796	796	975	—
	51.00	J-55	0.450	9.850	9.694	11.750	11.250	9.694	11.460	2.700	801	4,030	4,030	—	4,030	4,030	3,290	4,030	565	—	891	891	822	891	1,092	—
	40.50	K-55	0.350	10.050	9.894	11.750	11.250	—	—	1,580	629	3,130	3,130	—	3,130	3,130	3,130	3,130	450	—	819	819	819	819	—	—
	45.50	K-55	0.400	9.950	9.794	11.750	11.250	9.794	11.460	2,090	715	3,580	3,580	—	3,580	3,580	3,200	3,580	528	—	931	931	931	931	1,236	—
	51.00	K-55	0.450	9.850	9.694	11.750	11.250	9.694	11.460	2,700	801	4,030	4,030	—	4,030	4,030	3,290	4,030	606	—	1,043	1,043	1,041	1,043	1,383	—
	51.00	C-75	0.450	9.850	9.694	11.750	11.250	9.694	11.460	3,100	1092	5,490	5,490	—	5,490	—	4,490	—	756	—	1,160	—	1,041	—	1,383	—
	55.50	C-75	0.495	9.760	9.604	11.750	11.250	9,604	11.460	3,950	1196	6,040	6,040	—	6,040	—	4,490	—	843	—	1,271	—	1,041	—	1,515	—
	51.00	L-80	0.450	9.850	9.694	11.750	11.250	9.694	11.460	3,220	1165	5,860	5,860	—	5,860	—	4,790	—	794	—	1,190	—	1,041	—	1,383	—
	55.50	L-80	0.495	9.760	9.604	11.750	11.250	9.604	11.460	4,020	1276	6,450	6,450	—	6,450	—	4,790	—	884	—	1,303	—	1,041	—	1,515	—
	51.00	N-80	0.450	9.850	9.694	11.750	11.250	9.694	11.460	3,220	1165	5,860	5,860	—	5,860	5,860	4,790	5,860	804	—	1,228	1,228	1,096	1,228	1,456	—
	55.50	N-80	0.495	9.760	9.604	11.750	11.250	9.604	11.460	4,020	1276	6,450	6,450	—	6,450	6,450	4,790	6,450	895	—	1,345	1,345	1,096	1,345	1,595	—
	51.00	C-95	0.450	9.850	9.694	11.750	11.250	9.694	11.460	3,490	1383	6,960	6,960	—	6,960	—	5,680	—	927	—	1,354	—	1,151	—	1,529	—
	55.50	C-95	0.495	9.760	9.604	11.750	11.250	9.604	11.460	4,300	1515	7,660	7,660	—	7,660	—	5,680	—	1,032	—	1,483	—	1,151	—	1,675	—
	51.00	P-110	0.450	9.850	9.694	11.750	11.250	9.694	11.460	3,670	1602	8,060	8,060	—	8,060	8,060	6,580	8,060	1,080	—	1,594	1,594	1,370	1,594	1,820	—
	55.50	P-110	0.495	9.760	9.604	11.750	11.250	9.604	11.460	4,630	1754	8,860	8,860	—	8,860	8,860	6,580	8,860	1,203	—	1,745	1,745	1,370	1,745	1,993	—
	60.70	P-110	0.545	9.660	9.504	11.750	11.250	9.504	11.460	5,860	1922	9,760	9,760	—	9,760	9,760	6,580	8,970	1,338	—	1,912	1,912	1,370	1,754	2,000	—
	65.70	P-110	0.595	9.560	9.404	11.750	11.250	—	—	7,490	2088	10,650	10,650	—	10,650	10,650	6,580	8,970	1,472	—	2,077	2,077	1,370	1,754	—	—
11¾	42.00	H-40	0.333	11.084	10.928	12.750	—	—	—	1,070	478	1,980	1,980	—	—	—	—	—	307	—	—	—	—	—	—	—
	47.00	J-55	0.375	11.000	10.844	12.750	—	—	—	1,510	737	3,070	3,070	—	3,070	3,070	—	—	477	—	807	807	—	—	—	—
	54.00	J-55	0.435	10.880	10.724	12.750	—	—	—	2,070	850	3,560	3,560	—	3,560	3,560	—	—	568	—	931	931	—	—	—	—
	60.00	J-55	0.489	10.772	10.616	12.750	—	—	—	2,660	952	4,010	4,010	—	4,010	4,010	—	—	649	—	1,042	1,042	—	—	—	—
	47.00	K-55	0.375	11.000	10.844	12.750	—	—	—	1,510	737	3,070	3,070	—	3,070	3,070	—	—	509	—	935	935	—	—	—	—
	54.00	K-55	0.435	10.880	10.724	12.750	—	—	—	2,070	850	3,560	3,560	—	3,560	3,560	—	—	606	—	1,079	1,079	—	—	—	—
	60.00	K-55	0.489	10.772	10.616	12.750	—	—	—	2,660	952	4,010	4,010	—	4,010	4,010	—	—	693	—	1,208	1,208	—	—	—	—
	60.00	C-75	0.489	10.772	10.616	12.750	—	—	—	3,070	1298	5,460	5,460	—	5,460	—	—	—	869	—	1,361	—	—	—	—	—
	60.00	L-80	0.489	10.772	10.616	12.750	—	—	—	3,180	1384	5,830	5,830	—	5,830	—	—	—	913	—	1,399	—	—	—	—	—
	60.00	N-80	0.489	10.772	10.616	12.750	—	—	—	3,180	1384	5,830	5,830	—	5,830	—	—	—	924	—	1,440	1,440	—	—	—	—
	60.00	C-95	0.489	10.772	10.616	12.750	—	—	—	3,440	1644	6,920	6,920	—	6,920	—	—	—	1,066	—	1,596	—	—	—	—	—

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
					Threaded and Coupled			Extreme Line				Internal Yield Pressure at Minimum Yield, psi							★ Joint Strength — 1,000 lbs: Threaded and Coupled						Extreme Line	
													Round Thread		Buttress Thread: Regular Coupling		Buttress Thread: Special Clearance Coupling		Round Thread		Buttress Thread					
Size: Outside Diameter in. *D*	Nominal Weight, Threads and Coupling lb per ft	Grade	Wall Thickness in. *t*	Inside Diameter in. *d*	Drift Diameter in.	Outside Diameter of Coupling in. *W*	Outside Diameter Special Clearance Coupling in. W_c	Drift Diameter in.	Outside Diameter of Box Power-tight in. *M*	Collapse Resistance psi	Pipe Body Yield Strength 1000 lbs	Plain End or Extreme Line	Short	Long	Same Grade	Higher Grade	Same Grade	Higher Grade	Short	Long	Regular Coupling	Regular Coupling Higher Grade	Special Clearance Coupling	Special Clearance Coupling Higher Grade ‡	Standard Joint	Optional Joint
13⅜	48.00	H-40	0.330	12.715	12.559	14.375	—	—	—	770	541	1,730	1,730	—	—	—	—	—	322	—	—	—	—	—	—	—
	54.50	J-55	0.380	12.615	12.459	14.375	—	—	—	1,130	853	2,730	2,730	—	2,730	2,730	—	—	514	—	909	909	—	—	—	—
	61.00	J-55	0.430	12.515	12.359	14.375	—	—	—	1,540	962	3,090	3,090	—	3,090	3,090	—	—	595	—	1,025	1,025	—	—	—	—
	68.00	J-55	0.480	12.415	12.259	14.375	—	—	—	1,950	1069	3,450	3,450	—	3,450	3,450	—	—	675	—	1,140	1,140	—	—	—	—
	54.50	K-55	0.380	12.615	12.459	14.375	—	—	—	1,130	853	2,730	2,730	—	2,730	2,730	—	—	547	—	1,038	1,038	—	—	—	—
	61.00	K-55	0.430	12.515	12.359	14.375	—	—	—	1,540	962	3,090	3,090	—	3,090	3,090	—	—	633	—	1,169	1,169	—	—	—	—
	68.00	K-55	0.480	12.415	12.259	14.375	—	—	—	1,950	1069	3,450	3,450	—	3,450	3,450	—	—	718	—	1,300	1,300	—	—	—	—
	68.00	C-75	0.480	12.415	12.259	14.375	—	—	—	2,220	1458	4,710	4,710	—	4,710	—	—	—	905	—	1,496	—	—	—	—	—
	72.00	C-75	0.514	12.347	12.191	14.375	—	—	—	2,590	1558	5,040	5,040	—	5,040	—	—	—	978	—	1,598	—	—	—	—	—
	68.00	L-80	0.480	12.415	12.259	14.375	—	—	—	2,260	1556	5,020	5,020	—	5,020	—	—	—	952	—	1,545	—	—	—	—	—
	72.00	L-80	0.514	12.347	12.191	14.375	—	—	—	2,670	1661	5,380	5,380	—	5,380	—	—	—	1,029	—	1,650	—	—	—	—	—
	68.00	N-80	0.480	12.415	12.259	14.375	—	—	—	2,260	1556	5,020	5,020	—	5,020	—	—	—	963	—	1,585	1,585	—	—	—	—
	72.00	N-80	0.514	12.347	12.191	14.375	—	—	—	2,670	1661	5,380	5,380	—	5,380	—	—	—	1,040	—	1,693	1,693	—	—	—	—
	68.00	C-95	0.480	12.415	12.259	14.375	—	—	—	2,320	1847	5,970	5,970	—	5,970	—	—	—	1,114	—	1,772	—	—	—	—	—
	72.00	C-95	0.514	12.347	12.191	14.375	—	—	—	2,820	1973	6,390	6,390	—	6,390	—	—	—	1,204	—	1,893	—	—	—	—	—
16	65.00	H-40	0.375	15.250	15.062	17.000	—	—	—	670	736	1,640	1,640	—	—	—	—	—	439	—	—	—	—	—	—	—
	75.00	J-55	0.438	15.124	14.936	17.000	—	—	—	1,020	1178	2,630	2,630	—	2,630	2,630	—	—	710	—	1,200	1,200	—	—	—	—
	84.00	J-55	0.495	15.016	14.822	17.000	—	—	—	1,410	1326	2,980	2,980	—	2,980	2,980	—	—	817	—	1,351	1,351	—	—	—	—
	75.00	K-55	0.438	15.124	14.936	17.000	—	—	—	1,020	1178	2,630	2,630	—	2,630	2,630	—	—	752	—	1,331	1,331	—	—	—	—
	84.00	K-55	0.495	15.010	14.822	17.000	—	—	—	1,410	1326	2,980	2,980	—	2,980	2,980	—	—	865	—	1,499	1,499	—	—	—	—
18⅝	87.50	H-40	0.435	17.755	17.567	20.000	—	—	—	*630	994	1,630	1,630	—	—	—	—	—	559	—	—	—	—	—	—	—
	87.50	J-55	0.435	17.755	17.567	20.000	—	—	—	*630	1367	2,250	2,250	—	—	—	—	—	754	—	1,329	1,329	—	—	—	—
	87.50	K-55	0.435	17.755	17.567	20.000	—	—	—	*630	1367	2,250	2,250	—	—	—	—	—	794	—	1,427	1,427	—	—	—	—
20	94.00	H-40	0.438	19.124	18.936	21.000	—	—	—	*520	1077	1,530	1,530	—	—	—	—	—	581	—	—	—	—	—	—	—
	94.00	J-55	0.438	19.124	18.936	21.000	—	—	—	*520	1480	2,110	2,110	2,110	—	—	—	—	784	907	1,402	1,402	—	—	—	—
	106.50	J-55	0.500	19.000	18.812	21.000	—	—	—	*770	1685	2,410	2,410	2,410	—	—	—	—	913	1,057	1,596	1,596	—	—	—	—
	133.00	J-55	0.635	18.730	18.542	21.000	—	—	—	1,500	2125	3,060	3,060	3,060	—	—	—	—	1,192	1,380	2,012	2,012	—	—	—	—
	94.00	K-55	0.438	19.124	18.036	21.000	—	—	—	*520	1480	2,100	2,110	2,110	—	—	—	—	824	955	1,479	1,479	—	—	—	—
	106.50	K-55	0.500	19.000	18.812	21.000	—	—	—	*770	1685	2,410	2,410	2,410	—	—	—	—	960	1,113	1,683	1,683	—	—	—	—
	133.00	K-55	0.625	18.730	18.542	21.000	—	—	—	1,500	2125	3,060	3,060	3,060	—	—	—	—	1,253	1,453	2,123	2,123	—	—	—	—

‡ For P-110 casing the next higher grade is 150YS, a non-API steel grade having a minimum yield strength of 150,000 psi.

★ Some joint strengths listed in Col. 20 through 27 are greater than the corresponding pipe body yield strength listed in Col. 12.

* Collapse resistance values calculated by elastic formula.

Taken from API Bul 5C2, 17th Edition, March 1980. Source: From Ref. [155]

Table 4-151
D/t Ranges for Collapse Pressures

Steel Grade	D/t range for formula (1)	D/t range for formula (2)	D/t range for formula (3)	D/t range for formula (4)
H–40	16.44 & less	16.44 to 26.62	26.62 to 42.70	42.70 & greater
J–K–55	14.80 & less	14.80 to 24.39	24.39 to 37.20	37.20 & greater
C–75	13.67 & less	13.67 to 23.09	23.03 to 32.05	32.05 & greater
L–N–80	13.38 & less	13.38 to 22.46	22.46 to 31.05	31.05 & greater
C–95	12.83 & less	12.83 to 21.21	21.21 to 28.25	28.25 & greater
P–105	12.56 & less	12.56 to 20.66	20.66 to 26.88	26.88 & greater
P–110	12.42 & less	12.42 to 20.29	20.29 to 26.20	26.20 & greater

Table 4-152
Factors for Collapse Pressure Formulas

Steel Grade	Formula Factor A	B	C	F	G
H–40	2.950	0.0463	755	2.047	0.03125
J–K–55	2.990	0.0541	1205	1.990	0.03360
C–75	3.060	0.0642	1805	1.985	0.0417
L–N–80	3.070	0.0667	1955	1.998	0.0434
C–95	3.125	0.0745	2405	2.047	0.0490
P–105	3.162	0.0795	2700	2.052	0.0515
P–110	3.180	0.0820	2855	2.075	0.0535

(text continued from page 1147)

The factors F and G and applicable D/t range for the transition collapse pressure formula are shown in Tables 4-152 and 4-151, respectively.

Elastic Collapse Pressure Formula. The minimum collapse pressure of the elastic range of collapse is calculated by

$$p_E = \frac{46.95 \times 10^6}{(D/t)|(D/t - 1)|^2} \tag{4-311}$$

Collapse Pressure under Axial Tension Stress. The reduced minimum collapse pressure caused by the action of axial tension stress is calculated by

$$p_{CA} = p_{CO}\left[\sqrt{1 - 0.75[(S_A + P_i)/Y_p]^2} - 0.5(S_A + P_i/Y_p)\right] \tag{4-312}$$

Equation 4-312 is not applicable if P_{CO} is calculated from the elastic collapse formula.

Symbols in Equations 4-308 to 4-312 are as follows:

D = nominal outside diameter in in.
t = nominal wall thickness in in.
Y_p = minimum yield strength of the pipe in psi
p_{Yp} = minimum yield strength collapse pressure in psi
p_p = minimum plastic collapse pressure in psi
p_T = minimum plastic/elastic transition collapse pressure in psi
p_E = minimum elastic collapse pressure in psi
p_{CA} = minimum collapse pressure under axial tension stress in psi
p_{CO} = minimum collapse pressure without axial tension stress in psi
S_A = axial tension stress in psi
p_i = internal pressure in psi

Internal Yield Pressure (Section 3 of API 5C3)

Internal Yield Pressure for Pipe. Internal yield pressure for pipe is calculated from formula 4-133. The factor 0.875 appearing in formula 4-313 allows for minimum wall thickness.

$$p_i = 0.875\left|\frac{2Y_p t}{D}\right| \tag{4-313}$$

where p_i = minimum internal yield pressure in psi
Y_p = minimum yield strength in psi
t = nominal wall thickness in in.
D = nominal outside diameter in in.

Internal Yield Pressure for Couplings. Internal yield pressure for threaded and coupled pipe is the same as for plain end pipe, except where a lower pressure is required to avoid leakage due to insufficient coupling strength. The lower pressure is based on

$$p = Y_c\left|\frac{D_c - d_1}{D_c}\right| \tag{4-314}$$

where Y_c = minimum yield strength at coupling in psi
D_c = nominal outside diameter of coupling
d_1 = diameter of the root of the coupling thread at the end of the pipe in the powertight position (see API Bulletin 5C3, 3rd Edition, March 1980)

Pipe Body Yield Strength (Section 2 of API 5C3). Pipe body yield strength is the axial load required to yield the pipe. It is taken as the product of the cross-sectional area and the specified minimum yield strength for the particular grade of pipe. Values for pipe body yield strength were calculated by means of the following formula:

$$p_Y = 0.7854(D^2 - d^2)Y_p \tag{4-315}$$

where p_Y = pipe body yield strength in psi
Y_p = minimum yield strength

D = specified outside diameter in in.
d = specified inside diameter in in.

Joint Strength (Section 9 of API 5C3)

Round Thread Casing Joint Strength. Round thread casing joint strength is calculated from formulas 4-316 and 4-317. The lesser of the values obtained from the two formulas governs. Formulas 4-316 and 4-317 apply to both short and long threads and couplings. Formula 4-316 is for minimum strength of a joint failing by fracture and formula 4-317 for minimum strength of a joint failing by thread jumpout or pullout.

The fracture strength is

$$T_j = 0.95A_{jp}U_p \tag{4-316}$$

The pull-out strength is

$$T_j = 0.95A_{jp}L\left|\frac{0.74^{-0.59}U_p}{0.5L+0.14D}+\frac{Y_p}{L+0.14D}\right| \tag{4-317}$$

where T_j = minimum joint strength in lb
A_{jp} = cross-sectional area of the pipe wall under the last perfect thread in in.^{2}
= $0.7854\ |(D - 0.1425)^2 - d^2|$ for eight round threads
D = nominal outside diameter of pipe in in.
d = nominal inside diameter of pipe in in.
L = engaged thread length in in.
= L_4 − M for nominal makeup, Standard 5B
Y_p = minimum yield strength of pipe in psi
U_p = minimum ultimate strength of pipe in psi

Buttress Thread Casing Joint Strength. Buttress thread casing joint strength is calculated from formulas 4-318 and 4-319. The lesser of the values obtained from the two formulas governs.

Pipe thread strength is

$$T_j = 0.95A_pU_p\left|1.08-0.0386(1.083-Y_p/U_p)D\right| \tag{4-318}$$

Casing thread strength is

$$T_j = 0.95A_cU_c \tag{4-319}$$

where T_j = minimum joint strength in lb
Y_p = minimum yield strength of pipe in lb
U_p = minimum ultimate strength of pipe in psi
U_c = minimum ultimate strength of coupling in psi
A_p = cross-sectional area of plain end pipe in in.^{2} where $A_p = 0.7854(D^2 - d^2)$
A_c = cross-sectional area of coupling in in.^{2} where $A_c = 0.7854(D_c^2 - d_i^2)$
D = outside diameter of pipe in in.
D_c = outside diameter of coupling in in.

d = inside diameter of pipe in in.
d_j = diameter of the root of the coupling thread at the end of the pipe in the powertight position

Extreme-line Casing Joint Strength. Extreme-line casing joint strength is calculated from

$$T_j = A_{cr}U_p \tag{4-320}$$

where T_j = minimum joint strength in lb
A_{cr} = critical section area of box, pin or pipe, whichever is least, in in.^{2} (see API Bulletin 5C3)
U_p = specified minimum ultimate strength in psi (Table 4-142)

Combination Casing Strings

The term *combination casing string* is generally applied to a casing string that is composed of more than one weight per foot, or more than one grade of steel, or both.

Design Consideration

Solving the problem of casing string design for known type and size of casing string relies on selection of the most economical grades and weights of casing that will withstand, without failure, the loads to which the casing will be subjected throughout the life of the well.

There are various established methods of designing a technically satisfactory combination casing string. The differences between these methods rely upon different design models, different values of the safety factors and different sequences of calculations. There are no commonly accepted methods of combination casing string design nor accepted values for the safety factors. Some suggestions are offered below; however, the decision is left to the person responsible for the design.

In general, the following loads must be considered: i.e., tension, collapse, burst and compression, and the reasonably worst working conditions ought to be assumed.

Collapse

The casing must be designed against collapse to withstand the hydrostatic pressure of the fluid behind the casing at any depth, decreased by anticipated pressure inside the casing at the corresponding level. Usually, the maximum collapse pressure to be imposed on the casing string is considered to be the hydrostatic pressure of the heaviest mud used to drill to the lending depth of the casing string, acting on empty string. Depending upon design model, it is recommended to use a design factor of 1.0 to 1.2. For example, if it is known that casing will never be empty inside, this fact should be considered for collapse pressure evaluation and selection of the magnitude of safety factor.

Burst

Casing must be designed to resist expected burst pressure at any depth. In burst pressure consideration, it is suggested to consider different design models depending upon the type of casing string.

Conductor String

It is assumed that the external pressure is zero. In any case, the maximum expected internal pressure cannot be greater than fracture pressure at the open hole below the conductor casing shoe; usually, it is the first formation right below the casing shoe. If this pressure is not known (in exploratory drilling), the burst pressure of gas equivalent to 0.9 or 1.0 psi/ft can be assumed. Hydrostatic head due to gas weight is neglected. For example, if the setting depth of conductor string is 1,100 ft, then the maximum expected burst pressure is even along the string and equal to (1,100)(1.0) = 1,100 psi; Safety factor = 1.1 to 1.15.

Surface and Intermediate Strings

It is suggested to evaluate the burst load based on the internal pressure expected, reduced by the external pressure of the drilling fluid outside the string. Internal pressure is based on the expected bottomhole pressure of the next string with the hole being evacuated from drilling fluid up to a minimum of 50%. In exploratory wells, a reasonable assumption of expected formation pore pressure gradient is required.

Example 3

Evaluate an expected burst pressure acting on surface casing string in exploratory drilling if setting depth of the string is 5,000 ft, mud specific gravity is 1.2 and setting depth of the next string is 11,000 ft.

Solution

Step 1: Internal pressure in the borehole. Because the next string is set at 11,000 ft, the formation pore pressure gradient is assumed to be 0.65 psi/ft. Thus, the bottomhole pressure (at a depth of 11,000 ft) is (11,000)(0.65) = 7,150 psi.

Assume 50% of evacuation; thus, (11,000)(0.5) = 5,500 ft.

Note: It is assumed that below 5,500 ft, the hole is filled with mud which exerts a pressure gradient of 0.65 psi/ft. The hole above 5,500 ft is filled with gas, the weight of which is ignored.

The internal pressure at a depth of 5,500 ft is (5,500)(0.65) = 3,575 psi. Since the weight of gas is ignored for this type of string, the internal pressure at the top of the hole is also 3,575 psi.

Step 2: External pressure. It is assumed that there is drilling fluid with specific gravity of 1.2 outside the casing. Thus, the external pressure at the surface = 0.0 psi and at 5,000 ft is (5,000)(1.2)(8.34)(0.052) ≃ 2,600 psi.

Step 3: Burst load (P_b). The burst pressure is equal to internal pressure reduced by the external pressure of the drilling fluid outside the casing. Therefore,

at surface: P_b = 3,575 − 0.0 = 3.575 psi

at 5,000 ft: P_b = 3,575 − 2600 = 987 psi

The burst pressure line equation is as below:

$$P_b = 3575 - 0.52(D) \text{ (psi)}$$

D = depth at the hole (ft) (from 0 to 5,000 ft)

Note: For practical purposes, a graphic solution is very advisable.

Production String

In exploratory drilling, it is assumed that internal pressure acting on the casing is reduced by external saltwater pressure gradient of about 0.5 psi/ft. Internal pressure is based on expected gas pressure gradient. For long strings, the weight of gas is not ignored.

Tension Load

The maximum tensile load acting on the casing string is often considered as the static weight of the casing string as measured in air.

Casing must be designed to satisfy these equations:

$$p_j = (W)(N_j) \tag{4-321}$$

$$p_y = (W)(N_p) \tag{4-322}$$

where p_j = casing joint strength in lb
p_y = pipe body strength in lb
W = weight of casing suspended below the cross-section under consideration in lb
N_j = safety factor for joint
N_p = safety factor for pipe body

Safety factors (N_j, N_p) of 1.6 to 2.0 are used and should be applied to the minimum joint tensile strength or the minimum pipe body tensile strength, whichever is the smallest.

Compression Load

Under certain conditions, casing can be subjected to the compression load, e.g., if the weight of the inner strings (conductor or surface casing string) is transferred to the outer string or if the portion of the casing weight is slacked off on the bottom of the hole. This load may result in casing failure and, therefore, must also be considered.

It should be pointed out that hydrostatic pressure does not produce an effective compression and, therefore, is not considered. If the casing string is suspended at the top of the hole that is filled with fluid, then the only effect of hydrostatic pressure is reduction of casing weight per foot and the string is effectively under tension.

An example is offered on the following pages of the design procedure to be followed in designing a combination casing string.

Example 4

Design a combination casing string if data are as below:

Type of well: exploration well
Type of casing: production for testing purposes

Casing setting depth: 12,000 ft
Casing size: 7 in.

Design model:

Collapse: Assumed external fluid pressure gradient of 0.52 psi/ft and casing empty inside. Safety factor for collapse = 1.0. Reduction of collapse pressure resistance due to the axial load is considered.
Burst: Assumed external pressure gradient of saltwater = 0.465 psi/ft and formation pore (gas) pressure gradient = 0.65 psi/ft. Gas weight is neglected. Safety factor = 1.1.
Tension: Casing suspended at the surface. Weight reduction due to buoyancy effect is ignored. Safety factor = 1.6.
Compression: Casing not subjected to compression load.

The selected coupling should be long with round thread. The available casing grade is N-80 and unit weights as given in table below.

Steel grade	Unit weight (lb/ft)	Cross-sectional area (in.)	Collapse pressure resistance	Burst pressure resistance	Joint strength (10^3 lb)	Pipe body strength (10^3 lb)
N-80	26.0	7.548	5,410	7,240	519	604
N-80	29.0	8.451	7,020	8,160	597	676
N-80	32.0	9.315	8,600	9,060	672	745

Solution

Part 1. Consider collapse pressure and tension load.

Step 1. Determine the lightest weight of casing to resist collapse pressure for a setting depth of 12,000 ft. Because the maximum collapse pressure is (12,000)(0.52) = 6,240 psi, select N-80, 29-lb/ft casing with collapse pressure resistance of 7,020 psi. (*Note:* assumed safety factor for collapse = 1.0.) This is Section 1.

Step 2. The next section (above Section 1) is to consist of the next lighter casing, i.e., N-80, 26 lb/ft. This is Section 2. Neglecting the effect of the axial load due to the weight of Section 1 suspended below it, the setting depth of Section 2 is

$$D_2' = \frac{5,410}{(1.0)(0.52)} = 10,403\,\text{ft}$$

Under this assumption, the weight of Section 1 is

$$W_1' = (12000 - 10403)(29) = 46,313 \text{ lb}$$

For this axial load, the reduced minimum collapse pressure resistance of Section 2 can be calculated from formula 4-312. (*Note:* internal pressure P_i for considered case = 0.)

$$p_{CA}(2) = 5,410\left[\sqrt{1 - 0.75\left(\frac{6,136}{80,000}\right)^2} - (0.5)\frac{6,136}{80,000}\right] = 5,190\,\text{psi}$$

(*Note:* S_A = W′/A = 46,313/7.548 = 6,136 psi)

Step 3. Using the obtained reduced minimum collapse pressure resistance of Section 2, calculate the setting depth of this section:

$$D_2'' = \frac{5,190}{(1.0)(0.52)} = 9,980\,\text{ft}$$

For obtained setting depth of Section 2, the weight of Section 1 is

W_1'' = (12,000 - 9,980)(29) = 58,580 lb

and corresponding reduced minimum collapse pressure resistance of Section 2 is

$$p_{CA}''(2) = 5,410\left|\sqrt{1 - 0.75\left(\frac{7,761}{80,000}\right)^2} - (0.5)\frac{7,761}{80,000}\right| = 5,128\,\text{psi}$$

Step 4. The third assumed setting depth of Section 2 is usually taken as a correct setting depth, i.e.,

$$D_2''' = \frac{5,128}{(1.0)(0.52)} = 9,861\,\text{ft}$$

Then, the length and weight of Section 2 is

L_1 = 12,000 - 9,861 = 2,128 ft and W_1 = 62,015 lb

(*Note:* If the next lighter casing were available, then Steps 2 through 4 must be repeated for this casing and that would be Section 3, etc.). The maximum length of Section 2 is limited by coupling load capacity and is calculated below:

$$L_{2\,max} = \frac{519,000 - (62,015)(1.6)}{(1.6)(26)} = 10,090\,\text{ft}$$

which is greater than its setting depth (9,861 ft). So, Section 2 extends to the top of the hole. (*Note:* If Section 2 would not cover the entire length of the hole, then the next stronger casing should be applied. That would be Section 3. The setting depth of Section 3 is governed by joint strength, not by collapse pressure.)

Part 2. Check casing string on burst pressure obtained in Part 1 and make necessary corrections.

Step 1. Determine external pressure.
At top: 0.0 psi.
At bottom: (12,000)(0.465) = 5,580 psi.

Step 2. Determine internal pressure.
At bottom: (12,000)(0.65) = 7,800 psi.
At top: 7,800 psi (weight of gas is ignored).

Step 3. Determine burst pressure.
At top: 7,800 psi.
At bottom: 7,800 - 5,580 = 2,220 psi.
Burst pressure line equation is

$$P_b = 7,800 - \frac{7,800 - 2,220}{12,000}(D) = 7,800 - (0.456)(D)$$

D = hole depth in ft

Graphical solution is presented in Figure 4-377.

Step 4. It is apparent that Section 1 is capable of withstanding the expected burst pressure.

Step 5. Section 2 can withstand the expected burst pressure up to the depth calculated below:

$$\frac{7,240}{1.1} = 7,800 - (0.456)(D)$$

$$D = \frac{7,800 - 6,581}{0.456} = 2,673\,\text{ft}$$

Therefore, the length and weight of Section 2 is

9,861 - 2,673 = 7,188 ft

W_2 = (7,188)(26) = 186,888 lb

To cover the upper part of the hole (2,673 ft), stronger casing must be used.

Step 6. Take the next stronger casing, i.e., N-80, 29 lb/ft with burst pressure resistance of (8,160)/(1.1) = 7,418 psi.
This is Section 3. This casing can be used up to the hole depth of

7,418 = 7,800 - (0.456)(D)

$$D = \frac{7,800 - 7,418}{0.456} = 837\,\text{ft}$$

Then, the length and weight of Section 3 is 2,673 - 837 = 1,836 ft

W_3 = (1,836)(26) = 53,244 lb

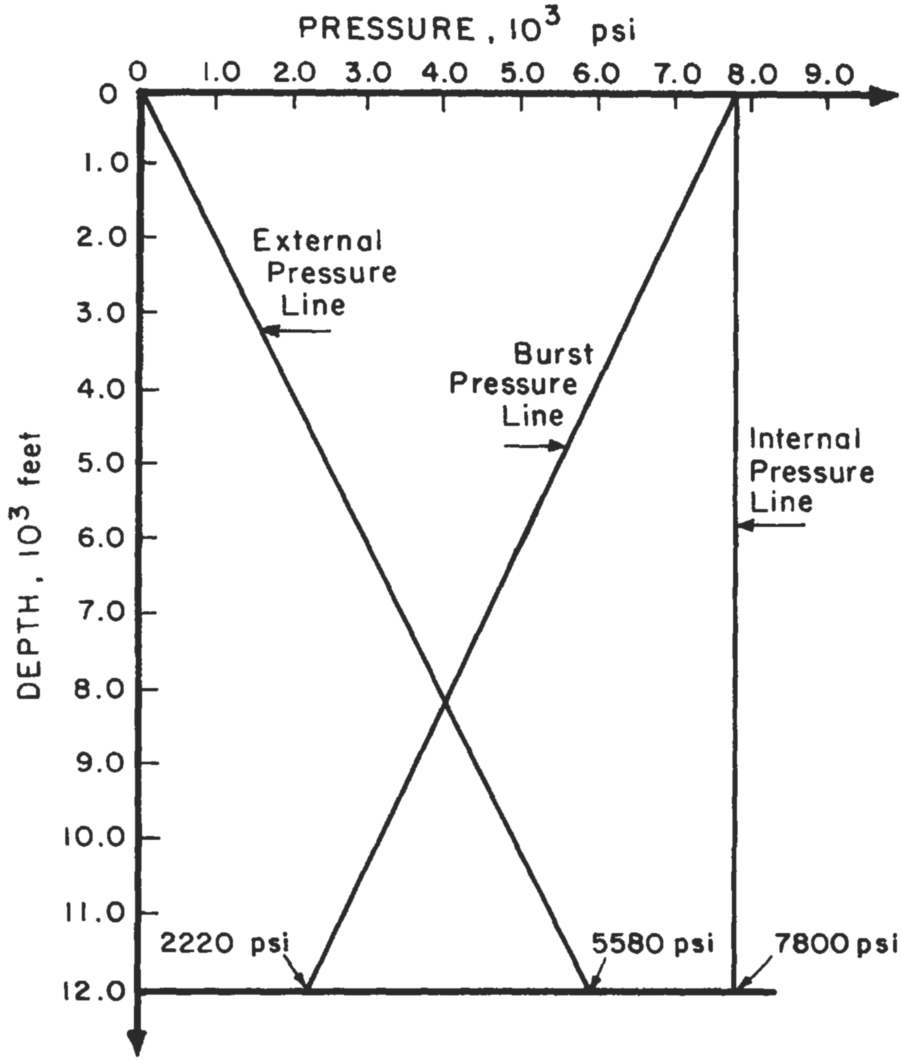


Figure 4-377. Graphical representation of burst load determination.

To cover the remaining 837 ft of the hole, stronger casing must be used.

Step 7. The next stronger casing is N-80, 32 lb/ft with burst pressure resistance of (9,060)/(1.1) = 8,236 psi. This is Section 4.

N-80, 32-lb/ft casing is strong enough to cover the remaining part of the hole (*Note:* 8,236 psi > 7,800 psi). Therefore, the length and weight of Section 4 is 837 − 0.0 = 837 ft and W_4 = 26,784 lb.

Part 3. Check casing string on tension obtained in Part 2 and, if necessary, make corrections to satisfy the required magnitude of safety factor.

Step 1. Maximum length of Section 3 due to its joint strength is

$$L_3 = \frac{597,000-(1.6)(62,015+186,888)}{(1.6)(29)} = 4,283\,\text{ft}$$

Since the length of Section 3 is 1,836, the safety factor is even greater than required.

Step 2. Maximum length of Section 4 due to its joint strength is

$$L_4 = \frac{67,200-(1.6)(62,015+186,888+26,784)}{(1.6)(32)} = 3,682\,\text{ft}$$

Because L_4 is greater than the length obtained in Part 2 (837), it may be concluded that the requirements for tension are satisfied.

A summary of the results obtained is presented in the table below and in Figure 4-378.

Section no.	Setting depth (ft)	Length	Weight (lb)	Grade	Unit weight	Coupling
4	837	837	26,784	N-80	32	Long with round thread
3	2,637	1,836	53,244	N-80	29	Long with round thread
2	9,861	7,188	186,888	N-80	26	Long with round thread
1	12,000	2,138	62,015	N-80	29	Long with round thread

Running and Pulling Casing

The following excerpts are taken from API Recommended Practice: 5C1, "Care and Use of Casing and Tubing," 12th Edition, March 1981 (Section 1, Preparation and Inspection before Running).

1.1 New casing is delivered free of injurious defects. Various nondestructive inspection services have been employed by users to assure that the desired quality of casing is being run.

1.2 All casing, whether new or used or reconditioned, should be handled with thread protectors in place. Casing should be handled at all times on racks or on wooden or metal surfaces free of rocks, sand or dirt other than normal drilling fluid.

1.3 Slip elevators are recommended for long strings. Both spider and elevator slips should be clean and sharp and should fit properly. Slips should be extra long for heavy casing strings. The spider must be level. *Note:* slip and tong marks are injurious. Every possible effort must be made to keep such damage at a minimum by using up-to-date equipment.

1.4 If collar-pull elevators are used, the bearing surface should be carefully inspected for (1) uneven wear, which may produce a side lift on the coupling with danger of jumping it off, and (2) uniform distribution of the load where applied over the bearing face of the coupling.

1.5 Spider and elevator slips should be examined and watched to see that all lower together. If they lower unevenly, there is danger of denting the pipe or badly slip-cutting it.

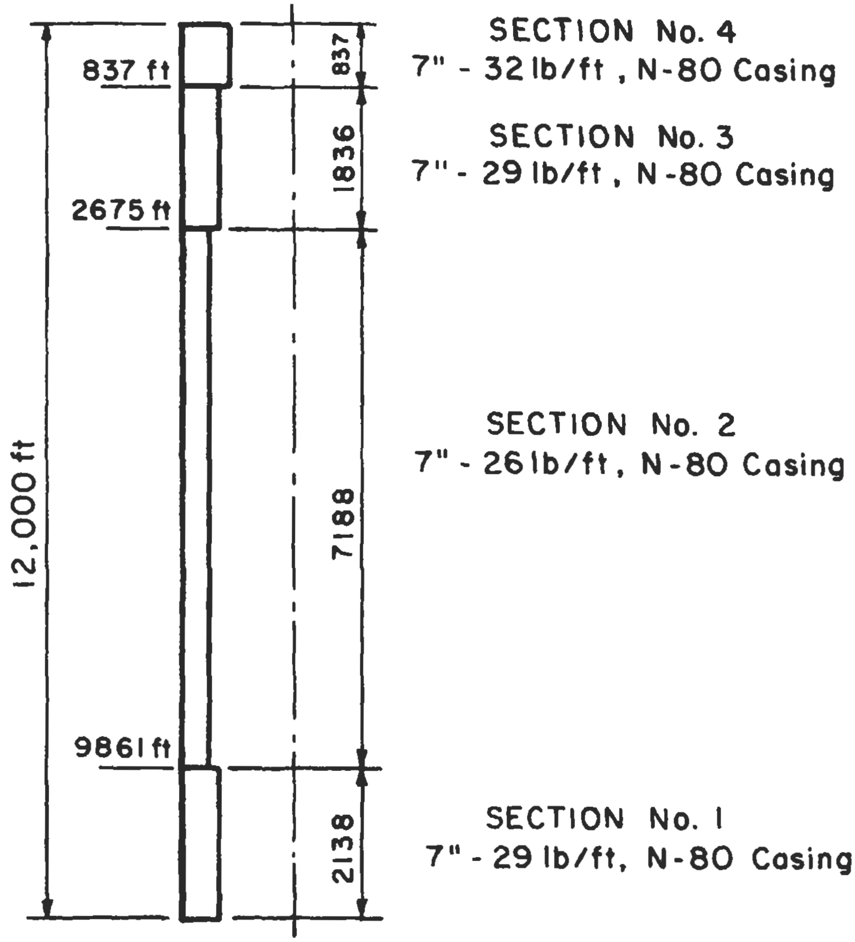


Figure 4-378. Schematic diagram of combination casing string.

1.6 Care must be exercised, particularly when running long casing strings, to ensure that slip bushing or bowl is in good condition. Tongs should be examined for wear on hinge pins and hinge surfaces. The backup line attachment to the backup post should be corrected if necessary to be level with the tong in the backup position, so as to avoid uneven load distribution on the gripping surfaces of the casing. The length of backup line should be such as to cause minimum bending stresses on the casing and to allow full stroke movement of the makeup tong.

1.7 The following precautions should be taken in the preparation of casing threads for makeup in the casing strings:

a. Immediately before running, remove thread protectors from both field and coupling ends and clean the threads thoroughly, repeating as additional rows become uncovered.

b. Carefully inspect the threads. Those found damaged, even slightly, should be laid aside unless satisfactory means are available for correcting thread damage.
c. The length of each piece of casing shall be measured prior to running. A steel tape, calibrated in decimal feet to the nearest 0.01 ft, should be used. The measurements should be made from the outermost face of the coupling or box to the position on the externally threaded end where the coupling or the box stops when the joint is made up powertight. On round-thread joints this position is the place of the vanish point on the pipe; on buttress-thread casing, this position is to the base of the triangle stamp on the pipe; on extreme line casing to the shoulder on the externally threaded end. The total of the individual length so measured will represent the unloaded length of the casing string.
Note: The actual length under tension in the hole can be obtained from pertinent graphs or approximately calculated from

$$\Delta L = \frac{L^2}{2E}\left[\gamma_s - 2\gamma_f(1-\mu)\right] \tag{4-323}$$

where ΔL = casing elongation in in.
L = original length of casing in in.
E = casing modulus of elasticity (for steel $E = (30)(10^6)$ lb/in.^{2})
γ_s = casing specific weight, lb/in.^{3} (for steel $\gamma_s = 0.283$ lb/in.^{3})
γ_f = fluid specific weight in lb/in.^{3}
μ = Poisson's ratio (for steel $\mu = 0.28$)

Formula 4-323 is valid for any consistent system of units and is applicable to vertical holes.
d. Check each coupling for makeup. If the stand-off is abnormally great, check the coupling for tightness. Tighten any loose couplings after thoroughly cleaning the threads and applying fresh compound over the entire thread surfaces, and before pulling the pipe into the derrick.
e. Before stubbing, liberally apply thread compound to the entire internally and externally threaded areas. It is recommended that high-pressure modified thread compound as specified in API Bulletin 5A2: "Bulletin on Thread Compounds" be used except in special cases where severe conditions are encountered; it is recommended that high pressure silicone thread compound as specified in Bulletin 5A2 be used.
f. Place a clean thread protector on the field end of the pipe so that the thread will not be damaged while rolling pipe on the rack and pulling into the derrick. Several thread protectors may be cleaned and used repeatedly for this operation.
g. If a mixed string is to be run, check to determine that appropriate casing will be accessible on the pipe rack when required according to program.
h. Connectors used as tensile and lifting members should have their thread capacity carefully checked to assure that the connector can safely support the load.
i. Care should be taken when making up pup joints and connectors to assure that the mating threads are of the same size and type.

1.8 Drifting of casing. It is recommended that each length of casing be drifted for its entire length just before running with mandrels conforming to the requirements of Standard 5A: "Specification for Casing, Tubing and Drill Pipe." Casing that will not pass the drift test should be laid aside.

1.9 Lower or roll each piece of casing carefully to the walk without dropping. Use rope snubber if necessary. Avoid hitting casing against any part of derrick or other equipment. Provide a hold back rope at window. For mixed and unmarked strings, a drift or "jack rabbit" should be run through each length of casing when it is picked up from the catwalk and pulled onto the derrick floor, to avoid running a heavier length or one with a lesser inside diameter than called for in the casing string.

Stubbing, Making Up, and Lowering

1.10 Do not remove thread protector from field end of casing until ready to stub.

1.11 If necessary, apply thread compound over entire surface of threads just before stubbing.

1.12 In stubbing, lower casing carefully to avoid injuring threads. Stub vertically, preferably with assistance of someone on the stubbing board. If the casing stub tilts to one side after stubbing, lift up, clean and correct any damaged thread with three-cornered file, then carefully remove any filings to ensure that threads are engaging properly and not cross-threading. If spinning line is used, it should pull close to the coupling.
Note: Recommendations in paragraphs 1.13 and 1.14 for casing makeup apply to the use of power tongs. For recommendations of makeup of casing with spinning lines and conventional tongs, see paragraph 1.15.

1.13 The use of power tongs for making up casing made desirable the establishment of recommended torque values for each size, weight and grade of casing. Early studies and tests indicated that torque values are affected by a large number of variables, such as variations in taper, lead, thread height and thread form; surface finish; type of thread compound; length of thread; weight and grade of pipe; etc. In view of the number of variables and the extent that those variables, alone or in combination, could affect the relationship of torque versus madeup positions, it was evident that both applied torque and madeup position must be considered. Since the API joint pullout strength formula in API Bulletin 5C2 contains several of the variables believed to affect torque, the use of a modification of this formula to obtain torque values was investigated. Torque values obtained by taking 1% of the calculated pull-out value were found to be generally comparable to values obtained by field makeup tests using API-modified thread compound in accordance with API Bulletin 5A2. This procedure was, therefore, used to establish the optimum makeup torque values listed in Table 4-153. Maximum torque values listed are 75% of optimum values and maximum values listed are 125% of optimum values. These values must necessarily be considered a guide only, due to the very wide variations in torque requirements that can exist for a specific connection. Because of this, it is essential that torque be related to makeup position as outlined in paragraph 1.14.

(text continued on page 1174)

Table 4-153
Recommended Casing Make-Up Torque

1	2	3	4	5	6	7	8	9
Size: Outside Diameter in.	Nominal Weight Threads, and Coupling lb per ft	Grade	Torque, ft-lb					
			Short Thread			Long Thread		
			Optimum	Minimum	Maximum	Optimum	Minimum	Maximum
4½	9.50	H-40	770	580	960	—	—	—
	9.50	J-55	1010	760	1260	—	—	—
	10.50	J-55	1320	990	1650	—	—	—
	11.60	J-55	1540	1160	1930	1620	1220	2030
	9.50	K-55	1120	840	1400	—	—	—
	10.50	K-55	1460	1100	1830	—	—	—
	11.60	K-55	1700	1280	2130	1800	1350	2250
	11.60	C-75	—	—	—	2150	1610	2690
	13.50	C-75	—	—	—	2600	1950	3250
	11.60	L-80	—	—	—	2230	1670	2790
	13.50	L-80	—	—	—	2710	2030	3390
	11.60	N-80	—	—	—	2280	1710	2850
	13.50	N-80	—	—	—	2760	2070	3450
	11.60	C-95	—	—	—	2580	1940	3230
	13.50	C-95	—	—	—	3130	2350	3910
	11.60	P-110	—	—	—	3020	2270	3780
	13.50	P-110	—	—	—	3660	2750	4580
	15.10	P-110	—	—	—	4400	3300	5500
5	11.50	J-55	1330	1000	1660	—	—	—
	13.00	J-55	1690	1270	2110	1820	1370	2280
	15.00	J-55	2070	1550	2590	2230	1670	2790
	11.50	K-55	1470	1100	1840	—	—	—
	13.00	K-55	1860	1400	2330	2010	1510	2510
	15.00	K-55	2280	1710	2850	2460	1850	3080
	15.00	C-75	—	—	—	2960	2220	3700
	18.00	C-75	—	—	—	3770	2830	4710
	21.40	C-75	—	—	—	4660	3500	5830
	24.10	C-75	—	—	—	5390	4040	6740
	15.00	L-80	—	—	—	3080	2310	3850
	18.00	L-80	—	—	—	3950	2950	4910
	21.40	L-80	—	—	—	4860	3650	6080
	24.10	L-80	—	—	—	5610	4210	7010
	15.00	N-80	—	—	—	3140	2360	3930
	18.00	N-80	—	—	—	4000	3000	5000
	21.40	N-80	—	—	—	4950	3710	6190
	24.10	N-80	—	—	—	5720	4290	7150
	15.00	C-95	—	—	—	3560	2670	4450
	18.00	C-95	—	—	—	4550	3410	5690
	21.40	C-95	—	—	—	5620	4220	7030
	24.10	C-95	—	—	—	6500	4880	8130
	15.00	P-110	—	—	—	4170	3130	5210
	18.00	P-110	—	—	—	5310	3980	6640
	21.40	P-110	—	—	—	6580	4940	8230
	24.10	P-110	—	—	—	7600	5700	9500

Table 4-153
(continued)

1	2	3	4	5	6	7	8	9
Size: Outside Diameter in.	Nominal Weight, Threads and Coupling lb. per ft.	Grade	Torque, ft-lb					
			Short Thread			Long Thread		
			Optimum	Minimum	Maximum	Optimum	Minimum	Maximum
5½	14.00	H-40	1300	980	1630	—	—	—
	14.00	J-55	1720	1290	2150	—	—	—
	15.50	J-55	2020	1520	2530	2170	1630	2710
	17.00	J-55	2290	1720	2860	2470	1850	3090
	14.00	K-55	1890	1420	2360	—	—	—
	15.50	K-55	2220	1670	2780	2390	1790	2990
	17.00	K-55	2520	1890	3150	2720	2040	3400
	17.00	C-75	—	—	—	3270	2450	4090
	20.00	C-75	—	—	—	4030	3020	5040
	23.00	C-75	—	—	—	4730	3550	5910
	17.00	L-80	—	—	—	3410	2560	4260
	20.00	L-80	—	—	—	4200	3150	5250
	23.00	L-80	—	—	—	4930	3700	6160
	17.00	N-80	—	—	—	3480	2610	4350
	20.00	N-80	—	—	—	4280	3210	5350
	23.00	N-80	—	—	—	5020	3770	6280
	17.00	C-95	—	—	—	3960	2970	4950
	20.00	C-95	—	—	—	4870	3650	6090
	23.00	C-95	—	—	—	5720	4290	7150
	17.00	P-110	—	—	—	4620	3470	5780
	20.00	P-110	—	—	—	5690	4270	7110
	23.00	P-110	—	—	—	6680	5010	8350
6⅝	20.00	H-40	1840	1380	2300	—	—	—
	20.00	J-55	2450	1840	3060	2660	2000	3330
	24.00	J-55	3140	2360	3930	3400	2550	4250
	20.00	K-55	2670	2000	3340	2900	2180	3630
	24.00	K-55	3420	2570	4280	3720	2790	4650
	24.00	C-75	—	—	—	4530	3400	5660
	28.00	C-75	—	—	—	5520	4140	6900
	32.00	C-75	—	—	—	6380	4790	7980
	24.00	L-80	—	—	—	4730	3550	5910
	28.00	L-80	—	—	—	5760	4320	7200
	32.00	L-80	—	—	—	6660	5000	8330
	24.00	N-80	—	—	—	4810	3610	6010
	28.00	N-80	—	—	—	5860	4400	7330
	32.00	N-80	—	—	—	6770	5080	8460
	24.00	C-95	—	—	—	5490	4120	6860
	28.00	C-95	—	—	—	6690	5020	8360
	32.00	C-95	—	—	—	7740	5810	9680
	24.00	P-110	—	—	—	6410	4810	8010
	28.00	P-110	—	—	—	7810	5860	9760
	32.00	P-110	—	—	—	9040	6780	11300

Table 4-153 (continued)

1	2	3	4	5	6	7	8	9
Size: Outside Diameter in.	Nominal Weight, Threads and Coupling lb. per ft.	Grade	Torque, ft-lb					
			Short Thread			Long Thread		
			Optimum	Minimum	Maximum	Optimum	Minimum	Maximum
7	17.00	H-40	1220	920	1530	—	—	—
	20.00	H-40	1760	1320	2200	—	—	—
	20.00	J-55	2340	1760	2930	—	—	—
	23.00	J-55	2840	2130	3550	3130	2350	3910
	26.00	J-55	3340	2510	4180	3670	2750	4590
	20.00	K-55	2540	1910	3180	—	—	—
	23.00	K-55	3090	2320	3860	3410	2560	4260
	26.00	K-55	3640	2730	4550	4010	3010	5010
	23.00	C-75	—	—	—	4160	3120	5200
	26.00	C-75	—	—	—	4890	3670	6110
	29.00	C-75	—	—	—	5620	4220	7030
	32.00	C-75	—	—	—	6330	4750	7910
	35.00	C-75	—	—	—	7030	5270	8790
	38.00	C-75	—	—	—	7670	5750	9590
	23.00	L-80	—	—	—	4350	3260	5440
	26.00	L-80	—	—	—	5110	3830	6390
	29.00	L-80	—	—	—	5870	4400	7340
	32.00	L-80	—	—	—	6610	4960	8260
	35.00	L-80	—	—	—	7340	5510	9180
	38.00	L-80	—	—	—	8010	6010	10010
	23.00	N-80	—	—	—	4420	3320	5530
	26.00	N-80	—	—	—	5190	3890	6490
	29.00	N-80	—	—	—	5970	4480	7460
	32.00	N-80	—	—	—	6720	5040	8400
	35.00	N-80	—	—	—	7460	5600	9330
	38.00	N-80	—	—	—	8140	6110	10180
	23.00	C-95	—	—	—	5050	3790	6310
	26.00	C-95	—	—	—	5930	4450	7410
	29.00	C-95	—	—	—	6830	5120	8540
	32.00	C-95	—	—	—	7680	5760	9600
	35.00	C-95	—	—	—	8530	6400	10660
	38.00	C-95	—	—	—	9310	6980	11640
	26.00	P-110	—	—	—	6930	5200	8660
	29.00	P-110	—	—	—	7970	5980	9960
	32.00	P-110	—	—	—	8970	6730	11210
	35.00	P-110	—	—	—	9960	7470	12450
	38.00	P-110	—	—	—	10870	8150	13590
7⅝	24.00	H-40	2120	1590	2650	—	—	—
	26.40	J-55	3150	2360	3940	3460	2600	4330
	26.40	K-55	3420	2570	4280	3770	2830	4710
	26.40	C-75	—	—	—	4610	3460	5760
	29.70	C-75	—	—	—	5420	4070	6780
	33.70	C-75	—	—	—	6350	4760	7940
	39.00	C-75	—	—	—	7510	5630	9390
	42.80	C-75	—	—	—	8520	6390	10650
	47.10	C-75	—	—	—	9530	7150	11910

Table 4-153 (continued)

1	2	3	4	5	6	7	8	9
Size: Outside Diameter in.	Nominal Weight, Threads and Coupling lb. per ft.	Grade	Torque, ft-lb					
			Short Thread			Long Thread		
			Optimum	Minimum	Maximum	Optimum	Minimum	Maximum
	26.40	L-80	—	—	—	4820	3620	6030
	29.70	L-80	—	—	—	5670	4250	7090
	33.70	L-80	—	—	—	6640	4980	8300
	39.00	L-80	—	—	—	7860	5900	9830
	42.80	L-80	—	—	—	8910	6680	11140
	47.10	L-80	—	—	—	9970	7480	12460
	26.40	N-80	—	—	—	4900	3680	6130
	29.70	N-80	—	—	—	5750	4310	7190
	33.70	N-80	—	—	—	6740	5060	8430
	39.00	N-80	—	—	—	7980	5980	9980
	42.80	N-80	—	—	—	9060	6800	11330
	47.10	N-80	—	—	—	10130	7600	12660
	26.40	C-95	—	—	—	5600	4200	7000
	29.70	C-95	—	—	—	6590	4940	8240
	33.70	C-95	—	—	—	7720	5790	9650
	39.00	C-95	—	—	—	9140	6860	11430
	42.80	C-95	—	—	—	10370	7780	12960
	47.10	C-95	—	—	—	11590	8690	14490
	29.70	P-110	—	—	—	7690	5770	9610
	33.70	P-110	—	—	—	9010	6760	11260
	39.00	P-110	—	—	—	10660	8000	13330
	42.80	P-110	—	—	—	12100	9080	15130
	47.10	P-110	—	—	—	13530	10150	16910
8⅝	28.00	H-40	2330	1750	2910	—	—	—
	32.00	H-40	2790	2090	3490	—	—	—
	24.00	J-55	2440	1830	3050	—	—	—
	32.00	J-55	3720	2790	4650	4170	3130	5210
	36.00	J-55	4340	3260	5430	4860	3650	6080
	24.00	K-55	2630	1970	3290	—	—	—
	32.00	K-55	4020	3020	5030	4520	3390	5650
	36.00	K-55	4680	3510	5850	5260	3950	6580
	36.00	C-75	—	—	—	6480	4860	8100
	40.00	C-75	—	—	—	7420	5570	9280
	44.00	C-75	—	—	—	8340	6260	10430
	49.00	C-75	—	—	—	9390	7040	11740
	36.00	L-80	—	—	—	6780	5090	8480
	40.00	L-80	—	—	—	7760	5820	9700
	44.00	L-80	—	—	—	8740	6560	10930
	49.00	L-80	—	—	—	9830	7370	12290
	36.00	N-80	—	—	—	6880	5160	8600
	40.00	N-80	—	—	—	7880	5910	9850
	44.00	N-80	—	—	—	8870	6650	11090
	49.00	N-80	—	—	—	9970	7480	12460
	36.00	C-95	—	—	—	7890	5920	9860
	40.00	C-95	—	—	—	9040	6780	11300
	44.00	C-95	—	—	—	10170	7630	12710
	49.00	C-95	—	—	—	11440	8580	14300
	40.00	P-110	—	—	—	10550	7910	13190
	44.00	P-110	—	—	—	11860	8900	14830
	49.00	P-110	—	—	—	13350	10010	16690

Table 4-153
(continued)

1	2	3	4	5	6	7	8	9
Size: Outside Diameter in.	Nominal Weight, Threads and Coupling lb. per ft.	Grade	Torque, ft-lb					
			Short Thread			Long Thread		
			Optimum	Minimum	Maximum	Optimum	Minimum	Maximum
9⅝	32.30	H-40	2540	1910	3180	—	—	—
	36.00	H-40	2940	2210	3680	—	—	—
	36.00	J-55	3940	2960	4930	4530	3400	5660
	40.00	J-55	4520	3390	5650	5200	3900	6500
	36.00	K-55	4230	3170	5290	4890	3670	6110
	40.00	K-55	4860	3650	6080	5610	4210	7010
	40.00	C-75	—	—	—	6940	5210	8680
	43.50	C-75	—	—	—	7760	5820	9700
	47.00	C-75	—	—	—	8520	6390	10650
	53.50	C-75	—	—	—	9990	7490	12490
	40.00	L-80	—	—	—	7270	5450	9090
	43.50	L-80	—	—	—	8130	6100	10160
	47.00	L-80	—	—	—	8930	6700	11160
	53.50	L-80	—	—	—	10470	7850	13090
	40.00	N-80	—	—	—	7370	5530	9210
	43.50	N-80	—	—	—	8250	6190	10310
	47.00	N-80	—	—	—	9050	6790	11310
	53.50	N-80	—	—	—	10620	7970	13280
	40.00	C-95	—	—	—	8470	6350	10590
	43.50	C-95	—	—	—	9480	7110	11850
	47.00	C-95	—	—	—	10400	7800	13000
	53.50	C-95	—	—	—	12200	9150	15250
	43.50	P-110	—	—	—	11060	8300	13830
	47.00	P-110	—	—	—	12130	9100	15160
	53.50	P-110	—	—	—	14220	10670	17780
10¾	32.75	H-40	2050	1540	2560	—	—	—
	40.50	H-40	3140	2360	3930	—	—	—
	40.50	J-55	4200	3150	5250	—	—	—
	45.50	J-55	4930	3700	6160	—	—	—
	51.00	J-55	5650	4240	7060	—	—	—
	40.50	K-55	4500	3380	5630	—	—	—
	45.50	K-55	5280	3960	6600	—	—	—
	51.00	K-55	6060	4550	7580	—	—	—
	51.00	C-75	7560	5670	9450	—	—	—
	55.50	C-75	8430	6320	10540	—	—	—
	51.00	L-80	7940	5960	9930	—	—	—
	55.50	L-80	8840	6630	11050	—	—	—
	51.00	N-80	8040	6030	10050	—	—	—
	55.50	N-80	8950	6710	11190	—	—	—
	51.00	C-95	9270	6950	11590	—	—	—
	55.50	C-95	10320	7740	12900	—	—	—
	51.00	P-110	10800	8100	13500	—	—	—
	55.50	P-110	12030	9020	15040	—	—	—
	60.70	P-110	13380	10040	16730	—	—	—
	65.70	P-110	14720	11040	18100	—	—	—

Table 4-153 (continued)

1	2	3	4	5	6	7	8	9
Size: Outside Diameter in.	Nominal Weight, Threads and Coupling lb. per ft.	Grade	Torque, ft-lb					
			Short Thread			Long Thread		
			Optimum	Minimum	Maximum	Optimum	Minimum	Maximum
11¾	42.00	H-40	3070	2300	3840	—	—	—
	47.00	J-55	4770	3580	5960	—	—	—
	54.00	J-55	5680	4260	7100	—	—	—
	60.00	J-55	6490	4870	8110	—	—	—
	47.00	K-55	5090	3820	6360	—	—	—
	54.00	K-55	6060	4550	7580	—	—	—
	60.00	K-55	6930	5200	8660	—	—	—
	60.00	C-75	8690	6520	10860	—	—	—
	60.00	L-80	9130	6850	11410	—	—	—
	60.00	N-80	9240	6930	11550	—	—	—
	60.00	C-95	10660	8000	13330	—	—	—
	60.00	P-110	12420	9320	15530	—	—	—
13⅜	48.00	H-40	3220	2420	4030	—	—	—
	54.50	J-55	5140	3860	6430	—	—	—
	61.00	J-55	5950	4460	7440	—	—	—
	68.00	J-55	6750	5060	8440	—	—	—
	54.50	K-55	5470	4100	6840	—	—	—
	61.00	K-55	6330	4750	7910	—	—	—
	68.00	K-55	7180	5390	8980	—	—	—
	68.00	C-75	9060	6800	11330	—	—	—
	72.00	C-75	9780	7340	12230	—	—	—
	68.00	L-80	9520	7140	11900	—	—	—
	72.00	L-80	10290	7720	12860	—	—	—
	68.00	N-80	9630	7220	12040	—	—	—
	72.00	N-80	10400	7800	13000	—	—	—
	68.00	C-95	11140	8360	13930	—	—	—
	72.00	C-95	12040	9030	15050	—	—	—
	68.00	P-110	12970	9730	16210	—	—	—
	72.00	P-110	14020	10520	17530	—	—	—
*16	65.00	H-40	—	4390	—	—	—	—
	75.00	J-55	—	7100	—	—	—	—
	84.00	J-55	—	8170	—	—	—	—
	75.00	K-55	—	7520	—	—	—	—
	84.00	K-55	—	8650	—	—	—	—
*18⅝	87.50	H-40	—	5590	—	—	—	—
	87.50	J-55	—	7540	—	—	—	—
	87.50	K-55	—	7940	—	—	—	—
*20	94.00	H-40	—	5810	—	—	—	—
	94.00	J-55	—	7840	—	—	9070	—
	106.50	J-55	—	9130	—	—	10570	—
	133.00	J-55	—	11920	—	—	13800	—
	94.00	K-55	—	8240	—	—	9550	—
	106.50	K-55	—	9600	—	—	11130	—
	133.00	K-55	—	12530	—	—	14530	—

* See Par. 1.14h for make-up instructions.

Source: Taken from API RP 5C1, 12th Edition, March 1981.

(text continued from page 1167)

The torque values listed in Table 4-153 apply to casing with zinc-plated couplings. When making up connections with tin-plated couplings, 80% of the listed value can be used as a guide.

Field Makeup

1.14 The following practice is recommended for field makeup casing.

Round Thread, $4\frac{1}{2}$ through $13\frac{3}{8}$ in. OD

a. It is advisable when starting to run casing from each particular mill shipment to make up sufficient joints to determine the torque necessary to provide proper makeup. See paragraph 1.15 for proper number of turns beyond hand-tight position. These values may indicate that a departure from the recommended optimum values listed in Table 4-153 is advisable. If other optimum torque values are chosen, the minimum torque should not be less than 75% of the optimum selected. The maximum torque should be not more than 125% of the optimum torque.

b. The power tong should be provided with a reliable torque gage of known accuracy. To prevent galling when making up connections in the field, the connections should be made up at a speed not to exceed 25 rpm.

c. Continue the makeup, observing both the torque gage and the approximate position of the coupling face with respect to the last scratch position.

d. The optimum torque values as shown in Table 4-153 have been selected to give optimum makeup under normal conditions and should be considered as satisfactory providing the face of the coupling is flush with the last scratch or within two thread turns plus or minus the last scratch.

e. If the makeup is such that the last scratch is buried, two thread turns and the minimum torque shown in Table 4-153 is not reached, the joint should be treated as a questionable joint as provided under paragraph 1.16.

f. If several threads remain exposed when the optimum torque is reached, apply additional torque up to the maximum shown in Table 4-153. If the stand-off (distance from face of coupling to last scratch) is greater than three thread turns when the maximum torque is reached, the joint should be treated as a questionable joint as provided under paragraph 1.16.

Buttress Thread, $4\frac{1}{2}$ through $13\frac{3}{8}$-in. OD

g. Makeup torque for buttress thread casing connection in sizes $4\frac{1}{2}$ through $13\frac{3}{8}$ in. OD should be determined by carefully noting the torque to make up each of several connections to the base of the triangle; then, using the torque value thus established, make up the balance of the pipe of that particular weight and grade in the string.

Round Thread and Buttress Thread, 16, $18\frac{5}{8}$- and 20-in. OD

h. Makeup of 16, $18\frac{5}{8}$ and 20-in. OD shall be to a position on each connection represented by last scratch on 8-round thread and the base of the triangle on buttress thread using the minimum torque shown in Table 4-153 as a guide. Care must be taken to avoid cross-threading in starting these large connections.

1.15 When conventional tongs are used for casing makeup, tighten with tongs to proper degree of tightness. The joint should be made up beyond the hand-tight position at least three turns for sizes $4\frac{1}{2}$ through 7 in., and at least three-and-one-half turns for sizes $7\frac{5}{8}$ in. and larger, except of $9\frac{5}{8}$ and $10\frac{3}{4}$ in. grade, P-110 and 20 in grade J-55 and K-55 which should be made up four turns beyond hand-tight position. When using a spinning line, it is necessary to compare hand-tightness with spin-up tight position. Compare relative position of these two makeups and use this information to determine when the joint is made up the recommended number of turns beyond hand-tight.

1.16 Joints that are questionable as to their proper tightness should be unscrewed and the casing laid down for inspection and repair. Ported joints should never be reused without shopping or regaging, even though the joints may have little appearance of damage.

1.17 If casing has a tendency to wobble unduly at its upper end when making up, indicating that the thread may not be in line with the axis of the casing, the speed of rotation should be decreased to prevent galling of threads. If wobbling should persist despite reduced rotational speed, the casing should be laid down for inspection.

1.18 In making up the field joint it is possible for the coupling to make up slightly on the mill end. This does not indicate that the coupling on the mill end is too loose, but simply that the field end has reached the tightness with which the coupling was screwed on at the mill.

1.19 Casing strings should be picked up and lowered carefully, and care exercised in setting slips to avoid shock loads. Care should be exercised to prevent setting casing down on bottom, or otherwise placing it in compression because of the danger of buckling, particularly in the part of the well where hole enlargement has occurred.

1.20 Definite instructions should be available as to the design of the casing string, including the proper location of the various grades of steel, weight of casing and type of joint. Care should be exercised to run the string in exactly the order in which it was designed.

1.21 Casing should be periodically filled with mud while being run. In most cases, filling every 6–10 lengths should suffice. Filling should be done with mud of the proper weight, using a conveniently located hole of adequate size to expedite the filling operation. A quick-opening and closing plug valve on the mud hole will facilitate the operation and prevent overflow.

Casing Landing Procedure

1.22 Definite instructions should be provided for the proper string tension, also on the proper landing procedure after the cement has set. The purpose is to avoid critical stresses or excessive and unsafe tensile stresses at any time during the life of the well. In arriving at the proper tension and landing procedure, consideration should be given to all factors such as well temperature and pressure, temperature developed due to cement hydration, mud temperature, and changes of temperature during producing operations. The adequacy of the original tension safety factor of the string as designed will influence the landing procedure instructions (and this probably applies to a very large majority of the wells drilled), then the procedure should be followed of landing the casing in the casing head at exactly the position in which it was hanging when the cement plug reached its lowest point or "as cemented."

Care of Casing in Hole

1.23 Drillpipe run inside casing should be with suitable drillpipe protectors.

Recovery of Casing

1.24 Breakout tongs should be positioned close to the coupling but not too close since a slight squashing effect where tong dies contact the pipe surface cannot be avoided, especially if the joint is tight and/or the casing is light. Keeping a space of $\frac{1}{3}$ to $\frac{1}{4}$ of the diameter of the pipe between the tong and the coupling should normally prevent unnecessary friction in the threads. Hammering the coupling to break the joint is an injurious practice. If tapping is required, use the flat face, never the peak face of the hammer, and under no circumstance should a sledge hammer be used. Tap lightly near the middle and completely around the coupling, never near the end nor on opposite sides only.

1.25 Great care should be exercised to disengage all of the thread before lifting the casing out of the coupling. Do not jump casing out of coupling.

1.26 All threads should be cleaned and lubricated or should be coated with a material that will minimize corrosion.

1.27 Before casing is stored or reused, pipe and thread should be inspected and defective joints marked for shopping and regaging.

1.28 When casing is being retrieved because of casing failure it is imperative to future prevention of such failures that a thorough metallurgical study be made. Every attempt should be made to retrieve the failed portion in the "as-failed" condition.

1.29 Casing stacked in the derrick should be set on a firm wooden platform and without the bottom thread or protector since the design or most protectors is not such as to support the joint or stand without damage of the field thread.

Cause of Casing Troubles

1.30 The more common causes of casing troubles are as follows:

a. Improper selection for depth and pressures encountered.
b. Insufficient inspection of each length of casing or of field-shop threads.
c. Abuse in mill, transportation and field handling.
d. Nonobservance of good rules in running and pulling casing.
e. Improper cutting of field-shop threads.
f. The use of poorly manufactured couplings for replacements and additions.
g. Improper care and storage.
h. Excessive torquing of casing to force it through tight places in the hole.
i. Pulling too hard on a string (to free it). This may loosen the coupling at the top of the string. They should be retightened with tongs before finally setting the string.
j. Rotary drilling inside casing. Setting the casing with improper tension after cementing is one of the greatest contributing causes of such failures.

k. Drillpipe wear while drilling inside casing is particularly significant in drifted holes. Excess doglegs in deviated holes or occasionally in straight holes where corrective measurements are taken result in concentrated bending of the casing, which, in turn, results in excess internal wear, particularly when doglegs are high in the hole.
l. Wireline cutting, by swabbing or cable-tool drilling.
m. Buckling of casing in an enlarged, washed-out, uncemented cavity if too much tension is released in landing.
n. Dropping a string, even a very short distance.

WELL CEMENTING

Introduction

Cementing the casing and *liner cementing* (denoted as *primary cementing*) are probably the most important operations in the development of an oil and gas well. The drilling group is usually responsible for cementing the casing and the liner. The quality of these cementing operations will affect the success of follow-on drilling, completion, production and workover efforts in the well [160–162].

In addition to primary cementing of the casing and liner, there are other important well cementing operations. These are *squeeze cementing* and *plug cementing*. Such operations are often called *secondary* or *remedial cementing* [161].

Well cementing materials vary from basic Portland cement used in civil engineering construction of all types, to highly sophisticated special-purpose resin-based or latex cements. The purpose of all of these cementing materials is to provide the well driller with a fluid state slurry of cement, water and additives that can be pumped to specific locations within the well. Once the slurry has reached its intended location in the well and a setup time has elapsed, the slurry material can become a nearly impermeable, durable solid material capable of bonding to rock and steel casing.

The most widely used cements for well cementing are the Portland-type cements. The civil engineering construction industry uses Portland cement and water slurries in conjunction with clean rock aggregate to form concrete. The composite material formed by the addition of rock aggregate forms a solid material that has a compressive strength that is significantly higher than the solid formed by the solidified cement and water slurry alone. The rock of the aggregate usually has a very high compressive strength (of the order of 5,000 to 20,000 psi). The cement itself will have a compressive strength of about 1,000 to 3,000 psi. Therefore, the rock aggregate together with the matrix of solid cement can form a high-strength composite concrete with compressive strengths of the order of 4,000 to 15,000 psi.

The well drilling industry does not generally use aggregate with the cement except for silica flour and Ottawa sand. This is mainly due to the tight spacing within a well that precludes the passage of the larger particles of aggregate through the system. Thus, the well drilling industry refers to this material as simply cement. The slurry pumped to wells is usually a slurry of cement and water with appropriate additives. Because of the lack of aggregate, the compressive strength of well cements are restricted to the order of 200 to about 3,000 psi [160].

Through the past half century the well cementing industry has considered cement compressive strengths of about 500 psi to be acceptable. However, such low compressive strengths plus some of the past cementing practices may not be adequate for future wells.

Chemistry of Cements

Cement is made of calcareous and argillaceous rock materials that are usually obtained from quarries. The process of making cement requires that these raw rock materials be ground, mixed and subjected to high temperatures.

The calcareous materials contain calcium carbonate or calcium oxide. Typical raw calcareous materials are as follows:

Limestone. This is a sedimentary rock that is formed by the accumulation of organic marine life remains (shells or coral). Its main component is calcium carbonate.

Cement rock. This is a sedimentary rock that has a similar composition as the industrially produced cement.

Chalk. This is a soft limestone composed mainly of marine shells.

Marl. This is loose or crumbly deposit that contains a substantial amount of calcium carbonate.

Alkali waste. This is a secondary source and is often obtained from the waste of chemical plants. Such material will contain calcium oxide and/or calcium carbonate.

The argillaceous materials contain clay or clay minerals. Typical raw argillaceous materials are as follows:

Clay. This material is found at the surface of the earth and often is the major component of soils. The material is plastic when wetted, but becomes hard and brittle when dried and heated. It is composed mainly of hydrous aluminum silicates as well as other minerals.

Shale. This sedimentary rock is formed by the consolidation of clay, mud and silt. It contains substantial amounts of hydrous aluminum silicates.

Slate. A dense fine-grained metamorphic rock containing mainly clay minerals. Slate is obtained from metamorphic shale.

Ash. This is a secondary source and is the by-product of coal combustion. It contains silicates.

There are two processes used to manufacture cements: the dry process and the wet process. The dry process is the least expensive of the two, but is the more difficult to control.

In the dry process the limestone and clay materials are crushed and stored in separate bins and their composition analyzed. After the composition is known, the contents of the bins are blended to achieve the desired ultimate cement characteristics. The blend is ground to a mesh size of 100–200. This small mesh size maximizes the contact between individual particles.

In the wet process the clay minerals are crushed and slurried with water to allow pebbles and other rock particles to settle out. The limestone is also crushed and slurried. Both materials are stored in separate bins and analyzed. Once the desired ultimate composition is determined, the slurry blend is ground and then partially dried out.

After the blends have been prepared (either in the dry or wet process), these materials are fed at a uniform rate into a long rotary kiln. The materials are gradually heated to a liquid state. At temperatures up to about 1,600°F the free water evaporates, the clay minerals dehydroxylate and crystallize, and $CaCO_3$ decomposes. At temperatures above 1,600°F the $CaCO_3$ and CaO react with aluminosilicates and the materials become liquids. Heating is continued to as high as 2,800°F.

When the kiln material is cooled it forms into crystallized clinkers. These are rather large irregular pieces of the solidified cement material. These clinkers are ground and a small amount of gypsum is added (usually about 1.5 to 3%). The gypsum prevents flash setting of the cement and also controls free CaO. This final cement product is sampled, analyzed and stored. The actual commercial cement is usually a blend of several different cements. This blending ensures a consistent product.

There are four chemical compounds that are identified as being the active components of cements.

Tricalcium aluminate ($3CaO \bullet Al_2O_3$) hydrates rapidly and contributes most to heat of hydration. This compound does not contribute greatly to the final strength of the set cement, but it sets rapidly and plays an important role in the early strength development. This setting time can be controlled by the addition of gypsum. The final hydrated product of tricalcium aluminate is readily attacked by sulfate waters. High-sulfate-resistant (HSR) cements have only a 3% or less content of this compound. High early strength cements have up to 15% of this compound.

Tricalcium silicate ($3CaO \bullet SiO_2$) is the major contributor to strength at all stages, but particularly during early stages of curing (up to 28 days). The average tricalcium silicate content is from 40% to maximum of 67%. The retared cements will contain from 40% to 45%. The high early strength cements will contain 60% to 67%.

Dicalcium silicate ($2CaO \bullet SiO_2$) is very important in the final strength of the cement. This compound hydrates very slowly. The average dicalcium silicate content is 25% to 35%.

Tetracalcium aluminoferrite ($4CaO \bullet Al_2O_3 \bullet FeO_3$) has little effect on the physical properties of the cement. For high-sulfate-resistant (HSR) cements, API specifications require that the sum of the tetracalcium aluminoferrite content plus twice the tricalcium aluminate may not exceed a maximum of 24%.

In addition to the four compounds discussed above, the final Portland cement may contain gypsum, alkali sulfates, magnesia, free lime and other components. These do not significantly affect the properties of the set cement, but they can influence rates of hydration, resistance to chemical attack and slurry properties.

When the water is added to the final dry cement material, the hydration of the cement begins immediately. The water is combined chemically with the cement material to eventually form a new immobile solid. As the cement hydrates, it will bond to the surrounding surfaces. This cement bonding is complex and depends on the type of surface to be cemented. Cement bonds to rock by a process of crystal growth. Cement bonds to the outside of a casing by filling in the pit spaces in the casing body [163].

Cementing Principles

There are two basic oil well cementing activities: primary cementing and secondary cementing.

Primary cementing refers to the necessity to fix the steel casing or liner (which is placed in the drilled borehole) to the surrounding formations adjacent to the casing or liner. The purposes of primary cementing are the following:

1. support vertical and radial loads applied to casing;
2. isolate porous formations from producing zone formations;
3. exclude unwanted subsurface fluids from the producing interval;

4. protect casing from corrosion;
5. resist chemical deterioration of cement;
6. confine abnormal formation pressures.

When applied to the various casing or liner strings used in oil well completions, the specific purposes of each string are as follows:

- Conductor casing string is cemented to prevent the drilling fluid from escaping and circulating outside the casing.
- Surface casing string must be cemented to protect fresh-water formations near the surface and provide a structural connection between the casing and the subsurface competent rock formations. This subsurface structural connection will allow the blowout preventor to be affixed to the top of this casing to prevent high-pressure fluids from being vented to the surface. Further, this structural connection will give support for deeper run casing or liner strings.
- Intermediate casing strings are cemented to seal off abnormal pressure formations and cover both incompetent formations, which could cave or slough, and lost circulation formations.
- Production casing string is cemented to prevent the produced fluids from migrating to nonproducing formations and to exclude other fluids from the producing interval.

Cementing operations are carried out with surface equipment specially designed to carry out the primary and secondary cementing operations in oil wells. The key element in any well cementing operation is the recirculating blender (Figure 4-379). The recirculating blender has replaced the older jet mixing hopper. The blender provides a constant cement slurry specific weight that could never be achieved by the older equipment. A very careful control of the slurry weight is critical to a successful cementing operation. The recirculating blender is connected to a cement pump that in turn pumps the cement at low

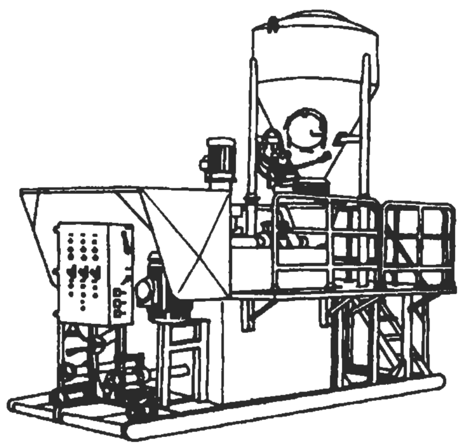


Figure 4-379. Recirculating blender [161].

circulation rates (and high pressures if necessary) to the cementing head at the top of the casing string (see Figures 4-380 and 4-381 and Ref. [162]).

The proper amount of water must be mixed with the dry cement product to ensure only sufficient water for hydration of the cement. Excess water above that needed for hydration will reduce the final strength of the set cement and leave voids in the cement column that are filled with unset liquid. Insufficient water for proper hydration will leave voids filled with dry unset cement, or result in a slurry too viscous to pump.

The cement is usually dry mixed with the additives that are usually added for a particular cementing application. Often this mixing of additives is carried out at the service company central location. Depending on the application of the well cement, there are a variety of additives that can be used to design the cement slurry characteristics. These are accelerators, retarders, dispersants, extenders, weighting agents, gels, foamers and fluid-loss additives. With these additives the cement slurry and ultimately the set cement can be designed for the particular cementing operation. It is necessary that the engineer in charge of the well carry out the necessary engineering design of the slurries to be used in the well. In addition, the engineer should ensure that the work is carried out in accordance to the design specifications. This critical activity should not be left to the service company technicians.

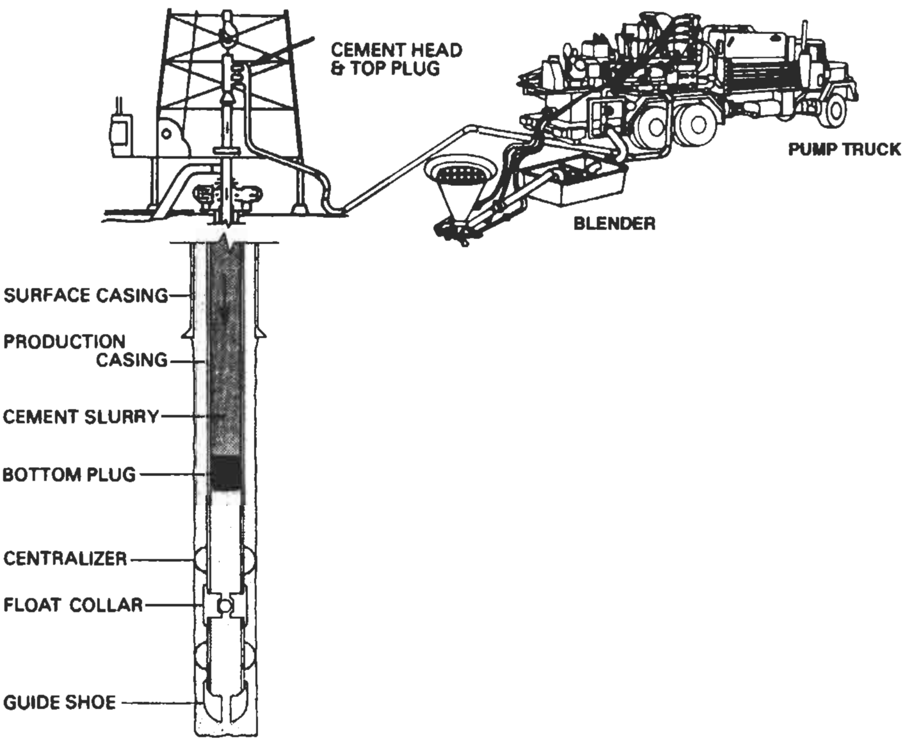


Figure 4-380. Blender, pump truck, cementing head and subsurface equipment [161].

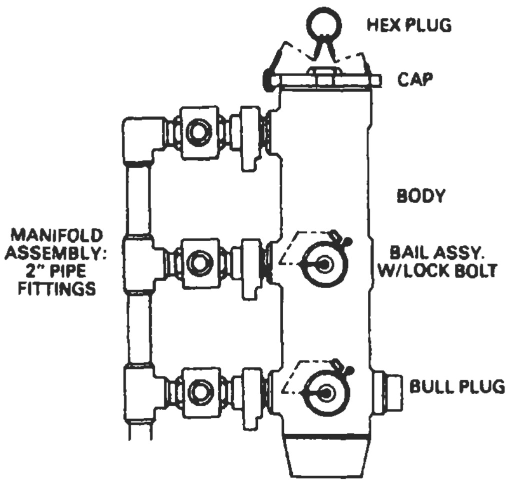


Figure 4-381. Cementing head [161].

Standardization and Properties of Cements

The American Petroleum Institute (API) has nine classes of well cements. These are as follows [164]:

Class A: Intended for use from surface to 6,000 ft (1,830 m) depth, when special properties are not required. Available only in ordinary type (similar to ASTM C 150, Type I*).

Class B: Intended for use from surface to 6,000 ft (1,830 m) depth, when conditions require moderate to high sulfate resistance. Available in both moderate (similar to ASTM C 150, Type II) and high-sulfate-resistant types.

Class C: Intended for use from surface to 6,000 ft (1,830 m) depth, when conditions require high early strength. Available in ordinary and moderate (similar to ASTM C 150, Type III) and high-sulfate-resistant types.

Class D: Intended for use from 6,000 to 10,000 ft (1,830 to 3,050 m) depth, under conditions of moderately high temperatures and pressures. Available in both moderate and high-sulfate-resistant types.

Class E: Intended for use from 10,000 to 14,000 ft (3,050 to 4,270 m) depth, under conditions of high temperatures and pressures. Available in both moderate and high-sulfate-resistant types.

Class F: Intended for use from 10,000 to 16,000 ft (3,050 to 4,880 m) depth, under conditions of extremely high temperatures and pressures. Available in both moderate and high-sulfate-resistant types.

Class G: Intended for use from surface to 8,000 ft (2,440 m) depth as manufactured, or can be used with accelerators and retarders to cover a wide range of well depths and temperatures. No additions other than calcium sulfate or water, or both, shall be interground or blended with the clinker during

*American Society for Testing and Materials (ASTM).

manufacture of Class G well cement. Available in moderate and high-sulfate-resistant types.

Class H: Intended for use as a basic well cement from surface to 8,000 ft (2,440 m) depth as manufactured, and can be used with accelerators and retarders to cover a wide range of well depths and temperatures. No additions other than calcium sulfate or water, or both, shall be interground or blended with the clinker during manufacture of Class H well cement. Available in moderate and high- (tentative) sulfate-resistant types.

Class J: Intended for use as manufactured from 12,000 to 16,000 ft (3,660 to 4,880 m) depth under conditions of extremely high temperatures and pressures or can be used with accelerators and retarders to cover a range of well depths and temperatures. No additions of retarder other than calcium sulfate or blended with the clinker during manufacture of Class J well cement.

The ASTM specifications provide for five types of Portland cements: Types I, II, III, IV and V; they are manufactured for use at atmospheric conditions [165]. The API Classes A, B and C correspond to ASTM Types I, II and III, respectively. The API Classes D, E, F, G, H and J are cements manufactured for use in deep wells and to be subject to a wide range of pressures and temperatures. These classes have no corresponding ASTM types.

Sulfate resistance is an extremely important property of well cements. Sulfate minerals are abundant in some underground formation waters that can come into contact with set cement. The sulfate chemicals, which include magnesium and sodium sulfates, react with the lime in the set cement to form magnesium hydroxide, sodium hydroxide and calcium sulfate. The calcium sulfate reacts with the tricalcium aluminate components of cement to form sulfoaluminate, which causes expansion and ultimately disintegrates to the set cement. To increase the resistance of a cement to sulfate attack, the amount of tricalcium aluminate and free lime in the cement should be decreased. Alternatively, the amount of pozzolanic material can be increased in the cement to obtain a similar resistance. The designations of ordinary sulfate resistance, moderate sulfate resistance and high sulfate resistance in the cement classes above indicate decreasing amounts of tricalcium aluminate.

Table 4-154 gives the basic properties of the various classes of the dry API cements [165].

Properties of Cement Slurry and Set Cement

In well engineering and applications, cement must be dealt with in both its slurry form and in its set form. At the surface the cement must be mixed and then pumped with surface pumping equipment through tubulars to a designated location in the well. After the cement has set, its structure must support the various static and dynamic loads placed on the well tubulars.

Specific Weight

Specific weight is one of the most important properties of a cement slurry. A neat cement slurry is a combination of only cement and water. The specific weight of a neat cement slurry is defined by the amount of water used with the dry cement. The specific weight range for a particular class of cement is, therefore, limited by the minimum and maximum water-to-cement ratios permissible by API standards.

The minimum amount of water for any class of cement is defined as that amount of water that can be used in the slurry with the dry cement that will

Table 4-154
Properties of the Various Classes of API Cements

Cement Class	A	B	C	D	E	F	G	H
Specific Gravity	3.14	3.14	3.14	3.16	3.16	3.16	3.16	3.16
Surface Area (cm^2/gm)	1500	1600	2200	1200	1200	1200	1400	1200
Weight Per Sack, lb	94	94	94	94	94	94	94	94
Bulk Volume, Ft^3/sk	1	1	1	1	1	1	1	1
Absolute Volume, gal/sk	3.59	3.59	3.55	3.57	3.57	3.57	3.62	3.57

still produce a neat slurry with a consistency that is below 30 Bearden units of consistency.* Note that the minimum water content defined in this manner is much greater than the stoichiometric minimum for cement hydration and setting.

The maximum amount of water for any class of cement is usually defined as the ratio that results in the cement particles remaining in suspension until the initial set of the slurry has taken place. If more than the maximum amount of water is used, then the cement particles will settle in such a manner that there will be pockets of free water within the set cement column.

Table 4-155 gives the API recommended optimum water-to-cement ratios and the resulting neat slurry specific weight (lb/gal) and specific volume, or yield (ft^3/sack) for the various classes of API cements [161].

Table 4-156 gives the maximum and minimum water-to-cement ratios and the resulting neat slurry specific weight and specific volume, or yield for three classes of API cement.

The specific weight of a cement slurry $\bar{\gamma}$ (lb/gal) is

$$\bar{\gamma} = \frac{\text{Cement (lb)} + \text{Water (lb)} + \text{Additive (lb)}}{\text{Cement (gal)} + \text{Water (gal)} + \text{Additive (gal)}} \tag{4-324}$$

The volumes above are absolute volumes. For example, a 94-lb sack of cement contains 1 ft^3 of bulk cement powder, yet the actual or absolute space occupied by the cement particles is only 0.48 ft^3.

For dealing with other powdered materials that are additives to cement slurries, the absolute volume must be used. The absolute volume (gal) is

$$\text{Absolute value (gal)} = \frac{\text{Additive material (lb)}}{(8.34\ \text{lb/gal})(\text{S.G. of material})} \tag{4-325}$$

The volume of slurry that results from 1 sack of cement additives is defined as the yield. The yield (ft^3/sack) is

$$\text{Yield} = \frac{\text{Cement (gal)} + \text{Water (gal)} + \text{Additives (gal)}}{7.48\ \text{gal/ft}^3} \tag{4-326}$$

where the cement, water and additives are those associated with 1 sack of cement, or 94 lb of cement.

*This was formally referred to as poise.

Table 4-155
Properties of Neat Cement Slurries for Various Classes of API Cements

API Class	API Rec'd Water (gal/sk)	API Rec'd Water (gal/sk)	API Rec'd Water (gal/sk)	Percent of Mixing Water
A (Portland)	5.20	15.6	1.18	46
B (Portland)	5.20	15.6	1.18	46
C (High Early)	6.32	14.8	1.32	56
D (Retarded)	4.29	16.46	1.05	38
E (Retarded)	4.29	16.46	1.05	38
F (Retarded)	4.29	16.46	1.05	38
G (Basic)	4.97	15.8	1.15	44
H (Basic)	4.29	16.46	1.05	38

Table 4-156
Maximum and Minimum Water-to-Cement Ratios for API Classes A, C and E Neat Cement Slurries

	Maximum Water			Minimum Water		
API Class	Water (gal/sk)	Weight (lb/gal)	Volume (ft^3/sk)	Water (gal/sk)	Weight (lb/gal)	Volume (ft^3/sk)
A	5.5	15.39	1.22	3.90	16.89	1.00
C	7.9	13.92	1.53	6.32	14.80	1.32
E	4.4	16.36	1.07	3.15	17.84	0.90

Example 1

Calculate the specific weight and yield for a neat slurry of Class A cement using the maximum permissible water-to-cement ratio.

Table 4-156 gives the maximum water-to-cement ratio for Class A cement as 5.5 gal/sack. Thus using the absolute volume for Class A cement from Table 4-154 as 3.59 gal/sack, Equation 4-324 is

$$\bar{\gamma} = \frac{94 + 5.5(8.34)}{3.59 + 5.5}$$

$$= 15.39 \text{ lb/gal}$$

The yield determined from Equation 4-326 is

$$\text{Yield} = \frac{3.59 + 5.5}{7.48}$$

$$= 1.22 \text{ ft}^3/\text{sack}$$

It is often necessary to decrease the specific weight of a cement slurry to avoid fracturing weak formations during cementing operations. There are basically two methods for accomplishing lower specific weights. These are

1. Adding clay or chemical silicate type extenders together with their required extra water.
2. Adding large quantities of pozzolan, ceramic microspheres or nitrogen. These materials lighten the slurry because they have lower specific gravities than the cement.

When using the first method above great care must be taken not to use too much water. There is a maximum permissible water-to-cement ratio for each cement class. This amount of water can be used with the appropriate extra water required for the added clay or chemical silicate material. Using too much water will result in a very poor cement operation.

It also may be necessary to increase the specific weight of a cement slurry, particularly when cementing through high-pressure formations. There are basically two methods for accomplishing higher specific weights. These are

1. Using the minimum permissible water-to-cement ratio for the particular cement class and adding dispersants to increase the fluidity of the slurry.
2. Adding high-specific-gravity materials to the slurry together with optimal or slightly reduced (but not necessarily the minimum) water-to-cement ratio for the particular cement class.

The first method above is usually restricted to setting plugs in wells since it results in high strength cement that is rather difficult to pump. The second method is used for primary cementing, but these slurries are difficult to design since the settling velocity of the high-specific-gravity additive must be taken into consideration.

Thickening Time

It is important that the thickening time for a given cement slurry be known prior to using the slurry in a cementing operation. When water is added to dry cement and its additives, a chemical reaction begins that results in an increase in slurry viscosity. This viscosity increases over time, which will vary in accordance with the class of cement used, the additives placed in the dry cement prior to mixing with water and the temperature and confining pressure in the location where the cement slurry is placed. When the viscosity becomes too large, the slurry is no longer pumpable. Thus, if the slurry has not been placed in its proper location within the well prior to the cement slurry becoming unpumpable, the well and the surface equipment would be seriously damaged.

Thickening time T_t (hr) is defined as the time required for the cement slurry to reach the limit of 100 Bearden units of consistency.* This thickening time must be considerably longer than the time necessary to carry out the actual cementing operation. This can be accomplished by choosing the class of cement that has a sufficiently long thickening time, or placing the appropriate additives in the slurry that will retard the slurry chemical reaction and lengthen the thickening time.

*70 Bearden units of consistency is considered to be maximum pumpable viscosity.

It is necessary that an accurate estimate of the total time be for the actual operation. A safety factor should be added to this estimate. Usually this safety factor is from 30 min (for shallow operations) to as long as 2 hr (for deep complex operations).

The cementing operation time T_o (hr) is the time required for the cement slurry to be placed in the well:

$$T_o = T_m + T_d + T_p + T_s \tag{4-327}$$

where T_m = time required to mix the dry cement (and additives) with water in hr
T_d = the time required to displace the cement slurry (that was pumped to the well as mixing took place) by mud or water from inside the casing in hr
T_p = plug release time in hr
T_s = the safety factor of 0.5 to 2 hr

The mixing time T_m is

$$T_m = \frac{\text{Volume of dry cement}}{\text{Mixing rate}}$$

$$= \frac{V_c\ (\text{sacks})}{\text{Mixing rate (sack/min)(60)}} \tag{4-328}$$

where V_c is the dry volume of cement (sacks).

The displacement time T_d is

$$T_d = \frac{\text{Volume of fluid required to displace top plug}}{\text{Displacement rate}}$$

$$= \frac{\text{Volume (ft}^3\text{)}}{\text{Displacement rate (ft}^3\text{/min)(60)}} \tag{4-329}$$

The cement slurry chosen must have a thickening time that is greater than the estimated time obtained for the actual cementing operation using Equation 4-327. Thus, $T_t > T_o$.

Figure 4-382 gives a relationship between well depth and cementing operation time and the specifications for the various cement classes. This figure can be used to approximate cementing operation time.

The cement slurry thickening time can be increased or decreased by adding special chemicals to the dry cement prior to mixing with water. Retarders are added to increase the thickening time and thus increase the time when the cement slurry sets. Some common retarders are organic compounds such as lignosulphonate, cellulose derivatives and sugar derivatives. Accelerators are added to decrease the thickening time and thus decrease the time when the cement slurry sets. Accelerators are often used when it is required to have the cement obtain an early compressive strength (usually of the order of 500 psi). Early setting cement slurries are used to cement surface casing strings or directional drilling plugs where waiting-on-cementing (WOC) must be kept to a

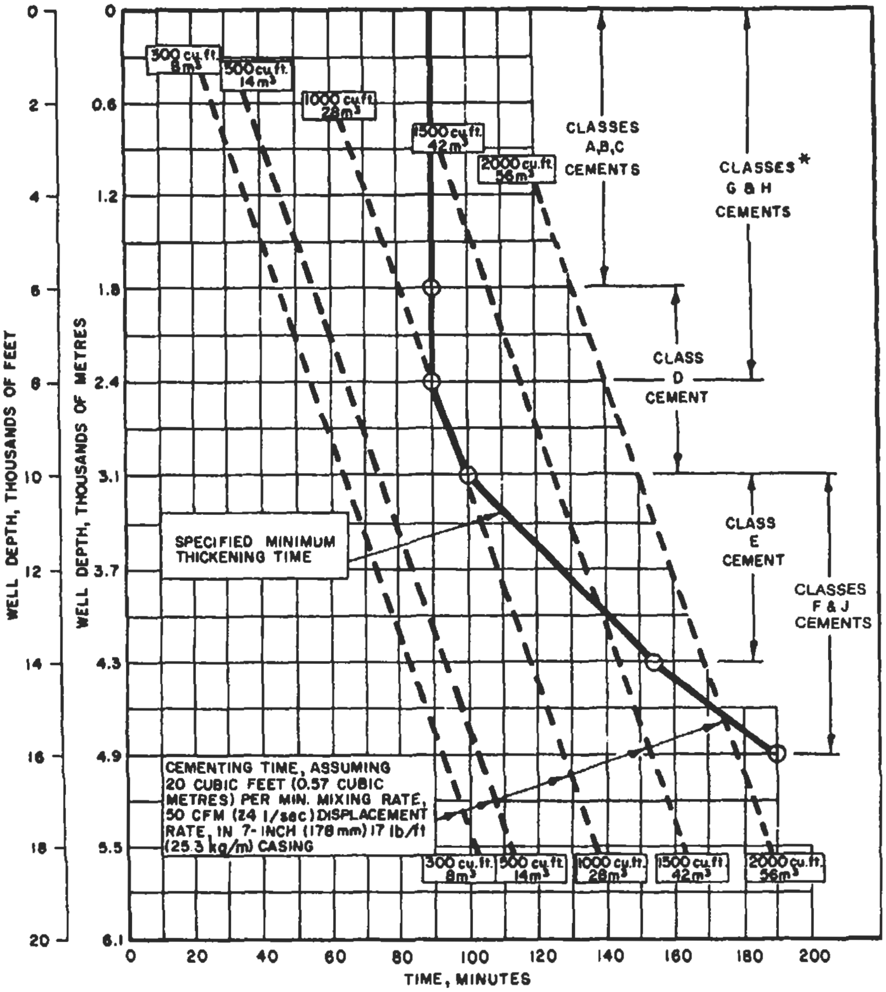


Figure 4-382. Well depth and cementing time relationships [164].
*Specified maximum thickening time—120 minutes.

minimum. The most common accelerators are calcium chloride and to a lesser extent sodium chloride.

In general the thickening time for all neat cement slurries of all classes of cement will decrease significantly with increasing well environment temperature and/or confining pressure. Thus it is very important that the well extreme temperatures and confining pressures be defined for a particular cementing operation before the cement slurry is designed. Once the well temperature and pressure conditions are known, either from offset wells or from actual well logs during the drilling operation, and the estimated cementing operation time is known, the cement slurry with the required thickening time can be designed. After the initial cement slurry design has been made it is usually necessary to carry out laboratory tests to verify that the actual cement batch mix used will give the required thickening time (and other characteristics needed). Such tests are usually carried out by the cementing service company laboratory using the operator's specifications.

Example 2

Calculate the minimum thickening time required for a primary cementing operation to cement a long intermediate casing string. The intermediate casing string is a $9\frac{5}{8}$-in., 53.5-lb/ft casing set in a $12\frac{1}{4}$-in. hole. The string is 12,000 ft in length from the top of the float collar to the surface. The cementing operation will require 1,200 sacks of Class H cement. The single cementing truck has a mixing capacity of 25 sacks per minute. The rig duplex mud pump has an 18-in. stroke (2.5 in. rod) and $6\frac{1}{2}$-in. liners and will be operated at 50 strokes per minute with a 90% volumetric efficiency. The plug release time is estimated to be about 15 min.

The mixing time is obtained from Equation 4-328. This is

$$T_m = \frac{1,200}{25(60)}$$

$$= 0.80 \text{ hr}$$

The inside diameter of $9\frac{5}{8}$-in., 53.50-lb/ft casing is 8.535 in. (see Section titled "Fishing Operations"). The internal capacity per unit length of casing is

$$\text{Casing internal capacity} = \frac{\pi}{4}\frac{(8.535)^2}{12}$$

$$= 0.3973 \text{ ft}^3/\text{ft}$$

Thus the total internal capacity of the casing is

$$\text{Total volume} = 12,000(0.3973)$$

$$= 4768 \text{ ft}^3$$

The volume capacity of the mud pump per stroke q (ft^3/stroke) (see section titled "Mud Pumps") is

$$q = \left\{2\left[\frac{\pi}{4}(6.5)^2\right](18.0)\right.$$

$$\left. + 2\left[\frac{\pi}{4}(6.5)^2 - \frac{\pi}{4}(2.5^2)\right](18.0)\right\}\frac{(0.90)}{(12)^3}$$

$$= 1.1523 \text{ ft}^3/\text{stroke}$$

The displacement time is obtained from Equation 4-329. This is

$$T_d = \frac{4,768}{50(1.1523)(60)}$$

$$= 1.38 \text{ hr}$$

The plug release time is

$$T_p = \frac{15}{60}$$

$$= 0.25 \text{ hr}$$

Using a safety factor of 1 hr, the cementing operation time is obtained from Equation 4-327. This is

$$T_o = 0.80 + 1.38 + 0.25 + 1.00$$

$$= 3.43 \text{ hr}$$

Thus the minimum thickening time for the cement slurry to be used this cementing operation is

$$T_t = 3.43 \text{ hr}$$

or 3 hr and 26 min.

Strength of Set Cement

A properly designed cement slurry will set after it has been placed in its appropriate location within the well. Cement strength is the strength the set cement has obtained. This usually refers to compressive strength, but can also refer to tensile strength. Cement having a compressive strength of 500 psi is considered adequate for most well operations.

The compressive strength of set cement is dependent upon the water-to-cement ratio used in the slurry, curing time, the temperature during curing, the confining pressure during curing and the additives in the cement. As part of the cement design procedures, samples of the cement slurry to be used in the cementing operation are cured and compression strength tested and often shear bond strength tested. These tests are usually carried out by the cementing company laboratory at the request of the operator.

In the compression test, four or five sample cubes of the slurry are allowed to cure for a specified period of time. The cement cubes are placed in a compression testing machine and the compressive strength of each sample cube obtained experimentally. The average value of the samples is obtained and reported as the compressive strength of the set cement.

In the shear bond strength test, the cement slurry is allowed to cure in the annulus of two concentric steel cylinders. After curing the force to break the bond between the set cement and one of the cylinders (usually the inner one) is obtained experimentally. The shear force, F_s (lb), which can be supported on the inner cylinder, or by the casing, is [163]

$$F_s = 0.969\ \sigma_c dl \tag{4-330}$$

where σ_c = compressive strength of cement in psi
d = outside of the casing in in.
l = length of the cement column in in.

Table 4-157 shows the influence of curing time, temperature and confining pressure on the compressive strength of Class H cement [162]. Also shown is the effect of the calcium chloride accelerator. In general, for the ranges of temperature and other values considered, the compressive strength increases with increasing curing time, temperature, confining pressure and the amount of calcium chloride accelerator. In general, the other classes of cement follow the same trend in compressive strength versus curing time, temperature and confining pressure.

At curing temperatures of about 200°F or greater, the compressive strength of nearly all classes of cement cured under pressure will decrease. This decrease in compressive strength is often denoted as strength retrogression. An increase in cement permeability accompanies this decrease in strength. This strength retrogression usually continues for up to 15 days and, thereafter, the strength level will remain constant. This strength retrogression problem can be solved by adding from 30 to 50% (by weight) of silica flour or silica sand to the cement slurry. This silica additive prevents the strength retrogression and the corresponding increase in permeability of the cement.

Table 4-158 shows the effect of silica flour on the compressive strength of Class G cement cured at 700°F.

Example 3

For Example 2, determine the total weight that can be supported by the set cement bonded to the 9 $\frac{5}{8}$-in. casing. Assume the cement slurry yield is 1.05 ft^3/sack

Table 4-157
Influence of Time and Temperature on the Compressive Strength of API Class H Cement [162]

Curing Time (hours)	Calcium Chloride (percent)	Compressive Strength (psi) at Curing Temperature and Pressure of			
		95°F 800 psi	110°F 1,600 psi	140°F 3,000 psi	170°F 3,000 psi
6	0	100	350	1,270	1,950
8		500	1,200	2,500	4,000
12		1,090	1,980	3,125	4,700
24		3,000	4,050	5,500	6,700
6	1	900	1,460	2,320	2,500
8		1,600	1,950	2,900	4,100
12		2,200	2,970	3,440	4,450
24		4,100	5,100	6,500	7,000
6	2	1,100	1,700	2,650	2,990
8		1,850	2,600	3,600	4,370
12		2,420	3,380	3,900	5,530
24		4,700	5,600	6,850	7,400

Table 4-158
Effect of Silica Flour on the Compressive Strength of Class G Cement Cured at Pressure

Curing Time (days)	Silica Flour (percent)	Compressive Strength (psi), 1 Day at 130°F, All Remaining Days at 700°F
1	0	3525
2		988
4		1012
8		1000
1	40	2670
2		3612
4		3188
8		4588

Source: Courtesy B. J. Hughes

(see Table 4-155) and there are 120 ft of casing below the float collar. A compressive strength of 500 psi is to be used.

The total volume of the cement slurry to be pumped to the well is

$$\text{Volume of slurry} = 1{,}200(1.05)$$

$$= 1{,}260 \text{ ft}^3$$

The volume of slurry that is pumped to the annulus between the open borehole and the outside of the casing is

$$\text{Volume of annulus} = 1{,}260 - (0.3973)(120)$$

$$= 1{,}212 \text{ ft}^3$$

The height in the annulus to where the casing is cemented to the borehole well and casing is

$$h = \frac{1{,}212}{\pi/4\left[(12.5/12)^2 - (9.625/12)^2\right]}$$

$$= 3{,}493 \text{ ft}$$

The total weight that can be supported is calculated from Equation 4-330. This is

$$F_s = 0.969(500)(9.625)(3{,}493)(12)$$

$$= 195.5 \times 10^6 \text{ lb}$$

Cement Additives

There are many chemical additives that can be used to alter the basic properties of the neat cement slurry and its resulting set cement. These additives are to alter the cement so that it is more appropriate to the surface cementing equipment and the subsurface environment.

Additives can be subdivided into six functional groups that are [167,168]:

1. specific weight control
2. thickening and setting time control
3. loss of circulation control
4. filtration control
5. viscosity control
6. special problems control

Specific Weight Control

As in drilling mud design, the cement slurry must have a specific weight that is high enough to prevent high pore pressure formation fluids from flowing into the well. But also the specific weight must not be so high as to cause fracturing of the weaker exposed formations. In general, the specific weight of the neat cement slurry of any of the various API classes of cement are so high that most exposed formations will fracture. Thus, it is necessary to lower the specific weight of nearly all cement slurries. The slurry specific weight can be reduced by using a higher water-to-cement ratio. But this can only be accomplished within the limits for maximum and minimum water-to-cement ratios set by API standards (see Table 4-156). The reduction in specific weight is normally accomplished by adding low-specific-gravity solids to the slurry. Table 4-159 gives the specific gravity properties of a number of the low-specific-gravity solids that are used to reduce the specific weight of cement slurries.

The most common low-specific-gravity solids used to reduce cement slurry specific weight are bentonite, diatomaceous earth, solid hydrocarbons, expanded perlite and pozzolan. It may not be possible to reduce the cement slurry specific weight enough with the above low-specific-weight materials when very weak formations are exposed. In such cases nitrogen is used to aerate the mud column above the cement slurry to assist in further decreasing the hydrostatic pressure.

Nearly all materials that are added to a cement slurry require the addition of additional water to the slurry. Table 4-160 gives the additional requirements for the various cement additives [167].

Bentonite. Bentonite without an organic polymer can be used as an additive to cement slurries. The addition of bentonite requires the use of additional water, thereby, further reducing the specific weight of the slurry (Table 4-160). Bentonite is usually dry blended with the dry cement prior to mixing with water. High percentages of bentonite in cement will significantly reduce compressive strength and thickening time. Also, high percentages of bentonite increase permeability and lower the resistance of cement to sulfate attack. At temperatures above 200°F, the bentonite additive promotes retrogression of strength in cements with time. Bentonite has been used in 25% by weight of cement. Such high concentrations are not recommended. In general, the bentonite additive makes a poor well cement.

Table 4-159
Physical Properties of Cementing Materials

Material	Bulk Weight (lbm/cu ft)	Specific Gravity	Weight 3.6[a] Absolute Gal	Absolute Volume gal/lbm	Absolute Volume cu ft/lbm
API cements	94	3.14	94	0.0382	0.0051
Ciment Fondu	90	3.23	97	0.0371	0.0050
Lumnite cement	90	3.20	96	0.0375	0.0050
Trinity Lite-Wate	75	2.80	75.0[e]	0.0429	0.0057
Activated charcoal	14	1.57	47.1	0.0765	0.0102
Barite	135	4.23	126.9	0.0284	0.0038
Bentonite (gel)	60	2.65	79.5	0.0453	0.0060
Calcium chloride, flake[b]	56.4	1.96	58.8	0.0612	0.0082
Calcium chloride, powder[b]	50.5	1.96	58.8	0.0612	0.0082
Cal-Seal, gypsum cement	75	2.70	81.0	0.0444	0.0059
CFR-1[b]	40.3	1.63	48.9	0.0736	0.0098
CFR-2[b]	43.0	1.30	39.0	0.0688	0.0092
DETA (liquid)	59.5	0.95	28.5	0.1258	0.0168
Diacel A[b]	60.3	2.62	78.6	0.0458	0.0061
Diacel D	16.7	2.10	63.0	0.0572	0.0076
Diacel LWL[b]	29.0	1.36	40.8	0.0882	0.0118
Diesel Oil No. 1 (liquid)	51.1	0.82	24.7	0.1457	0.0195
Diesel Oil No. 2 (liquid)	53.0	0.85	25.5	0.1411	0.0188
Gilsonite	50	1.07	32	0.1122	0.0150
HALDAD®-9[b]	37.2	1.22	36.6	0.0984	0.0131
HALDAD®-14[b]	39.5	1.31	39.3	0.0916	0.0122
Hematite	193	5.02	150.5	0.0239	0.0032
HR-4[b]	35	1.56	46.8	0.0760	0.0103
HR-7[b]	30	1.30	39	0.0923	0.0123
HR-12[b]	23.2	1.22	36.6	0.0984	0.0131
HR-L (liquid)[b]	76.6	1.23	36.9	0.0976	0.0130
Hydrated lime	31	2.20	66	0.0545	0.0073
Hydromite	68	2.15	64.5	0.0538	0.0072
LA-2 Latex (liquid)	68.5	1.10	33	0.1087	0.0145
LAP-1 Latex[b]	50	1.25	37.5	0.0960	0.0128
LR-11 Resin (liquid)	79.1	1.27	38.1	0.0945	0.0126
NF-1 (liquid)[b]	61.1	0.98	29.4	0.1225	0.0164
NF-P[b]	40	1.30	39.0	0.0923	0.0123
Perlite regular	8[c]	2.20	66.0	0.0546	0.0073
Perlite Six	38[d]	—	—	0.0499	0.0067
Pozmix® A	74	2.46	74	0.0487	0.0065
Pozmix® D	47	2.50	73.6	0.0489	0.0065
Salt (dry NaCl)	71	2.17	65.1	0.0553	0.0074
Salt (in solution at 77°F with fresh water)					
6%, 0.5 lbm/gal	—	—	—	0.0384	0.0051
12%, 1.0 lbm/gal	—	—	—	0.0399	0.0053
18%, 1.5 lbm/gal	—	—	—	0.0412	0.0055
24%, 2.0 lbm/gal	—	—	—	0.0424	0.0057
Saturated, 3.1 lbm/gal	—	—	—	0.0445	0.0059
Salt (in solution at 140°F with fresh water) saturated, 3.1 lbm/gal	—	—	—	0.0458	0.0061
Sand (Ottawa)	100	2.63	78.9	0.0456	0.0061
Silica flour (SSA-1)	70	2.63	78.9	0.0456	0.0061
Coarse silica (SSA-2)	100	2.63	78.9	0.0456	0.0061
Tuf Additive No. 1	—	1.23	36.9	0.0976	0.0130
Tuf-Plug	48	1.28	38.4	0.0938	0.0125
Water	62.4	1.00	30.0	0.1200	0.0160

[a] Equivalent to one 94-lbm sack of cement in volume.
[b] When less than 5% is used, these chemicals may be omitted from calculations without significant error.
[c] For 8 lbm of Perlite regular use a volume of 1.43 gal at zero pressure.
[d] For 38 lbm of Perlite Six use a volume of 2.89 gal at zero pressure.
[e] 75 lbm = 3.22 absolute gal.

Source: Courtesy Halliburton Services

Table 4-160
Water Requirement of Cementing Materials

Material	Water Requirements
API Class A and B cements	5.2 gal (0.70 cu ft)/94-lbm sack
API Class C cement (Hi Early)	6.3 gal (0.84 cu ft)/94-lbm sack
API Class D and E cements (retarded)	4.3 gal (0.58 cu ft)/94-lbm sack
API Class G cement	5.0 gal (0.67 cu ft)/94-lbm sack
API Class H cement	4.3 to 5.2 gal/94-lbm sack
Chem Comp cement	6.3 gal (0.84 cu ft)/94-lbm sack
Ciment Fondu	4.5 gal (0.60 cu ft)/94-lbm sack
Lumnite cement	4.5 gal (0.60 cu ft)/94-lbm sack
HLC	7.7 to 10.9 gal/87-lbm sack
Trinity Lite-Wate cement	7.7 gal (1.03 cu ft)/75-lbm sack (maximum)
Activated charcoal	none at 1 lbm/sack of cement
Barite	2.4 gal (0.32 cu ft)/100-lbm sack
Bentonite (gel)	1.3 gal (0.174 cu ft)/2% in cement
Calcium chloride	none
Gypsum hemihydrate	4.8 gal (0.64 cu ft)/100-lbm sack
CFR-1	none
CFR-2	none
Diacel A	none
Diacel D	3.3 to 7.2 gal/10% in cement (see Lt Wt. Cement)
Diacel LWL	none (up to 0.7%) 0.8 to 1.0 gal/1% in cement (except gel or Diacel D slurries)
Gilsonite	2.0 gal (0.267 cu ft)/50 lbm/cu ft
HALAD -9	none (up to 0.5%) 0.4 to 0.5 gal/sack of cement at over 0.5%
HALAD -14	none
Hematite	0.36 gal (0.048 cu ft)/100-lbm sack
HR-4	none
HR-7	none
HR-12	none
HR-20	none
Hydrated lime	0.153 gal (0.020 cu ft)/lbm
Hydromite	3.0 gal (0.40 cu ft)/100-lbm sack
LA-2 Latex	0 to 0.8 gal/sack of cement
LAP-1 powdered latex	1.7 gal (0.227 cu ft)/1% in cement
NF-P	none
Perlite regular	4.0 gal (0.535 cu ft)/8 lbm/cu ft
Perlite Six	6.0 gal (0.80 cu ft)/38 lbm/cu ft
Pozmix A	3.6 gal (0.48 cu ft)/74 lbm/cu ft
Salt (NaCl)	none
Sand, Ottawa	none
Silica flour (SSA-1)	1.5 gal (0.20 cu ft)/35% in cement (32.9 lbm)
Coarse silica (SSA-2)	none
Tuf Additive No. 1	none
Tuf Plug	none

Source: Courtesy Halliburton Services

Diatomaceous Earth. Diatomaceous earth has a lower specific gravity than bentonite. Like bentonite this additive also requires additional water to be added to the slurry. This additive will affect the slurry properties similar to the addition of bentonite. However, it will not increase the viscosity as bentonite will do. Diatomaceous earth concentrations as high as 40% by weight of cement have been used. This additive is more expensive than bentonite.

Solid Hydrocarbons. Gilsonite (an asphaltite) and coal are used as very-low-specific-gravity solids additives. These additives do not require a great deal of water to be added to the slurry when they are used.

Expanded Perlite. Expanded perlite requires a great deal of water to be added to the slurry when it is used to reduce the specific weight of a slurry. Often perlite as an additive is used in a blend of additives such as perlite with volcanic glass fines, or with pozzolanic materials, or with bentonite. Without bentonite the perlite tends to separate and float in the upper part of the slurry.

Pozzolan. Diatomaceous earth is a type of pozzolan. Pozzolan refers to a finely ground pumice or fly ash that is marketed as a cement additive under that name. The specific gravity of pozzolans is slightly less than the specific gravity of cement. The water requirements for this additive are about the same as for cements. Only a slight reduction in specific weight of a slurry can be realized by using these additives. The cost of pozzolans is very low.

Where very high formations pore pressure are present the specific weight of the cement slurry can be increased by using the minimum water-to-cement ratio and/or by adding high-specific-gravity materials to the slurry. The most common high-specific-gravity solids used to increase cement slurry specific weight are hematite, ilmenite, barite and sand. Table 4-159 gives the specific gravity properties of a number of these high-specific-gravity additives.

Hematite. This additive can be used to increase the specific weight of a cement slurry to as high as 19 lb/gal. This is an iron oxide ore with a specific gravity of about 5.02. Hematite requires the addition of some water when it is used as an additive. Hematite has minimal effect on thickening time and compressive strength of the cement.

Ilmenite. This additive has a specific gravity of about 4.67. It is a mineral composed of iron, titanium and oxygen. It requires no additional water to be added to the slurry; thus, it can yield slurry specific weights as high as the hematite additive. Ilmenite also has mineral effect on thickening time and compressive strength of the cement.

Barite. This mineral additive requires much more water to be added to the cement slurry than does hematite. This large amount of added water required will decrease the compressive strength of the cement. This additive can be used to increase the specific weight of a cement slurry to as high as 19 lb/gal.

Sand. Ottawa sand has a low specific gravity of about 2.63. But since no additional water is required when using this additive, it is possible to use sand to increase the cement slurry specific weight. The sand has little effect on the pumpability of the cement slurry. When set the cement will form a very hard surface. Sand used as an additive can be used to increase the specific weight of a cement slurry to as high as 18 lb/gal.

Example 4

A specific weight of 12.8 lb/gal is required for a Class A cement slurry. It is decided that the cement be mixed with bentonite to reduce the specific weight of the slurry. Determine the weight of bentonite that should be dry blended with each sack of cement. Determine the yield of the cement slurry. Determine the volume (gal) of water needed for each sack of cement.

The weight of bentonite to be blended is found by using Equation 4-324. Taking the appropriate data from Tables 4-154, 4-155, 4-159 and 4-160, and letting x be the unknown weight of bentonite per sack of cement, Equation 4-324 is

$$12.8 = \frac{94 + x + 8.34(5.20 + 0.692x)}{\left(\frac{94}{3.14(8.34)} + \frac{x}{2.65(8.34)} + 5.20 + 0.692x\right)}$$

Solving the above for x gives

$$x = 9.35 \text{ lb}$$

Thus, 9.35 lb of bentonite will need to be added for each sack of Class A cement used.

The yield can be determined from Equation 4-326. This is

$$\text{Yield} = \frac{\frac{94}{3.14(8.34)} + \frac{9.35}{2.65(8.34)} + 5.20 + 0.692(9.35)}{7.48} = 2.10 \text{ ft}^3/\text{sack}$$

The volume of water needed is

$$\text{Volume of water} = 5.20 + 0.692(9.35)$$

$$= 11.69 \text{ gal/sack}$$

Example 5

A specific weight of 18.2 lb/gal is required for a Class H cement slurry. It is decided that the cement be mixed with hematite to increase the specific weight of the slurry. Determine the weight of hematite that should be dry blended with each sack of cement. Determine the yield of the cement slurry. Determine the volume (gal) of water needed for each sack of cement.

The weight of hematite to be blended is found by using Equation 4-324. Taking the appropriate data from Tables 4-154, 4-155, 4-159 and 4-160, and letting x be the unknown weight of hematite per sack of cement, Equation 4-324 is

$$18.2 = \frac{94 + x + 8.34(4.29 + 0.0036x)}{\frac{94}{3.14(8.34)} + \frac{x}{5.02(8.34)} + (4.29 + 0.0036x)}$$

Solving the above for x gives

$$x = 25.8 \text{ lb}$$

Thus, 25.8 lb of hematite will need to be added for each sack of Class H cement used.

The yield can be determined from Equation 4-326. This is

$$\text{Yield} = \frac{\dfrac{94}{3.14(8.34)} + \dfrac{25.8}{5.07(8.34)} + 4.29 + 0.0036(25.8)}{7.48} = 1.15\,\text{ft}^3/\text{sack}$$

The volume of water needed is

$$\text{Volume of water} = 4.29 + 0.0036(25.8)$$

$$= 4.38 \text{ gal/sack}$$

Example 6

A specific weight of 17.1 lb/gal is required for a Class H cement slurry. It is decided that the cement be mixed with sand in order to increase the specific weight of the slurry. Determine the weight of sand that should be added with each sack of cement. Determine the yield of the cement slurry. Determine the volume (gal) of water needed for each sack of cement.

The weight of sand to be added is found by using Equation 4-324. Taking the appropriate data from Tables 4-154, 4-155, 4-159 and 4-160, and letting x be the unknown weight of sand per sack of cement, Equation 4-324 is

$$17.1 = \frac{94 + x + 8.34(4.29)}{\dfrac{94}{3.14(8.34)} + \dfrac{x}{2.63(8.34)} + 4.29}$$

Solving the above for x gives

$$x = 31.7 \text{ lb}$$

Thus, 31.7 lb of sand will need to be added for each sack of Class H cement used.

The yield can be determined from Equation 4-326. This is

$$\text{Yield} = \frac{\dfrac{94}{3.14(8.34)} + \dfrac{31.7}{2.63(8.34)} + 4.29}{7.48} = 1.25\,\text{ft}^3/\text{sack}$$

Since no additional water is needed when using sand as the additive, the volume of water needed is

$$\text{Volume of water} = 4.29 \text{ gal/sack}$$

Thickening Setting Time Control

It is often necessary to either accelerate, or retard the thickening and setting time of a cement slurry.

For example, when cementing a casing string run to shallow depth or when setting a directional drilling kick-off plug, it is necessary to accelerate the cement hydration so that the waiting period will be minimized. The most commonly used cement hydration accelerators are calcium chloride, sodium chloride, a hemihydrate form of gypsum and sodium silicate.

Calcium Chloride. This accelerator may be used in concentrations up to 4% (by weight of mixing water) in wells having bottomhole temperatures less than 125°F. This additive is usually available in an anhydrous grade (96% calcium chloride). Under pressure curing conditions calcium chloride tends to improve compressive strength and significantly reduce thickening and setting time.

Sodium Chloride. This additive will act as an accelerator when used in cements containing no bentonite and when in concentrations of about 5% (by weight of mixing water). In concentrations above 5% the effectiveness of sodium chloride as an accelerator is reduced. This additive should be used in wells with bottomhole temperatures less than 160°F. Saturated sodium chloride solutions act as a retarder. Such saturated sodium chloride solutions are used with cement slurries that are to be used to cement through formations that are sensitive to freshwater. However, potassium chloride is far more effective in inhibiting shale hydration. In general, up to a 5% concentration, sodium chloride will improve compressive strength while reducing thickening and setting time.

Gypsum. Special grades of gypsum hemihydrate cement are blended with Portland cement to produce a cement with reduced thickening and setting time for low-temperature applications. Such cement blends should be used in wells with bottomhole temperatures less than 140°F (regular-temperature grade) or 180°F (high-temperature grade). There is a significant additional water requirement for the addition of gypsum (see Table 4-160). When very rapid thickening and setting times are required for low-temperature conditions (i.e., primary cementing of a shallow casing string or a shallow kick-off plug), a special blend is sometimes used. This is 90 lb of gypsum hemihydrate, 10 lb of Class A Portland cement and 2 lb of sodium chloride mixed with 4.8 gal of water. Such a cement slurry can develop a compression strength of about 1,000 psi in just 30 min (at 50°).

Sodium Silicate. When diatomaceous earth is used with the cement slurry, sodium silicate is used as the accelerator. It can be used in concentrations up to 7% by weight.

When it is necessary to cement casing or liner strings set at great depths, additives are often used in the design of the cement slurry to retard the thickening and setting time. Usually such retarding additives are organic compounds. These materials are also referred to as thinners or dispersants. Calcium lignosulfonate is one of the most commonly used cement retarders. It is very effective at increasing thickening and setting time in cement slurries at very low concentrations (of the order of 1% by weight or less). It is necessary to add an organic acid to the calcium lignosulfonate when high-temperature conditions are encountered.

Calcium-sodium lignosulfonate is a better retarding additive when high concentrations of bentonite are to be used in the design of the cement slurry.

Also sodium tetraborate (borax) and carborymethyl hydroxyethyl cellulose are used as retarding additives.

Filtration Control

Filtration control additives are added to cement slurries for the same reason that they are added to drilling muds. However, untreated cement slurries have much greater filtration rates than untreated drilling muds. It is thus important to limit the loss of water filtrate from a slurry to a permeable formation. This is necessary for several reasons:

- minimize hydration of formations containing water sensitive formations;
- limit the increase in slurry viscosity as cement is placed in the well;
- prevent annular bridges;
- allow for sufficient water to be available for cement hydration.

Examples of filtration control additives are latex, bentonite with a dispersant and other various organic compounds and polymers.

Viscosity Control

The viscosity of the cement slurry will affect the pumping requirements for the slurry and frictional pressure gradient within the well. The viscosity must be kept low enough to ensure that the cement slurry can be pumped to the well during the entire cementing operation period. High viscosity will result in high-pressure gradients that could allow formation fractures.

Examples of commonly used viscosity control additives are calcium lignosulfonate, sodium chloride and some long-chain polymers. These additives also act as accelerators or retarders so care must be taken in designing the cement slurry with these materials.

Special Problems Control

There are some special problems in the design of cement slurries for which additives have been developed. These are

- Gel strength additives for the preparation of spacers (usually fragile gel additives).
- Silica flour is used to form a stronger and less permeable cement, especially for high-temperature applications.
- Hydrazine is used to control corrosion of the casing.
- Radioactive tracers are used to assist in assessing where the cement has been placed.
- Gas-bubble-producing compounds to slowly create gas bubbles in the cement as the slurry sets and hardens. It is felt that such a cement will have less tendency to leak formation gas.
- Paraformaldehyde and sodium chromate that counteract organic contaminants left in the well from drilling operations.
- Fibrous materials such as nylon are added to increase strength, in particular, resistance to impact loads.

Primary Cementing

Primary cementing refers to the cementing of casing and liner strings in a well. The cementing of casing or liner string is carried out so that producible oil and gas, or saltwater will not escape from the producing formation to another formation, or pollute freshwater sands at shallower depth.

The running of long casing strings and liner strings to great depths and successfully cementing these strings required careful engineering design and planning.

Normal Single-Stage Casing Cementing

Under good rig operating conditions, casing can be run into the hole at the rate of from 1,000 to 2,000 ft/hr. It is often necessary to circulate the drilling mud in the hole prior to running the casing string. This is to assure that all the cutting and any borehole wall prices have been removed. However, the longer a borehole remains open, the more problems will occur with the well.

Prior to running the casing string into the well the mill varnish should be removed from the outer surface of the casing. The removal of the mill varnish is necessary to ensure that the cement will bond to the steel surface.

The casing string is run into the well with a guide shoe on the bottom of the string. Figures 4-383 and 4-384 show two typical guide shoes. Figure 4-383 is the regular pattern guide shoe that has no drillable internal materials. Figure 4-384 is the swirl guide shoe with drillable aluminum internal materials.

About one or two joints above the guide shoe is the float collar. The float collar acts as a back-flow valve that keeps the heavier cement slurry from flowing back into the casing string after it has been placed in the annulus between the outside of the casing and the borehole wall. Figures 4-385 and 4-386 show two typical float collars. Figure 4-385 is a ball float collar and Figure 4-386 is a flapper float collar. Both float collars are constructed with drillable aluminum internal parts (larger float collars have a combination of cement and aluminum internal parts).

Along the length of the casing string are placed scratchers and centralizers. Figure 4-387 shows the typical scratcher. Figure 4-388 shows the typical centralizer. The scratchers help remove the mud cake as the casing string is lowered into the well. The centralizers ensure that the casing is nearly centered in the borehole, thus allowing a more uniform distribution of cement slurry flow around the casing. This nearly uniform flow around the casing is necessary to remove the drilling mud in the annulus.

The casing string is lowered into the drilling mud in the well using the rig drawworks and elevators. The displaced drilling mud flows to the mud tanks

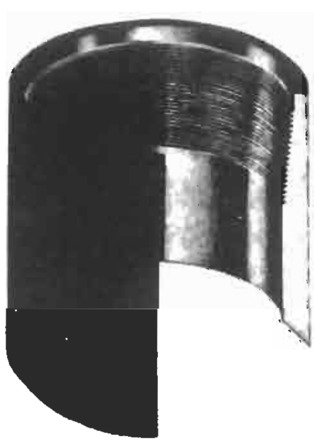


Figure 4-383. Regular pattern guide shoe [161].

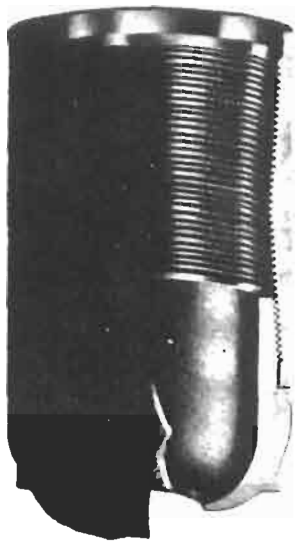


Figure 4-384. Swirl guide shoe [161].

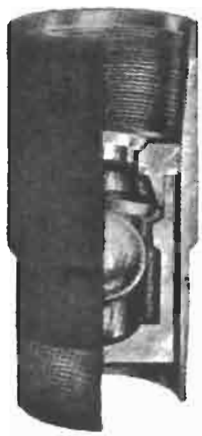


Figure 4-385. Ball float collar [161].

and is stored there for later use. Once the entire casing string is in place in the borehole, the casing string is left hanging in the elevators through the cementing operation. This allows the casing string to be reciprocated (moved up and down) and possibly rotated as the cement is placed in the annulus. This movement assists the removal of the drilling mud.

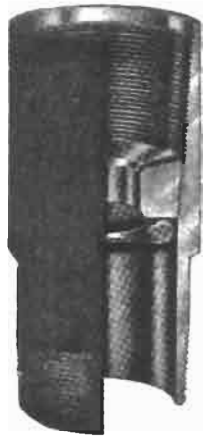


Figure 4-386. Flapper float collar [161].

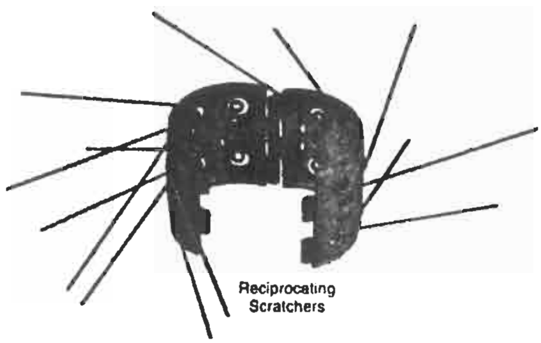


Figure 4-387. Scratcher [161].

While the casing string is hanging in the elevators a cementing head is made up to the upper end of the string (see Figure 4-381). The cementing head is then connected with flow liners that come from pump truck (see Figure 4-380). The blender mixes the dry cement and additives with water. The high-pressure, low-volume triplex cement pump on the pump truck pumps the cement slurry to the cementing head. Usually a preflush or spacer is initially pumped ahead of the cement slurry. This spacer (usually about 15 to 25 bbl) is used to assist in removing the drilling mud from the annular space between the outside of the casing and the borehole wall.

Figure 4-389 gives a series of schematics that show how the spacer and cement slurry displace drilling mud in the well. Two wiper plugs are usually used to separate the spacer and the cement slurry from the drilling mud in the well. The cementing head has two retainer valves that hold the two flexible rubber wiper plugs with two separate plug-release pins (see Figure 4-389a). When the spacer and the cement slurry are to be pumped to the inside of the casing

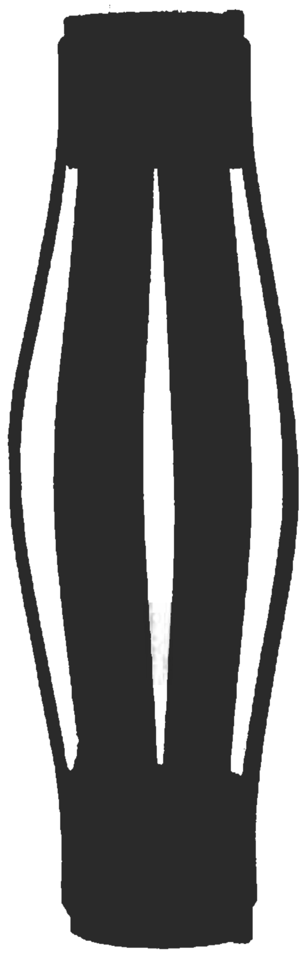


Figure 4-388. Centralizer [161].

through the cementing head, the bottom plug-release pin is removed. This releases the bottom wiper plug into the initial portion of the spacer flow to the well (see Figure 4-389b). This bottom wiper plug keeps the drilling mud from contaminating the spacer and the cement slurry while they pass through the inside of the casing. When all the cement slurry has passed through the cementing head, the top plug-release pin is removed releasing the top wiper plug into the flow to the well. At this point in the cementing operation the cement pump begins to pump drilling mud through the cementing head to the well (see Figure 4-389c).

When the bottom wiper plug reaches the float collar there is a brief sharp rise in pump pressure. The pump pressure increase breaks the diaphragm in the lower wiper plug. Once the diaphragm is broken the spacer and the cement slurry flow through the float collar into the final joint or two of the casing

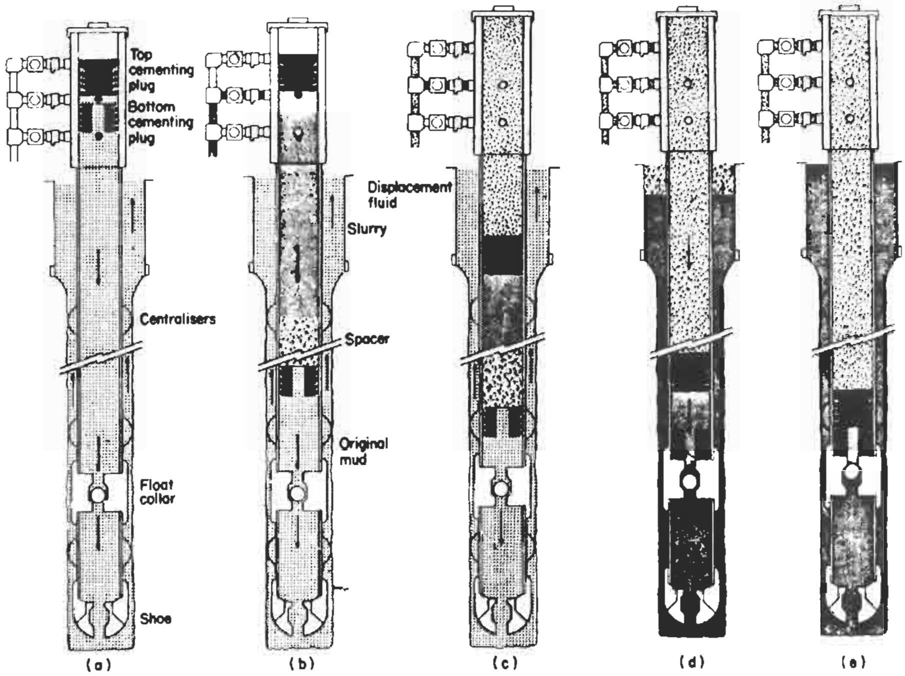


Figure 4-389. Single-stage cementing: (a) circulating mud; (b) pumping spacer and slurry; (c) and (d) displacing; (e) end of job. ●, plug-releasing pin in; ○, plug releasing pin out. (Courtesy Dowell Schlumberger.)

string and then into the annulus between the casing and the borehole wall (see Figure 4-389d). The spacer and the cement slurry displace the drilling mud below the float collar and the drilling mud in the annulus. The spacer is designed to efficiently displace nearly all the drilling mud in the annulus prior to the cement slurry entering the annulus.

When the top wiper plug reaches the float collar, the pump pressure rises sharply again. This wiper plug does not have a diaphragm and, therefore, no further flow into the well can take place. At this point in the cementing operation the cement pump is usually shut down and the pressure released on the cementing head. The back-flow valve in the float collar stops the heavier cement slurry from flowing back into the inside of the casing string (see Figure 4-389e). The volume of spacer and cement slurry is calculated to allow the cement slurry to either completely fill the annulus, or to fill the annulus to a height sufficient to accomplish the objectives of the casing and cementing operation.

Usually a lighter drilling mud is used to follow the heavy cement slurry. In this way the casing is under compression from a higher differential pressure on the outside of the casing. Thus when the cement sets and drilling or production operations continues, the casing will always have an elastic load on the cement-casing interface. This elastic load is considered essential for maintenance of the cement-casing bond and to keep leakage between the cement and casing (i.e., the microannulus) from occurring.

Since the early 1960s there has been a great deal of discussion regarding the desirability of using a low viscosity cement slurry to improve the removal of drilling mud from the annulus [161,162]. More recent studies have shown that low viscosity slurries do not necessarily provide effective removal of drilling mud in the annulus. In fact, there is strong evidence that spacers and slurries should have a higher gel strength and a higher specific weight than the mud that is being displaced [169]. Also, the displacement pumping rate should be low with an annular velocity of 90 ft/min or less [169]. Figure 4-390 gives the annular residual drilling mud removal efficiency as a function of differential gel strength and differential specific weight between the drilling mud and the spacer. The most successful spacer is an initial portion of the cement slurry that will give about 200 ft of annulus and that has a higher gel strength and a higher specific weight than the drilling mud it is to displace. The cement slurry spacer should have a gel strength that is about 10 to 15 lb/100 ft^2 greater and a specific weight that is about 2 to 4 lb/gal greater than the drilling mud to be displaced. The spacer should be a specially treated cement slurry.

It makes no sense to use a water or a drilling mud spacer. If such spacers are used, the follow-on normal cement slurry will be just as unsuccessful at removing the water or the drilling mud of the spacer as it would at removing drilling mud if no spacer were used.

The basic steps for planning the cement slurry to be used in the cementing operation are:

1. Determine the specific weight and gel strength of drilling mud to be displaced.
2. Estimate the approximate cementing operation time.

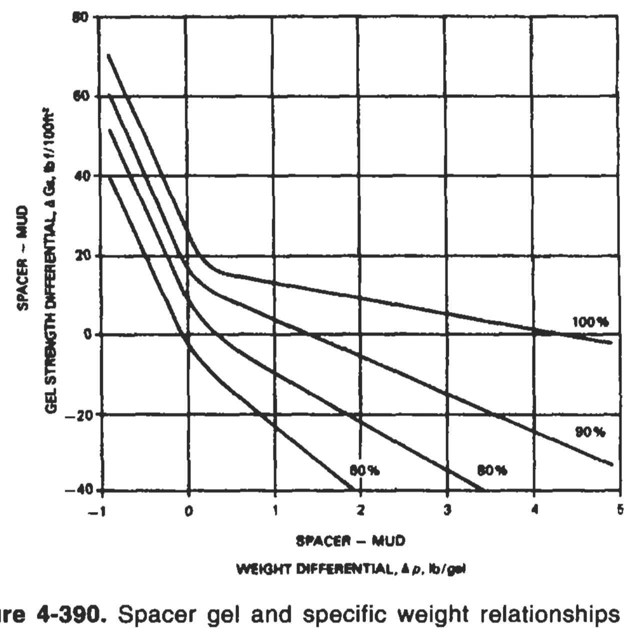


Figure 4-390. Spacer gel and specific weight relationships [168].

3. Select the most appropriate API class of cement that meets the depth, temperature, sulfate resistance and other well limitations. Select the cement class that has a natural thickening time that most nearly meets the cementing operation time requirement, or that will require only small amounts of retarding additives.
4. Do not add water loss control additives unless drilling mud requires such additives.
5. If the major portion of the cement slurry must have a rather low specific weight, do not use excess water to lower specific weight. Try to use only the optimum recommended water-to-cement ratio. Utilize only additives that will require little or no added water to lower the specific weight of the cement slurry. Avoid using bentonite in large concentrations since it requires a great deal of added water and significantly reduces cement strength. Silicate flour should be used whenever possible to increase compressive strength and decrease permeability of set cement.
6. Design the cement slurry and its initial spacer to have the appropriate gel strength and specific weight. Have the cementing service company run laboratory tests on the cement slurry blend selected. These tests should be run at the anticipated bottomhole temperature and pressure. These tests should be carried out to verify thickening time, specific weight, gel strength (of spacer) and compressive strength.
7. The mixed cement slurry should be closely monitored with high-quality continuous specific weight monitoring and recording equipment as the slurry is pumped to the well.
8. When cementing operation is completed, bleed off internal pressure inside the casing and leave valve open on cementing head.

Example 7

A 7-in., N-80, 29-lb/ft casing string is to be run in a 13,900-ft borehole. The casing string is to be 13,890 ft in length. Thus the guide shoe will be held about 10 ft from the bottom of the hole during cementing. The float collar is to be 90 ft from the bottom of the casing string. The borehole is cased with $9\frac{5}{8}$-in., 32.30-lb/ft casing to 11,451 ft of depth. The open hole below the casing shoe of the $9\frac{5}{8}$-in. casing is $8\frac{1}{2}$ in. in diameter. The 7-in. casing string is to be cemented to the top of the borehole. Class E cement with 20% silica flour is to be used from the bottom of the hole to a height of 1,250 ft from the bottom of the borehole. Class G cement with at least 10% silica flour is to be used from 1,250 ft from the bottom to the top of the borehole. The drilling mud in the borehole has a specific weight of 12.2 lb/gal and an initial gel strength of 15 lb/100 ft^2. A spacer sufficient to give 200 ft of length in the open-hole section will be used. An excess factor of 1.2 will be applied to the Class G cement slurry volume.

1. Determine the specific weight and gel strength for the spacer.
2. Determine the number of cement sacks for each class of cement to be used.
3. Determine the volume of water to be used.
4. Determine the cementing operation time and thus the minimum thickening time. Assume a cement mixing rate of 25 sacks/min. Also assume an annular displacement rate no greater than 90 ft/min while the spacer is moving through the open-hole section and a flowrate of 300 gal/min thereafter. A safety factor of 1.0 hr is to be used.
5. Determine the pressure differential prior to bumping the plug.
6. Determine the total mud returns during the cementing operation.

The basic properties of Class E and Class G cements are given in Table 4-154 and 4-155.

1. The drilling mud has a specific weight of 12.2 lb/gal and an initial gel strength of 15 lb/100 ft². Thus from Figure 4-390 it will be desirable to have a cement slurry spacer with a specific weight of at least 15.2 lb/gal and an initial gel strength of about 20 lb/100 ft².

The spacer will be designed with Class G cement. Taking the appropriate data from Tables 4-159 and 4-160, the weight of silica flour to be used is

$$0.10(94) = 9.4 \text{ lb/sack}$$

Equation 4-324 becomes

$$\bar{\gamma} = \frac{94 + 9.4 + 8.34[4.97 + 0.0456(9.4)]}{\frac{94}{3.14(8.34)} + \frac{9.4}{2.63(5.34)} + 4.97 + 0.0456(9.4)} = 16.4 \text{ lb/gal}$$

Thus the spacer will be designed to have a specific weight of 16.4 lb/gal and sufficient fragile gel additive to have an initial gel strength of about 20 lb/100 ft².

2. The spacer must have a volume sufficient to give 200 ft of length in the open-hole section. The annular capacity of the open hole is

$$\frac{\pi}{4}\left[\left(\frac{8.5}{12}\right)^2 - \left(\frac{7.0}{12}\right)^2\right] = 0.1268 \text{ft}^3/\text{ft}$$

Equation 4-326 gives the yield of the spacer as

$$\text{Yield} = \frac{3.59 + 0.4286 + 4.97 + 0.4286}{7.48} = 1.21 \text{ft}^3/\text{sack}$$

Thus the number of sacks to give that will be 200 ft in length in the open-hole section is

$$\text{Sacks} = \frac{0.1268(200)}{1.21} = 20.96$$

The above is rounded off to the next highest sack, or 21 sacks.

The volume of Class G cement is the sum of the spacer volume, annular volume in the cased portion of the borehole and the applicable annular volume in the open-hole section of the borehole. The annular capacity of the cased portion of the borehole is

$$\frac{\pi}{4}\left[\left(\frac{9.001}{12}\right)^2 - \left(\frac{7.0}{12}\right)^2\right] = 0.1746 \text{ft}^3/\text{ft}$$

Thus the volume of Class G cement slurry to be used to cement the well (excluding spacer, but considering the excess factor) is

$$\text{Volume} = [(2{,}449 - 1{,}250)(0.1268) + 11{,}451(0.1746)]\,1.2 = 2{,}581.66 \text{ ft}^3$$

The total number of Class G sacks of cement needed, including the spacer volume, is

$$\text{Total sacks} = \frac{2{,}581.66}{1.21} + 21 = 2{,}134 + 21 \approx 2{,}155$$

The volume of Class E cement is

$$\text{Volume} = \frac{\pi}{4}\left(\frac{8.5}{12}\right)^2(10) + \frac{\pi}{4}\left(\frac{6.184}{12}\right)^2 90 + (1{,}250 - 10)(0.1268)$$

$$= 3.94 + 18.77 + 157.23$$

$$= 179.94 \text{ ft}^3$$

The specific weight of the Class E cement is determined from Equation 4-324. The weight of silica flour to be added is

$$0.20(94) = 18.8 \text{ lb}$$

Equation 4-324 is

$$\bar{\gamma} = \frac{94 + 18.8 + 8.34[4.29 + 0.0456(18.8)]}{\dfrac{94}{3.14(8.34)} + \dfrac{18.8}{2.63(8.34)} + 4.29 + 0.0456(18.8)} = 16.2 \text{ lb/gal}$$

Equation 4-326 is

$$\text{Yield} = \frac{3.59 + 0.8571 + 4.29 + 0.8573}{7.48} = 1.28 \text{ ft}^3/\text{sack}$$

The total number of Class E sacks of cement needed are

$$\text{Total sacks} = \frac{179.94}{1.28} \approx 141$$

3. The volume of water to be used in the cementing operation is

$$\text{Volume} = 2{,}155(4.97 + 0.4286) + 141(4.29 + 0.8573)$$

$$\approx 11{,}634 + 726$$

$$\approx 12{,}360 \text{ gal}$$

or about 295 bbl

4. The total cementing operation time is somewhat complicated since it is desired to reduce the rate of flow when the spacer passes through the open hole section of the well. The mixing time is

$$T_m = \frac{2,155 + 141}{25(60)} = 1.53\,\text{hr}$$

During this time the volume of cement slurry pumped to the casing string is

$$\text{Volume pumped} = 2,155(1.21) + 141(1.28)$$
$$= 2,788.1\ \text{ft}^3$$

The internal volume of the casing string above the float collar is

$$\text{Internal volume} = 13,800(0.2086)$$
$$= 2,878.7\ \text{ft}^3$$

Therefore, the mud pumps are used to fill the remainder of the casing string at a rate of 300 gal/min. Thus, the additional time T_1 to get the bottom plug to the float collar is

$$T_1 = \frac{2,878.7 - 2,788.1}{\left(\frac{300}{7.48}\right)(60)} = 0.04\,\text{hr}$$

The time to release the plug is

$$T_p = 15/60$$
$$= 0.25\ \text{hr}$$

From the time the plug is released the pumping rate is reduced to 85.4 gal/min, which gives a velocity of 90 ft/min in the openhole section of the annulus. The time for the spacer to pass through the open-hole section T_2 (including the 90 ft of internal volume of the casing below the float collar and the 10 ft of open hole below the casing string guide shoe) is

$$T_2 = \frac{3.94 + 18.77 + (2,449 - 10)(0.1268)}{\left(\frac{85.4}{7.48}\right)(60)} = 0.49\,\text{hr}$$

After this time period, the rate of pumping to the well can be returned to 300 gal/min rate.

Equation 4-327 becomes

$$T_0 = 1.53 + 0.04 + 0.25 + 0.49 + \frac{2,878.7 - 91 - 332}{\left(\frac{300}{7.48}\right)(60)} + 1.0 = 4.33\,\text{hr}$$

Therefore, the thickening time for the cement slurries to be used must be greater than 4.33 hr.

5. The pressure differential prior to the top plug reaching the float collar is

$$p = \frac{122.7}{144}(13{,}900 - 1{,}250) + \frac{121.2}{144}(1{,}250 - 90 - 10) - \frac{91.3}{144}(13{,}900 - 90 - 10)$$

$$= 2997 \text{ psi}$$

6. The total volume of the mud returns will be

Mud volume = Total volume well without steel − Steel volume

$$= \frac{\;}{4}\frac{(8.5)^2}{12}(13{,}900 - 11{,}451) + \frac{\;}{4}\frac{(9.001)^3}{12}(11{,}451) - \frac{29.0}{490}(13{,}890)$$

$$= 5{,}203 \text{ ft}^3 \text{ or about } 927 \text{ bbl}$$

Large-Diameter Casing Cementing

Often when large-diameter casing strings are to be cemented, an alternate method to cementing through inside diameter of the casing is used. This alternative method requires that the inner string of drillpipe be placed inside the large-diameter casing string. The drillpipe string is centralized within the casing and a special stab-in unit is made up to the bottom of the drillpipe string. After the drillpipe string is run into the well through the casing, the special unit on the bottom of the drillpipe string is stabbed into the stab-in cementing collar located a joint or two above the casing string guide shoe, or the unit is stabbed into a stab-in cementing shoe (a combination of the stab-in cementing collar and the guide shoe). Figure 4-391 shows the schematic of a large-diameter casing with the inner drillpipe string used for cementing [163]. Figure 4-392 shows the stab-in cementing shoe and a stab-in cementing collar. Also shown is a flexible latch-in plug used to follow the cement slurry. Once the stab-in unit of the drillpipe string is seated in the stab-in cementing shoe or cementing collar, a special circulating head is made up to the top of the drillstring. Circulation is established through the circulating head to the drillpipe and up the annular space between the casing and the borehole wall. The stab-in unit may be locked into the cementing collar. The collar will act as a back-flow valve [162]. After the cement slurry has been run, the drillpipe string can be unlocked and withdrawn from the casing.

The advantages of cementing large diameter casing using an inner drillpipe string are:

- avoids excessive mud contaminations of the cement slurry with drilling mud prior to reaching the annulus;
- allows cement slurry to be added if wash out zones are excessive (avoids top-up job).

When cementing large-diameter casing loss of circulation can occur. The solution to such problems is to recement down the annulus. This technique of cementing is denoted as a top-up job. This type of cementing can be accomplished by running small-diameter tubing called "spaghetti" into the annulus space from the surface. Under these conditions usually low-specific-weight cement is used so that the formations will not fracture.

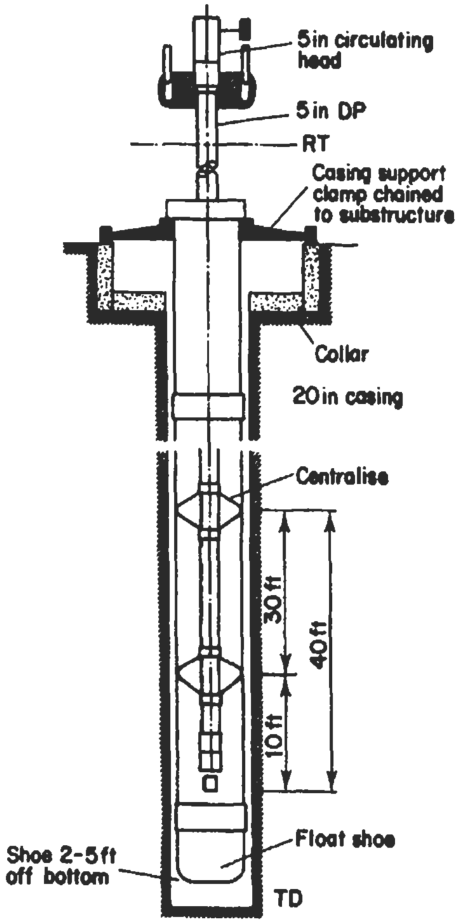


Figure 4-391. Cementing of large-diameter casing with and inner drillpipe string. (Courtesy Dowell Schlumberger.)

The hook load after a cementing operation (but prior to the cement setting) is important to know, particularly when large-diameter casings are to be cemented. Figure 4-393 shows the schematic of a casing string with the cement slurry height shown. Prior to the cement slurry setting the hook load V_h (lb) will be

$$V_h = w_c\left\{1 - \left[\left(\frac{\gamma_c}{\gamma_s}\right)\frac{h}{l} + \left(\frac{\gamma_m}{\gamma_s}\right)\left(\frac{l-h}{l}\right)\right]\right\}l - A_i[(\gamma_c - \gamma_m)h] \tag{4-331}$$

where w_c = unit weight of the casing in lb/ft
γ_c = specific weight of the cement slurry in lb/ft^3
γ_m = specific weight of the drilling mud in lb/ft^3

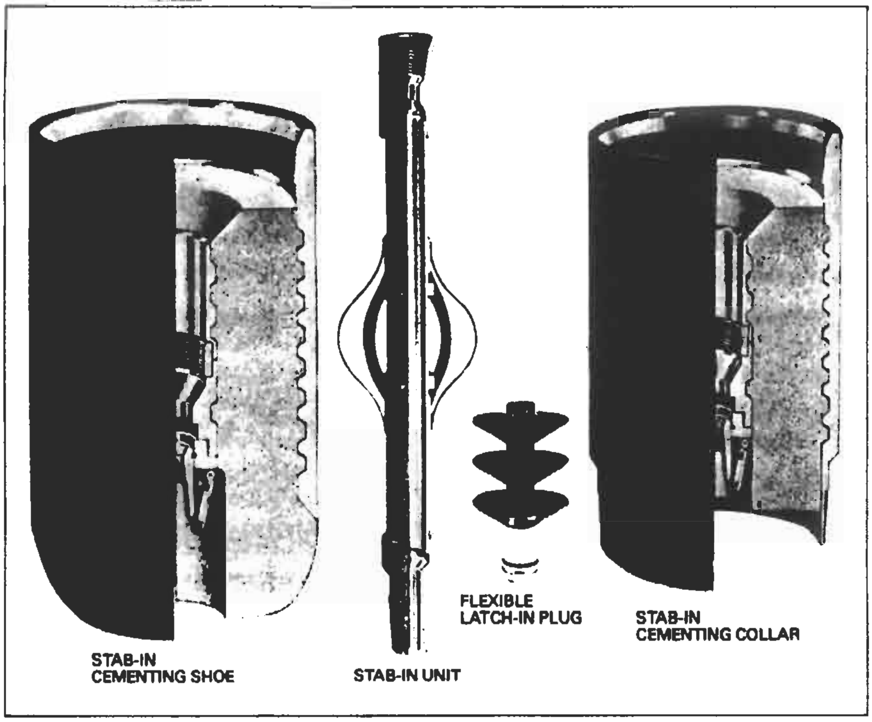


Figure 4-392. Stab-in unit, stab-in cementing shoe, stab-in cementing collar and flexible latch-in plug [161].

γ_s = specific weight of the steel, which is 490 lb/ft^3
l = length of the casing in ft
h = height the cement slurry is placed in the annular in ft
A_i = internal cross-sectional diameter of the casing in ft^2

Equation 4-331 is valid for a back-flow valve at the bottom of the casing string such as a float collar. It is also valid for a back-flow valve located at the top of the casing string.

Example 8

A large-diameter casing is to be cemented to the top in a 2,021-ft borehole. The casing string is 2,000 ft long and has ball float shoe at the bottom of the cement slurry. The borehole is 26 in. in diameter. The casing is to be 20 in., J-55, 94 lb/ft. The drilling mud in the borehole prior to running the casing has a specific weight of 10.0 lb/gal. The cement slurry is to have a specific weight of 17.0 lb/gal. The conventional cementing operation will be utilized, i.e., cementing through the inside of the casing. Determine the hook load after the cementing operation has been completed.

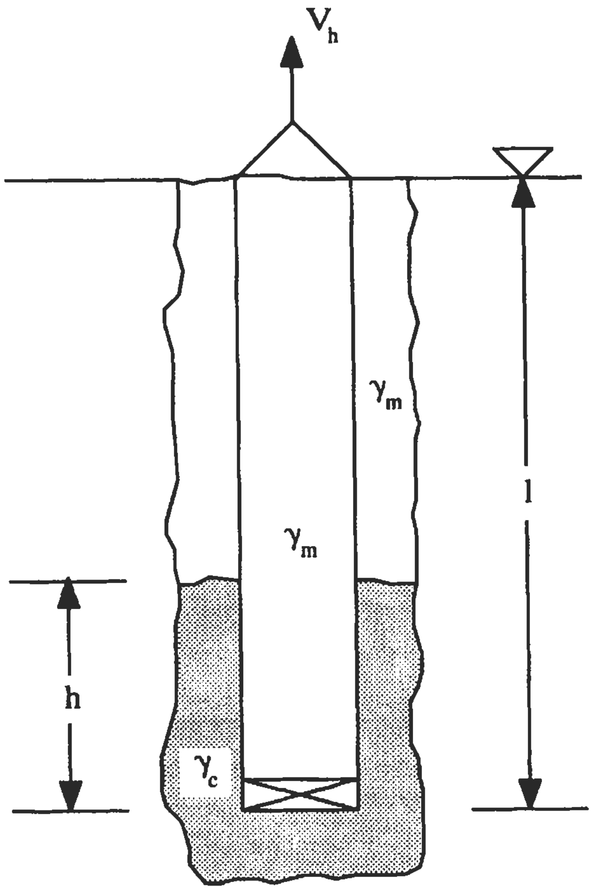


Figure 4-393. Schematic of casing after a cementing operation.

The hook load may be determined from Equation 4-331. This is

$$V_u = 94.00\left\{1 + \left[\left(\frac{127.3}{490.00}\right)\frac{2{,}000}{2{,}000} + \left(\frac{75.0}{490.0}\right)\frac{0}{2{,}000}\right]\right\}2{,}000$$

$$-\frac{\pi}{4}\left(\frac{19.124}{12}\right)^2[(127.3 - 75.0)2{,}000]$$

$$= -69{,}557 \text{ lb}$$

This answer means that the casing and its contained drilling mud will float when the cementing operation is completed and the back-flow valve is actuated. Thus the casing should be secured down at the well head prior to initiating the cementing operation.

It should be noted that if this cementing operation were to be carried out using an inner drillpipe string to place the cement in the annulus, the above force of buoyancy would be reduced by the buoyed weight of the drillpipe. However, unless very heavy drillpipe were used, the casing and drillpipe would still float on the cement slurry.

As the cement slurry sets the hook load will decrease from its value just after the cementing operation. If the cement slurry sets up to the height (freeze point) indicated in Figure 4-392, and if the casing has been landed as cemented, then the hook load V_h will be approximately

$$V_h = W_c\left(1 - \frac{\gamma_m}{\gamma_s}\right)(1 - h) \tag{4-332}$$

Example 9

A $5\frac{1}{2}$-in., J-55, 17.00-lb/ft casing string 16,100 ft in length is to be run into a 16,115-ft deep well. The float collar is 30 ft from the bottom of the casing string. The drilling mud in the well has a specific weight of 12.0 lb/gal. The casing string is to be cemented to a height of 4,355 ft above the bottom of the borehole with a cement slurry having a specific weight of 16.4 lb/gal.

1. Determine the approximate hook load prior to the cementing operation.
2. Determine the approximate hook load just after the cement slurry has been run.
3. Determine the approximate hook load after the cement has set.

1. The hook load prior to the cementing operation should read approximately

$$V_h = 17.00\left(1 - \frac{89.9}{490.0}\right)16{,}100 = 223{,}540\ \text{lb}$$

2. The hook load just after the cementing operation can be approximated by Equation 4-331. This is

$$V_h = 17.00\left\{1 - \left[\left(\frac{122.7}{490.0}\right)\frac{4{,}340}{16{,}100} + \left(\frac{89.9}{490.0}\right)\left(\frac{16{,}100 - 4{,}340}{16{,}100}\right)\right]\right\}16{,}100$$

$$- \frac{\pi}{4}\left(\frac{4.892}{12}\right)^2[(122.7 - 89.8)4{,}340]$$

$$= 199{,}943\ \text{lb}$$

3. The hook load after the cement has set can be approximated by Equation 4-332. This is

$$V_h - 17.00\left(1 - \frac{89.8}{490.0}\right)(16{,}100 - 4{,}340) = 163{,}282\ \text{lb}$$

Multistage Casing Cementing

Multistage casing cementing is used to cement long casing strings. The reasons that multistage cementing techniques are necessary are as follows:

- reduce the pumping pressure of the cement pumping equipment;
- reduce the hydrostatic pressure on weak formations to prevent fracture;
- selected formations can be cemented;
- entire length of a long casing string may be cemented;
- casing shoe of the previous casing string may be effectively cemented to the new casing string;
- reduces cement contamination.

There are methods for carrying out multistage cementing. These are

1. regular two-stage cementing
2. continuous two-stage cementing
3. regular three-stage cementing

Regular two-stage cementing requires the use of stage cementing collar and plugs in addition to the conventional casing cementing equipment. The stage cementing collar is placed in the casing string at near the mid point, or at a position in the casing string where the upper cementing of the casing is to take place. Figure 4-394 shows a schematic of a regular two-stage casing cementing operation. The stage cementing collar is a special collar with ports to the annulus that can be opened and closed (sealed off) by pressure operated sleeves (see Figure 4-394).

The first stage of cementing (the lower section of the annulus) is carried out similar to a conventional single-stage casing cementing operation. The exception is that a wiper plug is generally not run in the casing prior to the spacer and the cement slurry.* During the pumping of the spacer and cement slurry for the lower section of the annulus, the ports on the stage cementing collar are closed. After the appropriate volume of spacer and cement slurry has been pumped to the well (lower section) the first stage plug is released. This plug is pumped to its position on the float collar at the bottom of the casing string with drilling mud or completion fluid as the displacement fluid. This first plug is designed to pass through the stage cementing collar without actuating it. When the first plug is landed on the float collar, there is a pressure rise at the pump. This plug seals the float collar such that further flow throughout the collar cannot take place.

After the first-stage cementing operation has been carried out the opening bomb can be dropped and allowed to fall by gravity to the lower seal of the stage cementing collar. This can be done immediately after the first-stage cementing has taken place, or at some later time when the cement slurry in the lower section of the well has had time to set. In the later case great care must be taken that the first-stage cement slurry has not risen in the annulus to a height above the stage cementing collar. Once the opening bomb is seated, pump pressure is applied that allows hydraulic force to be applied to the lower sleeve of the stage cementing collar. This force shears the lower sleeve retaining pins and exposes the ports to the annulus.

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*A wiper plug may be accommodated with special equipment [161].

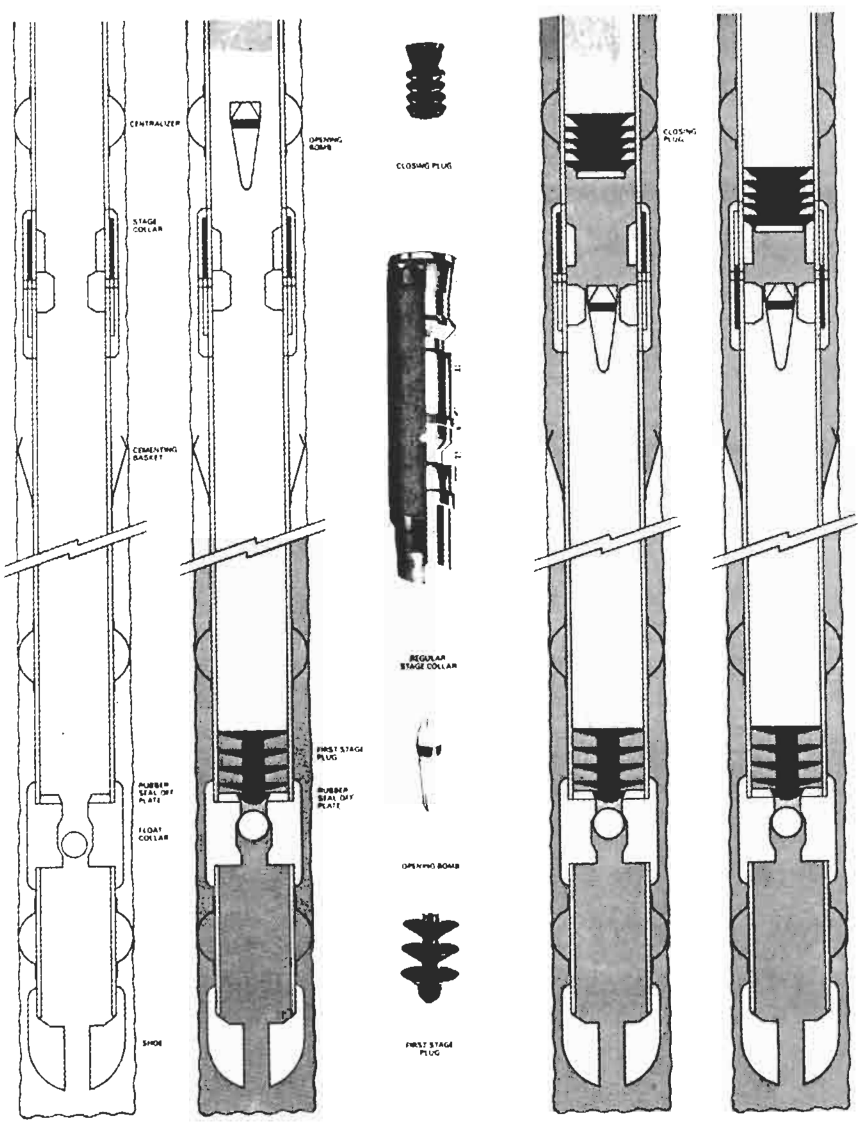


Figure 4-394. Regular two-stage cementing [161].

Once the ports are opened the well is circulated until the appropriate drilling mud or other fluid is in the well. The second-stage cementing slurry is mixed and pumped to the well (again without a wiper plug separating the spacer from the drilling mud). This cement slurry passes through the stage cementing float ports to the upper section of the annulus. The closing plug is released at the end of the second-stage cement slurry and is displaced to the stage cementing collar with drilling mud or other fluid. The closing plug seats on the upper sleeve of the stage cementing collar. A pressure of about 1,500 psi causes the retaining pins in the upper sleeve to shear thus forcing the sleeve downward to close the ports in the stage cementing collar.

Continuous two-stage cementing is an operation that requires that the cement slurry be mixed and displaced to the lower and upper sections of the annulus in sequence without stopping to wait for an opening bomb to actuate the stage cementing collar. In this operation the first-stage cement slurry is pumped to the well and a wiper plug released behind it (see Figure 4-395). Displacing the wiper plug is a volume of drilling mud or completion fluid that will displace the cement slurry out of the casing and fill the inside of the casing string from the float collar at the bottom of the casing string to a height of the stage cementing collar. A bypass insert allows fluid to pass through the wiper plug and float collar after the plug is landed. The opening plug is pumped immediately behind the volume of drilling mud. Immediately behind the opening plug is the second-stage spacer and cement slurry. The opening plug sits on the lower sleeve of the stage cementing collar, opening the ports to the annulus. At the end of the second-stage cement slurry the closing plug is run. This plug sits on the upper sleeve of the stage cementing collar and with hydraulic pressure, closing the ports in the stage cementing collar.

Three-stage cementing is carried out using the same procedure as the regular two-stage cementing operation discussed above. In this case, however, two-stage cementing collars are placed at appropriate locations in the casing string above the float collar. Each stage of cementing is carried out in sequence, the lower annulus section cemented first, the middle annulus section next and the top annulus section last. Each stage of cement can be allowed to set, but great care must be taken in not allowing the lower stage of cement to rise above the stage cementing collar of the next stage above.

Liner Cementing

A liner is a short string of casing that does not reach the surface. The liner is hung from the bottom of the previous casing string using a line hanger that grips the bottom of the previous casing with a set of slips. Figure 4-396 shows typical liner types. The liner is run into the borehole on the drillpipe and the cementing operation for the liner is carried out through the same drillpipe. The placing of liners and their cementing operations are some of the most difficult operations in well drilling and completions. Great care must be taken in designing and planning these operations to ensure a seal between the liner and the previous casing.

Figure 4-397 shows a typical liner assembly. The liner assembly is made up with the following components:

Float shoe. The float shoe may be placed at the bottom of the liner. This component is a combination of a guide shoe and a float collar.

Landing collar. This is a short sub placed in the string to provide a seat for the casing string.

Liner. This is a string of casing used to case off the open hole without bringing the end of the string to the surface. Usually the liner overlaps the previous casing string (shoe) by about 200 to 500 ft.

Liner hanger. This special tool is installed on the top of the liner string. The top of the liner hanger makes up to the drillpipe on which the entire liner assembly is lowered into the well. Liner hangers can be either mechanically or hydraulically actuated.

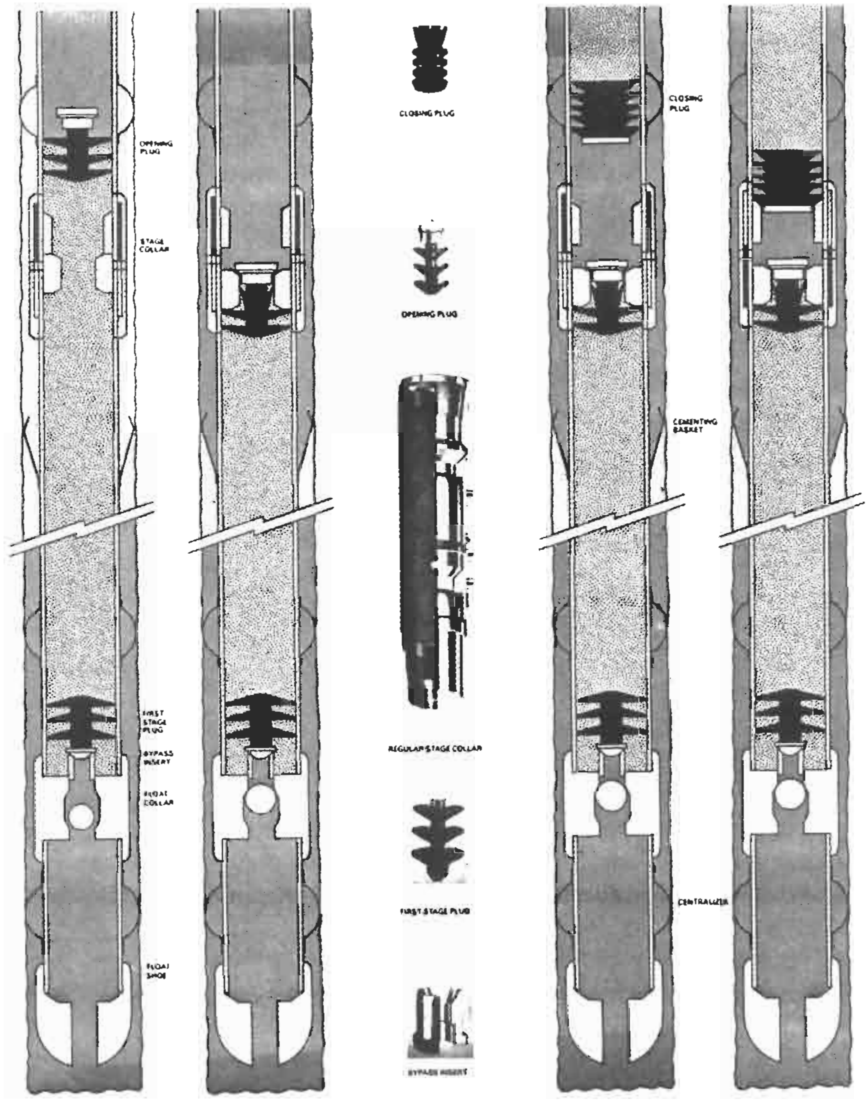


Figure 4-395. Continuous two-stage cementing [161].

The liner hanger is the key element in the running of liners and the follow-on cementing operations. The liner hanger allows the liner to be hung from the bottom section of the previous casing. After the cementing operation the upper part of the liner hanger is retrievable, thus allowing the residual cement above the liner hanger to be cleaned out of the annulus between the drillpipe and the previous casing (by reverse circulation) and the liner left in the well.

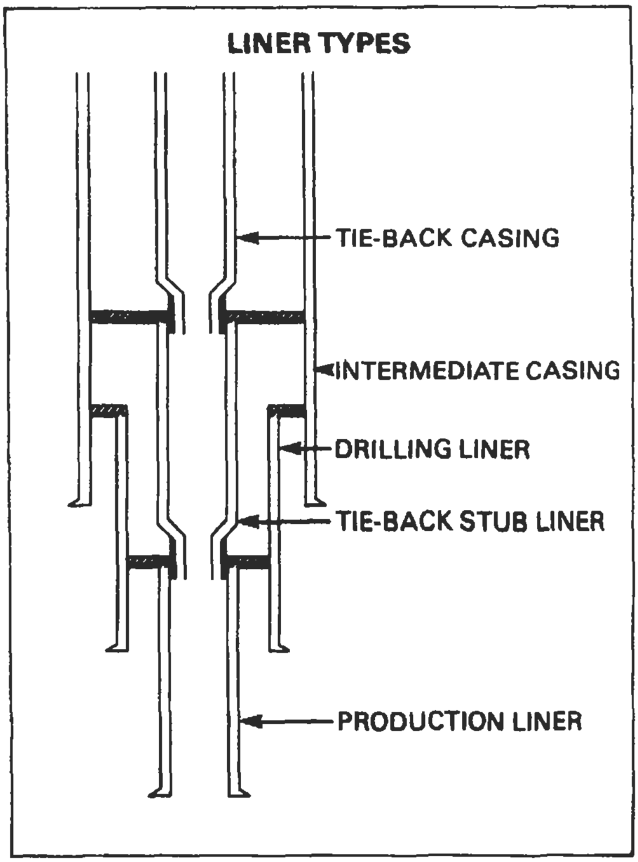


Figure 4-396. Liner types [161].

The casing joints of the liner are placed in the well in the normal manner. The liner hanger is made up to the top of the liner. The top of the liner hanger is made up to drillpipe and the whole assembly (i.e. liner, liner hanger and drillpipe) is lowered into the well. After the liner is at the desirable location in the well, the drillpipe is connected to the rig pumps and mud circulation carried out. This allows conditioning of the drilling mud in the well prior to the cementing operation and ensures that circulation is possible before the liner is hung and cemented.

The liner hanger is set (either mechanically or hydraulically) and the drillpipe with the upper part of the liner hanger (the setting tool) released. The drillpipe and the setting tool are raised to make sure that the setting tool and the drillpipe can be released from the lower part of the liner hanger and the liner. The drillpipe and the setting tool are lowered to make a tight seal with the lower portion of the liner hanger.

After these tests of the equipment have been made, a liner cementing head is made up to the drillpipe at the surface. Figure 4-398 shows the liner cementing head and the pump-down plug. The pump-down plug is placed in the liner

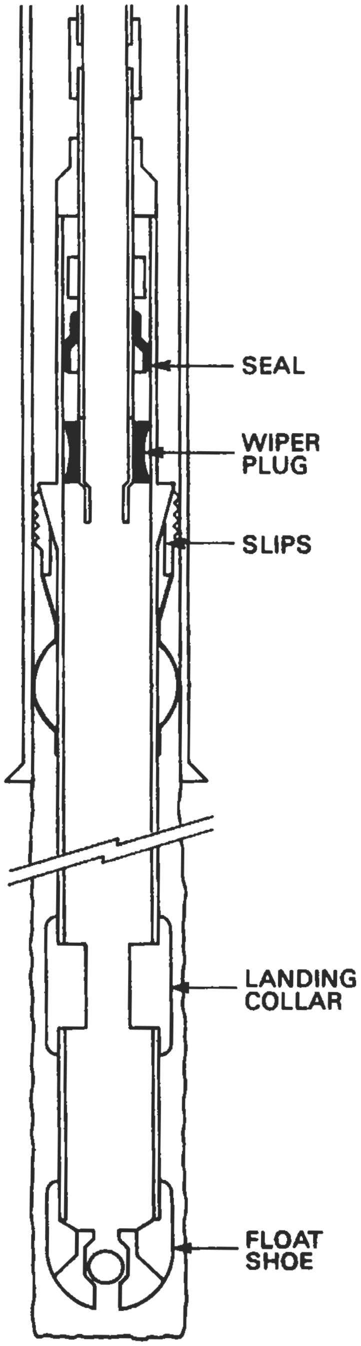


Figure 4-397. Liner assembly [161].

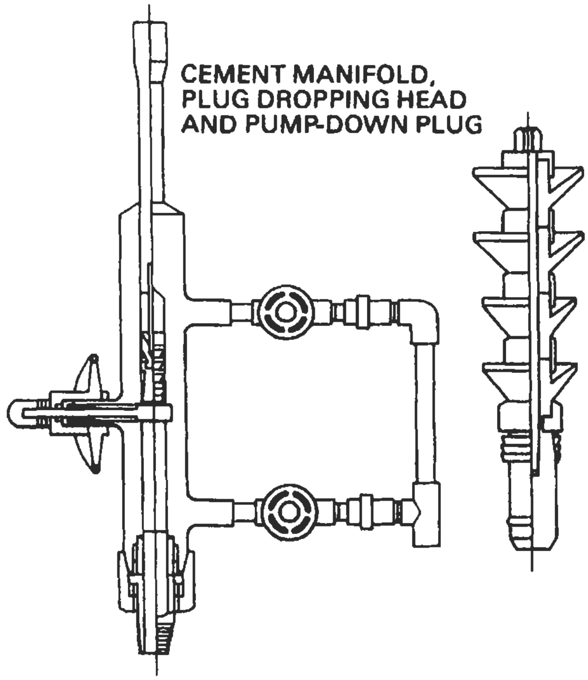


Figure 4-398. Liner cementing head [161].

cementing head, but not released. The cement pump is connected to the liner cementing head and the spacer and cement slurry pumped to the head (no wiper plug is run ahead of the spacer). The pump-down plug is released at the end of the cement slurry and separates the slurry from the follow-on drilling mud. Figure 4-399 shows the sequence of the liner cementing operation. The drilling mud displaces the pump-down plug to the liner hanger. As the pump-down plug passes through the liner hanger it latches into the liner wiper plug (see Figure 4-397). With increased surface pressure (1200 psi) by the pump the liner wiper plug with the pump-down plug coupled to it are released from the liner hanger and begin to move again downward. These two coupled plugs eventually seat on the landing collar or on the float collar. Another pressure rise indicates the cement is in place behind the liner.

Once the spacer and the cement slurry have been successfully pumped to their location in the well, the drillpipe and setting tool are released from the lower part of the liner hanger. Also, the liner cementing head is removed from the drillpipe. The drillpipe and setting tool are raised slightly and the excess cement slurry reverse circulated from around the liner hanger area. These steps should be taken immediately after the completion of the cementing operation, otherwise the excess cement slurry between the drillpipe and the previous casing could set and cause later drilling and completions problems if the excess cement is too great. If reverse circulation is not planned, then the excess cement slurry that comes through the liner hanger to the annulus between the drillpipe and the previous casing must be kept to a minimum so that it can be easily drilled out after it sets.

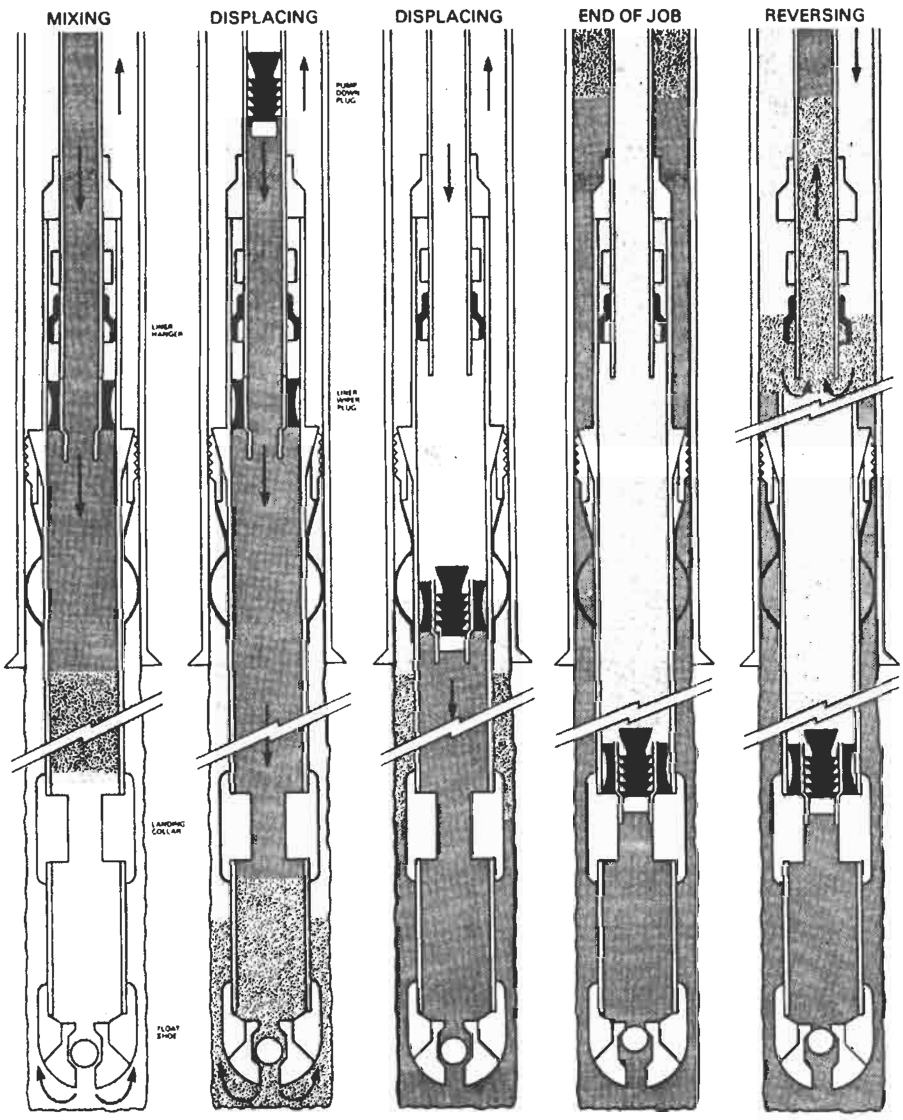


Figure 4-399. Liner cementing [161].

The determination of the excess cement slurry should be carefully calculated: too little and the cement seal at the liner hanger is contaminated with drilling mud; too much and there are problems removing it.

Secondary Cementing

Secondary or remedial cementing refers to cementing operations that are intended to use cement as a means of maintaining or improving the well's operation. There are two general secondary cementing operations, squeeze cementing and plug cementing.

Squeeze Cementing

Squeeze cementing operations utilize the mechanical power of the cement pumps to force a cement slurry into the annular space behind a casing and/or into a formation for the following purposes:

- repair a faulty primary cementing operation;
- stop loss of circulation in the open hole during drilling operations;
- reduce water-oil, water-gas and gas-oil ratios by selectively sealing certain fluid producing formations;
- seal abandoned or depleted formations;
- repair casing leaks such as joint leaks, split casing, parted casing or corroded casing;
- isolate a production zone by sealing off adjacent nonproductive zones.

There are numerous squeeze cementing placement methods that utilize many different special downhole tools. In general, the squeeze cementing operation forces the cement slurry into fractures under high pressure, or into casing perforations using low pressure. As a cement slurry is forced into rock fractures, or through casing perforations into the rock formations, the slurry loses part of its mix water. This leaves a filter cake of cement particles at the interface of the fluid and the permeable rock. As the filter cake builds up during the squeeze cementing operation, more channels into the formation are sealed and the pressure increases. If the cement slurry is poorly designed for an intended squeeze cementing operation, the filter cake builds up too fast and the cement pump capability is reached before the cement slurry has penetrated sufficiently to accomplish its purpose. Thus the squeeze cementing operation slurry should be designed to match the characteristics of the rock formation to be squeeze cemented and the equipment to be used.

There are five important considerations regarding the cement slurry design for a squeeze cementing operation:

1. *Fluid loss control.* The slurry should be designed to match the formation to be squeezed. Low permeability formations should utilize slurries with 100–200 ml/30 min water losses. High permeability formations should utilize slurries with 50–100 ml/30 min water losses.
2. *Slurry volume.* The cement slurry volume should be estimated prior to the squeeze operation. In general, high-pressure squeeze operations of high-permeability formations that have relatively low fracture strengths will require large volumes of slurry. Low-pressure squeeze operation through perforations will require low volumes.
3. *Thickening time.* High-pressure squeeze operations that pump large volumes in a rather short time period usually require accelerator additives. Low-pressure slow-pumping-rate squeeze operations usually require retarder additives.
4. *Dispersion.* Thick slurries will not flow well in narrow channels. Squeeze cement slurries should be designed to be thin and have low yield points. Dispersive agents should be added to these slurries.
5. *Compressive strength.* High compressive strength is not a necessary characteristic of squeeze cement slurries.

There are basically two squeeze cementing techniques used: the high-pressure squeeze operation and the low-pressure squeeze operation.

High-pressure squeeze cementing operations are utilized where the hydraulic pressure is used to make new channels in the rock formations (by fracturing the rock) and force the cement slurry into these channels.

Low-pressure squeeze cementing operations are utilized where the existing permeability structure is sufficient to allow the cement slurry to efficiently move in formation without making new fracture surfaces with the hydraulic pressure.

Hesitation method of applying pressure is applicable to both high and low-pressure squeeze cementing operations. This method of applying pressure (and thus volume) appears to be more effective than continuous pressure application. The hesitation method is the intermittent application of pressure, separated by a period of pressure leakoff caused by the loss of filtrate into the formation. The leakoff periods are short at the beginning of an operation but get longer as the operation progresses.

Figure 4-400 shows a typical squeeze cementing downhole schematic where a retrievable packer is used. The cement slurry is spotted adjacent to the perforations in the casing. The packer raise and set against the casing wall. Pressure is then applied to the drilling mud and cement slurry below the packer forcing the cement slurry into the perforations and into the rock formation or voids in the annulus behind the casing.

Figure 4-401 shows how a retrievable packer with a tail pipe can be used to precisely spot the cement slurry at the location of the perforations.

Figure 4-402 shows how a drillable cement retainer can be used to ensure that the cement slurry will be applied directly to the perforations.

Most squeeze cementing operations take place in cased sections of a well. However, open-hole packers can be used to carry out squeeze cement operations of thief zones during drilling operations.

Another technique for carrying out a squeeze cementing operation is the Bradenhead technique. This technique can be used to squeeze a cement slurry

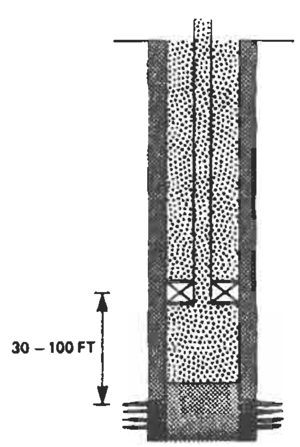


Figure 4-400. Retrievable packer near perforations [161].

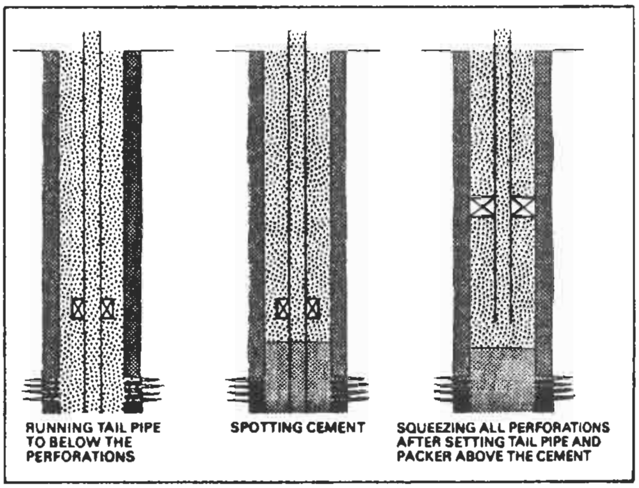


Figure 4-401. Retrievable packer with tail pipe [161].

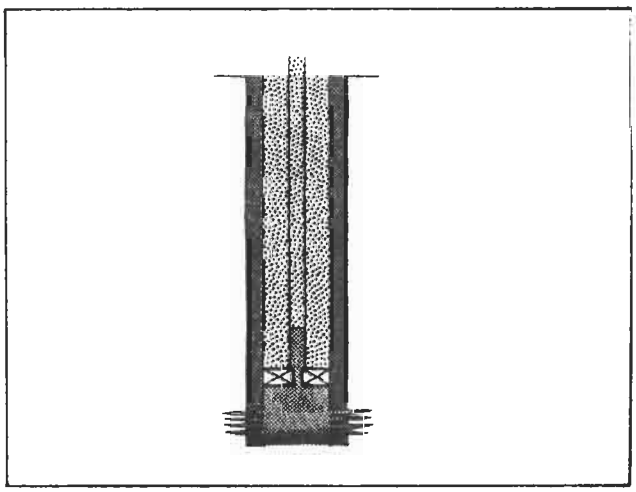


Figure 4-402. Drillable cement retainer [161].

in a cased hole or in an open hole. Figure 4-403 shows a schematic of the technique. Instead of using a downhole packer, the cement slurry is spotted by drillpipe (or tubing) adjacent to the perforations or a thief zone. After the drillpipe has been raised, the pipe rams are closed over the drillpipe and pump pressure applied to the drilling mud which in turn forces the cement slurry into the perforations or fractures.

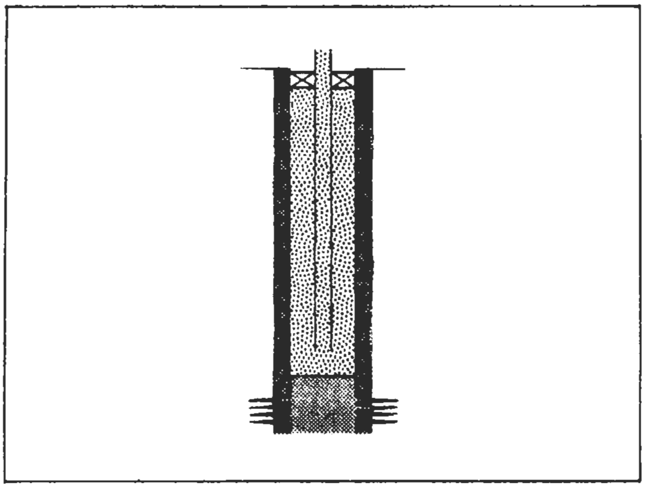


Figure 4-403. Bradenhead squeeze [161].

The basic squeeze cementing operation procedures are as follows [161]:

1. Lower zones are isolated by a retrievable or drillable bridge plug.
2. Perforations are washed using a perforation wash tool or they are reopened by surging. If there is no danger of damaging the lower perforations, this operation can be carried out before running the bridge plug, which may be run in one trip with the packer.
3. The perforation wash tool is retrieved and the packer run in the hole with a work string, set at the desired depth and tested. An annular pressure test of 1,000 psi is usually sufficient. The packer is run with or without a tail pipe, depending on the operation to be performed. If cement is to be spotted in front of the perforations, a tail pipe that covers the length of the zone plus 10 to 15 ft must be run with the packer.
4. An injectivity test is performed using clean, solids-free water or brine. If a low fluid loss completion fluid is in the hole, it must be displaced from the perforations before starting the injecting. This test will give an idea of the permeability of the formation to the cement filtrate.
5. A spearhead or breakdown fluid followed by the cement slurry is circulated downhole with the packer by-pass open. This is done to avoid the squeezing of damaging fluids ahead of the slurry. A small amount of back pressure must be applied on the annulus to prevent the slurry fall caused by U tubing. If no tail pipe has been run, the packer by-pass must be closed 2 or 3 bbl before the slurry reaches the packer. If the cement is to be spotted in front of the perforations, with the packer unset, circulation is stopped as soon as the cement covers the desired zone, the tail pipe pulled out of the cement slurry and the packer set at the desired depth. The depth at which the packer is set must be carefully decided.

   If tail pipe is run, the minimum distance between perforations and packer is limited to the length of the tail pipe. The packer must not be set too close to the perforations as pressure communication through the annulus above the packer might cause the casing to collapse. A safe setting depth must be decided on after seeing the logged quality of the cement bond. Casing conditions and possible cement contamination limit the maximum spacing between packer and treated zone.

6. Squeeze pressure is applied at the surface. If high-pressure squeezing is practiced, the formation is broken down and the cement slurry pumped into the fractures before the hesitation technique is applied. If low-pressure squeezing is desired, hesitation is started as soon as the packer is set.
7. Hesitation continues until no pressure leak-off is observed. A further test of about 500 psi over the final injection pressure will indicate the end of the injection process. Usually, well-cementing perforations will tolerate pressures above the formation fracture pressure, but the risk of fracturing is increased.
8. Pressure is released and returns are checked. If no returns are noticed, the packer by-pass is opened and excess cement reversed out. Washing off cement in front of perforations can be performed by releasing the packer and slowly lowering the work string during the reversing.
9. Tools are pulled out and the cement is left to cure for the recommended time, usually 4 to 6 hr.

Plug Cementing

The major reasons for plug cementing are:

Abandonment. State regulations have rules on plugging and abandoning wells. Cement plugs are normally used for that purpose (see Figure 4-404).

Kick-off plug. Usually an Ottawa sand-cement plug is used to plug off a section of the borehole. This plug uses a hard surface to assist the kick-off procedure (see Figure 4-405).

Lost circulation. A cement plug can be placed adjacent to a zone of lost circulation in the hope that the cement slurry will penetrate and seal fractures (see Figure 4-406).

Openhole completions. Often in openhole completions it is necessary to shut off water flows, or to provide an anchor for testing tools, or other maintenance operations (Figure 4-407).

There are three methods for placing cement plugs.

1. *Balance plug method* is the most commonly used. The cement slurry is placed at the desired depth through the drillpipe or tubing run to that depth. A spacer is placed below and above the slurry plug to avoid contamination of the cement slurry with surrounding drilling mud and to assist in balancing the plug.
2. *Dump bailing method* utilizes a bailing device that contains a measure volume of cement slurry. The bailer is run to the appropriate depth on a wireline and releases its load upon bumping the bottom or a permanent bridge plug set at the desired depth (see Figure 4-408).
3. *Two-plug method* is a rather new method and requires the use of the tell-tale catcher sub (Dowell Schlumberger) to set a cement plug in a well at a rather precise location with minimum contamination. The tell-tale catcher sub is made up to the lower end of a drillpipe string. The sub also has an aluminum tail pipe, a bottom wiper plug (which carries a dart) and a top wiper plug. The sub on the drillpipe is lowered to the depth desired for

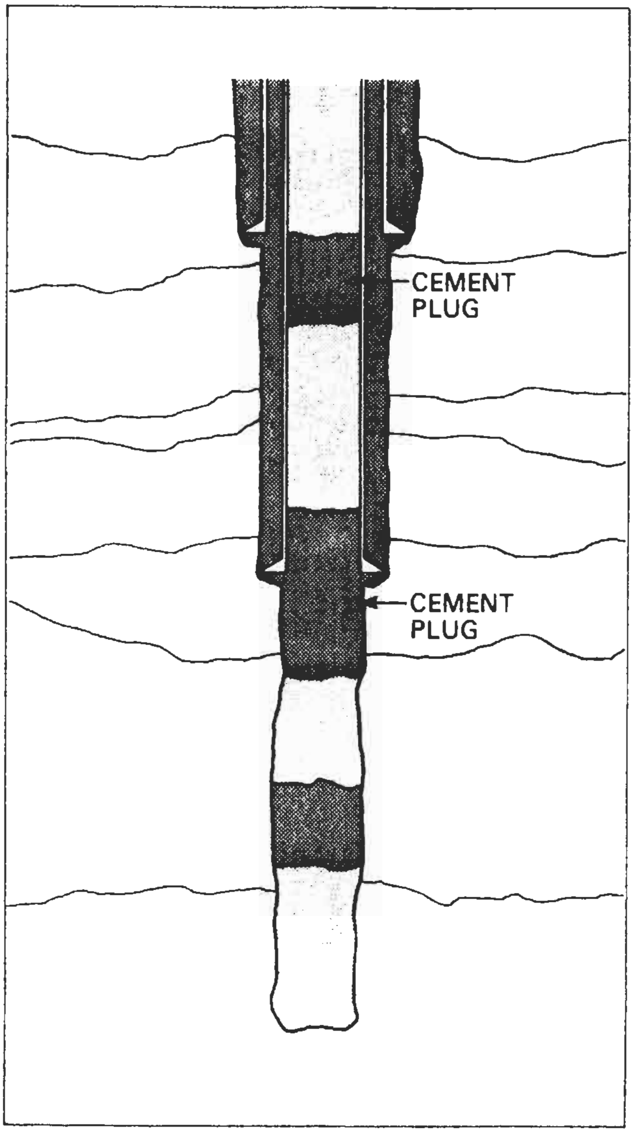


Figure 4-404. Abandonment plugs [161].

the location of the plug and similar to a primary cementing operation, the bottom plug (with dart) is run ahead of the cement slurry. When the plug reaches the sub, increased pump pressure separates the dart from the bottom plug. The dart continues through the tail pipe as a wiper plug. The top plug is pumped behind the cement slurry. When the top plug reaches the sub there is another increase in pressure. The cement slurry in the sub can be reverse-circulated from the sub.

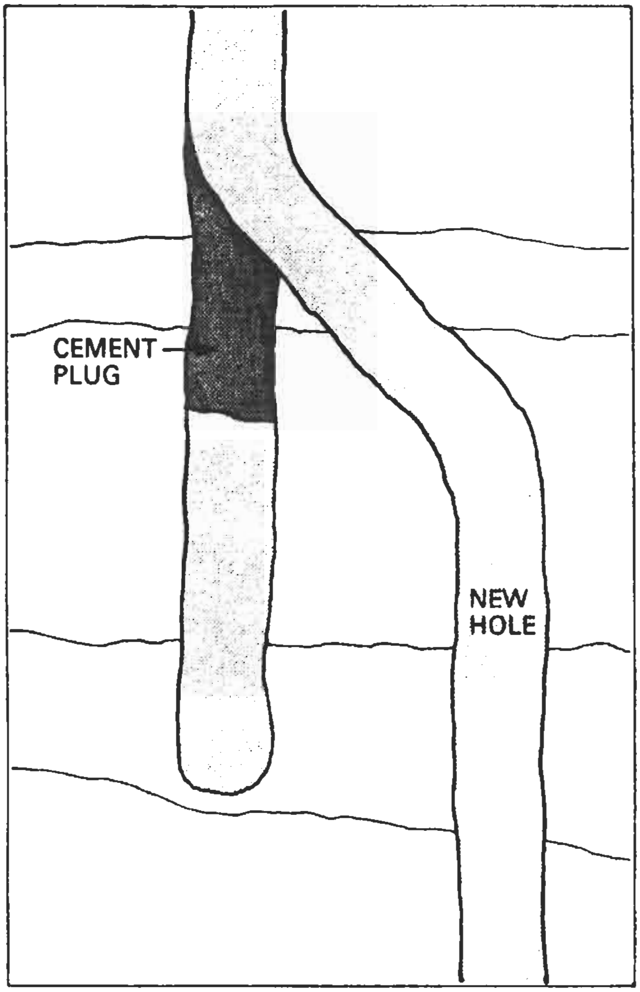


Figure 4-405. Kickoff plug [161].

Cement plugs fail for the following reasons:

- lack of hardness (kick-off plug)
- contaminated cement slurry
- placed at wrong depth
- plug migrates from intended depth location due to lack of balancing control

When placing a plug in well the fluid column in the tubing or drillpipe must balance the fluid column in the annulus. If the fluid in the drillpipe is heaviest, then the fluid from the drillpipe (the slurry) will continue down and out the end of the drillpipe and then up into the annulus after pumping has stopped. If the fluid in the annulus is heaviest, then the fluid from the drillpipe will

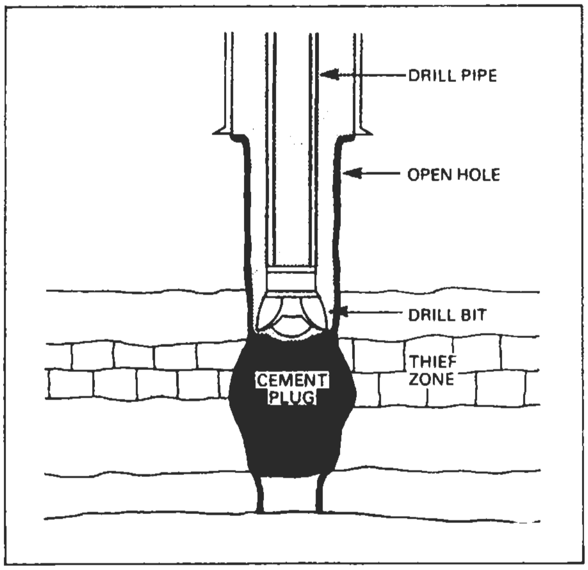


Figure 4-406. Lost circulation plug [161].

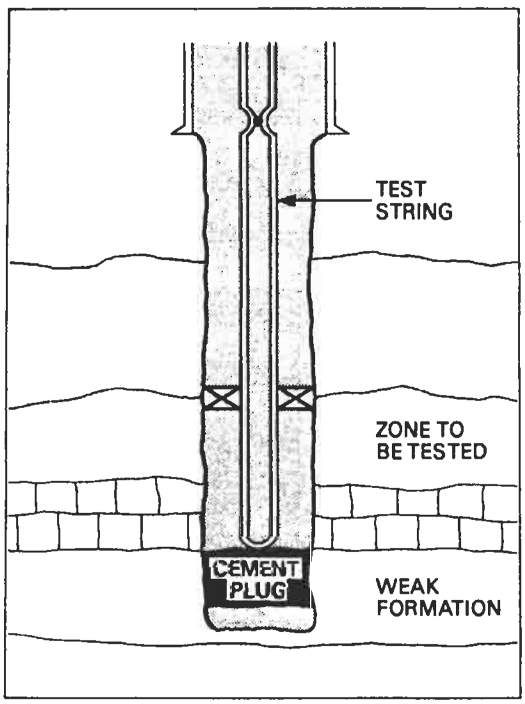


Figure 4-407. Openhole completions plug [161].

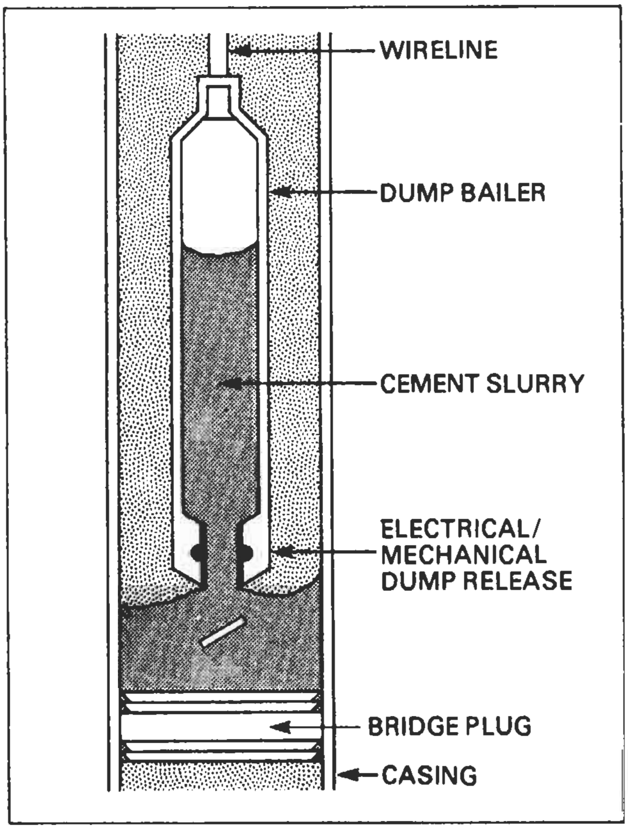


Figure 4-408. Dump bailer [161].

continue downward in the annulus after pumping. The two columns are balanced by using a spacer (or wash) above and below the cement slurry. Since the cement slurry usually has a higher specific weight than the annular fluid, the spacers are normally water.

Example 10

Balance a 100-sack cement plug of Class C neat cement slurry when 15 bbl of water are to be run ahead of the slurry. The plug is to be run through 2-in., 4.716-lb/ft, EUE tubing. The depth of placement is 4,000 ft.

1. Determine the top of the plug.
2. Determine the volume of water to run behind cement slurry to balance the 15 bbl of water run ahead of the slurry.
3. Determine the volume of mud to pump to balance plug.

The annular volume per ft is 0.1997 ft^3/ft. The tubing volume per ft is 0.0217 ft^3/ft). The yield of a Class C neat cement slurry is 1.32 $ft^3/sack$ (see Table 4-155). Thus the volume of the cement slurry is

Volume = 1.32(100) = 132 ft^3

1. The height H (ft) of the plug is

$$H = \frac{132}{0.1997 - 0.0217} = 742\,\text{ft}$$

Depth to top of plug D (ft) is

D = 4,000 - 742 = 3,258 ft.

2. The amount of water to place behind the plug to balance the 15 bbl ahead of the cement slurry is

$$\text{Volume} = 15\left(\frac{0.0217}{0.1997}\right) = 1.6\,\text{bbl}$$

3. The height h (ft) of trailing water in the tubing is

$$h = \frac{1.6(42)}{(0.0217)(7.48)} = 414\,\text{ft}$$

The volume of mud to pump to balance plug is

$$\text{Volume} = [4{,}000 - (742 + 414)]\frac{0.0217}{5.6146} = 11\,\text{bbl}$$

TUBING AND TUBING STRING DESIGN

Tubing string is installed in the hole to protect the casing from erosion and corrosion and to provide control for production. The proper selection of tubing size, steel grade, unit weight and type of connections is of critical importance in a well completion program. Tubing must be designed against failure from tensile/compressive forces, internal and external pressures and buckling.

Tubing is classified according to the outside diameter, the steel grade, unit weight (well thickness), length and type of joints. The API tubing list is given in Tables 4-161 and 4-162.

API Physical Property Specifications

The tubing steel grade data as given by API Specification 5A, 35th Edition (March 1981), Specification 5AX, 12th Edition (March 1982) and Specification 5AC, 13th Edition (March 1982), are listed in Table 4-163.

Dimensions, Weights and Lengths

The following are excerpts from API Specification 5A, 35th Edition (March 1981), "API Specification for Casing, Tubing, and Drill Pipe."

Table 4-161
API Tubing List

1		2	3	4		5
Size: Outside Diameter		Nominal Weight, Threads and Coupling, lb per ft	Grade	Wall Thickness		Type of Ends[1]
in.	mm			in.	mm	
D				t		
††*1.050*	*26,7*	*1.14*	*H, J, N*	*0.113*	*2.87*	*Non-Upset*
1.050	26,7	1.20	H, J, N	0.113	2.87	Ext. Upset
1.315	33,4	1.70	H, J, N	0.133	3,38	Non-Upset
1.315	33,4	1.72	H, J, N	0.133	3,38	Integral Joint
1.315	33,4	1.80	H, J, N	0.133	3,38	Ext. Upset
1.660	42,2	2.10	H, J	0.125	3,18	Integral Joint
1.660	42,2	2.30	H, J, N	0.140	3,56	Non-Upset
1.660	42,2	2.33	H, J, N	0.140	3,56	Integral Joint
1.660	42,2	2.40	H, J, N	0.140	3,56	Ext. Upset
1.900	48,3	2.40	H, J	0.125	3,18	Integral Joint
1.900	48,3	2.75	H, J, N	0.145	3,68	Non-Upset
1.900	48,3	2.76	H, J, N	0.145	3,68	Integral Joint
1.900	48,3	2.90	H, J, N	0.145	3,68	Ext. Upset
2.063	52,4	3.25	H, J, N	0.156	3,96	Integral Joint
2⅜	60,3	4.00	H, J, N	0.167	4,24	Non-Upset
2⅜	60,3	4.60	H, J, N	0.190	4,83	Non-Upset
2⅜	60,3	4.70	H, J, N	0.190	4,83	Ext. Upset
2⅜	60,3	5.80	N	0.254	6,45	Non-Upset
2⅜	60,3	5.95	N	0.254	6,45	Ext. Upset
2⅞	73,0	6.40	H, J, N	0.217	5,51	Non-Upset
2⅞	73,0	6.50	H, J, N	0.217	5,51	Ext. Upset
2⅞	73,0	8.60	N	0.308	7,82	Non-Upset
2⅞	73,0	8.70	N	0.308	7,82	Ext. Upset
3½	88,9	7.70	H, J, N	0.216	5,49	Non-Upset
3½	88,9	9.20	H, J, N	0.254	6,45	Non-Upset
3½	88,9	9.30	H, J, N	0.254	6,45	Ext. Upset
3½	88,9	10.20	H, J, N	0.289	7,34	Non-Upset
3½	88,9	12.70	N	0.375	9,52	Non-Upset
3½	88,9	12.95	N	0.375	9,52	Ext. Upset
4	101,6	9.50	H, J, N	0.226	5,74	Non-Upset
4	101,6	11.00	H, J, N	0.262	6,65	Ext. Upset
4½	114,3	12.60	H, J, N	0.271	6,88	Non-Upset
4½	114,3	12.75	H, J, N	0.271	6,88	Ext. Upset

[1]Non-upset tubing is available with regular couplings or special bevel couplings. External-upset tubing is available with regular, special bevel, or special clearance couplings.
††For information purposes only.
Source: From Ref. [173].

Table 4-162
API Work Tubing List

1		2	3		4	5		6
Size: Outside Diameter		Nominal Weight, lb/ft	Calculated Plain-end Weight,		Grade	Wall Thickness,		Upset Ends, for Weld-on Tool Joints
in.	mm		lb/ft	kg/m		in.	mm	
D			W_{pe}			t		
1.050	26,7	1.55	1.47	2,19	N	0.154	3,91	Int.-Ext. Upset
1.315	33,4	2.30	2.17	3,23	N	0.179	4,32	Int.-Ext. Upset
1.660	42,2	3.29	3.09	4,60	N	0.198	5,03	Int.-Ext. Upset
1.900	48,3	4.19	3.93	5,85	N	0.219	5,56	Int.-Ext. Upset

*Heavy wall tubing for small diameter work strings.
Source: From Ref. [173].

1. *Dimensions and weights.* Pipe shall be furnished in the sizes, wall thicknesses and weights (as shown in Tables 4-164, 4-165 and 4-166) as specified by API Specification 5A: "Casing, Tubing and Drill Pipe."
2. *Diameter.* The outside diameter of 4 in.-tubing and smaller shall be within ±0.79 mm. Inside diameters are governed by the outside diameter and weight tolerances.

Table 4-163
Tubing Steel Grade Data

Steel Grade	Yield Strength, psi Min.	Yield Strength, psi Max.	Min. Tensile Strength, psi	Elongation*
H-60	40,000	80,000	60,000	27
J-55	55,000	80,000	75,000	20
C-75	75,000	90,000	95,000	16
C-95	95,000	110,000	105,000	16
N-80	80,000	110,000	100,000	16
P-105	105,000	135,000	120,000	15

*The minimum elongation in 2 in. (50.80 mm) should be determined by the following formula:

$$e = 625,000\frac{A^{0.2}}{U^{0.9}}$$

where e = minimum elongation in 2 in. (50.80 mm) in percent rounded to the nearest ½%
A = cross-sectional area of the tensile specimen in in.², based on a specified outside diameter or nominal specimen width and specified wall thickness, rounded to the nearest 0.01 or 0.75 in.², whichever is smaller
U = specified tensile strength in psi.

Table 4-164
Nonupset Tubing—Dimensions and Weights

1	2	3	4	5	6
	Nominal Weight:[1]			Calculated Weight,	
Size: Outside Diameter, in. D	Threads and Coupling, lb. per ft.	Wall Thickness, in. t	Inside Diameter, in. d	Plain End lb/ft w_{pe}	Threads[2] and Coupling lb e_w
*1.050	1.14	0.113	0.824	1.13	0.20
1.315	1.70	0.133	1.049	1.68	0.40
1.660	2.30	0.140	1.380	2.27	0.80
1.900	2.75	0.145	1.610	2.72	0.60
2⅜	4.00	0.167	2.041	3.94	1.60
2⅜	4.60	0.190	1.995	4.43	1.60
2⅜	5.80	0.254	1.867	5.75	1.40
2⅞	6.40	0.217	2.441	6.16	3.20
2⅞	8.60	0.308	2.259	8.44	2.60
3½	7.70	0.216	3.068	7.58	5.40
3½	9.20	0.254	2.992	8.81	5.00
3½	10.20	0.289	2.922	9.91	4.80
3½	12.70	0.375	2.750	12.52	4.00
4	9.50	0.226	3.548	9.11	6.20
4½	12.60	0.271	3.958	12.24	6.00

[1]Nominal weights, threads and coupling (Col. 2), are shown for the purpose of identification in ordering.

*For information only.
[2]Weight gain due to end finishing.

See Figure 4-409. Source: From Ref. [173].

3. *Wall thickness.* Each length of pipe shall be measured for conformance to wall-thickness requirements. The wall thickness at any place shall not be less than the tabulated thickness minus the permissible undertolerance of 12.5%. Wall-thickness measurements shall be made with a mechanical caliper or with a properly calibrated nondestructive testing device of

(text continued on page 1238)

Table 4-165
External Upset Tubing—Dimensions and Weights

1	2	3	4	5	6	7	8	9	10	11
				Calculated Weight			Upset			
					Threads and Coupling[3]					
Size: Outside Diameter, Pipe Body, in.	Nominal Weight:[1] Threads and Coupling, lb. per ft.	Wall Thickness, in.	Inside Diameter, in.	Plain End lb/ft	Regular lb	Special Clearance lb	Outside[2] Diameter, in. +0.0625	Length, End of Pipe to start of taper, in. +0, −1	Length, End of Pipe to gaging plane, in.	Length, End of Pipe to start of pipe body, in. max.
D		t	d	w_{pe}	e_w	e_w	D_4	L_{eu}	L_4	L_b
1.050	1.20	0.113	0.824	1.13	1.40	....	1.315	2⅜	....	....
1.315	1.80	0.133	1.049	1.68	1.40	....	1.469	2½	....	....
1.660	2.40	0.140	1.380	2.27	1.60	....	1.812	2⅝	....	....
1.900	2.90	0.145	1.610	2.72	2.00	....	2.094	2 11/16	....	....
2⅜	4.70	0.190	1.995	4.43	4.00	2.96	2.594	4	6	10
2⅜	5.95	0.254	1.867	5.75	3.60	2.56	2.594	4	6	10
2⅞	6.50	0.217	2.441	6.16	5.60	3.76	3.094	4¼	6¼	10¼
2⅞	8.70	0.308	2.259	8.44	5.00	3.16	3.094	4¼	6¼	10¼
3½	9.30	0.254	2.992	8.81	9.20	5.40	3.750	4½	6½	10½
3½	12.95	0.375	2.750	12.52	8.20	4.40	3.750	4½	6½	10½
4	11.00	0.262	3.476	10.46	10.60	....	4.250	4½	6½	10½
4½	12.75	0.271	3.958	12.24	13.20	....	4.750	4¾	6¾	10¾

[1]Nominal weights, threads and coupling (Col. 2), are shown for the purpose of identification in ordering.

Source: From Ref. [173].

[2]The minimum outside diameter of upset, D_4, is limited by the minimum length of full-crest threads. See API Std 5B.

[3]Weight gain due to end finishing.

See Figure 4-410.

Table 4-166
Integral Joint Tubing—Dimensions and Weights

1	2	3	4	5	6
				Calculated Weight	
Size: Outside Diameter, in. D	Nominal[1] Weight: Upset and Threaded, lb per ft	Wall Thickness in. t	Inside Diameter, in. d	Plain End lb/ft w_{pe}	Upset[2] and Threads lb e_w
1.315	1.72	0.133	1.049	1.68	0.20
1.660	2.10	0.125	1.410	2.05	0.20
1.660	2.33	0.140	1.380	2.27	0.20
1.900	2.40	0.125	1.650	2.37	0.20
1.900	2.76	0.145	1.610	2.72	0.20
2.063	3.25	0.156	1.751	3.18	0.20

1	2	7	8	9	10	11	12	13	14	15
		Upset Dimensions, in.								
		Pin				Box				
Size: Outside Diameter, in. D	Nominal[1] Weight: Upset and Threaded, lb per ft	Outside[3] Diameter, +0.0625 D_4	Inside[4] Diameter, +0.015 d_{iu}	Length, Min., L_{iu}	Length of Taper Min., m_{iu}	Outside Diameter, +0.005 −0.025 W_b	Length Min., L_{eu}	Length of Taper, m_{eu}	Diameter of Recess Q	Width of Face Min., b
1.315	1.72	....	.970	1 3/8	1/4	1.550	1.750	1	1.378	1/32
1.660	2.10	....	1.301	1 1/2	1/4	1.880	1.875	1	1.723	1/32
1.660	2.33	....	1.301	1 1/2	1/4	1.880	1.875	1	1.723	1/32
1.900	2.40	....	1.531	1 5/8	1/4	2.110	2.000	1	1.963	1/32
1.900	2.76	....	1.531	1 5/8	1/4	2.110	2.000	1	1.963	1/32
2.063	3.25	2.094	1.672	1 11/16	1/4	2.325	2.125	1	2.156	1/32

[1]Nominal weights, upset and threaded (Col. 2), are shown for the purpose of identification in ordering.
[2]Weight gain due to end finishing.
[3]The minimum outside diameter D_4, is limited by the minimum length of full crest threads. See API Std 5B.
[4]The minimum inside diameter, d_{iu}, is limited by the drift test.
Source: From Ref. [173].
See Figure 4-411.

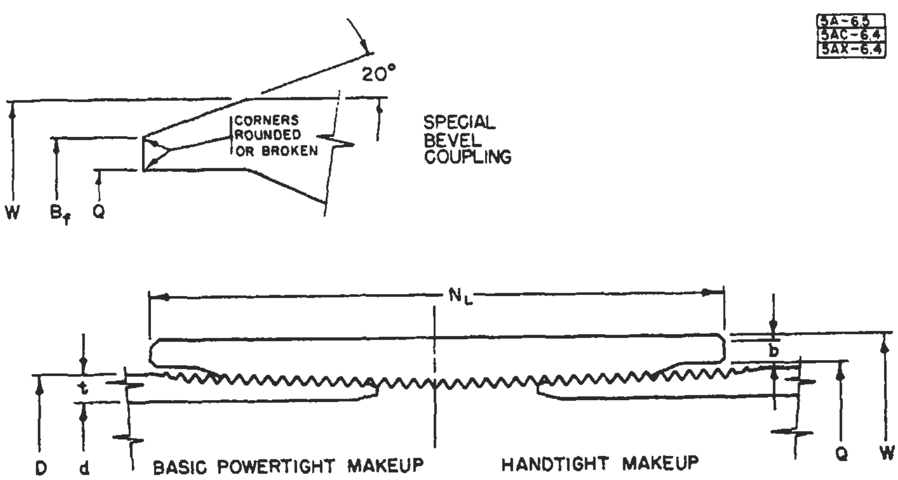


Figure 4-409. Non-upset tubing and coupling. *(From Ref. [173].)* See Table 4-164 for pipe dimensions.

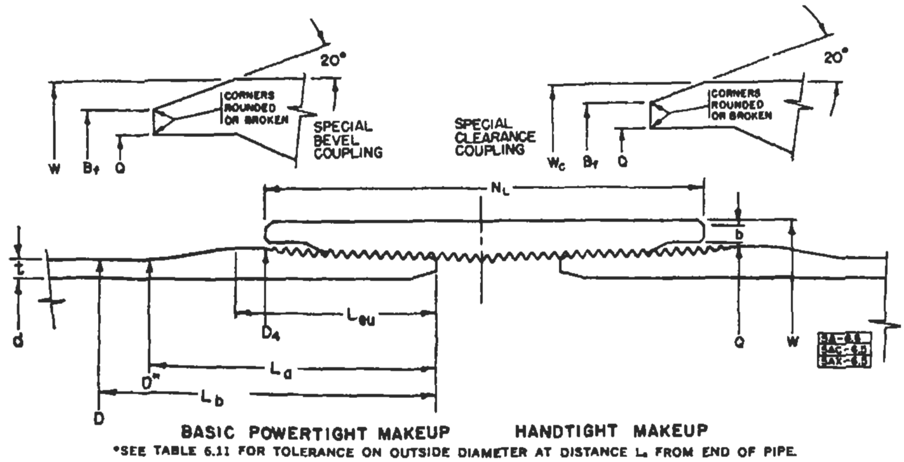


Figure 4-410. External-upset tubing and coupling. *(From Ref. [173].)* See Table 4-165 for pipe dimensions.

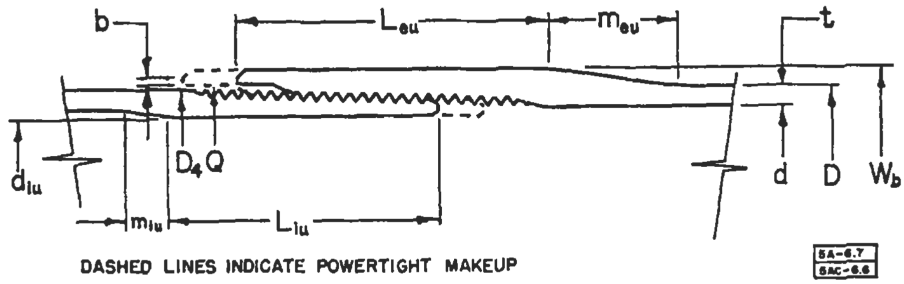


Figure 4-411. Integral-joint tubing. *(From Ref. [173].)* See Table 4-166 for pipe dimensions.

(text continued from page 1235)

appropriate accuracy. In case of dispute, the measurement determined by use of the mechanical caliper shall govern.

4. *Weight.* Each length of tubing in sizes 1.660 in. and larger shall be weighed separately. Lengths of tubing in sizes 1.050 and 1.315 in. shall be weighed either individually or in convenient lots.

   Threaded-and-coupled pipe shall be weighed with the couplings screwed on or without couplings, provided proper allowance is made for the weight of the coupling. Threaded-and-coupled pipe, integral joint pipe and pipe shipped without couplings shall be weighed without thread protectors except for carload weighings, for which proper allowances shall be made for the weight of the thread protectors.
5. Calculated weights shall be determined in accordance with the following formula:

$$W_L = (Wpe \times L) + e_W$$

where W_L = calculated weight of a piece of pipe of length L in lb (kg)
Wpe = plain-end weight in lb/ft (kg/m)
L = length of pipe, including end finish, as defined below in ft (m)
e_W = weight gain or loss due to end finishing in lb (kg) (for plain-end pipe, $e_W = 0$)

6. *Length.* Pipe shall be furnished in range lengths conforming to the following as specified on the purchase order: When pipe is furnished with threads and couplings, the length shall be measured to the outer face of the coupling, or if measured without couplings, proper allowance shall be made to include the length of the coupling. For integral joint tubing, the length shall be measured to the outer face of the box end.

Pipe Range Lengths

	Range 1	Range 2
Total range length*	20–24	28–32
Range length for 100% of carload:†		
Permissible variation, max.	2	2
Permissible length, min.	20	28

*By agreement between purchaser and manufacturer, the total range length for Range 1 tubing may be 6.10–8.53 m.

†Carload tolerances shall not apply to orders less than a carload shipped from the mill. For any carload of pipe shipped from the mill to the final destination without transfer or removal from the car, the tolerance shall apply to each car. For any order consisting of more than a carload and shipped from the mill by rail, but not to the final destination in the rail cars loaded at the mill, the carload tolerances shall apply to the total order, but not to the individual carloads.

Performance Properties

Tubing performance properties, according to API Bulletin 5C2, 18th Edition (March 1982), are given in Table 4-167. Formulas and procedures for calculating the values in Table 4-167 are given in API Bulletin 5C3, 3rd Edition (March 1980).

Running and Pulling Tubing

The following are excerpts from "API Recommended Practice for Care and Use of Casing and Tubing," API RP 5C1, 12th Edition (March 1981).

Preparation and Inspection before Running

1. New tubing is delivered free of injurious defects as defined in API Standard 5A, 5AC and 5AX, and within the practical limits of the inspection procedures therein prescribed. Some users have found that, for a limited number of critical well applications, these procedures do not result in casing sufficiently free of defects to meet their needs for such critical applications. Various nondestructive inspection services have been employed by users to assure that the desired quality of tubing is being run. In view of this practice, it is suggested that the individual user:

(text continued on page 1246)

Table 4-167
Minimum Performance Properties of Tubing

1	2	3	4	5	6	7	8	9	10	11
Size: Outside Diameter in.	Nominal Weight lb per ft: Threads and Coupling		Nominal Weight lb per ft	Grade	Wall Thickness in.	Inside Diameter in.	Threaded and Coupled: Drift Diameter in.	Threaded and Coupled: Outside Diameter of Coupling, in.		
	Non-Upset	Upset	Integral Joint					Non-Upset	Upset: Regular	Upset: Special Clearance
D					t	d		W	W	W_c
1.050	1.14	1.20	—	H-40	0.113	0.824	0.730	1.313	1.660	—
	1.14	1.20	—	J-55	0.113	0.824	0.730	1.313	1.660	—
	1.14	1.20	—	C-75	0.113	0.824	0.730	1.313	1.660	—
	1.14	1.20	—	L-80 & N-80	0.113	0.824	0.730	1.313	1.660	—
1.315	1.70	1.80	1.72	H-40	0.133	1.049	0.955	1.660	1.900	—
	1.70	1.80	1.72	J-55	0.133	1.049	0.955	1.660	1.900	—
	1.70	1.80	1.72	C-75	0.133	1.049	0.955	1.660	1.900	—
	1.70	1.80	1.72	L-80 & N-80	0.133	1.049	0.955	1.660	1.900	—
1.660	—	—	2.10	H-40	0.125	1.410	—	—	—	—
	2.30	2.40	2.33	H-40	0.140	1.380	1.286	2.054	2.200	—
	—	—	2.10	J-55	0.125	1.410	—	—	—	—
	2.30	2.40	2.33	J-55	0.140	1.380	1.286	2.054	2.200	—
	2.30	2.40	2.33	C-75	0.140	1.380	1.286	2.054	2.200	—
	2.30	2.40	2.33	L-80 & N-80	0.140	1.380	1.286	2.054	2.200	—
1.900	—	—	2.40	H-40	0.125	1.650	—	—	—	—
	2.75	2.90	2.76	H-40	0.145	1.610	1.516	2.200	2.500	—
	—	—	2.40	J-55	0.125	1.650	—	—	—	—
	2.75	2.90	2.76	J-55	0.145	1.610	1.516	2.200	2.500	—
	2.75	2.90	2.76	C-75	0.145	1.610	1.516	2.200	2.500	—
	2.75	2.90	2.76	L-80 & N-80	0.145	1.610	1.516	2.200	2.500	—

Table 4-167
(continued)

1	2	3	4	5	6	7	8	9	10	11
Size: Outside Diameter in.	Nominal Weight lb. per ft. Threads and Coupling		Nominal Weight lb. per ft.	Grade	Wall Thickness in.	Inside Diameter in.	Threaded and Coupled			
	Non-Upset	Upset	Integral Joint				Drift Diameter in.	Outside Diameter of Coupling, in. Non-Upset	Upset Regular	Upset Special Clearance
D					t	d		W	W	W_c
2.063	—	—	3.25	H-40	0.156	1.751	—	—	—	—
	—	—	3.25	J-55	0.156	1.751	—	—	—	—
	—	—	3.25	C-75	0.156	1.751	—	—	—	—
	—	—	3.25	L-80&N-80	0.156	1.751	—	—	—	—
2⅜	4.00	—	—	H-40	0.167	2.041	1.947	2.875	—	—
	4.60	4.70	—	H-40	0.190	1.995	1.901	2.875	3.063	2.910
	4.00	—	—	J-55	0.167	2.041	1.947	2.875	—	—
	4.60	4.70	—	J-55	0.190	1.995	1.901	2.875	3.063	2.910
	4.00	—	—	C-75	0.167	2.041	1.947	2.875	—	—
	4.60	4.70	—	C-75	0.190	1.995	1.901	2.875	3.063	2.910
	5.80	5.95	—	C-75	0.254	1.867	1.773	2.875	3.063	2.910
	4.00	—	—	L-80&N-80	0.167	2.041	1.947	2.875	—	—
	4.60	4.70	—	L-80&N-80	0.190	1.995	1.901	2.875	3.063	2.910
	5.80	5.95	—	L-80&N-80	0.254	1.867	1.773	2.875	3.063	2.910
	4.60	4.70	—	P-105	0.190	1.995	1.901	2.875	3.063	2.910
	5.80	5.95	—	P-105	0.254	1.867	1.773	2.875	3.063	2.910
2⅞	6.40	6.50	—	H-40	0.217	2.441	2.347	3.500	3.668	3.460
	6.40	6.50	—	J-55	0.217	2.441	2.347	3.500	3.668	3.460
	6.40	6.50	—	C-75	0.217	2.441	2.347	3.500	3.668	3.460
	8.60	8.70	—	C-75	0.308	2.259	2.165	3.500	3.668	3.460
	6.40	6.50	—	L-80&N-80	0.217	2.441	2.347	3.500	3.668	3.460
	8.60	8.70	—	L-80&N-80	0.308	2.259	2.165	3.500	3.668	3.460
	6.40	6.50	—	P-105	0.217	2.441	2.347	3.500	3.668	3.460
	8.60	8.70	—	P-105	0.308	2.259	2.165	3.500	3.668	3.460

Table 4-167 (continued)

12	13	14	15	16	17	18
Integral Joint				Joint Yield Strength, lb		
				Threaded and Coupled		
Drift Diameter	Outside Diameter of Box	Collapse Resist-ance	Internal Yield Pressure	Non-Upset	Upset	Integral Joint
in.	in. W_b	psi	psi			
—	—	7,680	7,530	6,360	13,310	—
—	—	10,560	10,360	8,740	18,290	—
—	—	14,410	14,130	11,920	24,950	—
—	—	15,370	15,070	12,710	26,610	—
0.955	1.550	7,270	7,080	10,960	19,760	15,970
0.955	1.550	10,000	9,730	15,060	27,160	21,960
0.955	1.550	13,640	13,270	20,540	37,040	29,940
0.955	1.550	14,550	14,160	21,910	39,510	31,940
1.286	1.880	5,570	5,270	—	—	22,180
1.286	1.880	6,180	5,900	15,530	26,740	22,180
1.286	1.880	7,660	7,250	—	—	30,500
1.286	1.880	8,490	8,120	21,360	36,770	30,500
1.286	1.880	11,580	11,070	29,120	50,140	41,600
1.286	1.880	12,360	11,810	31,060	53,480	44,370
1.516	2.110	4,920	4,610	—	—	26,890
1.516	2.110	5,640	5,340	19,090	31,980	26,890
1.516	2.110	6,640	6,330	—	—	36,970
1.516	2.110	7,750	7,350	26,250	43,970	36,970
1.516	2.110	10,570	10,020	35,800	59,960	50,420
1.516	2.110	11,280	10,680	38,180	63,960	53,780

Table 4-167
(continued)

12	13	14	15	16	17	18	19	20
Integral Joint			Internal Yield Pressure			Joint Yield Strength, lb.		
				Upset		Threaded and Coupled		
Drift Diameter in.	Outside Diameter of Box in. W_b	Collapse Resistance psi	Plain-end and Non-Upset psi	Regular Coupling psi	Special Clearance Coupling psi	Non-Upset	Upset	Integral Joint
1.657	2.325	5,590	5,290	—	—	—	—	35,700
1.657	2.325	7,690	7,280	—	—	—	—	49,000
1.657	2.325	10,480	9,920	—	—	—	—	66,900
1.657	2.325	11,180	10,590	—	—	—	—	71,400
—	—	5,230	4,920	—	—	30,100	—	—
—	—	5,890	5,600	5,600	5,600	36,000	52,200	—
—	—	7,190	6,770	—	—	41,400	—	—
—	—	8,100	7,700	7,700	7,700	49,500	71,700	—
—	—	9,520	9,230	—	—	56,500	—	—
—	—	11,040	10,500	10,500	10,500	67,400	97,800	—
—	—	14,330	14,040	13,960	10,720	96,600	126,900	—
—	—	9,980	9,840	—	—	60,300	—	—
—	—	11,780	11,200	11,200	11,200	71,900	104,300	—
—	—	15,280	14,970	14,890	11,440	103,000	135,400	—
—	—	15,460	14,700	14,000	14,700	94,400	136,900	—
—	—	20,060	19,650	19,540	15,010	135,200	177,700	—
—	—	5,580	5,280	5,280	5,510	52,800	72,500	—
—	—	7,680	7,260	7,200	7,260	72,600	99,700	—
—	—	10,470	9,910	9,910	9,910	99,000	135,900	—
—	—	14,350	14,060	14,010	10,340	149,400	186,300	—
—	—	11,160	10,570	10,570	10,570	105,600	145,000	—
—	—	15,300	15,000	14,940	11,030	159,300	198,700	—
—	—	14,010	13,870	13,870	13,870	138,600	190,300	—
—	—	20,090	19,690	19,610	14,480	209,100	260,800	—

Table 4-167
(continued)

1	2	3	4	5	6	7	8	9	10	11
Size: Outside Diameter in.	Nominal Weight lb. per ft. Threads and Coupling		Nominal Weight lb. per ft.	Grade	Wall Thickness in.	Inside Diameter in.	Threaded and Coupled			
	Threads and Coupling		Integral Joint				Drift Diameter in.	Outside Diameter of Coupling, in.		
									Upset	
	Non-Upset	Upset						Non-Upset	Regular	Special Clearance
D					t	d		W	W	W_c
3½	7.70	—	—	H-40	0.216	3.068	2.943	4.250	—	—
	9.20	9.30	—	H-40	0.254	2.992	2.867	4.250	4.500	4.180
	10.20	—	—	H-40	0.289	2.922	2.797	4.250	—	—
	7.70	—	—	J-55	0.216	3.068	2.943	4.250	—	—
	9.20	9.30	—	J-55	0.254	2.992	2.867	4.250	4.500	4.180
	10.20	—	—	J-55	0.289	2.922	2.797	4.250	—	—
	7.70	—	—	C-75	0.216	3.068	2.943	4.250	—	—
	9.20	9.30	—	C-75	0.254	2.992	2.867	4.250	4.500	4.180
	10.20	—	—	C-75	0.289	2.922	2.797	4.250	—	—
	12.70	12.95	—	C-75	0.375	2.750	2.625	4.250	4.500	4.180
	7.70	—	—	L-80&N-80	0.216	3.068	2.943	4.250	—	—
	9.20	9.30	—	L-80&N-80	0.254	2.992	2.867	4.250	4.500	4.180
	10.20	—	—	L-80&N-80	0.289	2.922	2.797	4.250	—	—
	12.70	12.95	—	L-80&N-80	0.375	2.750	2.625	4.250	4.500	4.180
	9.20	9.30	—	P-105	0.254	2.992	2.867	4.250	4.500	4.180
	12.70	12.95	—	P-105	0.375	2.750	2.625	4.250	4.500	4.180
4	9.50	—	—	H-40	0.226	3.548	3.423	4.750	—	—
	—	11.00	—	H-40	0.262	3.476	3.351	—	5.000	—
	9.50	—	—	J-55	0.226	3.548	3.423	4.750	—	—
	—	11.00	—	J-55	0.262	3.476	3.351	—	5.000	—
	9.50	—	—	C-75	0.226	3.548	3.423	4.750	—	—
	—	11.00	—	C-75	0.262	3.476	3.351	—	5.000	—
	9.50	—	—	L-80&N-80	0.226	3.548	3.423	4.750	—	—
	—	11.00	—	L-80&N-80	0.262	3.476	3.351	—	5.000	—
4½	12.60	12.75	—	H-40	0.271	3.958	3.833	5.200	5.563	—
	12.60	12.75	—	J-55	0.271	3.958	3.833	5.200	5.563	—
	12.60	12.75	—	C-75	0.271	3.958	3.833	5.200	5.563	—
	12.60	12.75	—	L-80&N-80	0.271	3.958	3.833	5.200	5.563	—

Table 4-167
(continued)

12	13	14	15	16	17	18	19	20
Integral Joint			Internal Yield Pressure			Joint Yield Strength, lb.		
				Upset		Threaded and Coupled		
Drift Diameter	Outside Diameter of Box	Collapse Resistance	Plain-end and Non-Upset	Regular Coupling	Special Clearance Coupling	Non-Upset	Upset	Integral Joint
in.	in. W_b	psi	psi	psi	psi			
—	—	4,630	4,320	—	—	65,100	—	—
—	—	5,380	5,080	5,080	5,080	79,500	103,600	—
—	—	6,060	5,780	—	—	92,600	—	—
—	—	5,970	5,940	—	—	89,500	—	—
—	—	7,400	6,990	6,990	6,990	109,400	142,500	—
—	—	8,330	7,950	—	—	127,300	—	—
—	—	7,540	8,100	—	—	122,000	—	—
—	—	10,040	9,530	9,530	9,530	149,100	194,300	—
—	—	11,360	10,840	—	—	173,500	—	—
—	—	14,350	14,060	14,060	9,990	231,000	276,100	—
—	—	7,870	8,640	—	—	130,100	—	—
—	—	10,530	10,160	10,160	10,160	159,100	207,200	—
—	—	12,120	11,560	—	—	185,100	—	—
—	—	15,310	15,000	15,000	10,660	246,400	294,500	—
—	—	13,050	13,340	13,340	13,340	208,900	272,000	—
—	—	20,090	19,690	19,690	13,990	323,400	386,600	—
—	—	4,060	3,960	—	—	72,000	—	—
—	—	4,900	4,590	4,590	—	—	123,100	—
—	—	5,110	5,440	—	—	99,000	—	—
—	—	6,590	6,300	6,300	—	—	169,200	—
—	—	6,350	7,420	—	—	135,000	—	—
—	—	8,410	8,600	8,600	—	—	230,800	—
—	—	6,590	7,910	—	—	144,000	—	—
—	—	8,800	9,170	9,170	—	—	246,100	—
—	—	4,500	4,220	4,220	—	104,400	144,000	—
—	—	5,720	5,800	5,800	—	143,500	198,000	—
—	—	7,200	7,900	7,900	—	195,700	270,000	—
—	—	7,500	8,430	8,430	—	208,700	288,000	—

Source: From Ref. [174].

(text continued from page 1239)

a. Familiarize her or himself with inspection practices specified in the standards and employed by the respective mills and with the definition of "injurious defect" contained in the standards.
b. Thoroughly evaluate any nondestructive inspection to be used by him or her on API tubular goods to ensure that the inspection does in fact correctly locate and differentiate injurious defects from other variables that can be and frequently are sources of misleading "defect" signals with such inspection methods.

Caution: Due to the permissible tolerance on the outside diameter immediately behind the tubing upset, the user is cautioned that difficulties may occur when wraparound seal-type hangers are installed on tubing manufactured on the high side of the tolerance; therefore, it is recommended that the user select the joint of tubing to be installed at the top of the string.

2. All tubing, whether new, used or reconditioned, should always be handled with thread protectors in place. Tubing should be handled at all times on racks or on wooden or metal surfaces free of rocks, sand or dirt other than normal drilling mud. When lengths of tubing are inadvertently dragged in the dirt, the threads should be recleaned.
a. Before running in the hole for the first time, tubing should be drifted with an API drift mandrel to ensure passage of pumps, swabs and packers.
b. Elevators should be in good repair and should have links of equal length.
c. Slip-type elevators are recommended when running special clearance couplings, especially those beveled on the lower end.
d. Elevators should be examined to note if latch fitting is complete.
e. Spider slips that will not crush the tubing should be used. Slips should be examined before using to see that they are working together.

Note: Slip and tong marks are injurious. Every possible effort should be made to keep such damage at a minimum by using proper up-to-date equipment.

f. Tubing tongs that will not crush the tubing should be used on the body of the tubing and should fit properly to avoid unnecessary cutting of the pipe wall. Tong dies should fit properly and conform to the curvature of the tubing. The use of pipe wrenches is not recommended.
g. The following precautions should be taken in the preparation of tubing threads:

1. Immediately before running, remove protectors from both field end and coupling end and clean threads thoroughly, repeating as additional rows become uncovered.
2. Carefully inspect the threads. Those found damaged, even slightly, should be laid aside unless satisfactory means are available for correcting thread damage.
3. The length of each piece of tubing should be measured prior to running. A steel tape calibrated in decimal feet to the nearest 0.01 ft should be used. The measurement should be made from the outermost face of the coupling or box to the position on the externally threaded end where the coupling or the box stops when the joint is made up powertight. The total of the individual length so measured will represent the unloaded length of the tubing string.
4. Place clean protectors on the field end of the pipe so that the thread will not be damaged while rolling the pipe onto the rack and pulling it into the derrick.
5. Check each coupling for makeup. If the stand off is abnormally great, check the coupling for tightness. Loose couplings should be removed,

the threads thoroughly cleaned and fresh compound should be applied over the entire thread surfaces. Then the coupling should be replaced and tightened before pulling the tubing into the derrick.
6. Before stabbing, liberally apply thread compound to the entire internally and externally threaded areas. It is recommended that high-pressure, modified thread compound as specified in API Bulletin 5A2: "Bulletin on Thread Compounds" be used except in special cases where severe conditions are encountered. In these special cases it is recommended that high-pressure silicone thread compound as specified in Bulletin 5A2 be used.

h. For high-pressure or condensate wells, additional precautions to insure tight joints should be taken as follows:
1. Couplings should be removed and both the mill-end pipe thread and coupling thread thoroughly cleaned and inspected. To facilitate this operation, tubing may be ordered with couplings handling tight, which is approximately one turn beyond hand tight, or may be ordered with the couplings shipped separately.
2. Thread compound should be applied to both the external and internal threads, and the coupling should be reapplied handling tight. Field-end threads and the mating coupling threads should have thread compounds applied just before stabbing.

Stabbing, Making Up and Lowering

1. Do not remove thread protector from field end of tubing until ready to stab.
2. If necessary, apply thread compound over entire surface of threads just before stabbing.
3. In stabbing, lower tubing carefully to avoid injuring threads. Stab vertically, preferably with the assistance of someone on a stabbing board. If the tubing tilts to one side after stabbing, lift up, clean and correct any damaged thread with a three-cornered file, then carefully remove any filings and reapply compound over thread surface. Intermediate supports may be placed in the derrick to limit bowing of the tubing.
4. After stabbing, start screwing the pipe together by hand or apply regular or power tubing tongs slowly. To prevent galling when making up connections in the field, the connections should be made up to a speed not to exceed 25 rpm. Power tubing tongs are recommended for high-pressure or condensate wells to ensure uniform makeup and tight joints. Joints should be made up tight, approximately two turns beyond the hand-tight position, with care being taken not to gall the threads.

Field Makeup

1. Joint life of tubing under repeated field makeup is inversely proportional to the field makeup torque applied. Therefore, in wells where leak resistance is not a great factor, minimum field makeup torque values should be used to prolong joint life. Table 4-168 contains recommended optimum makeup torque values for nonupset, external upset, and integral joint tubing, based on 1% of the calculated joint pullout strength determined from the joint pull-out strength formula for eight-round thread casing in Bulletin 5C3. Minimum torque values listed are 75% of optimum values, and maximum torque values listed are 125% of optimum values. All values are rounded to the nearest 10 ft-lb. The torque values listed in Table 4-168

Table 4-168
Recommended Tubing Makeup Torque

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Size: Outside Diameter In.	Nominal Weight lb. per ft.			Grade	Torque, ft-lb								
	Threads and Coupling		Integral Joint		Non-Upset			Upset			Integral Joint		
	Non-Upset	Upset			Opt.	Min.	Max.	Opt.	Min.	Max.	Opt.	Min.	Max.
1.050	1.14	1.20	—	H-40	140	110	180	460	350	580	—	—	—
	1.14	1.20	—	J-55	180	140	280	600	450	750	—	—	—
	1.14	1.20	—	C-75	230	170	290	780	590	980	—	—	—
	1.14	1.20	—	L-80	240	180	300	810	610	1010	—	—	—
	1.14	1.20	—	N-80	250	190	310	830	620	1040	—	—	—
1.315	1.70	1.80	1.72	H-40	210	160	260	440	330	550	310	230	390
	1.70	1.80	1.72	J-55	270	200	340	570	430	710	400	300	500
	1.70	1.80	1.72	C-75	360	270	450	740	560	930	520	390	650
	1.70	1.80	1.72	L-80	370	280	460	760	570	950	540	400	680
	1.70	1.80	1.72	N-80	380	290	480	790	590	990	550	410	690
1.660	—	—	2.10	H-40	—	—	—	—	—	—	380	280	480
	2.30	2.40	2.33	H-40	270	200	340	530	400	660	380	280	480
	—	—	2.10	J-55	—	—	—	—	—	—	500	380	630
	2.30	2.40	2.33	J-55	350	260	440	690	520	860	500	880	630
	2.30	2.40	2.33	C-75	460	350	580	910	680	1140	650	490	810
	2.30	2.40	2.33	L-80	470	350	590	940	710	1180	670	500	850
	2.30	2.40	2.33	N-80	490	370	610	960	720	1200	690	520	860
1.900	—	—	2.40	H-40	—	—	—	—	—	—	450	340	560
	2.75	2.90	2.76	H-40	320	240	400	670	500	840	450	840	560
	—	—	2.40	J-55	—	—	—	—	—	—	580	440	730
	2.75	2.90	2.76	J-55	410	310	510	880	660	1100	580	440	730
	2.75	2.90	2.76	C-75	540	410	680	1150	860	1440	760	570	950
	2.75	2.90	2.76	L-80	560	420	700	1190	890	1490	790	590	990
	2.75	2.90	2.76	N-80	570	430	710	1220	920	1530	810	610	1010

apply only to tubing with zinc-plated couplings. When making up connections with tin-plated couplings, 80% of the listed value can be used as a guide.

2. Spider slips and elevators should be cleaned frequently, and slips should be kept sharp.
3. Finding bottom should be accomplished with extreme caution. Do not set tubing down heavily.

Pulling Tubing

1. A caliper survey prior to pulling a worn string of tubing will provide a quick means of segregating badly worn lengths for removal.
2. Break-out tongs should be positioned close to the coupling. Hammering the coupling to break the joint is an injurious practice.
3. Great care should be exercised to disengage all of the thread before lifting the tubing out of the coupling. Do not jump the tubing out of the coupling.
4. Tubing stacked in the derrick should be set on a firm wooden platform without the bottom thread protector since the design of most protectors is not such as to support the joint or stand without damage to the field thread.

Table 4-168
(continued)

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Size: Outside Diameter in.	Nominal Weight lb. per ft.			Grade	Torque, ft-lb								
	Threads and Coupling		Integral Joint		Non-Upset			Upset			Integral Joint		
	Non-Upset	Upset			Opt.	Min.	Max.	Opt.	Min.	Max.	Opt.	Min.	Max.
2.063	—	—	3.25	H-40	—	—	—	—	—	—	570	430	710
	—	—	3.25	J-55	—	—	—	—	—	—	740	560	920
	—	—	3.25	C-75	—	—	—	—	—	—	970	730	1210
	—	—	3.25	L-80	—	—	—	—	—	—	1010	760	1260
	—	—	3.25	N-80	—	—	—	—	—	—	1030	770	1290
2⅜	4.00	—	—	H-40	470	350	590	—	—	—	—	—	—
	4.60	4.70	—	H-40	560	420	700	990	740	1240	—	—	—
	4.00	—	—	J-55	610	460	760	—	—	—	—	—	—
	4.60	4.70	—	J-55	730	550	910	1290	970	1610	—	—	—
	4.00	—	—	C-75	800	600	1000	—	—	—	—	—	—
	4.60	4.70	—	C-75	960	720	1200	1700	1280	2130	—	—	—
	5.80	5.95	—	C-75	1380	1040	1730	2120	1590	2650	—	—	—
	4.00	—	—	L-80	830	620	1040	—	—	—	—	—	—
	4.60	4.70	—	L-80	990	740	1240	1760	1320	2200	—	—	—
	5.80	5.95	—	L-80	1420	1070	1780	2190	1640	2740	—	—	—
	4.00	—	—	N-80	850	640	1060	—	—	—	—	—	—
	4.60	4.70	—	N-80	1020	770	1280	1800	1350	2250	—	—	—
	5.80	5.95	—	N-80	1460	1100	1830	2240	1680	2800	—	—	—
	4.60	4.70	—	P-105	1280	960	1600	2270	1700	2840	—	—	—
	5.80	5.95	—	P-105	1840	1380	2300	2830	2120	3540	—	—	—
2⅞	6.40	6.50	—	H-40	800	600	1000	1250	940	1560	—	—	—
	6.40	6.50	—	J-55	1050	790	1310	1650	1240	2060	—	—	—
	6.40	6.50	—	C-75	1380	1040	1730	2170	1630	2710	—	—	—
	8.60	8.70	—	C-75	2090	1570	2610	2850	2140	3560	—	—	—
	6.40	6.50	—	L-80	1430	1070	1790	2250	1690	2810	—	—	—
	8.60	8.70	—	L-80	2160	1620	2700	2950	2210	3690	—	—	—
	6.40	6.50	—	N-80	1470	1100	1840	2300	1730	2880	—	—	—
	8.60	8.70	—	N-80	2210	1660	2760	3020	2270	3780	—	—	—
	6.40	6.50	—	P-105	1850	1390	2310	2910	2180	3640	—	—	—
	8.60	8.70	—	P-105	2790	2090	3490	3810	2860	4760	—	—	—

5. Protect threads from dirt or injury when the tubing is out of the hole.
6. Tubing set back in the derrick should be properly supported to prevent undue bending. Tubing that is $2\frac{3}{8}$-in. OD and larger preferably should be pulled in stands approximately 60 ft long or in doubles of range 2. Stands of tubing 1.900-in. OD or smaller and stands longer than 60 ft should have intermediate support.

Causes of Tubing Trouble

The more common causes of tubing troubles are as follows:

1. Improper selection for strength and life required.
2. Insufficient inspection of finished product at the mill and in the yard.
3. Careless loading, unloading and cartage
4. Damaged threads resulting from protectors loosening and falling off.
5. Lack of care in storage to give proper protection.
6. Excessive hammering on couplings.

Table 4-168 (continued)

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Size: Outside Diameter in.	Nominal Weight lb. per ft.			Grade	Torque, ft-lb								
	Threads and Coupling		Integral Joint		Non-Upset			Upset			Integral Joint		
	Non-Upset	Upset			Opt.	Min.	Max.	Opt.	Min.	Max.	Opt.	Min.	Max.
3½	7.70	—	—	H-40	920	690	1150	—	—	—	—	—	—
	9.20	9.30	—	H-40	1120	840	1400	1730	1300	2160	—	—	—
	10.20	—	—	H-40	1310	980	1640	—	—	—	—	—	—
	7.70	—	—	J-55	1210	910	1510	—	—	—	—	—	—
	9.20	9.30	—	J-55	1480	1110	1850	2280	1710	2850	—	—	—
	10.20	—	—	J-55	1720	1290	2150	—	—	—	—	—	—
	7.70	—	—	C-75	1600	1200	2000	—	—	—	—	—	—
	9.20	9.30	—	C-75	1950	1460	2440	3010	2260	3760	—	—	—
	10.20	—	—	C-75	2270	1700	2840	—	—	—	—	—	—
	12.70	12.95	—	C-75	3030	2270	3790	4040	3030	5050	—	—	—
	7.70	—	—	L-80	1660	1250	2080	—	—	—	—	—	—
	9.20	9.30	—	L-80	2030	1520	2540	3130	2350	3910	—	—	—
	10.20	—	—	L-80	2360	1770	2950	—	—	—	—	—	—
	12.70	12.95	—	L-80	3140	2360	3930	4200	3150	5250	—	—	—
	7.70	—	—	N-80	1700	1280	2130	—	—	—	—	—	—
	9.20	9.30	—	N-80	2070	1550	2590	3200	2400	4000	—	—	—
	10.20	—	—	N-80	2410	1810	3010	—	—	—	—	—	—
	12.70	12.95	—	N-80	3210	2410	4010	4290	3220	5360	—	—	—
	9.20	9.30	—	P-105	2620	1970	3280	4050	3040	5060	—	—	—
	12.70	12.95	—	P-105	4060	3050	5080	5430	4070	6790	—	—	—
4	9.50	—	—	H-40	940	710	1180	—	—	—	—	—	—
	—	11.00	—	H-40	—	—	—	1940	1460	2430	—	—	—
	9.50	—	—	J-55	1240	930	1550	—	—	—	—	—	—
	—	11.00	—	J-55	—	—	—	2560	1920	3200	—	—	—
	9.50	—	—	C-75	1640	1230	2050	—	—	—	—	—	—
	—	11.00	—	C-75	—	—	—	3390	2540	4240	—	—	—
	9.50	—	—	L-80	1710	1280	2140	—	—	—	—	—	—
	—	11.00	—	L-80	—	—	—	3530	2650	4410	—	—	—
	9.50	—	—	N-80	1740	1310	2180	—	—	—	—	—	—
	—	11.00	—	N-80	—	—	—	3600	2700	4500	—	—	—
4½	12.60	12.75	—	H-40	1320	990	1650	2160	1620	2700	—	—	—
	12.60	12.75	—	J-55	1740	1310	2180	2860	2150	3580	—	—	—
	12.60	12.75	—	C-75	2300	1730	2880	3780	2840	4730	—	—	—
	12.60	12.75	—	L-80	2400	1800	3000	3940	2960	4930	—	—	—
	12.60	12.75	—	N-80	2440	1830	3050	4020	3020	5030	—	—	—

Source: From Ref. [175].

7. Use of worn-out and wrong types of handling equipment.
8. Nonobservance of proper rules in running and pulling tubing.
9. Coupling wear and rod cutting.
10. Excessive sucker rod breakage.
11. Fatigue that often causes failure at the last engaged thread. There is no positive remedy, but using external upset tubing in place of nonupset tubing greatly delays the start of this trouble.
12. Replacement of worn couplings with non-API couplings.
13. Dropping a string, even a short distance.
14. Leaky joints, under external or internal pressure.

15. Corrosion. Both the inside and outside of tubing can be damaged by corrosion. The damage is generally in the form of pitting, box wear, stress-corrosion cracking, and sulfide stress cracking, but localized attack like corrosion–erosion, ringworm and caliper tracks—can also occur. Since corrosion can result from many causes and influences and can take different forms, no simple and universal remedy can be given for control. Each problem must be treated individually, and the solution must be attempted in light of known factors and operating conditions.
    a. Where internal or external tubing corrosion is known to exist and corrosive fluids are being produced, the following measures can be employed:
        1. In flowing wells the annulus can be packed off and the corrosive fluid confined to the inside of the tubing. The inside of the tubing can be protected with special liners, coatings or inhibitors. Under severe conditions, special alloy steel or glass-reinforced plastics may be used. Alloys do not eliminate corrosion. When H_2S is present in the well fluids, tubing of high-yield strength may be subject to sulfide corrosion cracking. The concentration of H_2S necessary to cause cracking in different strength materials is not yet well defined. Literature on sulfide corrosion or persons competent in this field should be consulted.
        2. In pumping and gas-lifting wells, inhibitors introduced via the casing-tubing annulus afford appreciable protection. In this type of completion, especially in pumping wells, better operating practices can also aid in extending the life of tubing; viz., through the use of rod protectors, rotation of tubing and longer and slower pumping strokes.
    b. To determine the value and effectiveness of the above practices and measures, cost and equipment failure records can be compared before and after application of control measures.
    c. In general, all new areas should be considered as being potentially corrosive, and investigations should be initiated early in the life of a field, and repeated periodically, to detect and localize corrosion before it has done destructive damage. Where conditions favorable to corrosion exist, a qualified corrosion engineer should be consulted.

Selection of Wall Thickness and Steel Grade of Tubing

Tubing design relies on the selection of the most economical steel grades and wall thicknesses (unit weight of tubing) that will withstand, without failure, the forces to which tubing will be exposed throughout the expected tubing life.

Tubing must be designed on:

- collapse
- burst
- tension
- possibility of permanent corkscrewing

Tubing string design is very much the same as for casing. For shallow and moderately deep holes, uniform strings are preferable; however, in deep wells, a tapered tubing can be desirable. A design factor (safety factor) for tension should be about 1.6. The collapse design factor must not be less than 1.0, assuming an annulus filled up with fluid and tubing empty inside. The design factor for burst should not be less than 1.1.

Tubing Elongation/Contraction Due to the Effect of Changes in Pressure and Temperature

During the service life of a well, tubing can experience various combinations of pressures and temperatures that result in tubing length changes. The four basic effects to consider are as follows:

1. piston effect
2. helical buckling effect
3. ballooning and reverse ballooning effect
4. temperature change effect

The tubing movement due to piston effect is

$$\Delta L_1 = \frac{\left[(A_p - A_i)\Delta P_i - (A_p - A_o)\Delta P_o\right]}{EA_s} L \tag{4-333}$$

Tubing movement due to helical buckling is

$$\Delta L_2 = -\frac{r^2 F_f^2}{8EI(W_s + W_i - W_o)} \tag{4-334}$$

where

$$I = \frac{\pi}{64}(OD^4 - ID^4)$$

$$F_f = (\Delta P_i - \Delta P_o)A_p$$

Note: If $F_f < 0$, $\Delta L_2 = 0$.

The tubing movement due to ballooning effect is

$$\Delta L_3 = -\frac{\nu}{E}\frac{\Delta\rho_i - R^2\Delta\rho_o - \frac{1+2\nu}{2\nu}\delta}{R^2 - 1} L^2 - \frac{2\nu}{E}\frac{\Delta p_i - R^2\Delta p_o}{R^2 - 1} L \tag{4-335}$$

The tubing movement due to change in temperature is

$$\Delta L_4 = \beta L \Delta T \tag{4-336}$$

Two approaches can be used to handle tubing movement:

1. Provide seals of enough length.
2. Slack off enough weight of tubing to prevent movement.

For practical purposes a combination of the approaches mentioned above can be applicable.

If slack off weight is applied, the tubing compensating movement is calculated from

$$\Delta L_5 = \frac{LF_s}{EA_s} + \frac{r^2F_s^2}{8EI(W_s + W_i - W_o)} \tag{4-337}$$

Notations used in the above equations are

A_i = area corresponding to tubing ID in in.^{2}
A_o = area corresponding to tubing OD in in.^{2}
A_p = area corresponding to packer bore in in.^{2}
A_s = cross-sectional area of the tubing wall in in.^{2}
F_f = fictitious force in presence of no restraint in the packer in lb_f
I = moment of inertia of tubing cross-section with respect to its diameter in in.^{4}
L = length of tubing in in.
P_i, P_o = pressure inside and outside the tubing at the packer level respectively in psi
ΔP_i, ΔP_o = change in pressure inside and outside of tubing at the packer level in psi
Δp_i, Δp_o = change in pressure inside and outside of tubing at the surface in psi
R = ratio OD/ID of the tubing
r = tubing and casing radial clearance in in.
$\Delta\rho_i$ = change in density of liquid in the tubing in lbm/in.^{3}
$\Delta\rho_o$ = change in density of liquid in annulus in lbm/in.^{3}
β = coefficient of thermal expansion of the tubing material (for steel, $\beta = 6.9 \times 10^{-6}/°F$)
W_s = average (i.e., including coupling) weight of tubing per unit length in lb/in.
W_i = weight of liquid in the tubing per unit length in lb/in.
W_o = weight of outside liquid displaced per unit length
δ = drop of pressure in the tubing due to flow per unit length in psi/in.
ΔT = change in average tubing temperature in °F
ν = Poisson's ratio of the tubing material (for steel, $\nu = 0.3$)
E = Young's modulus (for steel, $E = 30 \times 10^6$ psi)

Nomenclature used is as in the paper by A. Lubinski et al. [171].

Example 1

Calculate the expected movement of tubing under conditions as specified below. Initially both tubing and annulus are filled with a crude of 30°API. Thereafter, the crude in the tubing is replaced by a 15-lb/gal cement slurry to perform a squeeze cementing operation. While the squeeze cementing job is performed, pressures p_i = 5,000 psi and p_o = 1,000 psi are applied at the surface on the tubing and annulus respectively.

Tubing: $2\frac{7}{8}$ in; 6.5 lb/ft
Casing: 7 in.; 32 lb/ft (r = 1.61 in.)
A_o = 6.49 in.^{2}, A_i = 4.68 in.^{2}, A_s = 1.81 in.^{2}
Ratio of OD/ID of tubing, R = 1.178

A_p = 8.30 in.^{2}
Length of tubing = 10,000 ft (120,000 in.)
Average change in temperature: -20°F
Pressure drop due to flow is disregarded (δ = 0)

Solution

Pressure changes are the following. At the surface the pressures are Δp_i = 5,000 psi and Δp_o = 1,000 psi. At the packer level, ΔP_i = 9,000 psi and ΔP_o = 1,000 psi. Liquid density changes are $\Delta \rho_i$ = 0.0332 psi/in. and $\Delta \rho_o$ = 0.0 psi/in. The unit weight of tubing is

$$W = W_s + W_i - W_o = \frac{6.5}{12} + \frac{1.5 \times 4.68}{231} - \frac{0.876 \times 8.34 \times 6.49}{231} = 0.64 \text{ lb/in.}$$

The moment of inertia is I = 1.61 in.^{4} The tubing movement due to piston effect is

$$\Delta L_1 = \frac{120,000}{30 \times 10^6}[(8.3 - 4.68)9,000 - (8.3 - 6.49)1,000] = -68.0 \text{ in.}$$

The tubing movement due to buckling effect is

$$\Delta L_2 = \frac{(1.61^2) \times (8.3^2)(9,000 - 1,000)^2}{8 \times 30 \times 10^6 \times 1.61 \times 0.64} = -46.23 \text{ in.}$$

The tubing movement due to ballooning effect is

$$L_3 = \frac{0.3}{30 \times 10^6}\frac{0.0332 - 0}{1.178^2 - 1}(120,000)^2 - \frac{2 \times 0.3}{30 \times 10^6}$$

$$\frac{5,000 - (1.178^2)1,000}{1.178^2 - 1}120,000 = -34.69 \text{ in.}$$

The temperature effect is

ΔL_4 = -6.9 × 10^{-6} × 120,000 × 20 = -16.56 in.

The total expected tubing movement is

$\Delta L_6 = \Delta L_1 + \Delta L_2 + \Delta L_3 + \Delta L_4$ = -165.48 in.

Packer-To-Tubing Force

Certain types of packers permit no tubing motion in either direction. Depending upon operational conditions, a tubing can be landed either in compression (slack off) or tension (pull up). Landing tubing in compression is desirable if the expected tubing movement would produce tubing shortening while landing in tension to compensate the expected tubing elongation.

Restraint of the tubing in the packer results in a packer-to-tubing force. To find the expected packer-to-tubing force, the following sequence of calculations is applicable:

$$F_f = A_p (P_i - P_o) \tag{4-338}$$

$$F_a = (A_p - A_i)P_i - (A_p - A_o)P_o \tag{4-339}$$

$$\Delta L_p = -\Delta L_6 \tag{4-340}$$

$$\Delta L_f = -\frac{LF_f}{EA_s} - \frac{r^2F_f^2}{8EI(W_s + W_i - W_o)} \tag{4-341}$$

$$\Delta\hat{L}_f = \Delta L_f + \Delta L_p \tag{4-342}$$

If $\Delta\hat{L}_f$ is positive, then

$$\hat{F}_f = -\frac{\Delta\hat{L}_f EA_s}{L} \tag{4-343}$$

If $\Delta\hat{L}_f$ is negative, then

$$\hat{F}_f = \frac{4I(W_s + W_i - W_o)}{A_s r^2}\left[-L \pm \left(L^2 - \frac{A_s^2 r^2 E\Delta\hat{L}_f}{2I(W_s + W_i - W_o)}\right)^{0.5}\right] \tag{4-344}$$

and finally

$$F_p = \hat{F}_f - F_f \tag{4-345}$$

Upon determining the packer to tubing force F_p, the actual force $\hat{F}_a$ immediately above the packer is given by

$$\hat{F}_a = F_a + F_p \tag{4-346}$$

The symbols used are

F_a = actually existing pressure force at the lower end of tubing subjected to no restraint in the packer in lb_f
F_f = fictitious force in presence of no restraint in the packer in lb_f
$\hat{F}_a$ = actually existing force at the lower end of tubing in lb_f
$\hat{F}_f$ = fictitious force in presence of packer restraint in lb_f
ΔL_6 = overall tubing length change in in.
ΔL_p = length change necessary to bring the end of the tubing to the packer

Other symbols are as previously used.

Example 2

The operating conditions are the same as those described in Example 1. Assume that the packer does not permit any tubing movement at the packer setting depth (10,000 ft). Since tubing shortening is expected, a 20,000-lb force is slacked off before the squeeze job. Find the tubing-to-packer force.

Solution

Length change due to slack-off force is

$$\Delta L_s = \frac{120,000 \times 20,000}{30 \times 10^6 \times 1.81} + \frac{(1.61)^2 \times (20,000^2)}{8 \times 30 \times 10^6 \times 1.61 \times 0.64} = 48.39 \text{ in.}$$

The overall tubing length change is

$$\Delta L_6 = -165.48 + 48.39 = -117.09 \text{ in.}$$

The fictitious and actual forces are

$$F_f = 8.3(12,800 - 4,800) = 66,400 \text{ lb}_f$$

$$F_a = 12,800(8.3 - 4.68) - 4,800(8.3 - 6.49) = 37,648 \text{ lb}_f$$

Note: A positive sign indicates a compressive-type force, a negative sign indicates a tensional force.

$$\Delta L_p = -\Delta L_6 = 115.5 \text{ in.}$$

$$\Delta \hat{L}_f = -\frac{120,000 \times 66,400}{30 \times 10^6 \times 1.81} - \frac{(1.61^2) \times (66,400^2)}{8 \times 30 \times 10^6 \times 1.61 \times 0.64} = -192.9 \text{ in.}$$

$$\Delta \hat{L}_f = -192.9 + 115.5 = -77.4 \text{ in.}$$

Since $\Delta \hat{L}_f$ is negative, the force $\hat{F}_f$ is calculated from Equation 4-344:

$$\hat{F}_f = \frac{4 \times 1.61 \times 0.66}{1.81 \times (1.61^2)} \left[-120,000 + \left((120,000)^2 - \frac{(1.81)^2 (1.61^2) \times 30 \times 10^6 (-77.4)}{2 \times 1.61 \times 0.64} \right)^{0.5} \right] = 30,050.45 \text{ lb}_f$$

Since the force $\hat{F}_f$ is positive (compressive), a helical buckling of tubing is expected above the packer.

The tubing-to-packer force is

$$F_p = \hat{F}_f - F_f = 30,050.45 - 66,400 = 36,349.55 \text{ lb}_f$$

Permanent Corkscrewing

To ensure that permanent corkscrewing will not occur, the following inequalities must be satisfied:

$$\left[3\left(\frac{P_i - P_o}{R^2 - 1}\right)^2 + \left(\frac{P_i - R^2 P_o}{R^2 - 1} + \sigma_a \pm \sigma_b\right)^2\right]^{0.5} \leq Ym \tag{4-347}$$

$$\left[3\left(\frac{R^2(P_i - P_o)}{R^2 - 1}\right)^2 + \left(\frac{P_i - R^2 P_o}{R^2 - 1} + \sigma_a \pm \frac{\sigma_b}{R}\right)^2\right]^{0.5} \leq Ym \tag{4-348}$$

where

$$\sigma_a = \frac{\hat{F}_a}{A_s} \quad \text{and} \quad \sigma_b = \frac{D \times r}{4I}\hat{F}_f$$

All symbols in Equations 4-347 and 4-348 are as previously stated.

CORROSION AND SCALING

Introduction

A great number of specialized tools have evolved in drilling of oil and gas wells utilizing very different alloys specially suited for their service requirements. The main concern in designing the drilling equipment is controlling the high pressures and the ability to resist fractures. The fractures can be induced by low-temperature service of the surface equipment and fatigue failures of subsurface equipment.

Most of the drilling equipment components are made from AISI 4,100 and 4,300 series of steel alloys that are heat treated to specific strength and hardness necessary to their particular operation conditions.

The blowout preventer (BOP) failures due to corrosion are very rare for two main reasons. First, as the BOP represents the primary method of preventing potential blowout under uncontrollable circumstances, they are usually overdesigned with very high structural integrity. The second reason is that the surface equipment usually does not experience very severe conditions in terms of corrosive environment and complexity of stress state. API Specifications 5A, 5AX, 5AC, 6A, and API RP 53 provide more information [176–180]. Although all equipment exposed to drilling fluids is affected by corrosion problems, it is beyond the scope of this discussion to cover all the corrosion problems. Thus, from this point on, the main discussion will be limited to subsurface equipment and concentrate on drillstring corrosion. This is due to the fact that the drillstring represents the largest capital expenditure, directly and indirectly during drilling operations. For more information on drillstring see the section titled "Drillstring Design" and API RP 76 [181].

The tool joints made by forging are heat treated by quenching and tempering. They are either friction welded or flash welded to the pipe body. For friction welding, the tool joint is rotated on the stationary pipe body. This results in elevated contact pressure and temperature with subsequent welding. In flash welding both the pipe and the tool joint are held stationary while an electric current is impressed across the joint. Electrical resistance at the tool joint and pipe interface generates enough heat to cause welding. After each welding process, postweld heat treatment is applied to the weld zone. The heat treatment improves the uniformity and structural properties of the welded zone.

Material used for tool joints is generally AIS 4100 series low-alloy steel, usually AISI 4135-4140, although AISI 4145H is also used sometimes. For a greater resistance to cracking in hydrogen sulfide environments AISI 4100 series alloy steels with introduction of molybdenum and niobium are used for tool joint construction. Tool joints are normally quenched and tempered to a hardness of 30 to 37 Rockwell C, with resulting yield strengths of 120–150 ksi. According to API Specification 7, Section 4, tempering is performed for 2 hr at 1,100–1,200°F to produce the mechanical properties of the new tool joints [182].

Drillstring subs are made from AISI 4140 or 4145H steel and sometimes from AISI 4340 or 4340H steel. The steel is quenched and tempered to a hardness range of 285 to 341 Bhn.

Drilling jars, stabilizers and, usually, core barrels are also made from AISI 4140 or 4145H steel and sometimes AISI 4340 or 4340H steel is also used. The steel is heat treated to the hardness level of 285 to 341 Bhn.

The drill collars are made of AISI 4135-4140 or 4145H steel. The steel is quenched and tempered to hardness of 285 to 341 Bhn.

Nonmagnetic drill collars are manufactured from various alloys, although the most common are Monel K500 (approximately 68% nickel, 28% copper with some iron and manganese, and 316L austenitic stainless steel). A stainless steel with the composition of 0.06% carbon, 0.50% silicon, 17–19% manganese, less than 3.50% nickel, 12% chromium, and 1.15% molybdenum, with mechanical properties of 110 to 115 Ksi tensile strength is also used.

The drillpipes are available in various grades according to API spec 5A and 5AX [176,177]. Grade E drillpipe is heat treated by normalizing and tempering and has almost the same chemistry as AISI 1040 or 1045 steels, with an addition of 1.50% manganese and 0.20% molybdenum. Grades X, G and S are quenched and tempered and contain 0.2 to 0.30% carbon, 1.20 to 1.5% manganese, 0.40 to 0.60% chromium and 0.20 to 0.50% molybdenum. However, grade S-135 may contain 0.27 to 0.35% carbon, 1.50 to 1.60% manganese, 0.10 to 0.50% chromium, 0.30 to 0.40% molybdenum and 0.012 to 0.016% of vanadium.

Heavy-wall drillpipe has approximately twice the usual wall thickness and is usually made from AISI 4140-4145H. The steel is quenched and tempered to the Rockwell C hardness of various grades from 20 to 28 for grade E, 27 to 30 for grade X-95, 30 to 34 for grade G-105 and 34 to 37 for S 135.

Aluminum drillpipe is generally made of 2014 type aluminum-copper alloy. Composition of this alloy is 0.50 to 1.20% silicon, 1.00% iron maximum, 3.90 to 5.0% copper, 0.40 to 1.20% manganese, 0.25% zinc maximum and 0.05% titanium. The alloy is heat treated to T6 conditions that represent 64 ksi tensile strength, 58 Ksi yield strength, 7% elongation and a Hbn of 135. Aluminum drillpipe generally comes with steel tool joints that are threaded on to ensure maximum strength that cannot be attained with aluminum joints.

The rotary drill bits are generally made from quench and tempered steel alloys such as AISI 3115 to 3120, 4620, 4815 to 4820 and 8620 to 8720. The only corrosion related problem that can arise may result from a hydrogen sulfide environment. The bearing in the roller rock bits can be damaged by H_2S contamination of drilling fluid. However, a well conditioned drilling fluid at all times can control the problem. Failure of bits due to corrosion has not been reported in the literature. Thus, from now on the discussion will not be concerned with drill bits.

Corrosion Theory

Corrosion is the deterioration of a substance or its properties because of a reaction with its environment. For our purposes, we can be a little more precise in this definition; therefore, corrosion is a destructive attack of a metal by either chemical or electrochemical reaction with a given environment [183].

Most metals naturally occur as "ores" in the form of metallic oxides or salts. Refining the pure metal from these ores requires energy input. Different metals require varying amounts of energy for refining and, hence, show different tendencies to corrode. These tendencies are related to the driving voltage when the metal is placed in an aqueous solution and are called the *potentials* or the *electromotive forces* (emf) of the metal. The emf values of some selected metals are given in Table 4-169 [184]. This stored energy is released when the metal converts back to its original state—its "ore." Therefore, it is the energy stored in the metal during the refining process that makes corrosion possible. Figure 4-412 illustrates this process schematically. Thus, we can say that metals in general are relatively unstable with respect to most environments and have a natural tendency to return to their original state, or corrode.

Electrochemical Aspects of Corrosion

In oilfield situations we are generally faced with corrosion attacks in aqueous environments. Basically all attacks in aqueous solutions are electrochemical in nature. This means that besides the chemical reaction there will also be a flow of electrons, resulting in a flow of current. The current flows from a higher potential to a lower one. Hence, there are two reactions taking place simultaneously in the system. One reaction occurs as the electrons are discharged from the surface, called the *anode*. The released electrons are consumed in the other

Table 4-169
Electromotive Force Series of Metals

	Metal	Volts[(1)]	
Most energy required for refining	Magnesium	-2.37	Greatest tendency to corrode
↑	Aluminum	-1.66	↑
	Zinc	-0.76	
	Iron	-0.44	
	Tin	-0.14	
	Lead	-0.13	
	Hydrogen	0.00	
↓	Copper	+0.34 to +0.52	↓
Least energy required for refining	Silver	+0.80	Least tendency to corrode
	Platinum	+1.20	
	Gold	+1.50 to +1.68	

(1) vs standard hydrogen electrode.
Source: From Ref. [184].

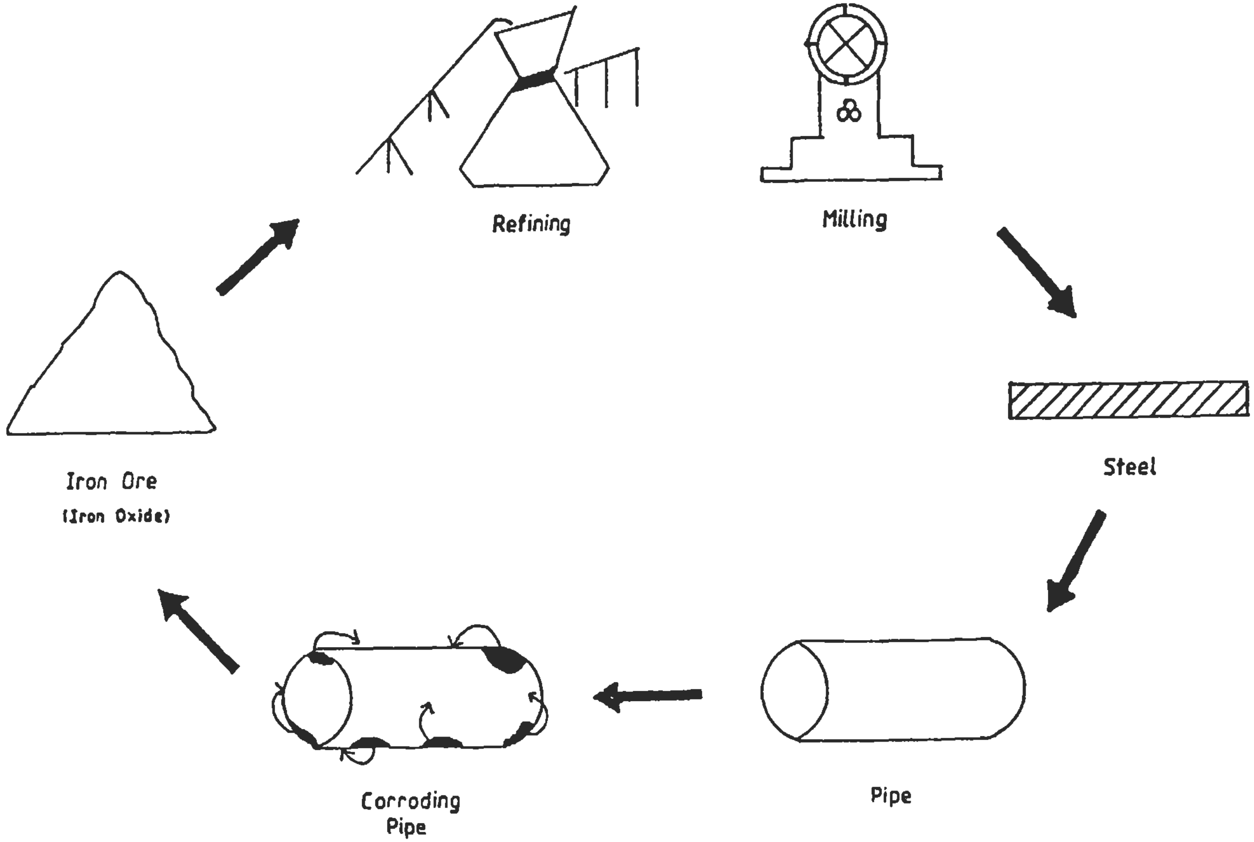


Figure 4-412. Metallurgical cycle; all metals eventually corrode. *(From Ref. [183].)*

reaction at the surface, called the *cathode*. In order to complete the electrical circuit of the corrosion cell shown schematically in Figure 4-413 we need an aqueous solution called an electrolyte. The corrosion cell shown in Figure 4-413 consists of the following four components:

1. *Anode*. As forementioned, an anode is the surface where oxidation or release of electrons takes place. Consequently the metal is corroded and goes into solution as metal ions. The chemical reaction for iron is

   $Fe \leftrightarrows Fe^{2+} + 2e^-$

2. *Cathode*. At the surface at which the reduction reaction occurs, the electrons are consumed by oxidizing agents present in the aqueous solution.

   $2H^+ + 2e^- \leftrightarrows H_2$ (in an acidic solution)

   However, if oxygen is present, two other reactions may occur:

   $O_2 + 4H^+ + 4e^- \leftrightarrows 2H_2O$ (acid solutions)

   $O_2 + 2H_2O + 4e^- \leftrightarrows 4OH^-$ (neutral and alkaline solutions)

3. *Electrolyte*. The aqueous solution that supports the above reactions and completes the electrical circuit is called an *electrolyte*. Both electrodes must be completely submerged.
4. *Electronic connector*. The metallic path between the anode and the cathode that conducts electricity outside the electrolyte. In practice, the two

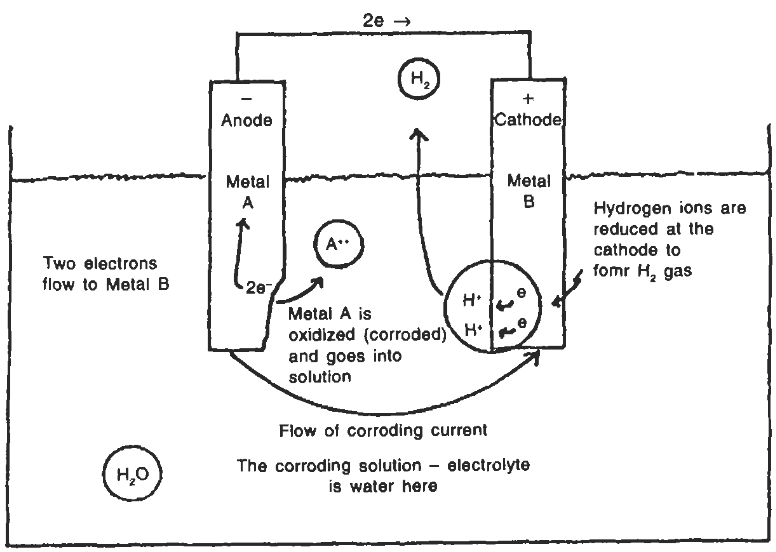


Figure 4-413. Electrochemical corrosion of metals.

electrodes may be a part of the same metal as shown in Figure 4-414. This could be due to small variations in metallurgy, surface films, etc.

Many factors and conditions affect the tendency of metals to corrode. Once the right conditions and all the components are present, corrosion may proceed in one or more forms. Before considering the aforementioned points, it is necessary to consider the concepts of polarization and passivity.

Polarization

An electrochemical reaction is said to be polarized or retarded when it is limited by various physical and chemical factors. In other words, the reduction in potential difference in volts due to net current flow between the two electrodes of the corrosion cell is termed polarization. Thus, the corrosion cell is in a state of nonequilibrium due to this *polarization.* Figure 4-415 is a schematic illustration of a Daniel cell. The potential difference (emf) between zinc and copper electrodes is about one volt. Upon allowing current to flow through the external resistance, the potential difference falls below one volt. As the current is increased, the voltage continues to drop and upon completely short circuiting ($R \approx 0$, therefore maximum flow of current) the potential difference falls toward about zero. This phenomenon can be plotted as a polarization diagram shown in Figure 4-416.

The overall reaction is controlled by the slowest reaction, anodic or cathodic. If the slower reaction is anodic or the polarization occurs mostly at the anode,

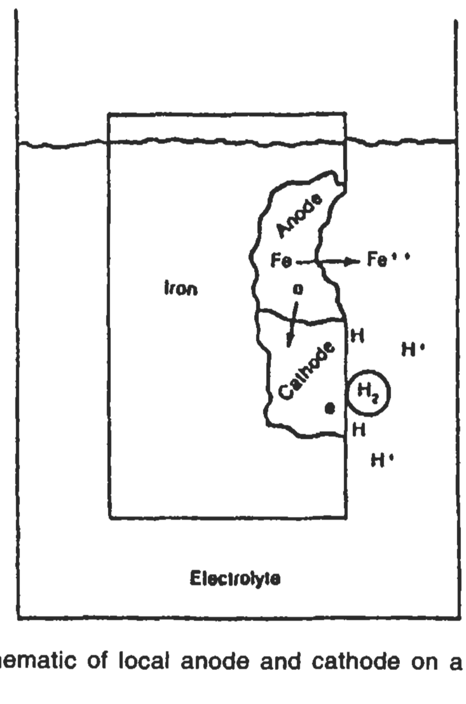


Figure 4-414. Schematic of local anode and cathode on a steel surface. *(From Ref. [185].)*

the corrosion reaction is said to be "anodically controlled". This can be seen in Figure 4-417a; as corrosion potential E_c is closer to the open circuit potential of the cathode, therefore, the reaction is under anodic control. When the slower reaction is cathodic and polarization occurs mostly at the cathode, the corrosion rate is "cathodically controlled". In this case corrosion potential is near the open-circuit anode potential, as shown in Figure 4-417b. Resistance of the electrolyte and resistance of polarization of the electrodes limits the current magnitude that can flow through the cell. Thus, the reaction is said to be under resistance control when the electrolyte resistance is so high that the current produced is not enough to polarize either the cathode or the anode, Figure 4-417c. The corrosion current in this situation is under the influence of IR drop. IR drop

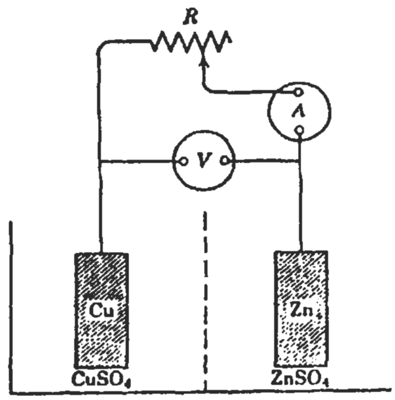


Figure 4-415. Polarized copper–zinc cell. *(From Ref. [186].)*

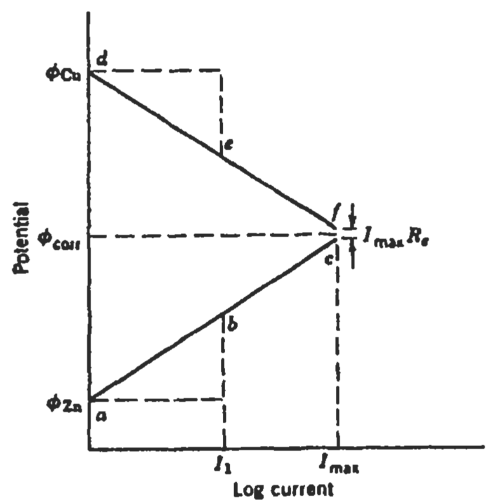


Figure 4-416. Polarization diagram for copper–zinc cell. *(From Ref. [186].)*

simply means a voltage or potential drop in an electrochemical cell due to the resistance in the electrolyte or other resistances, such as insulations or coatings. Finally, if the polarization occurs at both the anode and the cathode, the reaction is under mixed control as seen in Figure 4-417d.

Polarization can be divided into "activation polarization" and "concentration polarization". Activation polarization is an electrochemical reaction that is controlled by the reaction occurring on the metal-electrolyte interface. Figure 4-418 illustrates the concept of activation polarization where hydrogen is being reduced over a zinc surface. Hydrogen ions are adsorbed on the metal surface; they pick up electrons from the metal and are reduced to atoms. The atoms combine to

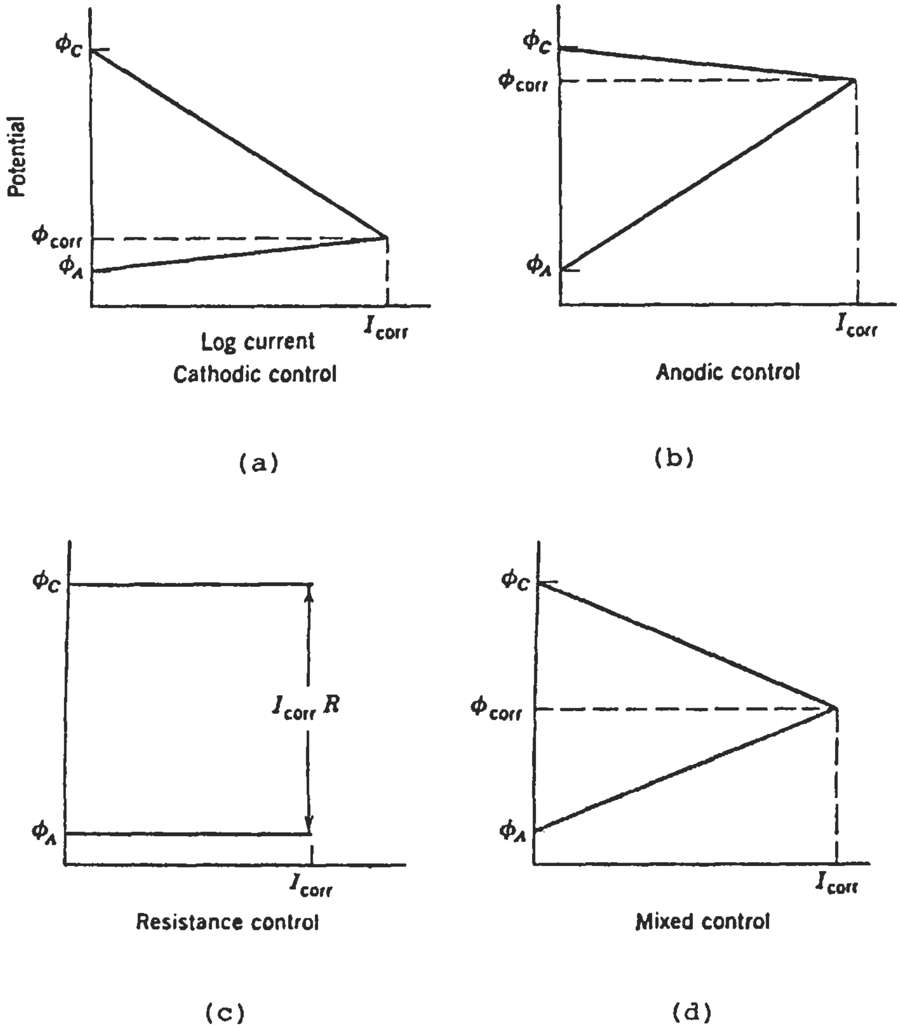


Figure 4-417. Types of corrosion control. *(From Ref. [186].)*

form molecules of hydrogen and the molecules combine to form gas bubbles. The gas bubbles at the cathode decrease the corrosion rate as they keep other hydrogen ions from reaching the metal surface. Activation polarization is usually the controlling factor when high concentrations of active species (i.e., concentrated acids) are involved in the corrosion process.

Concentration polarization is an electrochemical process controlled by the diffusion within the electrolyte.

Figure 4-419 illustrates the concept of corrosion process under concentration polarization control. Considering hydrogen evolution at the cathode, reduction rate of hydrogen ions is dependent on the rate of diffusion of hydrogen ions to the metal surface. Concentration polarization therefore is a controlling factor when reducible species are in low concentrations (e.g., dilute acids).

Although the above discussion is a very simplified picture of the polarization process, it does give a basic understanding of the processes involved. It is essential to determine which kind of polarization is controlling the reduction reaction. For example, any change in the system that increases the diffusion rate of the reducible species will increase the reduction rate under concentration polarization-control. However, this change will not have any affect on the activation-polarization-controlled reduction process. Thus, in order to control the corrosion rate, it is important to know exactly what types of reactions are occurring, and considering the polarization of the corrosion cell helps in doing just that.

Passivity

Passivity is the loss of chemical reactivity of certain metals and alloys under specific environmental conditions. In other words, certain metals (e.g., iron, nickel, chromium, titanium, etc.) become relatively inert and act as noble metals (e.g., gold and platinum). Figure 4-420 shows the behavior of a metal immersed in an air-free acid solution with an oxidizing power corresponding to point A

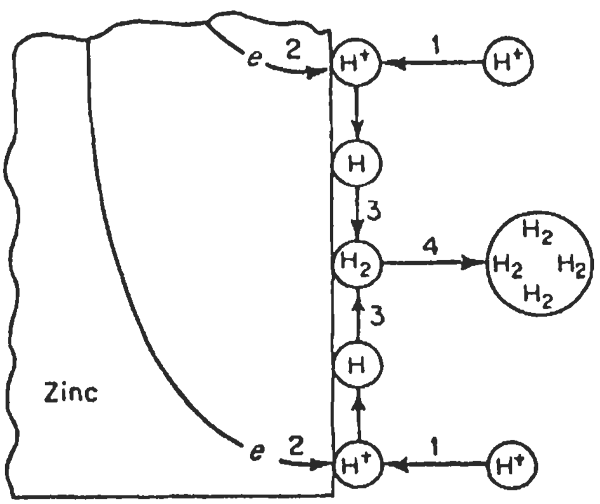


Figure 4-418. Hydrogen-reduction reaction under activation control (simplified). *(From Ref. [183].)*

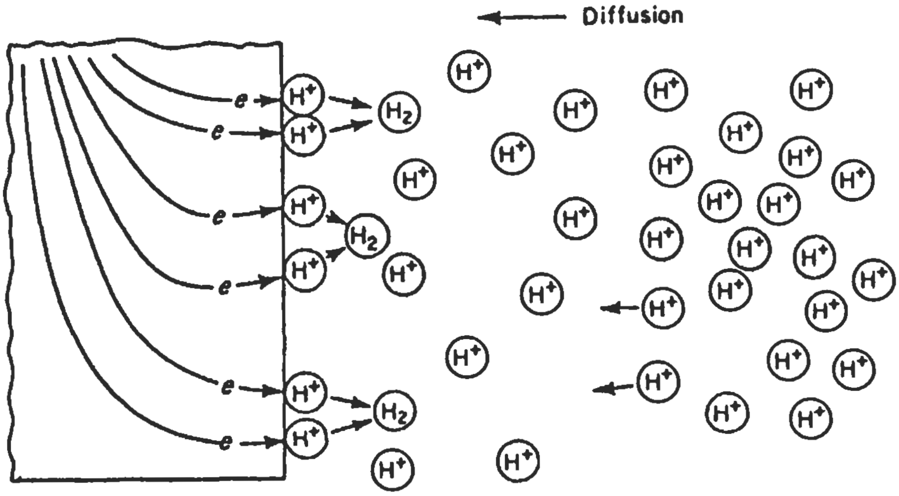


Figure 4-419. Concentration polarization during hydrogen reduction. *(From Ref. [183].)*

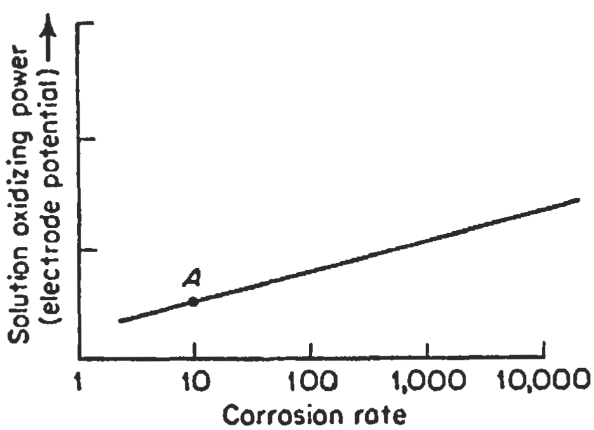


Figure 4-420. Corrosion rate of a metal as a function of solution oxidizing power (electrode potential). *(From Ref. [183].)*

and a corrosion rate corresponding to this point. If the oxidizing power of this solution is increased, by adding oxygen or ferric ions for example, the corrosion rate increases rapidly. This rate increase is exponential and often yields a straight line when it is plotted on a semilogarithmic scale. Figures 4-421 and 4-422 illustrate the behavior of a metal which demonstrates passivity effects. The behavior of this specimen exhibits three distinct regions: active, passive and transpassive. In the active region, the specimen behaves exactly the way it did in Figure 4-420. As the oxidizing power of the solution increases, the corrosion rate suddenly decreases and the

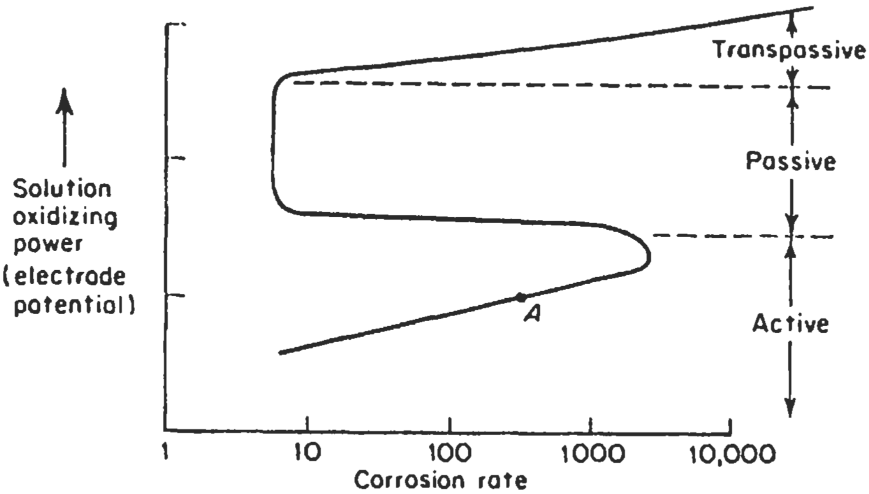


Figure 4-421. Corrosion characteristics of an active passive metal as a function of solution oxidizing power (electrode potential). *(From Ref. [183].)*

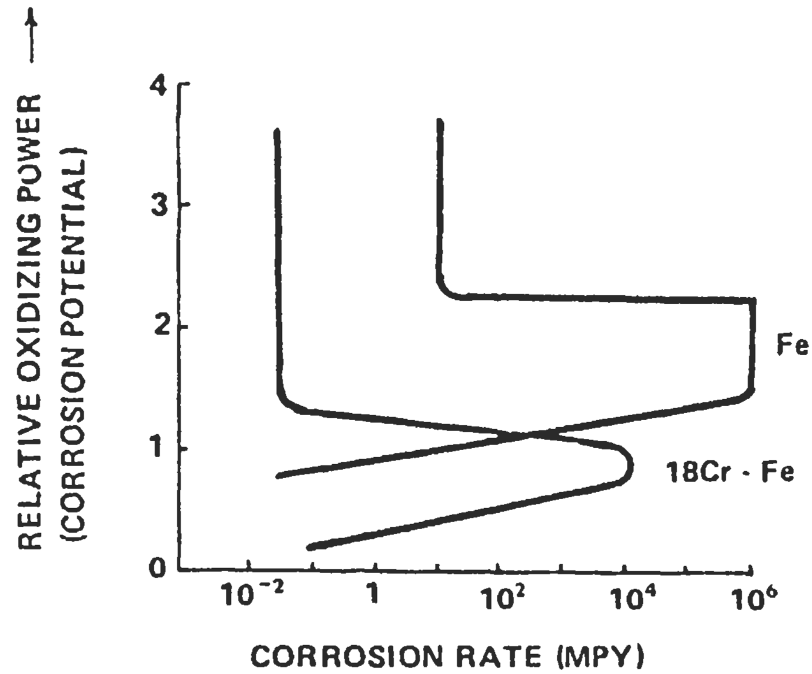


Figure 4-422. Corrosion characteristics of iron and 18% Cr stainless steel in dilute sulfuric acid and as a function of solution oxidizing power (corrosion potential). *(From Ref. [187].)*

behavior enters the passive zone. Further increase in oxidizing power of the solution produces hardly any change in corrosion rate. However, at a very high concentration of oxidizers the corrosion rate starts to increase rapidly as the behavior enters the transpassive zone. It is not fully understood why metals exhibit active-passive-transpassive transitions, although it is thought that it is a special case of activation polarization. This is due to the formation of an inert surface film or a protective barrier, which fails in very strong oxidizing solutions.

There are various theories on how passive films are formed; however, there are two commonly accepted theories. One theory is called the *oxide film theory* and states that the passive film is a diffusion-barrier layer of reaction products (i.e., metal oxides or other compounds). The barriers separate the metal from the hostile environment and thereby slow the rate of reaction. Another theory is the *adsorption theory of passivity*. This states that the film is simply adsorbed gas that forms a barrier to diffusion of metal ions from the substrata.

Considering Figure 4-421 we can conclude that metals that exhibit passivity can be used in moderately to strongly oxidizing environments. In highly oxidizing environments they will lose their corrosion-resistant properties and, therefore, cannot be used. It is, however, possible to passivate some metals by exposing them to passivating environments (i.e., iron in chromate or nitrite solutions) or by anodic polarization at sufficiently high-current densities (i.e., in H_2SO_4). The desired passivity can be achieved by appropriate alloy additions to the metal. Figure 4-422 illustrates the corrosion behavior of iron (Fe) and chromium stainless steel. Introducing 18% chromium to iron reduces the amount of oxidizer necessary to achieve passivity. The addition of chromium to iron also reduces the corrosion rate in the passive state. Proper alloying is an effective way to improve the corrosion resistance of a metal.

Forms of Corrosion Attack

Corrosion may take various forms and may combine other forms of damage (erosion, wear, fatigue, etc.) to cause equipment failure. The forms of corrosion most encountered in drilling equipment are *uniform corrosion* and *galvanic corrosion.*

Uniform Corrosion

All homogeneous metals without differences in potential between any points on their surfaces are subject to this type of general attack under some conditions. Uniform corrosion is usually characterized by a chemical or electrochemical attack over the entire exposed surface, Figure 4-423. Metal corrodes in an even

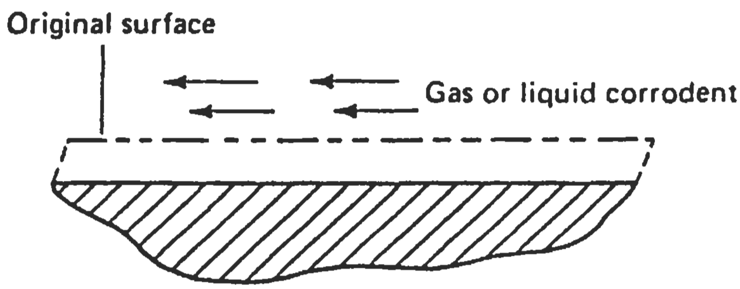


Figure 4-423. Uniform corrosion. *(From Ref. [185].)*

and regular manner becoming thinner, and consequently leads to failure due to reduction of the material's load-carrying capabilities. The rate of penetration or the thinning of a structural member can be used directly to predict the service life of a given component. Therefore, the expression mils penetration per year (mpy) is used to express corrosion resistance directly in terms of penetration. This expression can be calculated by the following [185,186]:

$$\text{mpy} = \frac{534\text{W}}{\text{DAT}} \tag{4-349}$$

where W = weight loss in mg
D = density of specimen in g/cm^3
A = area of specimen in $in.^2$
T = exposure time in hr

Using the value of mpy, we can classify metals into three categories according to the corrosion rates and intended applications. The categories are the following:

< 5 mpy—Metals in this group have good resistance to corrosion and are quite suitable for use as critical equipment parts.
5 to 50 mpy—Metals in this group have an acceptable corrosion resistance if a higher rate of corrosion is tolerable.
> 50 mpy—Metals of this group are usually not acceptable for service use.

This form of corrosion is not of great concern from the engineering aspect, as the service life can be estimated within reasonable accuracy. Uniform corrosion can be prevented or reduced by using the following methods singly or in combination:

- proper selection of materials for the types of service environments and conditions
- proper selection of inhibitor
- proper selection of coatings
- cathodic protection

These methods will be discussed in more detail.

Galvanic Corrosion

Galvanic corrosion occurs when two dissimilar metals or alloys come into electronic contact in a conductive solution. The severity of galvanic corrosion depends primarily on the difference in solution potentials between the two materials and the relative sizes of the cathode and the anode areas. The farther apart these metals are from each other in the galvanic series, the greater the possible corrosion of the anodic member of the galvanic couple. The galvanic series for some commercial metals and alloys in seawater is shown in Table 4-170. Figure 4-424 shows schematically an example of galvanic corrosion [186,187].

Species with more positive corrosion potential, located toward the bottom of the series, are called *noble* or *cathodic* metals and alloys. Those species with more negative corrosion potential located toward the top of the series are referred to as *active* or *anodic* metals and alloys.

Conductive films such as "magnetite" (Fe_3O_4) or "mill scale" on steel, and conductive nonmetals such as carbon can function as cathodes when in contact with anodes

Table 4-170
Galvanic Series of Some Commercial Metals and Alloys in Seawater

Active or Anodic (–)	Magnesium Magnesium Alloys Zinc Galvanized Steel
	Aluminum 1100
	Aluminum 2024 (4.5 Cu, 1.5 Mg, 0.6 Mn)
	Mild Steel Wrought Iron Cast Iron
	13% Chromium Stainles Steel Type 410 (Active) 18-8 Stainless Steel Type 304 (Active)
	Lead-Tin Solders Lead Tin
	Muntz Metal Manganese Bronze Naval Brass
	Nickel (Active) 76 Ni-16 Cr-7 Fe Alloy (Active)
	60 Ni-30 Mo-6 Fe-1 Mn
	Yellow Brass Admirality Brass
	Red Brass Copper Silicon Bronze
	70:30 Cupro Nickel G-Bronze Silver Solder Nickel (Passive) 76 Ni-16 Cr-7 Fe Alloy (Passive)
	13% Chromium Stainless Steel Type 410 (Passive) Titanium
	18-8 Stainless Steel Type 304 (Passive)
Noble or Cathodic (+)	Silver Graphite Gold Platinum

Source: From Ref. [187].

of clean metal. Aluminum drillpipes can experience localized galvanic corrosion attack resulting in pitting. The attack generally occurs immediately adjacent to the steel tool joints. Although not very severe, the extent of attack depends on salt concentration of the drilling fluid and exposure time. One reason the attack is not very severe is the favorable anode-to-cathode surface area ratio. That is, the

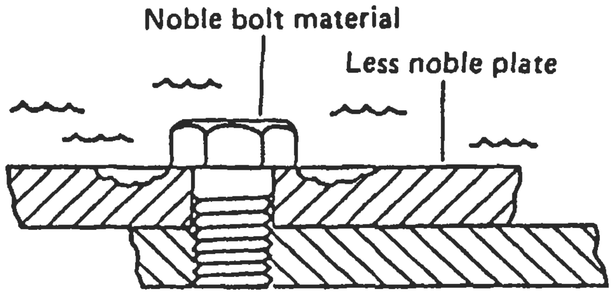


Figure 4-424. Galvanic corrosion. *(From Ref. [185].)*

aluminum anode surface area is much larger than the steel cathode surface area. Galvanic corrosion becomes very serious when an aluminum drillpipe string rests against the steel casing for long periods of time (e.g., stuck drillstring). The anode-to-cathode surface area ratio is unfavorable in this situation. The condition develops galvanic corrosion and results in severe pitting attack of aluminum.

To prevent or reduce galvanic corrosion we can employ several techniques. Any one of these techniques may be used either by itself or in combination of two or more of the techniques. These techniques are as follows:

1. If a metal or alloy combination is to be selected, choose combination of metals as close together in the galvanic series as possible.
2. Choose the metal or alloy so that the anode area is larger than the cathode area.
3. If any dissimilar metals are in contact with each other, isolate them electrically so that no electricity flows between them.
4. Apply proper coatings with caution, and keep the coatings in good repair.
5. Add proper inhibitors with appropriate practices.
6. Avoid threaded joints for materials that are far apart in galvanic series.
7. Anodic parts should be designed so that they are easily replaceable. They may also be designed thicker than what is required to extend their service life.

Localized Attack

Any normally-employed metal surface is a composite of anodic and cathodic sites. These electrodes are electrically short-circuited through the body of the metal itself (Figure 4-425). As long as the metal surface does not come into contact with an electrolytic solution, there is no flow of current. However, exposure of the metal to an electrolyte results in a flow of electric current accompanied by corrosion of the anodic areas. The flowing current of this kind is called a *local-action current* and the corresponding cell is called *local-action cell.* Local-action cells are set up when there are heterogeneities and/or other environmental variations present. Table 4-171 illustrates geometrical factors and heterogeneities in relation to localized attack [188].

Pitting Attack

Pitting is a form of extreme, localized attack. The rate of corrosion is greater at some areas than at others, resulting in holes in the metal. Heterogenous metal

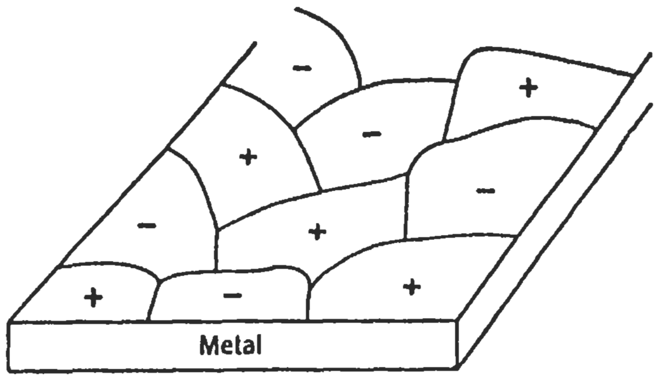


Figure 4-425. Metal surface enlarged, showing schematic arrangement of local action cells. *(From Ref. [195].)*

or alloy surfaces are susceptible to such kinds of attack. It can also occur under deposits of foreign matter or at imperfections in a protective film or coating. Pitting is one of the most deceptive and, hence, most destructive forms of corrosion. It is often very difficult to detect the problem. It is also very hard to predict the severity of the problem as it is to measure quantitatively since the pits vary in size, depth and frequency. Sudden failures occur as the perforations penetrate deep into the metal, creating weak areas. Depth of pitting can be expressed by a term known as the *pitting factor.* Pitting factor is the ratio of the deepest metal penetration to average metal penetration calculated by the overall weight loss. A pitting factor of unity represents uniform attack. This factor is [186]

$$\text{Pitting factor} = \frac{P}{d} \tag{4-350}$$

where P = deepest metal penetration in mm
d = average metal penetration in mm

Figure 4-426 shows a geometric representation of the terms in the above equation.

Although the pitting factor gives us a rough estimate of the pit damage, it by no means enables us to predict the service life of the equipment. Figure 4-427 illustrates the pitting form of attack. Pits can cause washouts and serve as initiating points for fatigue cracks. Chlorides, oxygen, carbon dioxide, and hydrogen sulfide or any combination of these corrodents may contribute to this form of attack. To minimize the attack or prevent it, the following points should be considered:

- Avoid materials prone to pitting.
- Use proper inhibitors carefully, as improper use may result in an accelerated attack.
- Avoid stagnant areas in design.
- Use proper handling practices for equipment—avoid damaging protective film, avoid nicks or scratches, etc.

Table 4-171
Geometric Factors and Heterogeneities in Relation to Localized Attack

System	Metal Area which Is Predominantly Anodic*
Metal	
Dissimilar metals in contact.	Metal which is more reactive in a given solution (i.e., metal which has a greater tendency to ionize).
Crevices, deposits on metal surface or any geometrical configuration which results in differences in the concentration of oxygen or other cathodic depolarizers (e.g., Cu^{2+}).	Metal in contact with the lower concentration—this follows from considerations of an equivalent reversible cell, although the situation is more complex in practice.
Differences in metallurgical structure.	Grain boundaries, more reactive phases (solid solutions, intermetallic compounds, etc.).
Differences in metallurgical condition due to thermal or mechanical treatment.	Cold worked areas anodic to annealed areas, metal subjected to external stress anodic to unstressed metal.
Discontinuities in conducting oxide film or scale or discontinuities in applied metallic or non-metallic coatings.	Exposed substrate (provided that this is more electrochemically active than the coating).
Environment	
Differences in aeration or concentration of cathodic depolarizers.	Metal in contact with lower concentration.
Differences in velocity.	Metal in contact with solution of higher velocity.
Differences in pH or salt concentration.	Metal in contact with solution of lower pH or higher salt concentation.

*The table gives a general indication of the area which is likely to be anodic. There are many exceptions, e.g., grain boundaries can be cathodic, the area of metal in contact with a higher salt concetration will be cathodic if the oxygen concentration is higher, etc.
Source: From Ref. [188].

- Maintain good and regular inspection programs and, at frequent intervals, remove any deposits formed.
- Arrange as uniform an environment as possible.

Intergranular Corrosion

Intergranular corrosion is a localized type of attack at the grain boundaries, with relatively little corrosion of the grains. The metal or the alloys lose their strength, ductility and eventually disintegrate (grains fall off). Relatively small areas of grain-boundary material act as anodes, and are in contact with larger areas of grain material, the cathodes. The attack can be caused by impurities at the grain boundaries, enrichment of one of the alloying elements, or depletion of one of these elements in the grain-boundary areas. This type of attack is

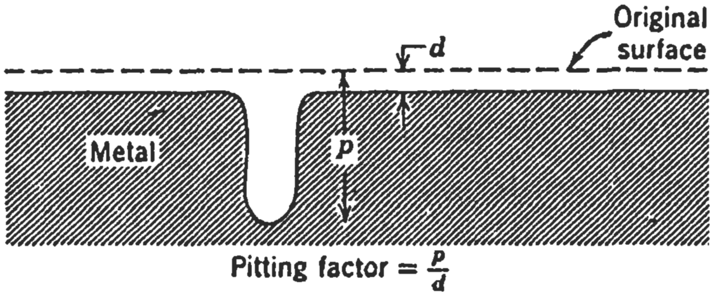


Figure 4-426. Sketch of deepest pit with relation to average metal penetration and the pitting factor. *(From Ref. [186].)*

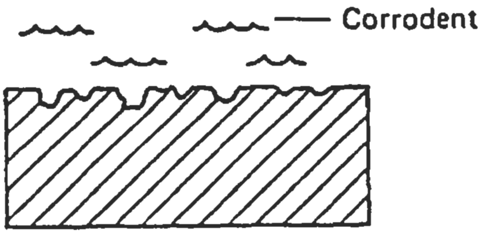


Figure 4-427. Pitting corrosion. *(From Ref. [185].)*

common on improperly heat-treated metals and their alloys. Figure 4-428 schematically illustrates intergranular corrosion [183].

Intergranular corrosion can be prevented or minimized by the following considerations:

- Choose proper treatment for the environmental conditions of service.
- Use properly heat-treated metals and alloys.
- Use materials that contain strong carbide-formers, the stabilizers.
- Use low-carbon grade materials.
- Avoid high-strength aluminum alloys.

Erosion Corrosion

Most metals and their alloys are susceptible to erosion corrosion as various media may provide the right conditions for it. Many metals depend on a protective oxide film or tightly-adherent deposit for corrosion resistance. Erosion corrosion occurs when the protective films or deposits are removed by mechanical wear effects of abrasion. Once the protective surface is damaged, accelerated corrosive attack occurs at the fresh metal surface. The damage done by this form of corrosion appears as grooves, gullies, waves, rounded holes, pits and valleys, and generally, exhibits a directional pattern. Figure 4-429 illustrates erosion corrosion schematically [185].

There are basically five ways to reduce or prevent erosion corrosion:

- Choose materials with a high resistance to erosion and wear.
- Keep erosion corrosion in mind while designing the equipment.

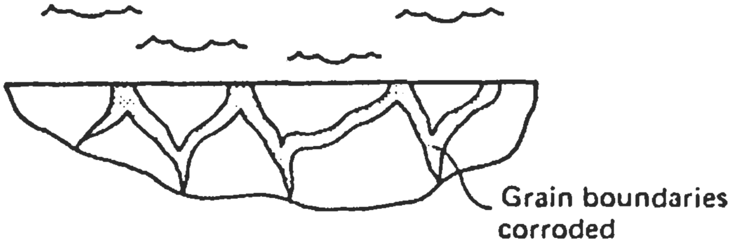


Figure 4-428. Intergranular corrosion of sensitized (improperly annealed) stainless steel. *(From Ref. [185].)*

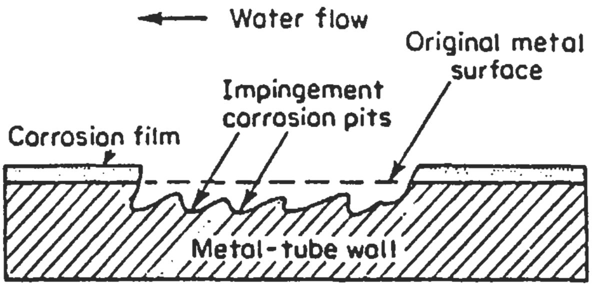


Figure 4-429. Erosion corrosion. *(From Ref. [183].)*

- Use proper coatings.
- Alter environment–add inhibitors, remove abrasives as soon as possible.
- Use cathodic protection wherever possible. This does not affect the erosion part of the attack, but it may reduce the corrosive attack.

Cavitation Corrosion

Cavitation is a special form of erosion corrosion results from formation and collapse of vapor bubbles in a liquid near a metal surface. Collapsing bubbles destroy the protective surface film, thereby exposing the metal to increased corrosion attack. Cavitation corrosion is shown schematically in Figure 4-430. Cavitation damage is often seen in high-velocity, turbulent, liquid-flow areas such as drilling-mud pumps. To reduce problems associated with this kind of corrosion, methods described for erosion corrosion should be considered. Also materials with high hardness and strength with tenacious passive films should be used. Titanium and corrosion-resistant, cobalt-base alloys appear to be suitable for a wide range of environments causing cavitation erosion [184].

Ringworm Corrosion

During hot-forming or upsetting operations the steel is subjected to large temperature gradients. These thermal gradients cause variations in metallurgical composition along the pipe. In the sections exposed to intermediate temperatures, the iron carbides condense, forming spheroids. During exposure to corrosive environment this narrow section corrodes preferentially, as shown in

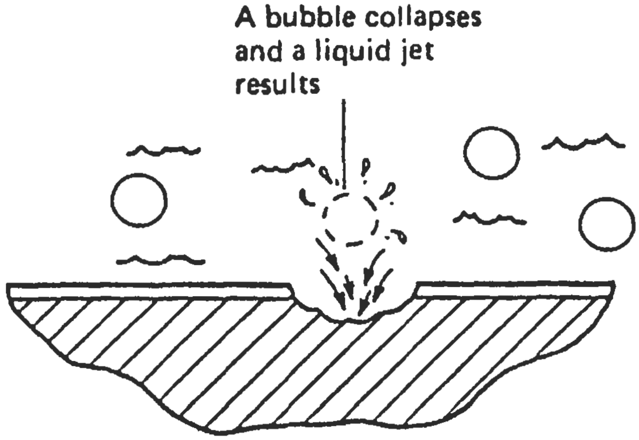


Figure 4-430. Cavitation corrosion. *(From Ref. [185].)*

Figure 4-431. This type of attack can be completely eliminated by annealing the entire length of the pipe after forging. The annealing results in a uniform metallurgical structure throughout the length of the pipe [184,186].

Concentration Cells

Concentration cells have two similar electrodes in contact with a solution of differing composition. The two kinds of concentration cells are salt concentration cells and differential aeration cells [186].

Salt Concentration Cells. In this type of cell the two electrodes are of the same metal (i.e., copper). These electrodes are immersed completely in electrolytes of the same salt solution (i.e., copper sulfate) but of different concentrations. When the cell is short circuited, the electrodes (anode) exposed to the dilute solution will dissolve into the solution and plate the electrode (cathode) exposed to the more concen-trated solution. These reactions will continue until the solutions are of the same concentration. Figure 4-432 shows a schematic of a salt concentration cell.

Differential Aeration Cells. This type of concentration cell is more important in practice than is the salt concentration cell. The cell may be made from two electrodes of the same metal (i.e., iron), immersed completely in dilute sodium chloride solution (Figure 4-433). The electrolyte around one electrode (cathode) is thoroughly aerated by bubbling air. Simultaneously the electrolyte around the other electrode is deaerated by bubbling nitrogen. The difference in oxygen concentration causes a difference in potential. This, in turn, initiates the flow of current. This type of cell exists in several forms. Some of them are as follows [188].

Crevice Corrosion. Crevices are formed at the interface of two coupled pipes, near the ends of drillpipe protectors, at threaded connections and where surface

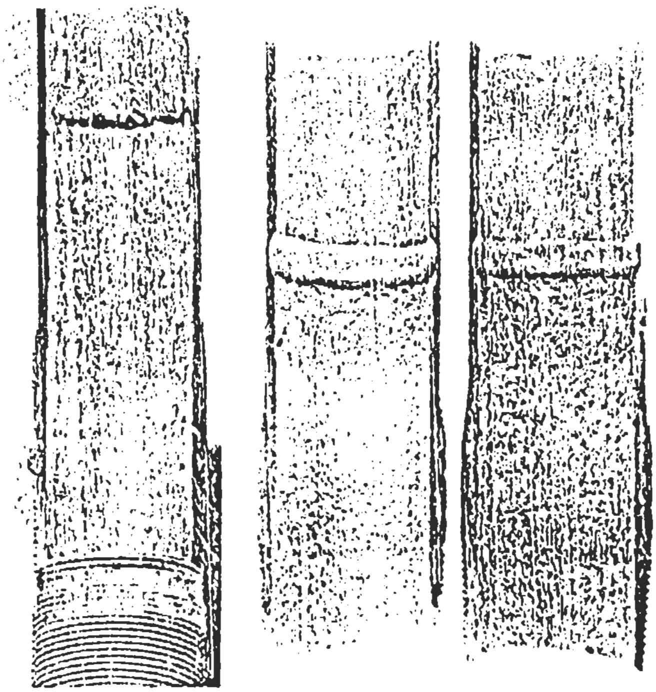


Figure 4-431. "Ringworm" corrosion—at the inside of the tubing where grain structure near edge of upset portion makes metal susceptible to rapid corrosion. *(From Ref. [219].)*

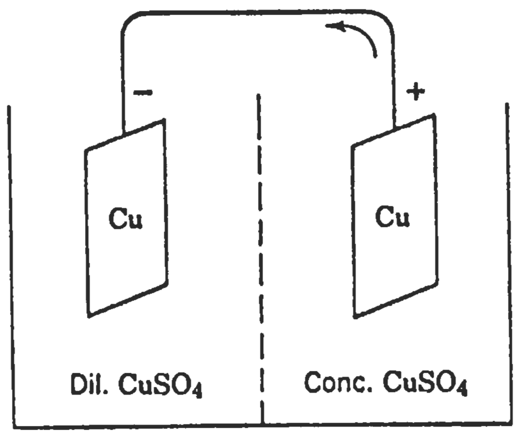


Figure 4-432. Salt concentration cell. *(From Ref. [186].)*

deposits create stagnant conditions. Oxygen may be consumed at a much faster rate inside the crevice than it can diffuse into the crevice. This lowers the pH in the crevice and creates acidic conditions, which, in turn, increases the corrosion rate. Figure 4-434 illustrates the concept schematically.

Oxygen Tubercles. Similar to crevice attack, it is encouraged by the deposit of a layer, semipermeable to oxygen (porous layer of iron oxide or hydroxide).

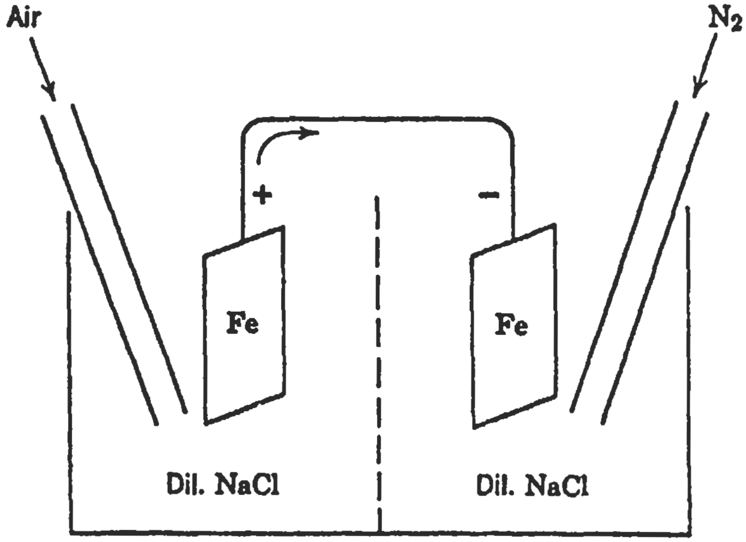


Figure 4-433. Differential aeration cell. *(From Ref. [186].)*

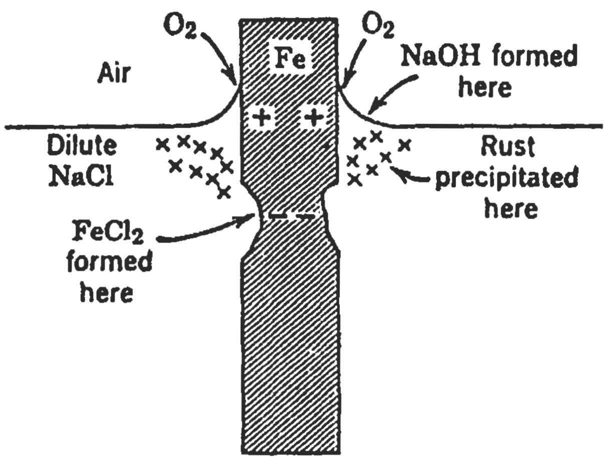


Figure 4-434. Differential aeration cell illustrated by waterline corrosion. *(From Ref. [185].)*

The deposit partially shields the steel surface, creating a differential aeration cell (Figure 4-435).

Air–Water Interface. This is another good example of a differential aeration cell (Figure 4-436). Here the water at the surface contains more oxygen than the water slightly below the surface. This difference in concentration can cause preferential attack just below the waterline.

Scale Deposits

Scale deposits create conditions for concentration-cell corrosion as they do not form uniformly over the metal surface. Sulfate-reducing bacteria thrive under these deposits, producing hydrogen sulfide and, consequently, increasing the rate of corrosion. Due to the following factors, the drilling fluid environment is ideal for scale deposition [189]. These factors are as follows:

- high pH levels
- turbulent flow
- pressure variations
- temperature changes

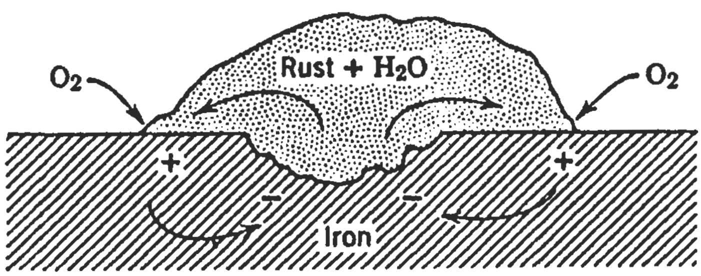


Figure 4-435. Differential aeration cell formed by rust on iron. *(From Ref. [184].)*

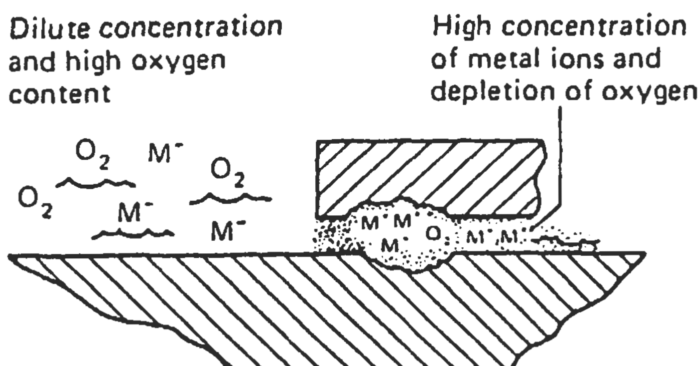


Figure 4-436. Crevice corrosion. *(From Ref. [186].)*

- dissolution of formation salts
- influx of carbon dioxide
- influx of low-pH, high-total-dissolved-solid content and formation fluids
- evaporation

Calcium carbonate ($CaCO_3$) calcium sulfate or gypsum ($CaSO_4$) and iron(II) carbonate ($FeCO_3$) are the most common types of scales formed in drilling. If hydrogen sulfide is present, then there is a possibility of iron sulfide (FeS) scale depositing.

From the corrosion point of view, it is very important to control the deposition of scale. Removal of deposited scale by mechanical means is the first step. Standard, industrial water-treating techniques can be used to control scale deposition in general. In deep, hot wells or geothermal wells it is best to avoid untreated makeup water (i.e., geothermal brines).

To prevent or minimize problems associated with concentration cells, the following methods should be considered in general:

- Welded butt joints should be used instead of bolted or riveted ones.
- Existing crevices should be closed by soldering, welding or caulking.
- Stagnant conditions should be avoided.
- "Solid" nonabsorbent gaskets such as Teflon should be used when necessary.
- Proper use of appropriate inhibitors.
- Maintain regular inspections of equipment and treatment status.
- Drillpipe protectors should be moved to different locations at each trip.

Hydrogen Damage

In general corrosion, hydrogen ions (H^+) are reduced to atomic hydrogen (H^0). These hydrogen atoms combine with each other and form molecular hydrogen [186,190,191].

$$2H^+ + 2e^- \rightarrow 2H^0 \rightarrow H_2 \uparrow$$

However, certain substances such as sulfide ions, phosphorus and arsenic compounds reduce the rate at which hydrogen combines to form molecules. Atomic hydrogen can diffuse through metal matrix and cause mechanical damage. Therefore, hydrogen damage is a mechanical damage of metal caused by the presence of atomic hydrogen. Some of the forms of hydrogen damage are as follows.

Hydrogen Blistering. Penetration of hydrogen in low-strength steel with any discontinuities in the steel such as laminations, inclusions or voids may result in hydrogen blistering. Hydrogen produced on the surface of the metal evolves in part as gas bubbles and the rest diffuses through the metal. The different hydrogen atoms collect in the void, and combine to form hydrogen gas molecules. Hydrogen gas molecules are too large to diffuse back through the metal. Thus, the gas concentration and pressure continue to increase within the void. In time, the gas pressure increases to sufficient levels to burst out to the metal surface. Figure 4-437 schematically illustrates the mechanism of hydrogen blistering [183].

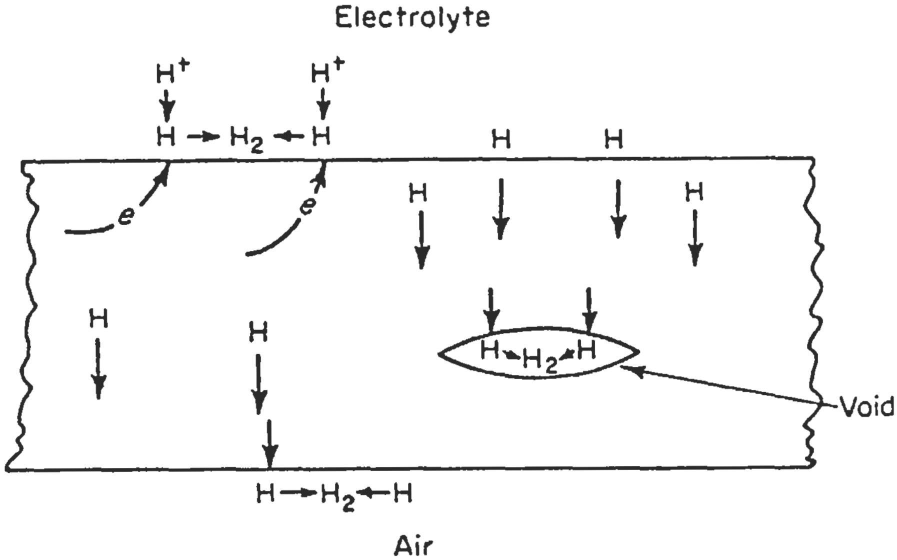


Figure 4-437. The simplified mechanism of hydrogen damage. *(From Ref. [191].)*

Hydrogen Embrittlement. Hydrogen embrittlement is a form of mechanical damage of a high-strength steel in a brittle failure. Limited to high-strength steels, the mechanism and cause are similar to those which cause hydrogen blistering. Hydrogen gas is trapped within the metal lattice, causing loss in ductility of the metal. Once ductility is reduced or lost, and the metal is subjected to high tensile stress, the metal will fail in a brittle manner. The path of the failure will be transgranular when the process takes place at normal temperatures when hydrogen diffuses through the grain, and gathers at inclusions, voids, or any other lattice defects. The failure is said to be integranular when the temperatures are higher and the absorbed hydrogen gathers at the grain boundaries [183,184].

To prevent or minimize hydrogen blistering and embrittlement, the following methods should be considered:

- Proper use of appropriate corrosion inhibitors.
- Using suitable, more-resistant alloys for the environment.
- Practice proper welding techniques.
- Apply suitable coatings in a proper manner.
- Select steels with minimum voids or discontinuities.
- Remove "poisons" such as sulfide ions from the environment.

Hydrogen Sulfide Cracking. Known also as sulfide stress cracking, the exact mechanism of this form of hydrogen damage is not yet fully understood. Thus, some experts classify it as a form of hydrogen embrittlement. Nevertheless, for hydrogen sulfide cracking of steel, the following conditions should be present [186,192]:

- Hydrogen sulfide gas, as low as one ppm.
- Even traces of water are sufficient.
- High-strength steel (i.e., yield strength above 90,000 psi or Rc 20 to 22).
- The steel must be under residual or applied tensile stress.

With the above conditions present, hydrogen sulfide stress cracking may take anywhere from a couple of hours to years to occur. Figure 4-438 illustrates some examples of typical hydrogen sulfide cracking failure, with a characteristic brittle fracture and final ductile-shear lip. Hydrogen sulfide stress cracking occurs on most nonferrous materials, for example aluminum alloys. Hydrogen sulfide stress cracking can occur if appropriate conditions such as high residual or applied tensile stress, corrosive environment and susceptible metallurgy exist simultaneously.

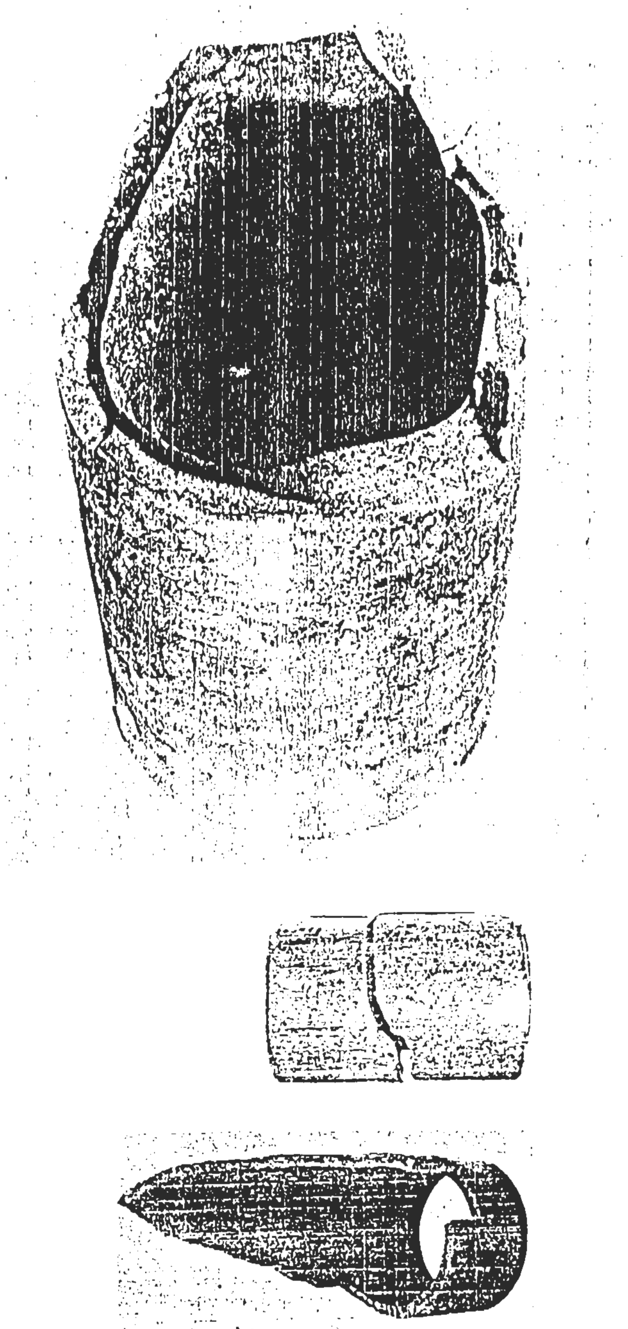
To prevent or reduce the possibilities of problems associated with hydrogen sulfide cracking the following measures should be considered [184]:

- Strength: steels with yield strength of 90,000 psi or less and Rc 22 hardness or less should be used. As the strength goes up, the time to failure goes down. This can be seen in Figure 4-439.
- Reduce applied or residual stress levels.
- Reduce hydrogen sulfide concentration levels.
- Increase pH levels. Figure 4-40 shows that steel fails much less rapidly when the pH is above 7 or 8.
- As the temperature goes above 150°F (66°C), the susceptibility to cracking decreases.

Stress Corrosion

Stress-corrosion cracking occurs when a metal is under constant tensile stress and exposed simultaneously to a corrosive environment. The source of this stress can be external (caused by slip or tong notch, etc., on drillpipes, collars and tool joints, and the weight of the drill stem) or it can be residual in the metal from heat treatment or cold working. The area damaged by the slips becomes stressed and undergoes accelerated corrosion as it becomes anodic to the rest of the unstressed area of the pipe. The appearance of damage is of a brittle mechanical fracture, but it results from local corrosion attack. The cracks are intergranular or transgranular depending on the metal structure and the corrosive environment. Most structural metals and alloys are susceptible to this form of attack in specific environments. Cracks generally develop perpendicular to the applied stresses. They are randomly oriented and can vary in degree of branching depending on the metal structure and composition, the nature of stress and the corrosive environment. Cracks vary from being totally branchless to extremely branched like the "river delta." As the stress increases, the time before failure decreases. It is speculated that there is a minimum stress required to prevent cracking, depending on the alloy composition, environmental temperature and composition. The minimum stress may vary from 10% of the yield stress to 70% of the yield stress. Thus, for each alloy-environment combination there is minimum or threshold stress [185].

It is very important to have some idea of the service life of a component. As stress corrosion proceeds, the cross-sectional area is reduced, resulting in a cracking failure due to mechanical action. Figure 4-441 shows the rate of cracking as a function of crack depth for a specimen subjected to constant tensile load. At first, the crack propagation rate is roughly constant; but as cracking progresses, the cross-sectional area decreases. Eventually, the rate of



(a)

(b)

(c)

Figure 4-438. SSC failure, drillpipe (a); tubing failure (b); coupling (c). *(From Refs. [184,218].)*

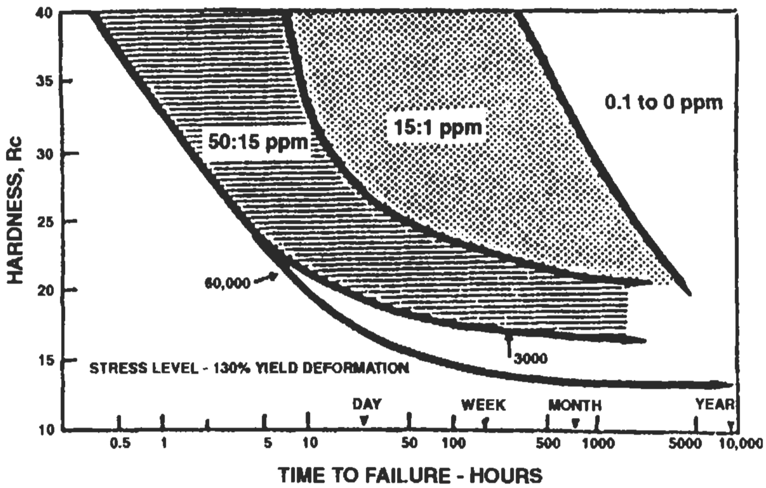


Figure 4-439. Approximate failure time of carbon steel is 5% NaCl and various parts per million of hydrogen sulfide. *(From Ref. [184].)*

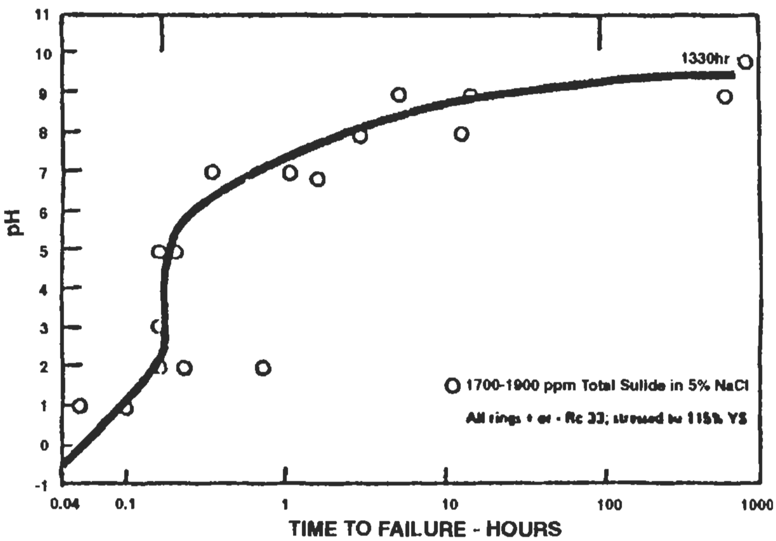


Figure 4-440. Steel fails much less rapidly when pH is above 7 or 8. *(From Ref. [184].)*

cracking increases to the crack depth, where rupture occurs as the applied stress is greater than the ultimate strength of the metal at that point. Figure 4-442 shows the relationship between the time of exposure and extension of a specimen during stress-corrosion cracking. The width of the crack does not increase as time progresses. However, just before rupture, conditions are

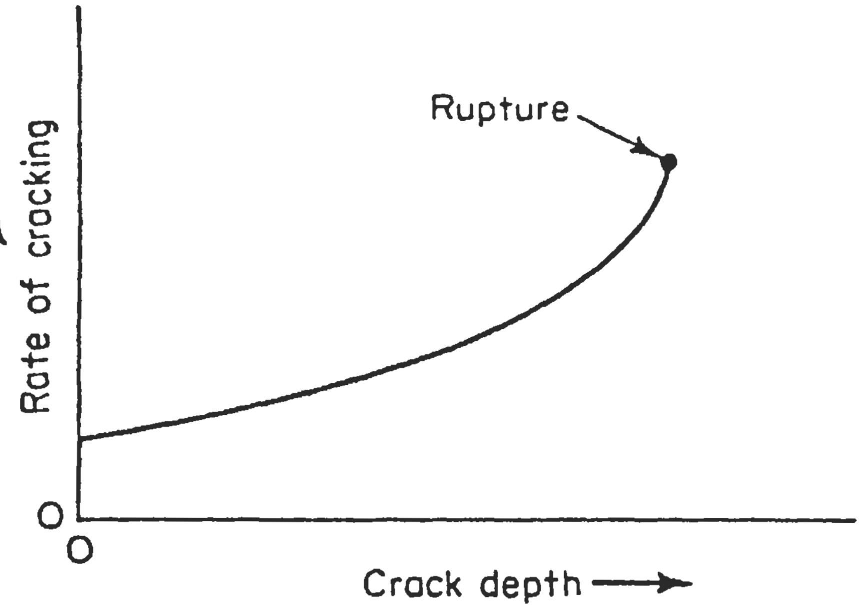


Figure 4-441. Rate of stress—corrosion crack propagation as a function of crack depth during tensile loading. *(From Ref. [183].)*

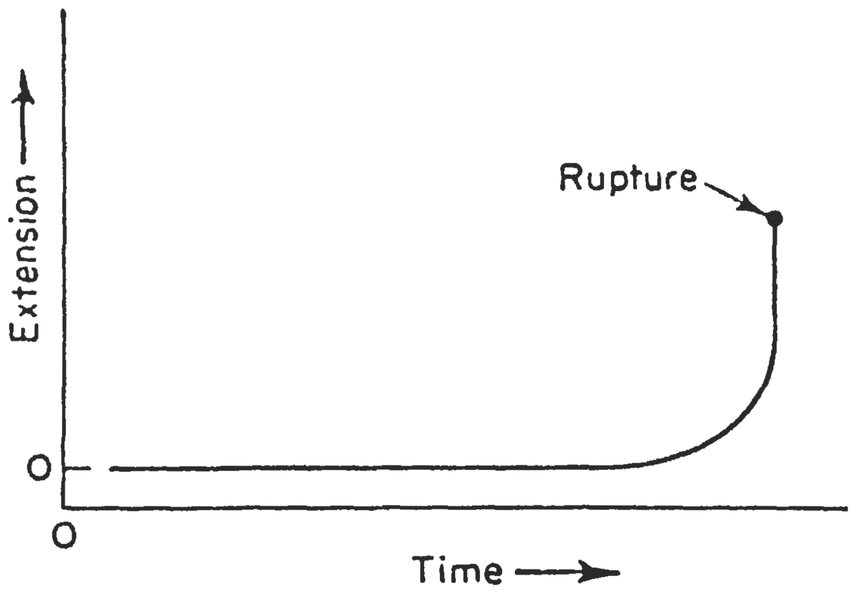


Figure 4-442. Specimen extension as a function of time during constant–load stress–corrosion cracking test. *(From Ref. [183].)*

different, extensive plastic deformation is observed and large increases in extension occur. This behavior further emphasizes the importance of better understanding this concept.

To reduce or prevent stress corrosion cracking the following methods can be employed:

- Eliminate unfavorable environments. The presence of oxygen and other oxidizers is a critical factor in stress corrosion cracking. For example, the cracking of austenitic stainless steel in chloride solutions can be reduced or completely eliminated if oxygen is removed.
- Avoid surface discontinuities such as pits, slip marks (notches) and other damage that act as stress risers. Stresses concentrate at the tip of the "notch." Therefore, stress-corrosion cracks usually originate from the base of a pit.
- Lower the stress levels below the "threshold" by annealing, using thicker sections or by reducing the load.
- Change the alloy to one which is more resistant to stress-corrosion cracking.
- Use suitable corrosion inhibitors in sufficient quantities.

Corrosion Fatigue

As pointed out earlier, even if the metal is under stress for an infinite number of cycles, it will not fail as long as the stress level is below its fatigue limit. However, if the metal is damaged or notched in any way, the fatigue resistance decreases, as shown in Figure 4-443. Note that the nonferrous metals have no

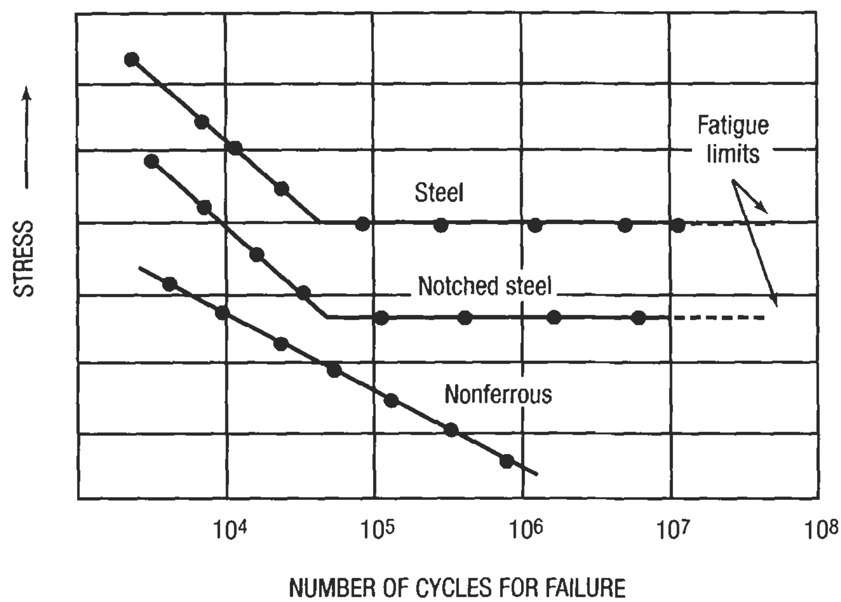


Figure 4-443. Fatigue behavior of ferrous and nonferrous alloys. *(From Ref. [183].)*

fatigue limit. Their fatigue resistance increases with reduction in applied stress, but continues to depend on the stress level [183,193].

The simultaneous action of cyclic stress alternating tensile and compressive and corrosive attack is known as corrosion fatigue. Corrosive attack can be in the form of pitting. These pits function as notches, acting as stress risers and initiate cracks. Once a crack is formed, the probability of pipe failure is enhanced by further corrosion as corrosion is accelerated by action of stress. The tip of the crack deep within the fracture, the area under the greatest stress, is anodic to the wider part of the crack. As corrosion progresses, the metal at the tip of the crack goes into the solution, the crack deepens and eventually penetrates the wall of the tube.

Figure 4-444 graphically illustrates the effect of corrosive environments on the fatigue life of a steel. It is evident in Figure 4-444 that the fatigue life is subject to considerable reduction under corrosive conditions. The reduction is severe in highly corrosive environments and behaves like nonferrous S-N behavior shown in Figure 4-443. This is due to the fact that in the presence of corrodents, there is no true endurance limit, and the failure may occur after any combination of number of stress cycle and stress level. Jamal Azar states that an increase in corrodents such as oxygen, carbon dioxide and chloride content result in decrease in fatigue strength of the drillpipe. Figures 4-445 to 4-450 summarize the corrosion fatigue data obtained and presented by Azar [194].

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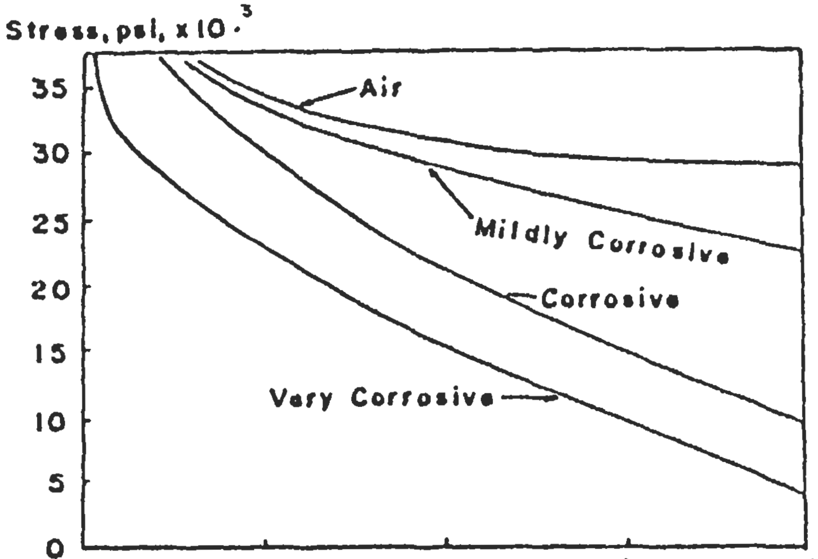

Figure 4-444. Effect of corrosive environment on fatigue life. *(From Ref. [193].)*

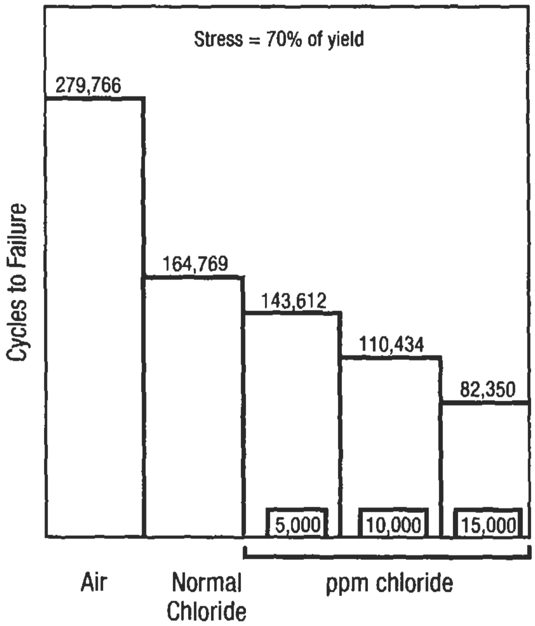

Figure 4-445. Chloride fatigue test—dispersed lignosulfonate mud, 9.5 pH. *(From Ref. [194].)*

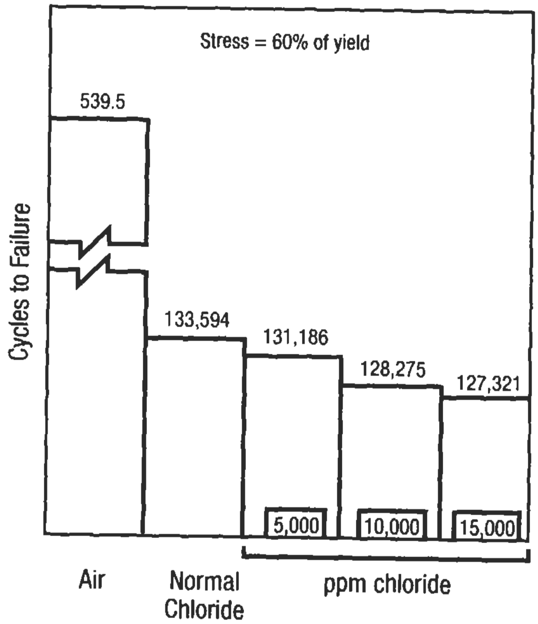

Figure 4-446. Chloride fatigue test—nondispersed bentonite mud, 8.5 pH. *(From Ref. [194].)*

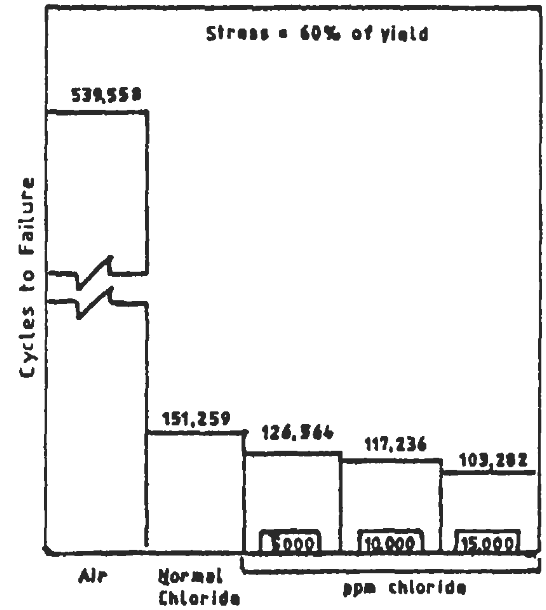

Figure 4-447. Chloride fatigue test—nondispersed bentonite mud, 9.5 pH. *(From Ref. [194].)*

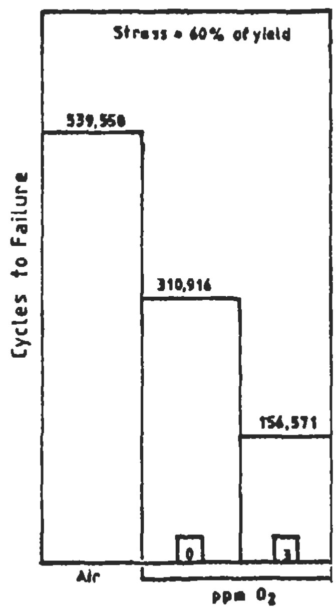

Figure 4-448. Oxygen fatigue test—lignosulfonate mud, 9.5 pH. *(From Ref. [194].)*

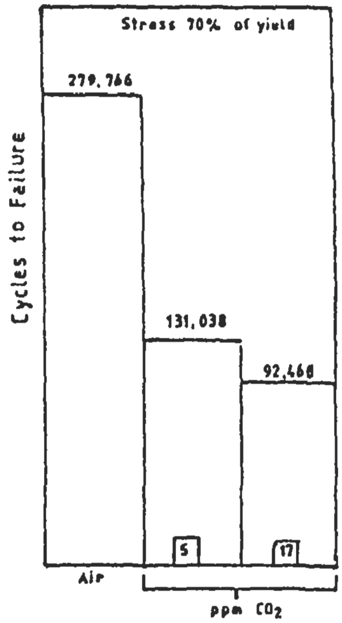

Figure 4-449. Fatigue test CO_2 in freshwater mud. *(From Ref. [194].)*

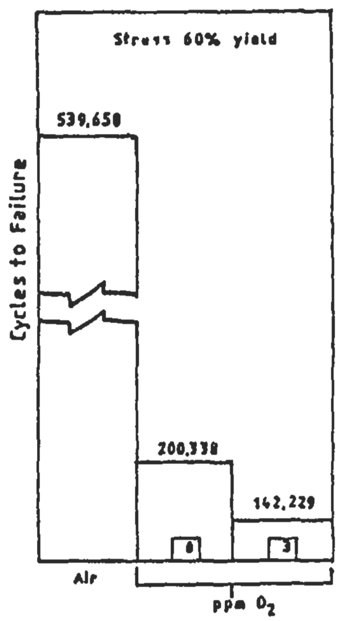

Figure 4-450. Oxygen fatigue test—nondispersed mud, 8.5 pH. *(From Ref. [194].)*

(text continued from page 1287)

Corrosion fatigue, therefore, is a special case of stress-corrosion cracking and fatigue failure. Figure 4-451 shows an example of pipe failures due to corrosion fatigue. Corrosion fatigue can be prevented or reduced by:

- reducing the stress on the metal by altering the design, heat treatment;
- use of proper corrosion inhibitors;
- use of proper coatings;
- keeping the drillstring under continuous tension.

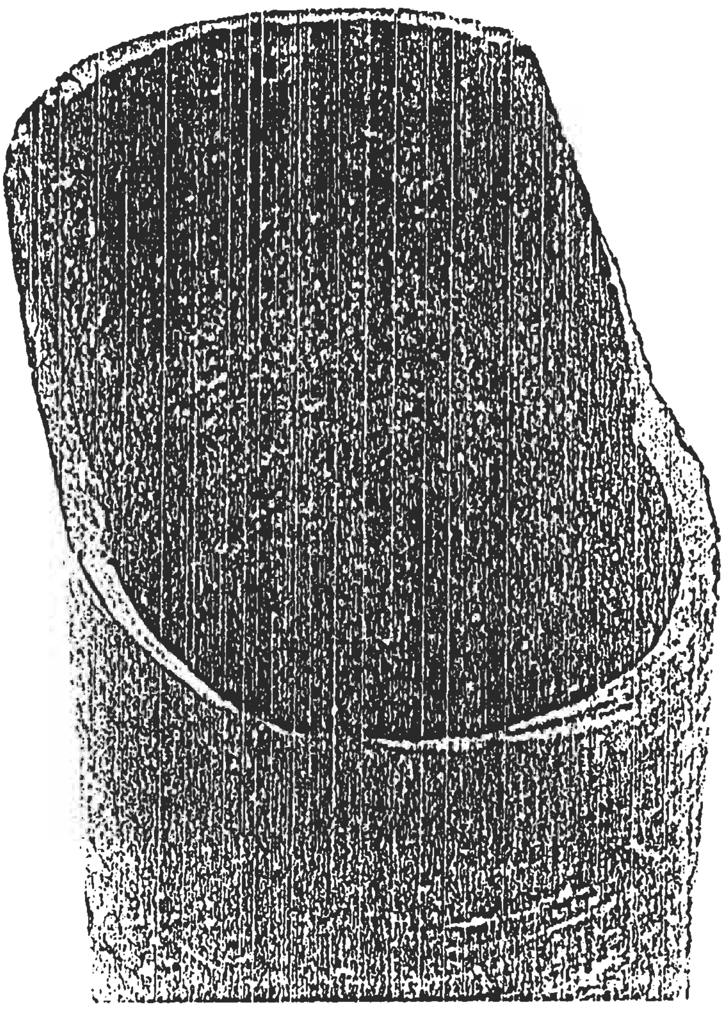

Figure 4-451. Corrosion—fatigue failure of drillpipe caused by internal pitting. *(From Ref. [218].)*

Factors Influencing Corrosion Rate

pH

The pH is one of the most important characteristics of an electrolyte, commonly expressed as a number between zero and fourteen, and is the negative logarithm of the hydrogen ion concentration [195].

$$pH = -\log[H^+]$$

The greater the concentration of hydrogen ions, the higher the acidity of the solution and the lower the pH. Solutions with pH of 7 are neutral solutions. Hydrogen ions (H^+) force the pH towards zero, and solutions with lower pH than pH of 7 are acidic. Hydroxyl ions (OH^-), on the other hand, make the solution alkaline and the solutions have higher pH values than 7. The pH variation affects the S-N curve behavior of steel. Figure 4-452 shows that as the pH is increased from 6.6 to 12.1 or 13, the threshold or fatigue limit of steel is restored in aerated saltwater systems. Figure 4-453 shows the relationship of failures by embrittlement (sulfide cracking) and pH. The time to failure increases as the pH rises. Below pH 7, failure occurs in less than an hour in the presence of hydrogen sulfide. As the pH rises above 7, time to failure rapidly increases. Both carbon dioxide and hydrogen sulfide lower the pH level to acidic regions and, consequently, increase the corrosion rate. Figure 4-454 illustrates the effect of pH on corrosion of mild steel. In high-pH range the corrosion reaction is anodically controlled. As the pH values decrease, the corrosion reaction gradually shifts to cathodic control. The resulting decrease in corrosion rate can be explained by the formation of a protective layer of hydrous ferrous oxide

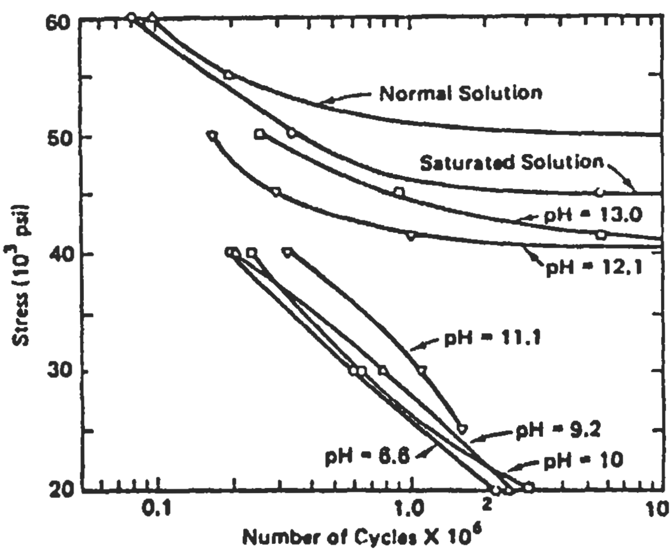

Figure 4-452. Effect of pH on corrosion fatigue in aerated saltwater. *(From Ref. [197].)*

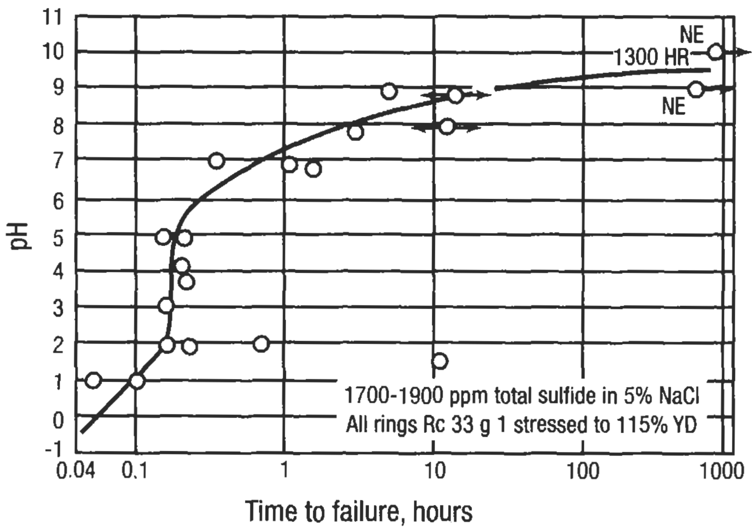

Figure 4-453. Relationship between pH and time to failure. *(From Ref. [219].)*

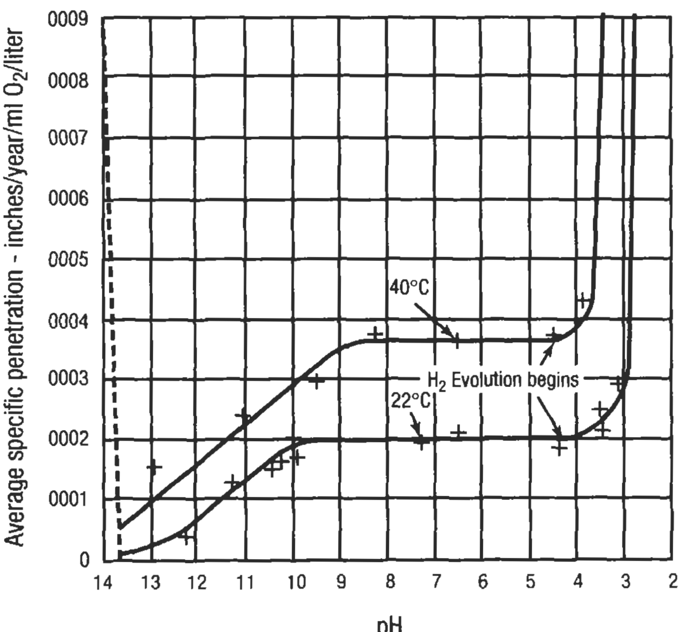

Figure 4-454. Effect of pH on corrosion of mild steel. *(From Ref. [219].)*

on the metal surface formed by the corrosion reaction. The corrosion rate is dependent upon the diffusion rate of oxygen through the protective layer to the metal surface. The corrosion rate increases with increasing oxygen concentration and with erosion of the protective layer in the presence of high-velocity, turbulent flow, which is often imposed on the drillstring. As the pH decreases further, the protective film breaks down, the reaction shifts back under anodic control, and the corrosion rate rises rapidly. During drilling operations it is very important to maintain high-pH levels around the flat part of the curve shown in Figure 4-451. However, it should be noted that aluminum alloys exhibit an increased rate of corrosion at pH higher than 10.5. Therefore, when aluminum drillpipes are used the pH values are generally kept between 7 and 10.5.

Temperature

The effect of temperature on corrosion rate is influenced by the following factors [188]:

1. Increase in temperature increases the redox reaction.
2. Solubility of gases in water decreases with increasing temperature.
3. Change in viscosity may affect the circulation, diffusion and other properties pertinent to the corrosion process.
4. Solubility of some reaction products can be affected by temperature variation.

When corrosion is due to the presence of mineral acids dissolved in water, resulting in hydrogen evolution, the corrosion rate generally increases with increasing temperatures. On the other hand, if the corrosion is due to dissolved oxygen in water, the rate decreases with increased temperature. This is due to the fact that oxygen solubility in water decreases with a rise in temperature. The forementioned fact will only be true in an open system where oxygen coming out of solution is free to escape. However, in a closed system, where oxygen coming out of solution cannot escape, the rate of corrosion increases with increase in temperature (Figure 4-455).

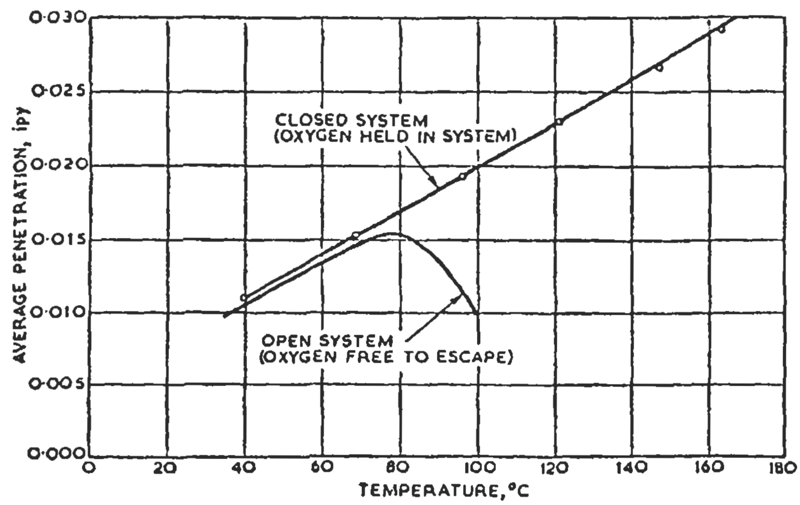

Figure 4-455. Effect of temperature on the corrosion of steel in water containing dissolved oxygen. *(From Ref. [188].)*

If calcium or magnesium bicarbonates are present in water, the rise in temperature decomposes them, and subsequent evolution of carbon dioxide will result in a higher corrosion rate, while at the same time calcium and magnesium carbonates may deposit on the metal surface. This scale may be protective, thus slowing the corrosion rate; however, it can create concentration cells if it is deposited loosely, exposing parts of the surface.

In systems with considerable temperature variations over the metal surface, the warmer areas will be anodic to the cooler areas. The creation of a cell usually leads to pitting corrosion of the anodic area.

Velocity

The velocity of fluids over the metal surface has an effect on the corrosion rate through influencing other factors responsible for corrosion. High velocity may increase erosion corrosion by either washing away the protective film or by mechanically agitating the metal surface. On the other hand, stagnant systems where the fluid velocity is zero may experience deposition of sludge and other suspended solids. This deposition may create concentration cells, resulting in pitting corrosion [196–198].

Systems with extremely high velocity of turbulent flow give rise to pressure variations that may result in cavitation corrosion. When oxygen is present, low-velocity areas receive less oxygen and become anodic to the high-velocity areas receiving high concentrations of oxygen and, thus, corrode. Figure 4-456 illustrates the effect of velocity and temperature on corrosion rates of steel pipes in an oxygen-contaminated system. Figure 4-457 shows the effect of velocity on the corrosion rate of steel pipes of various sizes. At 80°F the corrosion rates for all three pipes rise to their respective limits and become more or less constant. However, at higher temperatures the corrosion rates of $\frac{1}{4}$ and $\frac{1}{2}$-in.

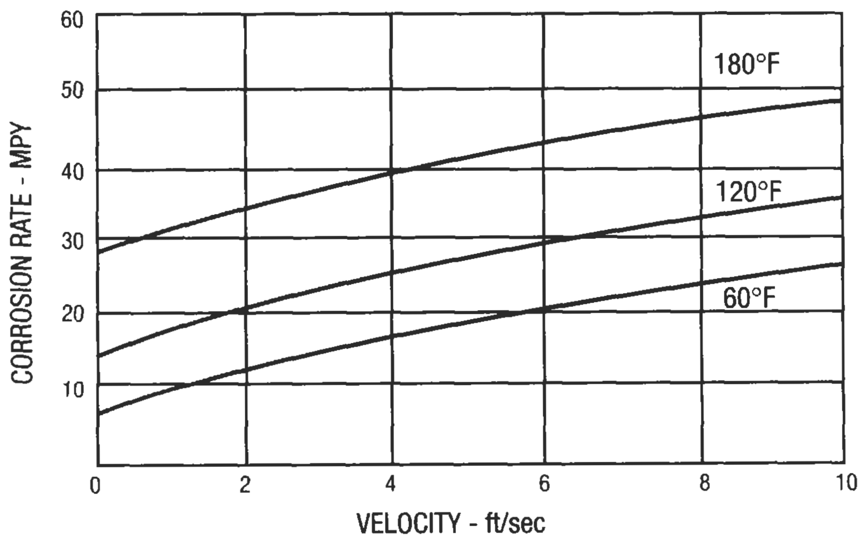

Figure 4-456. Velocity of the fluid accelerated corrosion rates. *(From Ref. [198].)*

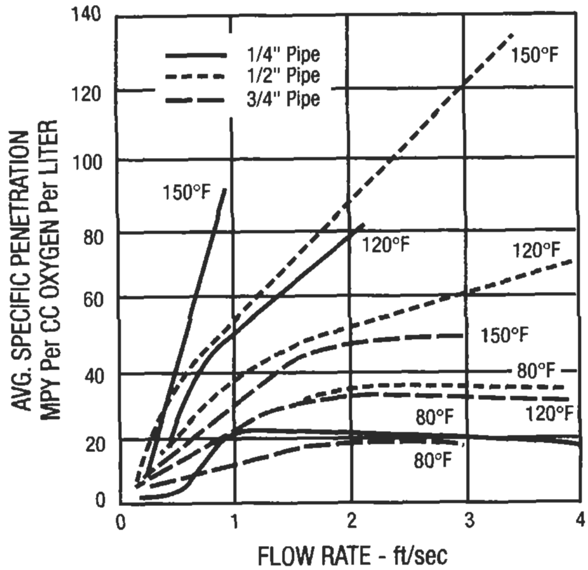

Figure 4-457. Effect of velocity of flow on the initial rate of corrosion of steel pipe. *(From Ref. [197].)*

pipes continue to increase as the velocity increases. A possible explanation for this behavior is that the corrosion rate increases with increasing velocity in small-diameter pipes, probably due to turbulence effects. Thus, the corrosion rate may be reduced by using either an oversized flow area or by reducing the velocity and, hence, minimizing the turbulence effect.

The critical velocity, which when exceeded may result in erosion corrosion, can be calculated by the equation presented in API RP 14E, which is [199]

$$V_c = \frac{C}{\sqrt{\gamma}} \tag{4-351}$$

where V_c = maximum allowable velocity in ft/s
C = a constant, typically 100–125
γ = fluid specific weight in lb/ft^3

Heterogeneity

Conditions necessary for the onset of corrosion are quite often provided by heterogeneities. These heterogeneities may very well exist within the metal or alloy or may be imposed by external factors. These heterogeneities can give rise to variations in potential on a metal surface immersed in an electrolytic fluid. The galvanic cell thus formed gives rise to flow of current that accompanies corrosion [188].

High Stresses

Highly stressed areas generally corrode faster than areas of lower stress. This is due to the fact that the more stressed areas are usually anodic and corrode more readily. The drillstem just above the drill collars is often susceptible to abnormal corrosion damage. High stresses and bending moments in this region may be partially responsible for this failure.

Microbial Activity

Microorganisms are present in most systems in one form or another. Their mere presence does not necessarily mean that they present a problem. Microbial-influenced corrosion is not a very significant problem in drilling operations. Their activity, however, does introduce corrodents in drilling fluids, reduces the pH of the environment and can attack the organic additives of the drilling fluids, thus producing corrosive products. Since the potential for problems does exist, it becomes necessary to consider the effects on metal corrosion resulting from microbial activity.

All microbes are classified into two main groups according to their oxygen requirements. These groups are:

Anaerobic organisms—Flourish in the absence of oxygen in environment with low redox potential.
Aerobic organisms—Require oxygen for survival.

The most common types of microorganisms found in oil fields that can cause corrosion related problem are now discussed.

Sulfate Reducers. Most of the oilfield corrosion problems arise from the activity of sulfate-reducing bacteria (SRB) belonging to genus *Desulfovibrio* and one of the genus *Clostridium.* They are anaerobic, but although inactive, they will survive in systems containing dissolved oxygen. They may grow under scale, debris or other bacterial masses where oxygen cannot penetrate, and in fresh or saltwater environments. SRB contribution to corrosion of metals is twofold; by direct corrosion attack, and by attack from products produced as a result of microbial activity. Figure 4-458 shows schematically the SRB direct corrosion of steel. A simplistic chemical mechanism of this process is as follows:

1. The metal goes into solution at the anode

   $$Fe \rightarrow Fe^{2+} + 2e^-$$

2. Reaction at the cathode results in molecular hydrogen that polarizes the cathode. Figure 4-459 shows the cathode polarization:

   $$H_2O \leftrightarrows H^+ + OH^-$$

   $$2H^+ + 2e^- \rightarrow H_2$$

3. Depolarization of the cathode by SRB. SRB contain an enzyme called hydrogenase, which allows the utilization of hydrogen to reduce sulfate to sulfide:

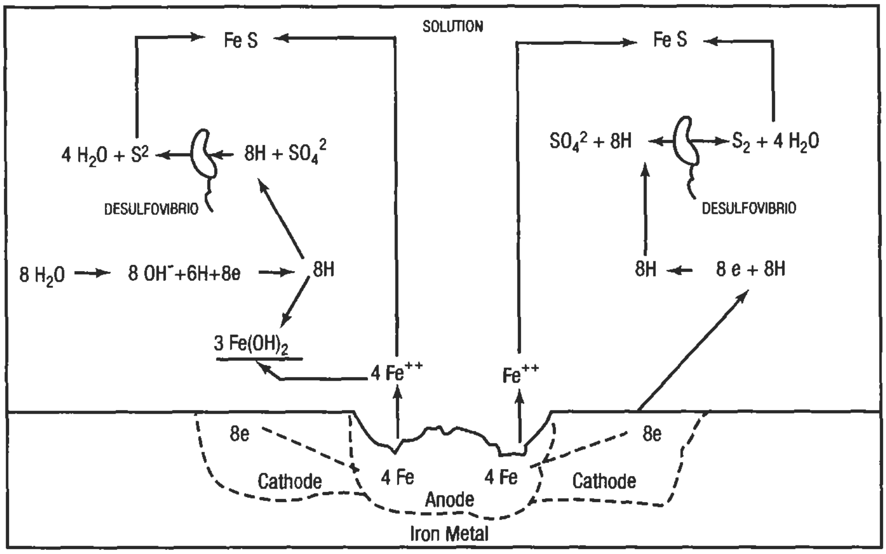

Figure 4-458. Diagram of polarization of local cathode by a film of hydrogen gas bubbles (cathodic area to right of anode is polarized). *(From Ref. [208].)*

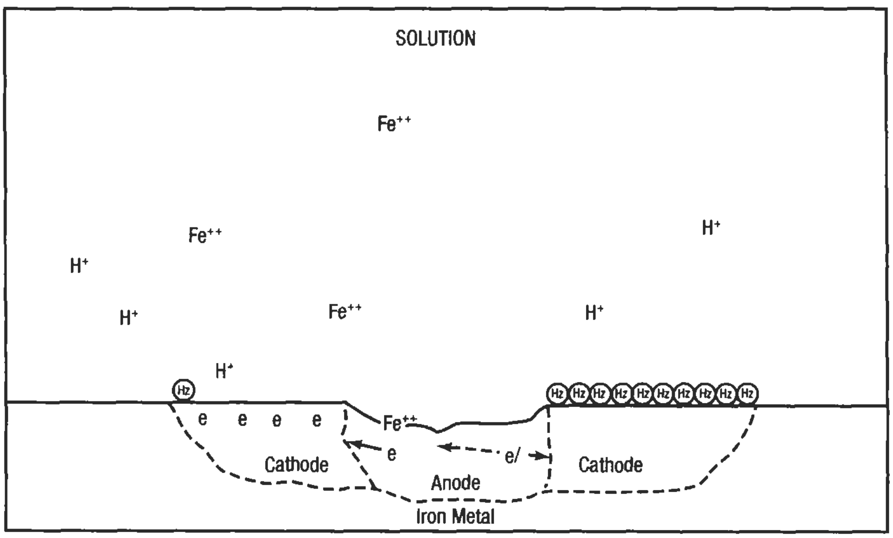

Figure 4-459. Diagram of the bacterial corrosion of steel or iron by *Desulfovibrio* bacteria (corrosion products are underlined). *(From Ref. [208].)*

$$SO_4^{2-} + 8H \xrightarrow{R} S^{2-} + 4H_2O$$

4. Corrosion product reaction

- Ferrous ions (Fe^{2+}) combine with sulfide ions (S^{2-}) to form black ferrous sulfide FeS):

$$Fe^{2+} + S^{2-} \rightarrow FeS$$

- Ferrous ions (Fe^{2+}) combine with hydroxyl ions (OH^-) to form brownish red ferrous hydroxide [$Fe(OH)_2$]:

$$3Fe^{2+} + 6OH^- \rightarrow 3Fe(OH)_2$$

The overall corrosion reaction equation is

$$4Fe + SO_4^{2-} + 3H_2O \rightarrow FeS + 3Fe(OH)_2 + 2OH^-$$

If the water system contains dissolved carbon dioxide, iron carbonate may form:

$$CO_2 + H_2O \rightarrow H_2CO_3$$

$$FeS + H_2CO_3 \rightarrow FeCO_3 \downarrow + H_2S \uparrow$$

The second way SRB influences corrosion of metal is by producing products such as hydrogen sulfide that can affect corrosion resistance of metals.

$$SO_4^{2-} + 2H^+ + 4H_2 \xrightarrow{SRB} H_2S + 4H_2O$$

In the above mechanism, both hydrogen ion and molecule are utilized by SRB to convert SO_4^{2-} to H_2S. The consumption of hydrogen depolarizes the cathode and leads to an increased rate of corrosion.

Slime-Forming Bacteria. Several forms of bacteria produce a slimy capsule under certain environmental conditions. These organisms are "heterotrophic," that is, they obtain their energy from organic sources such as sodium lactate. The reaction is

$$2CH_3CHOHCOONa + MgSO_4 \rightarrow 2CH_3COONa + CO_2 \uparrow + MgCO_3 + H_2S \uparrow + H_2O$$

These microorganisms become a problem when their numbers are large enough to produce corrosives such as carbon dioxide and hydrogen sulfide. *Pseudomonas, Bacillus* and *Flavobacterium* are examples of slime-forming bacteria.

Iron-Oxidizing Bacteria. These are aerobic organisms capable of growing in systems with less than 0.5 ppm oxygen. They oxidize iron from ferrous to the ferric state by the following mechanism:

$$4FeCO_3 + O_2 + 6H_2O \rightarrow 4Fe(OH)_3\downarrow + 4CO_2\uparrow$$

The carbon dioxide produced can contribute to the corrosion of metal. The deposits of ferric hydroxide that precipitate on the metal surface may produce oxygen concentration cells, causing corrosion under the deposits. *Gallionalla* and *Crenothrix* are two examples of iron-oxidizing bacteria.

Sulfur-Oxidizing Bacteria. These are aerobic bacteria that oxidize sulfur or sulfur-bearing compounds to sulfuric acid according to the following equation:

$$2S + 3O_2 + 2H_2O \rightarrow 2H_2SO_4$$

The resulting environment is low in pH and extremely corrosive. *Thiobacillus* and *Beggiatoa* are good examples of this form of bacteria.

Other Microorganisms. There are several other microorganisms that affect the corrosion of metal directly or indirectly. Some examples are yeasts and molds, algae and protozoa. For the present purposes it is sufficient to realize that there are other micro-organisms capable of presenting corrosion problems.

There are several methods of monitoring microbial-influenced corrosion. Some methods are as follows:

- Sample culturing
- Filtration technique
- Metal surface examination, i.e., use of coupons

Methods to prevent or reduce problems associated with microbial corrosion will be discussed later. Some of them are:

- use of effective microbiocides
- removal of nutrients
- pH adjustment
- proper coatings
- cathodic protection
- monitoring the effectiveness of microbiocides

Corrodents in Drilling Fluids

The principal corrosive agents affecting the drillstem components in drilling fluids are dissolved gases, dissolved salts and acids. These corrodents may enter the system from the formations being drilled. They can also be introduced by the addition of makeup water, or other treating chemicals and processes. Also, they can be products of thermal degradation of chemicals and microorganism activity, etc.

Dissolved Oxygen

Oxygen dissolved in aqueous solutions, even in very low concentrations, is a leading cause of corrosion problems (i.e., pitting) in drilling. Its presence also accelerates the corrosion rate of other corrodents such as hydrogen sulfide and carbon dioxide. Oxygen plays a dual role both as a cathodic depolarizer and an anodic polarizer or passivator. Within a certain range of concentration the attack is dependent on the oxygen concentration. Above a certain concentration it may act as anodic passivator for certain metals resulting in reduction of the corrosion rate. It is thought that the concentration of oxygen is sufficient to cause ferrous to ferric oxidation at a faster rate than ferric ion has a chance to diffuse away from the metal surface. This results in deposition of a protective film of ferrous hydroxide on the steel surface. The chemical reaction in oxygen corrosion of steel in an aqueous solution is

$$2Fe + 2H_2O + O_2 \rightarrow 2Fe^{2+} + 4OH^- \rightarrow 2Fe(OH)_2 \downarrow \qquad (4\text{-}352)$$

Ferrous hydroxide precipitates from solution. Since this compound is unstable in oxygenated solution, it is oxidized to the ferric salt:

$$2Fe(OH)_2 + H_2O + \tfrac{1}{2}O_2 \rightarrow 2Fe(OH)_3 \qquad (4\text{-}353)$$

Step-by-step explanation of the reaction is as follows:

1. Oxidation occurs at the anode, where the metal goes into solution as it corrodes:

$$Fe \rightarrow Fe^{2+} + 2e^- \qquad (4\text{-}354)$$

2. Reduction occurs at the cathode, where oxygen combines with water and gains electrons to form the hydroxyl ion:

$$O_2 + 2H_2O + 4e^- \rightarrow 4OH^- \qquad (4\text{-}355)$$

3. The hydroxyl ion thus produced combines with ferrous ion produced in the first step and forms ferrous hydroxide (rust), which coats the cathode:

$$Fe^{2+} + 2(OH)^- \rightarrow Fe(OH)_2 \qquad (4\text{-}356)$$

There are basically two ways through which oxygen promotes corrosion. First, it is a strong cathodic depolarizer; that is, it retards cathodic polarization. In a corrosion process with an acid substance (dissolved hydrogen sulfide or carbon dioxide), hydrogen molecules (H_2) formed during reduction reaction accumulate at the cathode. The gaseous hydrogen thus forms an insulating blanket that reduces the current flow and increases the corrosion rate. This process is called polarization.

$$2H^+ + 2e^- \rightarrow \underset{\substack{\text{Step 1} \\ \text{(atomic hydrogen)}}}{2H^0} \rightarrow \underset{\substack{\text{Step 2} \\ \text{(molecular hydrogen)}}}{H_2} \qquad (4\text{-}357)$$

However, if dissolved oxygen is present, the second step of Equation 4-357 does not occur and instead

$$2H^0 + \tfrac{1}{2}O_2 \rightarrow H_2O \qquad (4\text{-}358)$$

Second, oxygen attacks the metal directly by precipitating iron oxide at the anode, thus, preventing anodic polarization by Fe^{2+} ions.

The corrosion rate of steel has been found by some investigators to be approximately proportional to the oxygen content up to 5.5 cm^3/L (Figure 4-460). Beyond this concentration the corrosion rate begins to decline. The probable cause for this behavior is that the oxide layer deposited at low oxygen concentrations is much more protective than the one formed in high concentrations of oxygen. Oxygen solubility in an aqueous fluid tends to decrease with increasing salinity (Figure 4-461) and increases with decreasing temperatures (Figure 4-462).

As the drilling progresses, oxygen is continuously dissolved into the drilling fluid, through the surface part of the circulatory system. As the drilling mud flows through solid-removal systems, gas separators, chemical additions or treatment systems and mud mining systems, it picks up oxygen. The addition of makeup water also increases the oxygen content in the drilling mud. The increase of oxygen content is further enhanced if the viscosity is high enough to impede gas or air breakout from the mud. Low temperatures and low salinity also cause increased oxygen absorption in drilling muds.

The most effective method of controlling corrosion due to oxygen is by minimizing the contamination of oxygen into the drilling mud. With methods available it is possible to minimize the concentrations to reasonable levels. Further control can be obtained by use of oxygen scavengers and degasification of drilling fluids. The control methods will be discussed later.

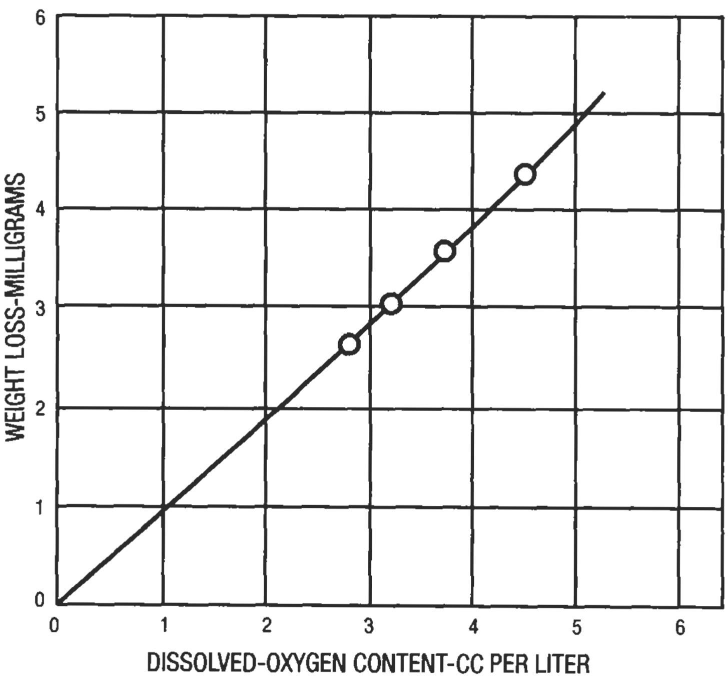

Figure 4-460. Corrosion in sodium chloride solution containing dissolved oxygen. *(From Ref. [197].)*

Dissolved Carbon Dioxide

Carbon dioxide dissolves in water to form a weak acid (carbonic acid), which reduces the pH of the solution and, consequently, increases its corrosivity. Corrosion caused by carbon dioxide is generally referred to as "sweet" corrosion, and results in pitting. The mechanism of carbon dioxide corrosion is as follows [197,198]:

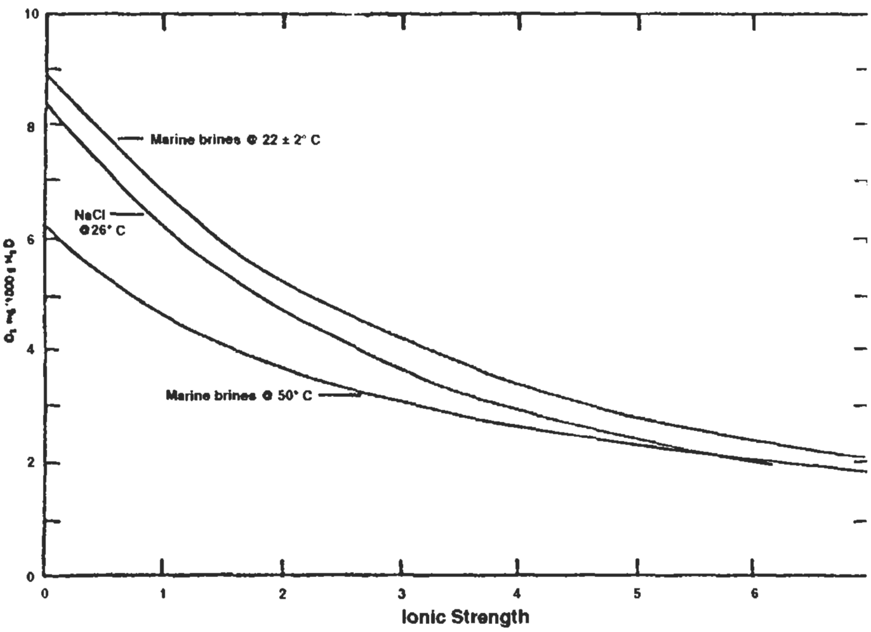

Figure 4-461. Oxygen solubility in salt solutions. *(From Ref. [197].)*

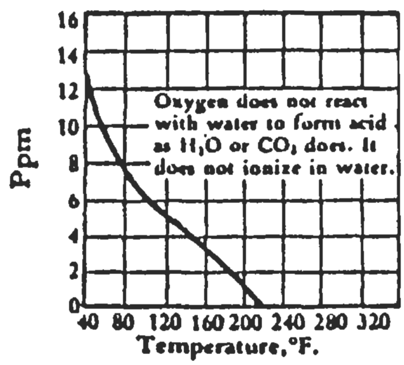

Figure 4-462. Oxygen solubility in water at varying temperatures. *(From Ref. [189].)*

1. Carbon dioxide dissolves in water to form carbonic acid

$$CO_2 + H_2O \rightarrow H_2CO_3 \quad (4\text{-}359)$$

2. which ionizes first to

$$H_2CO_3 \leftrightharpoons H^+ + HCO^-_3 \quad (3\text{-}360)$$

and then to

$$HCO^-_3 \rightarrow H^+ + CO_3^{2-} \quad (4\text{-}361)$$

3. At the anode metal goes into solution as it ionizes to

$$Fe \rightarrow Fe^{2+} + 2e^- \quad (4\text{-}362)$$

4. Finally, the carbonate ion combines with ferrous ion to form ferrous carbonate and hydrogen is evolved:

$$Fe^{2+} + 2e^- + 2H^+ + CO_3^{2-} \rightarrow FeCO_3 + H_2 \uparrow \quad (4\text{-}363)$$

The corrosion rate of carbon dioxide depends on metallurgy of the material, oxygen content and solubility of carbon dioxide in the aqueous solution. Solubility, in turn, depends on the amount of dissolved salt, temperature, partial pressure of carbon dioxide and oxygen, and velocity of the system. An aqueous solution that contains both oxygen and carbon dioxide in solution is more corrosive than the solution that contains only one of these gases equal in concentration to both gases. Figure 4-463 shows this fact by comparing various

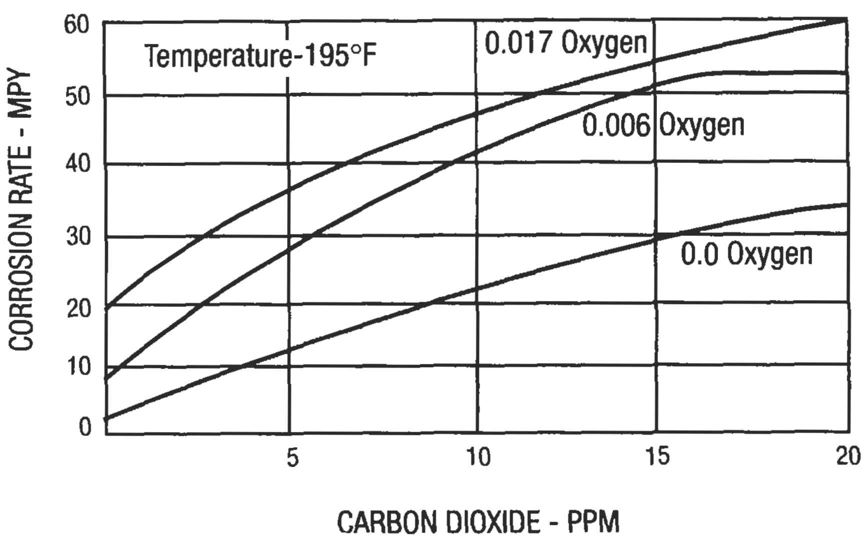

Figure 4-463. Effect of carbon dioxide concentration on corrosion rate. *(From Ref. [211].)*

combinations of carbon dioxide and oxygen in a static system. The corrosion rate increases with increasing oxygen and carbon dioxide concentrations. Figure 4-464 shows that the solubility of carbon dioxide increases with decreasing temperature. Figure 4-465 shows the influence of carbon dioxide partial pressure on the rate of corrosion. The corrosion rate increases rapidly until partial pressure of 200 psia, then gradually becomes more or less constant after 300 psia.

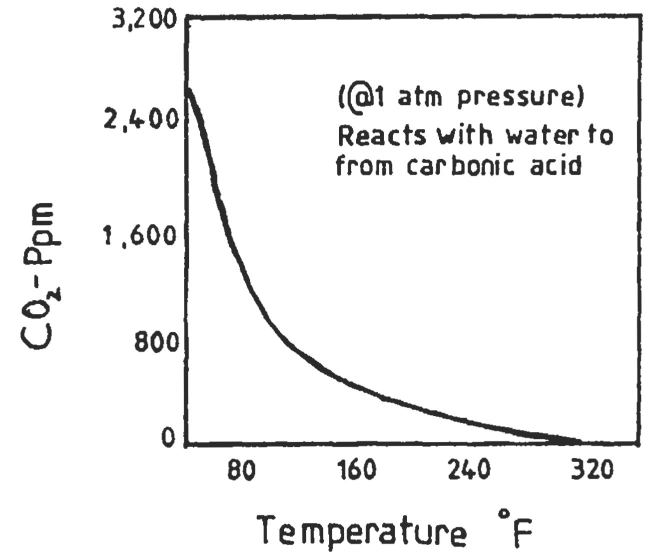

Figure 4-464. Effect of temperature on carbon dioxide solubility. *(From Ref. [189].)*

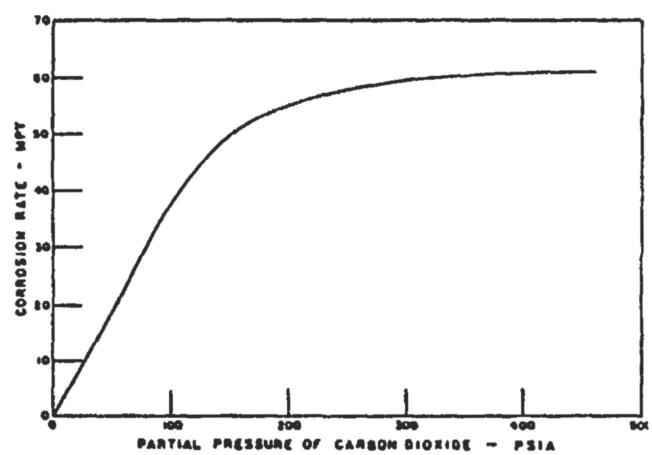

Figure 4-465. Corrosion of steel in distilled water containing carbon dioxide at various partial pressures. *(From Ref. [197].)*

Increase in partial pressure increases the solubility, consequently lowering the pH of the system (Figure 4-466).

When carbon dioxide is present during drilling operations, the corrosion rates are three to six times higher than those shown in Figure 4-463 according to EnDean [198]. Carbon dioxide may enter the drilling fluid through one or more of the following ways:

- influx of formation gas or formation water (primary way)
- through addition of makeup water
- thermal decomposition of dissolved salts and organic drilling fluid additives
- microbial activity

Carbon dioxide corrosion can be controlled by the use of caustic soda and lime and the addition of various inhibitors. Film-forming amine inhibitors are used to reduce the corrosion rates. The control measures will be discussed later.

Dissolved Hydrogen Sulfide

Hydrogen sulfide, as well as carbon dioxide, dissolves in water to form a weak acid. The acid solution thus formed is weaker than carbonic acid; nevertheless, it is corrosive enough to cause pitting problems on the drillstem. The presence of sulfide also promotes hydrogen absorption into the drillstem components. Hydrogen absorption, coupled with cyclic stressing of the drillstem, leads to failure such as sulfide stress cracking or hydrogen embrittlement of drillstem components. The chemical mechanism of hydrogen sulfide corrosion is as follows [191,197,200]:

1. Oxidation at the anode–where metal goes into solution:

$$Fe \rightarrow Fe^{2+} + 2e^- \tag{4-364}$$

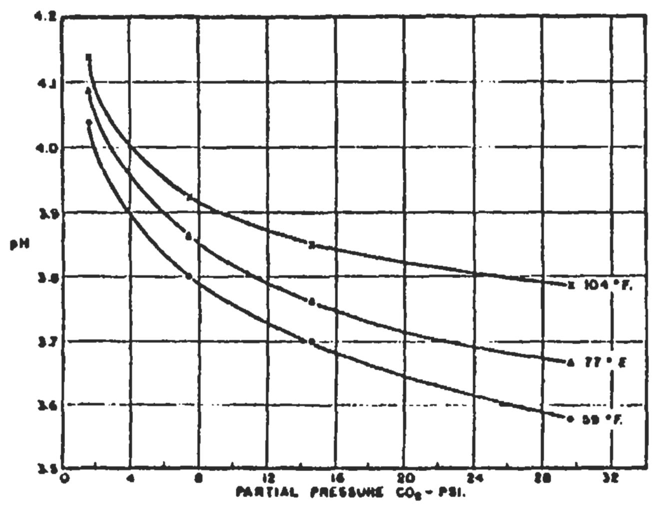

Figure 4-466. Effect of carbon dioxide partial pressure on the pH of condensate water. *(From Ref. [184].)*

2. Ionization of H_2S occurs

$$H_2S \rightarrow H^+ + SH^- \tag{4-365}$$

The anion, bisulfide SH^- further dissociates into anionic sulfide S^{2-} and cationic hydrogen ion H^+:

$$SH^- \leftrightarrows S^{2-} + H^+ \tag{4-366}$$

3. The ion S^{2-} reacts with ferrous Fe^{2+} ion to form black iron sulfide FeS corrosion product. The hydrogen ions are reduced by electrons produced by anodic reaction in step 1 and form hydrogen atom H^0:

$$Fe^{2+} + S^{2-} + 2H^+ + 2e^- \rightarrow FeS + 2H^0 \tag{4-367}$$

In absence of oxygen some hydrogen does manage to evolve and polarize the cathode to some extent. However, if oxygen is present, this polarization does not occur as discussed earlier, and results in accelerated corrosion attack. Hydrogen sulfide ionizes in two main stages when dissolved in fluid. The reactions mechanisms are

$$H_2S + NaOH \leftrightarrows Na^+ + HS^- + H_2O$$

$$NaHS + NaOH \leftrightarrows 2Na^+ + S^{2-} + H_2O$$

These reactions are easily reversed if the solution pH decreases, as can be seen in Figure 4-467.

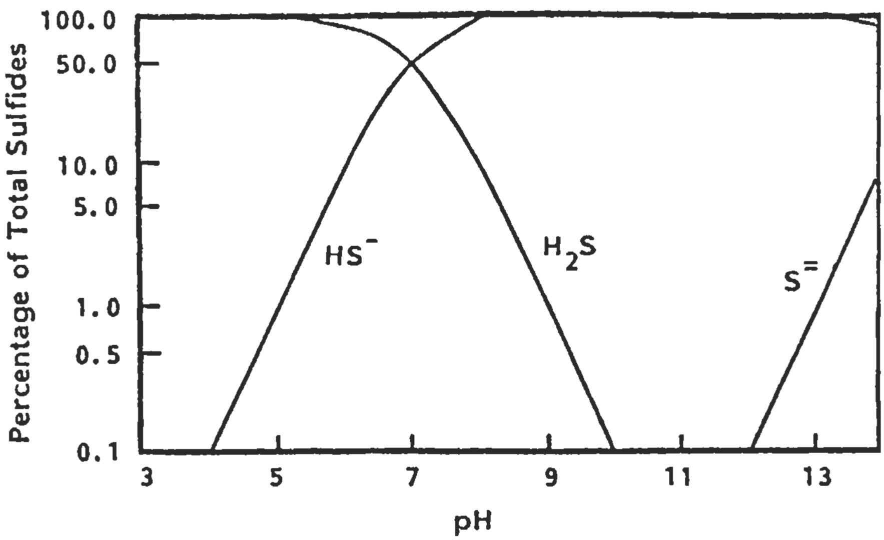

Figure 4-467. Approximate ionization of hydrogen sulfide in water at different pH values. *(From Ref. [191].)*

If carbon dioxide is present, the solution pH is reduced enough to convert the sulfide (S^{2-}) ion back into the dangerous bisulfide or molecular sulfide (H_2S) state. Figure 4-468 shows the variation of corrosion of a mild steel in distilled water containing varying concentrations of hydrogen sulfide. The corrosion rate increases sharply with increasing concentrations of hydrogen sulfide up to 150 ppm. It begins to decline rapidly after 400 ppm until it approaches 1,600 ppm. From 1,600 to 2,640 ppm it becomes approximately constant, due to inhibitive character of deposited iron sulfide at high concentrations.

Figure 4-469 shows the effect on corrosion rates of 1020 steel in different water systems with dissolved hydrogen sulfide. The difference in corrosion rates is due to different corrosion products formed in different solutions. In solution I, kansite forms. Kansite is widely protective as the pyrrhotite coats the surface giving slightly more protection until a very protective pyrite scale is formed. In solution II, only kansite scale forms, resulting in continued increase in the corrosion rate. Finally, in solution III, pyrite scale is formed as in solution I; however, continued corrosion may be due to the presence of carbon dioxide.

Hydrogen sulfide may enter the drilling fluid in one or more of the following ways:

- Influx of formation gas or formation water is the principal way.
- Through addition of makeup water.
- Thermal degradation of sulfur-containing drilling fluids additives (e.g., lignosulfates).
- Chemical reaction of sulfur-containing compounds (e.g., tool joint lubricants).
- Microbial activity.

The most effective methods of avoiding or reducing problems associated with hydrogen sulfide corrosion are:

- Avoid the use of high-strength steels. Relatively soft steels with low yield strength (up to Rc 22 and 90,000 psi) are resistant.

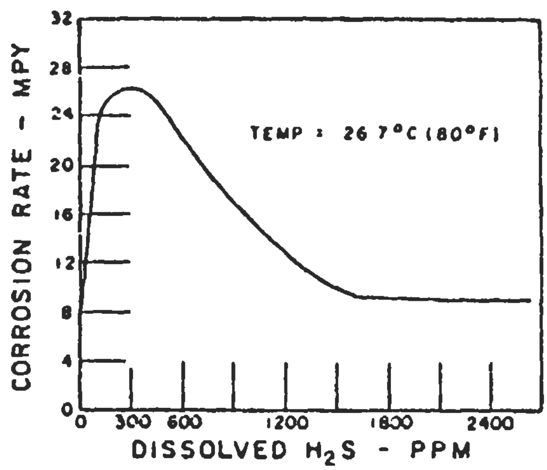

Figure 4-468. Corrosion rate of mild steel in distilled water containing varying concentrations of hydrogen sulfide. *(From Ref. [197].)*

- Use sulfide scavengers.
- Controlling pH with lime or caustic soda.

More detail on these control measures will be discussed later.

Dissolved Salts

Dissolved salts (chlorides, sulfates, carbonates or bicarbonates) generally increase the corrosivity of the drilling fluid. Figure 4-470 illustrates the influence of chloride, sulfate and bicarbonate on corrosion of steel. From the figure, it is obvious that the influence of dissolved salts is not governed by the salt concentration alone, but also by the type of dissolved salt. The ions can be divided into two groups: aggressive, such as chloride and sulfate, and inhibitive, such as carbonate or bicarbonate. All types of ions have inhibitive and aggressive properties to some extent, depending on their concentrations [197,201,202].

The aggressive ions either break down the protective films or prevent their formation and, in effect, increase the corrosion rate. In presence of chloride and sulfate ions the corrosion attack is more localized and, as a result, causes deep pitting. Inhibitive ions, on the other hand, tend to limit the attack and decrease the corrosion rate by forming protective films. The film is similiar to adherent carbonate-containing rust, which polarizes the anodic areas. When aggressive and inhibitive ions are present together, the aggressive ions, if present in sufficient quantities, interfere with the deposition of the protective layers.

Generally, the corrosivity of water containing dissolved salts increases at low-salt concentrations, until some maximum is reached, and beyond this maximum the corrosion rate decreases. Corrosion rate throughout the salt concentration range is under the influence of oxygen's ability to depolarize. It is believed that

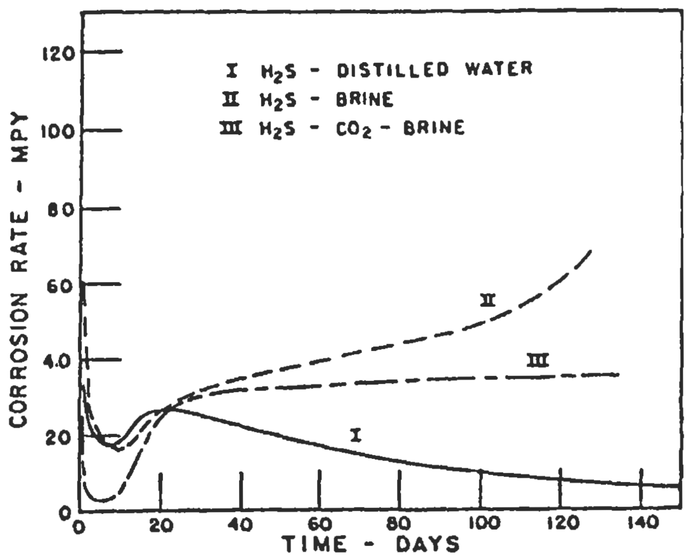

Figure 4-469. Corrosion rates in hydrogen sulfide–water systems. *(From Ref. [197].)*

the initial increase in corrosion rate is because oxygen solubility is not greatly decreased by low-salt concentrations. However, an increase in dissolved salt concentration increases the ion concentration of the electrolyte, which, in turn, reduces electrolytic resistance. This increases the electrical conductivity of the solution, allowing more current to flow between the anode and the cathode of the corrosion cell (Figure 4-471), resulting in an increased corrosion rate. Once the salt concentration is great enough to cause an appreciable decrease in oxygen solubility, there is a decreased rate of depolarization and the corrosion rate begins to decrease. Dissolved salts may also serve as a source of carbon dioxide or hydrogen sulfide contamination.

Chloride Salts

Chloride salts (sodium chloride, potassium chloride) tend to interfere with the formation of a protective layer over metals. Chloride salts destroy the passivity of some stainless steels and cause them to fail by rapid cracking under tensile stress at temperatures higher than about 176°F (80°C). This type of failure is called chloride stress cracking (CSC) [186,194].

Salts are sometimes added to drilling muds to obtain certain desired mud characteristics. They can also enter the drilling fluid through contamination by addition of makeup water, formation-fluid inflow, and drilled formations such as salt domes, gypsum or anhydride formations. In freshwater systems, if salt contamination reaches undesirable levels, the following methods should be considered for control.

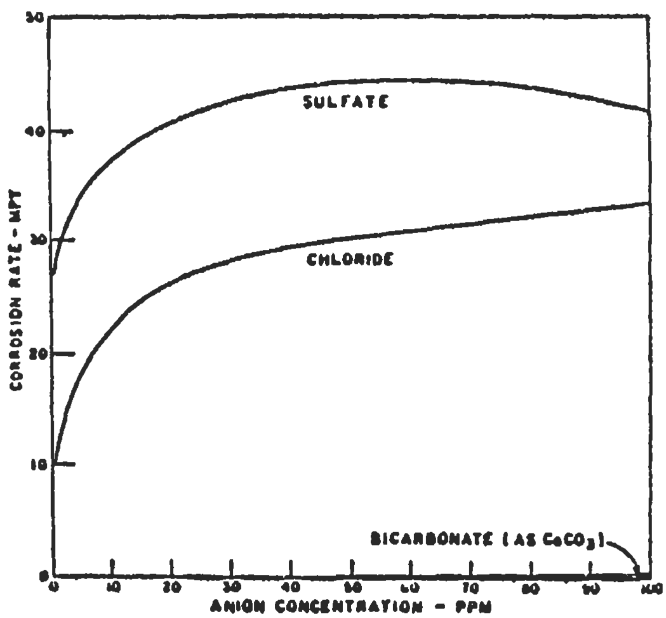

Figure 4-470. Influence of sulfate, chloride and bicarbonate on the corrosion of steel. *(From Ref. [197].)*

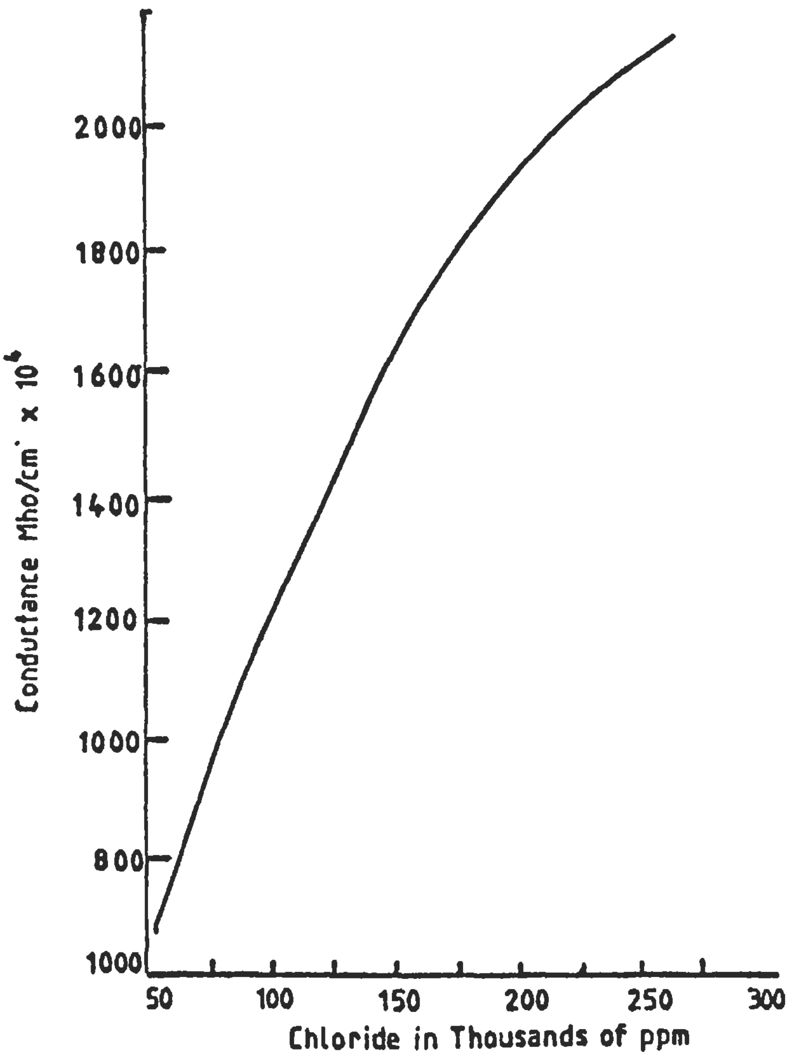

Figure 4-471. Conductivity of brine. *(From Ref. [202].)*

- Add chemical thinners such as Quebracho compounds, modified tannins and modified lignosulfonates.
- Use corrosion inhibitors such as film-forming inhibitors.
- Use internally plastic-coated drillpipes.
- Dilute the drilling fluid with fresh makeup water.

However, dilution can lead to added costs of mud-treating materials such as weighting materials and other chemical additives. In addition, disposal of excess drilling fluid can create problems on offshore drilling projects.

Organic Acids

Acids are substances that increase the hydrogen ion (H^+) concentration of the solution they are dissolved in. This, in turn, reduces the pH of the solution, and the corrosion rate increases. Acids may also attack the metal by dissolving the protective film on the metal surface. Presence of acid aggravates the oxygen-influenced attack and also hydrogen sulfide-promoted hydrogen embrittlement [203].

Organic acids can enter the drilling fluid through microbial activity or by thermal degradation of organic, drilling-fluid additives. The acids may also be formed by chemical reactions between drilling-mud additives or a result of other contamination. Some common acids found in drilling fluids are formic (HCOOH), acetic (CH_3COOH) and carbonic [H_2CO_3 (CO_2 in H_2O)].

Corrosion Monitoring and Equipment Inspections

The best way to combat corrosion is to maintain an effective corrosion-monitoring program to supplement good preventative measures. It is also very important to keep complete records of monitoring programs, control programs and failures that occur. The importance of well-qualified responsible personnel cannot be overemphasized as effective corrosion control depends on their efforts [201,204,205].

An effective corrosion control program should be able to detect evidence of corrosion and early identify the causes. Therefore, continuous monitoring is essential during drilling operations because the nature of drilling fluid corrosivity changes as the hole is drilled and different formations are penetrated. It is very important never to rely on any single method of monitoring corrosion. Several techniques should be used simultaneously whenever possible, and complete records should be kept.

Linear Polarization Instruments

Linear polarization instruments provide an instantaneous corrosion-rate data, by utilizing polarization phenomena. These instruments are commercially available as two-electrode "Corrater" and three electrode "Pairmeter" (Figure 4-472). The instruments are portable, with probes that can be utilized at several locations in the drilling fluid circulatory systems. In both Corrater and Pairmeter, the technique involves monitoring electrical potential of one of the electrodes with respect to one of the other electrodes as a small electrical current is applied. The amount of applied current necessary to change potential (no more than 10 to 20 mV) is proportional to corrosion intensity. The electronic meter converts the amount of current to read out a number that represents the corrosion rate in mpy. Before recording the data, sufficient time should be allowed for the electrodes to reach equilibrium with the environment. The corrosion-rate reading obtained by these instruments is due to corrosion of the probe element at that instant [184].

The limitation of these instruments is that they only indicate overall corrosion rate. Their sensitivity is affected by deposition of corrosion products, mineral scales or accumulation of hydrocarbons. Corrosivity of a system can be measured only if the continuous component of the system is an electrolyte.

Galvanic Probe

The galvanic probe continuously monitors the corrosion characteristics of the drilling fluid. The probe (Figure 4-473) consists of two dissimilar metal electrodes, usually brass and steel. The electrodes are mounted on, but insulated

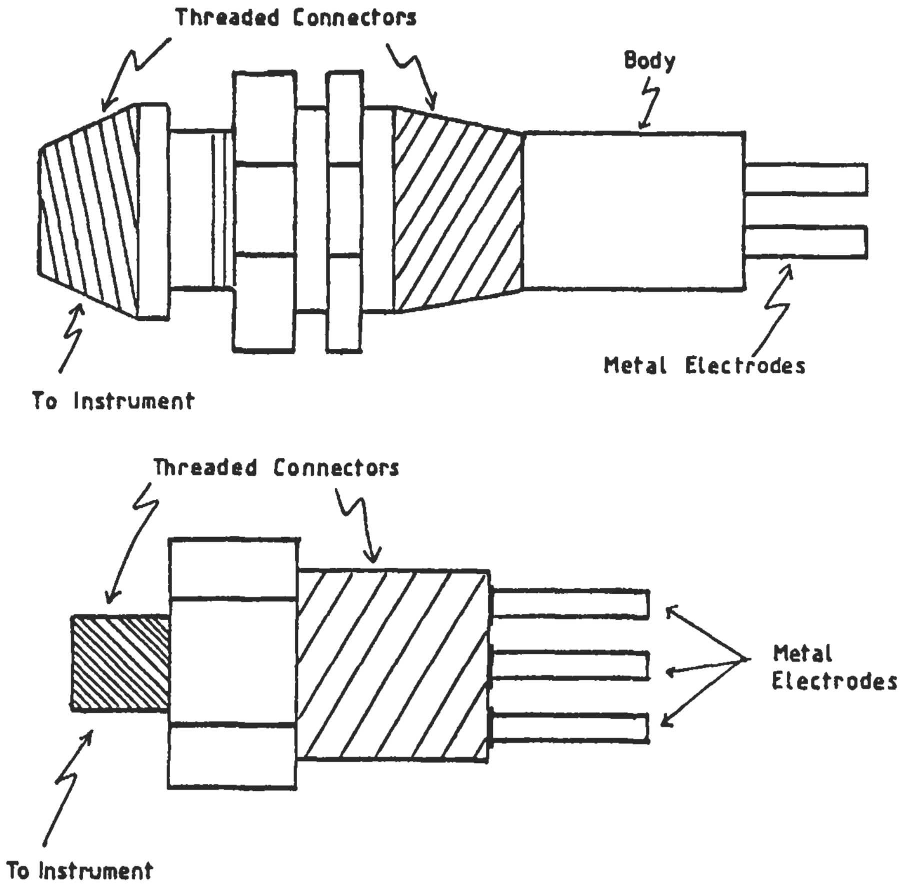

Figure 4-472. Linear polarization instrument probes.

from, a threaded high-pressure plug. These electrodes are connected to each other through a DC ammeter capable of detecting microamperes when the probes are immersed in an electrolyte. Enough time is allowed for the electrodes to reach equilibrium and read the current flow through the external loop. The current is generated by the corrosion process occurring on the electrodes [184].

The amount of current flow related to the environment is a measure of its corrosiveness. The probe generally registers low-current flow (0–10 mA) in slightly corrosive environments. However, high-current flows (40–100 mA) have been recorded in severely corrosive environments. The current intensity generally depends on oxygen concentration of the system, since oxygen depolarizes the brass cathode, thereby continuing the corrosion process of the cell. Among various locations of surface circulatory system, the instrument can be installed downstream of deaerator and in the standpipe.

If the instrument indicates current surge in an air-free system, it generally implies hydrogen sulfide contamination, but the galvanic probe is usually best suited to detect corrosion influenced by oxygen contamination.

Other limitations of this instrument are the same as those of linear polarization instruments discussed earlier.

Hydrogen Probe

The hydrogen probe (Figure 4-474) basically consists of a hollow, thin-walled, steel tube that is sealed on one end and the other end is equipped with a pressure gage. Once mounted in the system, the probe body corrodes. Some of the nascent hydrogen generated by the corrosion process in the presence of hydrogen sulfide diffuses through the tube wall. Once inside the void space in the tube, the hydrogen atoms combine to form molecular hydrogen gas. As these hydrogen gas molecules are too large to diffuse back through the tube wall, the pressure in the tube rises. The

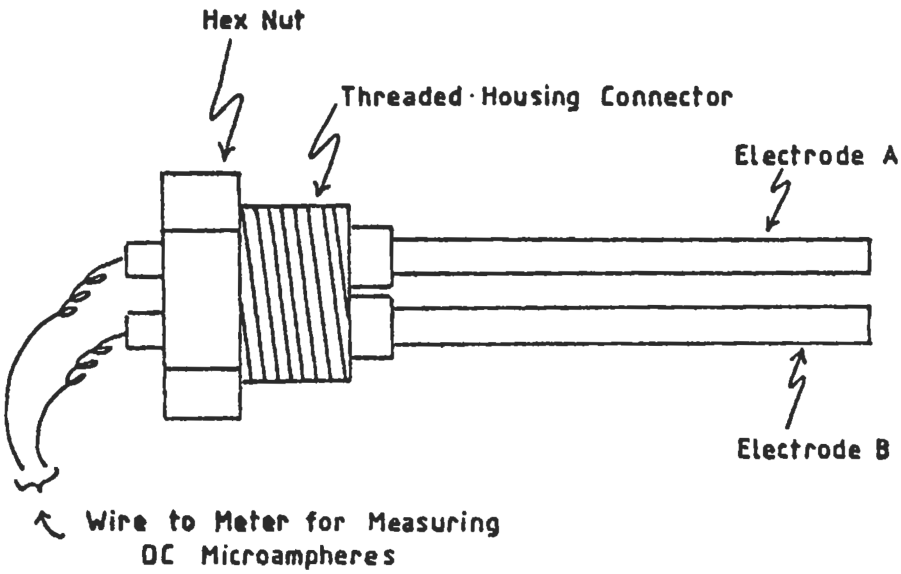

Figure 4-473. Typical galvanic probe.

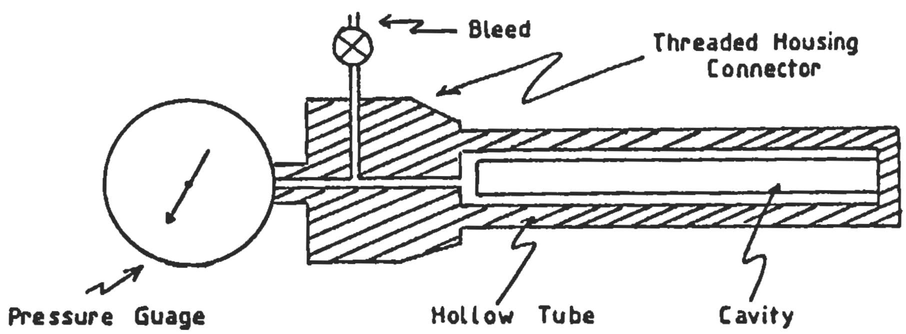

Figure 4-474. Hydrogen probe.

pressure rise in the tube is a function of the amount of hydrogen generated by the corrosion process. Hydrogen probes do not perform well in aerated fluids [201].

Corrosion Coupons

The most direct method of evaluating the corrosivity of a drilling fluid is the use of corrosion coupons (Figure 4-475). A drillstring corrosion coupon is a ring coupon machined from a section of tubing and sized to fit into the relief groove in the tool joint box (Figure 4-475). The coupons are normally installed

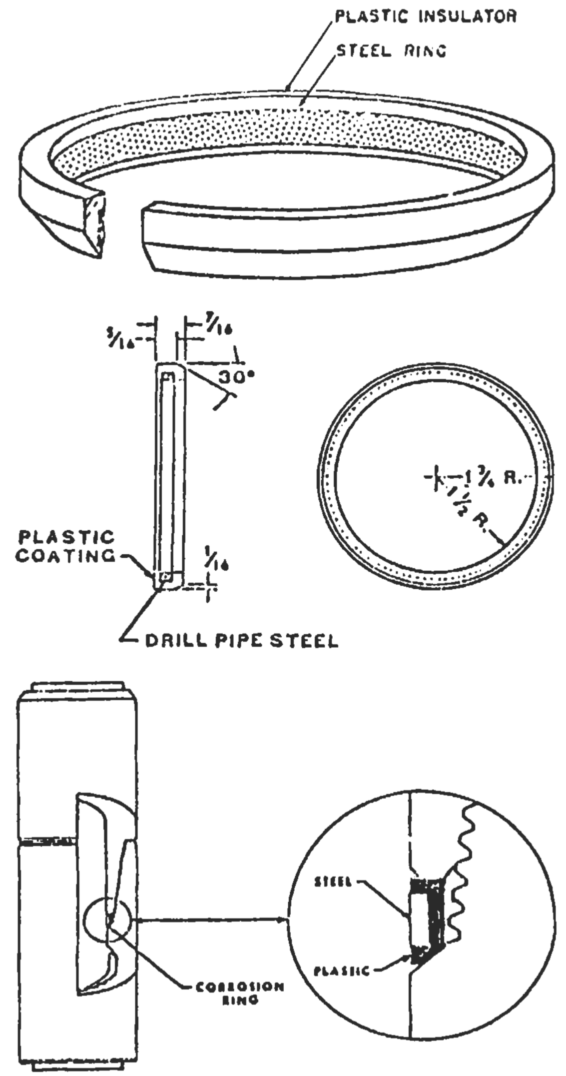

Figure 4-475. Drillstring corrosion coupon ring. *(From Ref. [220].)*

at the beginning of the drilling operations and changed at predetermined intervals (minimum of 40 h). The surface of the coupon is finished smoothly so that the effect of corrosion attack can be easily seen. The coupon is mounted in thermoplastic, which insulates it from the drillpipe to prevent the formation of a galvanic cell (Figure 4-472). Two coupons are generally used per string, one in the kelly-saver sub and a second at crossover sub above the drill collars [205].

To employ effective control measure, it is very important to determine the type of corrosion attack. Spot analysis of the corrosion film and careful visual examination of the coupon surface can help in determining the type and severity of corrosion attack. Generalized corrosion is represented by continuous attack over the entire surface, but no pitting. The pitting type of corrosion is represented by a high concentration of pits on the coupon surface. This type of corrosion attack is the most serious attack resulting from drilling fluids, as discussed earlier. The severity of general corrosion attack can be determined by weighing the coupon before and after exposure, and comparing the change in weight. Before installation the coupon must be clean (i.e., free of any corrosion, grease marks, drops of perspiration, etc.) and weighed. After exposure to the system for a minimum of 40 hr, the coupon is retrieved, visually examined, then cleaned and reweighed within one tenth of a milligram. The difference between the initial and final weights is attributed to corrosion and converted to the corrosion rate (mpy) using Equation 4-349. There are several factors such as handling, surface preparation and cleaning, etc., which can affect the results of the tests. The results obtained from this test assume uniform corrosion. Therefore, for proper analysis it becomes very important to include a complete description of the exposed coupons. One of the most important factors is visual inspection of the coupon, describing the form of attack and identifying the corrosion by-product. It is not very difficult to reduce the corrosion rate, so that overall mpy drops; however, the problem may still persist due to some pitting. It only takes a few sharp pits to cause failure. API Standard RP 13B contains complete information on this test [206]. Procedures provided by manufacturers of corrosion coupons should be followed.

To an appreciable degree the coupon experiences the same downhole conditions as the drillstring does. Therefore, its condition represents the corrosive effects of the downhole environment. However, limitations of this test are that the coupon is only exposed to the inside of the drillstring and not subject to the same stresses. The results obtained are only for certain depth of exposure, while the corrosion may vary appreciably up and down the hole. Finally, the results are not available until the tool is pulled out of the hole.

Chemical Testing

During drilling operations, chemical testing of drilling fluids is routinely carried out in the field. API has published recommended test procedures such as API RP 13B: "Standard Field Procedure for Testing Drilling Fluids" [206]. A number of service companies such as NL Baroid, Milchem and IMCO Services supply test kits for chemical analysis with procedures. These tests conform to standards set in API RP 13B [206]. The tests monitor the pH of drilling fluids and detect contaminants, such as dissolved gases and salts. These results are used either to detect any potential problem or to verify the effectiveness of remedial measures.

pH Determination

The two most commonly used methods of measuring the pH of a drilling fluid are a modified coalorimetric method, such as the pHydrion Dispenser;

and the electrometric method using a glass-electrode instrument, such as the Beckman or Analytical pH meter.

pHydrion Dispenser. pHydrion Dispenser provides a series of paper indicator strips that determine pH from 1 through 14. Once exposed to the medium, the strip changes color. The color, is matched to the range of color and corresponding pH values provided on the dispenser. The method is sufficiently accurate to allow the operator to read within 0.5 pH units. A limitation of this method is that it does not give accurate measurements in fluids containing high-salt concentrations. The method depends on the operator's ability to distinguish between different shades of color, and can vary from one operator to another [195].

pH Meter. The analytic pH meter offers a greater degree of accuracy than the coalorimetric method does. The instrument consists of two half-cells, the glass electrode half-cell and a reference half-cell. The glass electrode consists of a platinum electrode in a solution of fixed acidic pH. The electrode is placed inside a thin, glass membrane permeable to H^+ ions. The reference half-cell is usually either a calomel electrode [mercury in contact with mercury(I) chloride in saturated potassium chloride] or a silver/silver chloride electrode (silver wire coated with silver chloride saturated potassium chloride). The two half-cells are connected to each other through a sensitive voltmeter to form a pH meter. The electrode potential of the glass electrode is dependent on the (H^+) concentration of the solution in which it is placed. The electrical potential is amplified by means of a vacuum tube. In a commercial pH meter the glass electrode and reference half-cell are normally combined in a single unit that can be dipped into the solution under test, and the pH reading is indicated directly on a meter.

Oxygen Meter

Several commercially available oxygen meters are capable of reading dissolved oxygen concentrations in seconds. They can be calibrated for temperature and salt concentration for accurate readings. The oxygen probe operates on the same principle as the pH electrode, and develops a potential proportional to oxygen concentration. The potential is read on the oxygen meter as mg/L or ppm of dissolved oxygen present in solution [207].

Gas and Specific Ion Meters

Similar to the pH meter, gas meters employ specific ion electrodes. The electrodes generate a potential proportional to the activity of a specific ion in solution. The calibration is achieved in standard solution and results read in mV or concentration in mg/L or ppm on the meter. The water can be adapted to monitor the concentration of carbon dioxide, hydrogen sulfide, ammonia, chloride, calcium, potassium and sodium to name a few.

Hydrogen Sulfide Detection

Hydrogen sulfide can be detected by several tests. Some of the most commonly used in the drilling industry are as follows [195,207]:

1. An Alka-Seltzer® tablet gives off carbon dioxide when dissolved in aqueous solution. The gas is used to drive hydrogen sulfide out of drilling fluid samples. The H_2S then reacts with lead acetate paper in the bottle cap. The degree of discoloration is related to hydrogen sulfide concentrations.
2. The presence of sulfate-reducing bacteria can be detected by using API sulfate-reducing broth. If the broth is inoculated with drilling fluid and the color changes from yellow to black, the result is positive.
3. Sodium arsenite can be used to detect the presence of iron sulfide on the metal surface. Iron sulfide is the corrosion product of the reaction between hydrogen sulfide in drilling fluid and iron in the drillpipe. An acid solution of sodium arsenite reacts with the sulfide to form a bright yellow precipitate.
4. When present in low concentrations the presence of hydrogen sulfide can be detected by its characteristic odor of rotten eggs.
5. A prestressed roller bearing is used to detect the presence of hydrogen sulfide, but more specifically it is used to test for hydrogen embrittlement tendency of the drilling fluid. When introduced to the environment, the bearing has sufficient residual stresses to cause failure if sufficient hydrogen sulfide concentration is present.
6. A slight drop in pH level of the drilling fluid.

Carbon Dioxide Detection

If the pH level of drilling fluid drops and the hydrogen sulfide test result is negative, there is a good possibility that carbon dioxide will be present. Positive results of microbial activity tests (described later) also indicate the possibility of carbon dioxide presence. Carbon dioxide meters are also available commercially and can be used.

Ultrasonic Inspection

Ultrasonic inspection is a method of measuring the pipe wall thickness. The basic concept of the test is that the sound travels through metals at a constant speed and does not travel well through air. The method consists of cleaning the test surface smooth and coating the surface with a layer of coupling fluid such as oil and glycerine. The coupling fluid facilitates the transmission of sound from the test unit to the metal surface. A curved face transducer is used to convert a high-frequency electrical impulse to sound vibration. The sound travels through the metal at a known speed and is reflected or echoed back to the sending transducer. The time interval between the initial pulse and the return echo is calibrated electronically, and the instrument displays a digital readout of the metal thickness. Limitation of this method is that it only gives the wall thickness at the point tested. Thus, there is a possibility of missing the damaged area. Also, this method does not indicate the type of damage occurring [201].

Magnetic Particle Inspection

When iron filings are sprinkled on a bar magnet, they are attracted to the poles of the magnet (Figure 4-476). If the bar magnet is notched, each side of the notch becomes a pole of a magnet as seen in Figure 4-473. Cracks on drillpipe and collars behave the same way when magnetized. Magnetic particle inspection is based on this concept. The method consists of magnetizing the pipe with a suitable field to cause a magnetic flux. The pipe is then sprayed

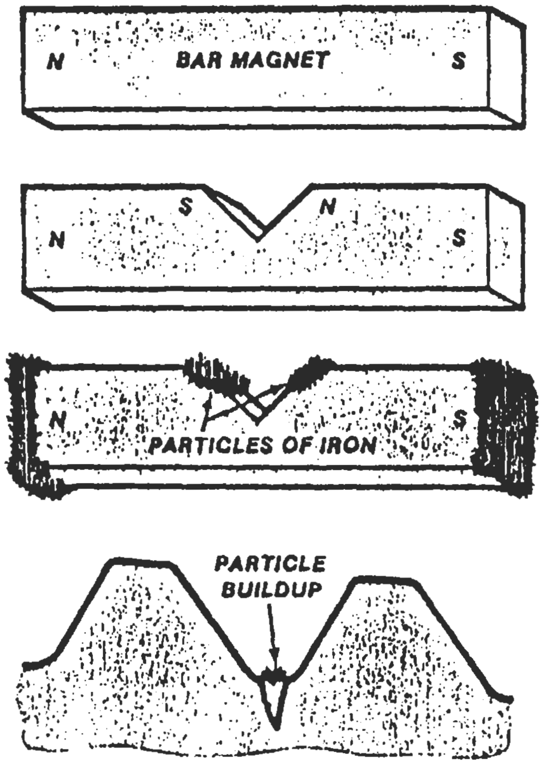

Figure 4-476. Principle of magnetic particle inspection. *(From Ref. [220].)*

with a fluid containing fluorescent magnetic particles (fluorescent tracer). Ultraviolet light is used to detect any buildup of magnetic particles. A limitation of this method is that it is only good for surface or near-surface areas of the pipe.

Visual Inspection

On-site visual inspection of drilling equipment before use is one of the most important factors for successful operation. The drillpipes and drill collars are inspected for signs of pitting corrosion, corrosion fatigue cracks and other damage or defects that may lead to their failure. It is very important to take notes describing the problem and measuring anything relevant, taking pictures (with reference scale) of the damage and making sketches. Any other relevant information helpful in later analysis should be included.

The inspection consists of two tests, the external pipe body and internal pipe surface. The external pipe body is inspected by thoroughly cleaning the surface and examining the whole length with either the naked eye or with the help of a magnifying glass. Internal inspection is carried out by cleaning the internal surface and using a borescope optical instrument. This method is limited by the fact that small cracks may be two small to be detected, and by the speed of the procedure.

Plastic Coating Inspection

Frequent and regular inspections are very important at every step of the process. Inspections should ensure adequate surface preparation and uniform coating application with proper curing process. Once completed, the coatings must be inspected for any pinholes or "holidays" (discontinuities). A low-voltage, wet-sponge, holiday-detector can be used to detect any pinholes in the coating (Figure 4-477). The water in the sponge should contain about 0.5% sodium chloride. The power source imposes a voltage of up to 90 V (usually 67.5 V) across the coating through the wet sponge. The wet sponge is pulled through a joint of internally coated pipe at low enough speed to detect the holidays. When the sponge is pulled across a holiday, an alarm in the detector is activated, and the electrical resistance of the coating drops below 80,000 Ω. Care must be taken to avoid burning holes in the coating by using high voltage. Voltage and current must be regulated to safe levels for the film thickness involved. Manufacturer's instructions should be carefully followed [201].

Monitoring Microbial Activity

As mentioned earlier, microorganisms can attack drilling fluid additives and introduce corrosive agents to the system. Therefore, it is very important to monitor their activity and detect any source of problem as early as possible. API RP 38 is probably the most widely used testing procedure in the industry [201]. The methods that can be used to monitor the microbial activity can include the following [201,208]:

- filtration technique
- bacterial population count
- metal-surface examinations
- evaluation of current microbiocide treatment

Filtration Technique. A measured quantity of sample water is filtered through the membrane filter. The filter is dried, cut and sections placed on microscope slides. The filter sections are rendered transparent by a drop of immersion oil. The slides are examined to identify the microorganisms listed below:

a. algae and protozoa
b. bacteria
c. fungi

Bacterial Population Count. Commercially available test media following API specifications are available for field testing. The two test sera available for inoculation by sample solutions are:

1. clear-yellow broth for anaerobic bacteria (i.e., sulfate-reducing bacteria).
2. phenol red broth for general aerobic bacterial counts.

The test sera are available in sealed bottles. Six bottles are used in each test. They are labeled 1, 10^{-1}, 10^{-2}, 10^{-3}, 10^{-4}, and 10^{-5}, indicating the dilution factor. One unit (one ml or one cm^3) of the sample solution is collected in a disposable,

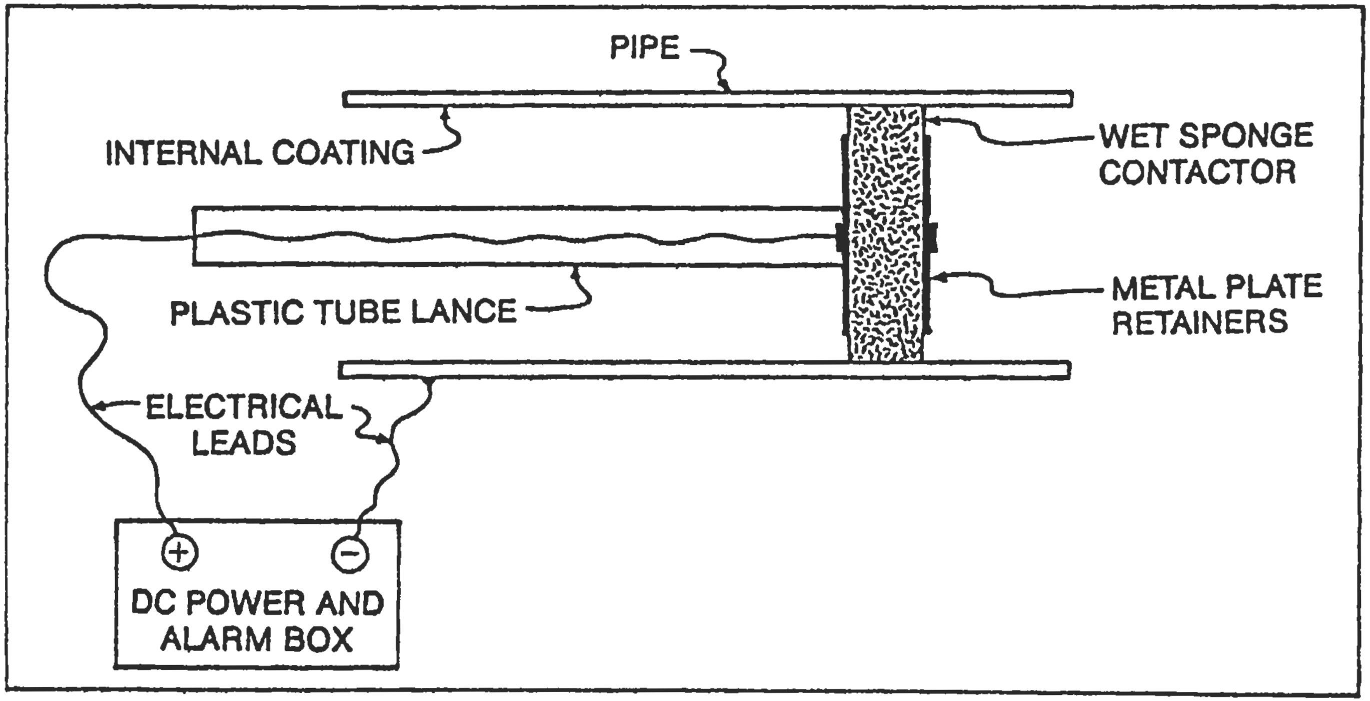

Figure 4-477. Holiday detector for thin film internal pipe coatings. *(From Ref. [201].)*

sterile, plastic syringe. The syringe needle is inserted all the way through the rubber seal of the bottle labeled *1*, and the contents are injected. One unit from bottle *1* is drawn into the syringe and injected into the bottle labeled 10^{-1}. One unit of mixture from bottle *1* contains 0.1 or 10^{-1} units of the original sample solution concentration. The procedure is repeated for the remaining four bottles with one unit of liquid each. Once all six bottles are inoculated, the syringe is discarded. The inoculated bottles are held at a temperature of within 41°F (or 5°C) of the original sample temperature in the system being tested. When phenol red broth changes from red to yellow and looks turbid, it indicates growth of aerobic bacteria. When clear-yellow broth turns dark, it shows the growth of anaerobic bacteria. The main disadvantage of these methods is the time these tests take, five days for phenol red broth and 21 days for clear-yellow broth.

Although not very precise, the bacterial count can be estimated by using the information provided in Table 4-172.

Table 4-172
Bacterial Count

Type of Bacteria—Anaerobic		
Bottle No. Positive After Twenty-one Days	**Estimated Count Cells/Unit**	**Possible Interpretation**
1	1–10	Acceptable count—not a problem.
10^{-1}	$10–10^2$	Low count—repeat tests.
10^{-2}	$10^2–10^3$	Moderate count—possibility of some problem.
10^{-3}	$10^3–10^4$	Slightly high count—great possibility of problems.
10^{-4}	$10^4–10^5$	High count—serious problems. Prepare remedial action.
10^{-5}	$10^5–10^6$	Very high count—extremely serious problems. Initiate remedial action.
Type of Bacteria—Aerobic		
Bottle No. Positive After Five Days	**Estimated Count Cells/Unit**	**Possible Interpretation**
1	1–10	Acceptable count—not a problem.
10^{-1}	$10–10^2$	Low count—generally not a problem.
10^{-2}	$10^2–10^3$	Low count—generally not a problem.
10^{-3}	$10^3–10^4$	Moderate count—repeat tests.
10^{-4}	$10^4–10^5$	High count—probable problem. Prepare remedial action.
10^{-5}	$10^5–10^6$	Very high count—problem imminent. Initiate remedial action.

Source: Adapted from Ref. [201].

Corrosion Control

Corrosion-related problems can be either prevented or reduced by considering the following important factors:

- material selection
- design
- environmental control
- corrosion barriers
- personnel training

To have an effective corrosion-control program, the following suggestions should be considered:

- Review the previous records of similar situations.
- Collect and analyze as much information and corrosion data as possible.
- Select compatible materials for the service environment and conditions.
- Maintain a program to monitor any signs of corrosion problems, and keep good records.

Material Selection

One of the most effective methods of preventing corrosion is the selection of the proper metal or alloy for a particular corrosive service. Once the conditions of service and environment have been determined that the equipment must withstand, there are several materials available commercially that can be selected to perform an effective service in a compatible environment. Some of the major problems arise from popular misconceptions; for example, the use of stainless steel. "Stainless" steel is not stainless and is not the most corrosion-resistant material. Compatibility of material with service environment is therefore essential. For example, in a hydrogen sulfide environment, high-strength alloys (i.e., yield strength above 90,000 psi or Rc 20 to 22) should be avoided. In material selection some factors that are important to consider are material's physical and chemical properties, economics and availability.

Material Properties. Materials possess various mechanical and chemical properties, and, therefore, it is possible to select materials appropriate for severe corrosion conditions. For example, if the equipment is under cyclic loading, a material with high fatigue strength is desired. Similarly, it is desirable to have corrosion-resistant materials for the corrosive environments. There are several sources for obtaining information on materials properties. Some are listed in Table 4-173.

Once materials have been selected, the next step is to compare the required properties with a large data of material properties that look promising for the application. One should then analyze test data (i.e., corrosion test data) to obtain the most suitable material for services.

Economics of Material Selection. Cost is an overpowering consideration in material selection. The basic cost of a material depends upon:

Table 4-173
Some Sources of Material Properties

NACA Corrosion Engineer's Reference Book, Treseder, R. S., 2nd Edition, National Association of Corrosion Engineers, Houston, Texas, 1991.
Corrosion Control in Petroleum Production, TPC5, National Association of Corrosion Engineers, Houston, Texas, 1984.
Engineering Materials Properties and Selection, Budinski, K., 2nd Edition, Reston Publishing Co., Reston, Virginia, 1983.
Betz Handbook of Industrial Water Conditioning, Betz Laboratories Inc., Trevose, Pennsylvania, 1980.

1. scarcity, as determined by concentrations of metal in the ore
2. the cost and amount of energy required to process the material
3. the basic supply and demand for the material; large-volume-usage materials generally have low prices

The more work invested in the processing of a material, the higher the cost (value is added). Increases in properties, such as corrosion resistance and yield strength, beyond the basic material properties, require structural changes. These structural changes occur due to chemical composition change and additional processing steps. For example, the cost goes up as expensive alloying elements are added to the steel or when the steel is heat treated.

As the materials used in drilling processes are produced from depletable mineral resources, there is a continuous upward trend of cost with time. The field engineer must make a detailed cost comparison of materials available within the target cost of the project. The final choice may be a tradeoff between cost and performance. This is because the choice may narrow down to two or more materials with different initial costs and different expected service lives. Transportation costs of selected materials must also be included in the final cost.

Availability of Materials. Availability of the candidate material is a very essential consideration in the decision-making process. There is no sense in specifying the use of a particular material if it cannot be obtained within the time constraints of the project. It is also advisable to select materials that are available from more than one supplier. If proprietary materials that are only available from one supplier are used, one can become a captive customer at the mercy of the supplier on cost and delivery.

It is very important to consult the available literature pertinent to the project. API and NACE among others publish standards on material selection that should be reviewed before the material is selected.

Design

The design of service equipment (i.e., drillstring) is quite often as important as the choice of materials for the equipment. The system can be designed to minimize or totally eliminate the factors contributing to corrosion problems. Therefore, when designing, it is very important to consider corrosion along with mechanical and strength requirements. Materials selected for their corrosion resistance vary widely in their characteristics. Therefore, the design of the system should be based on selected materials.

Most drillstem failures can be attributed to corrosion. Therefore, the following discussion will focus on design considerations to minimize drillstem failures. Tool joints suffer high wear on the outer surface because of the abrasion against the hole wall. This abrasive effect worsens with an increase in hole deviation. As the tool joint rotates under high lateral force against the hole wall, its temperature rises and may reach lower critical levels. The tool joint is alternatively heated by friction and quenched by drilling fluid. This leads to formation of small cracks, leading to hard-facing corrosion fatigue or sulfide stress cracking. Hard-facing the

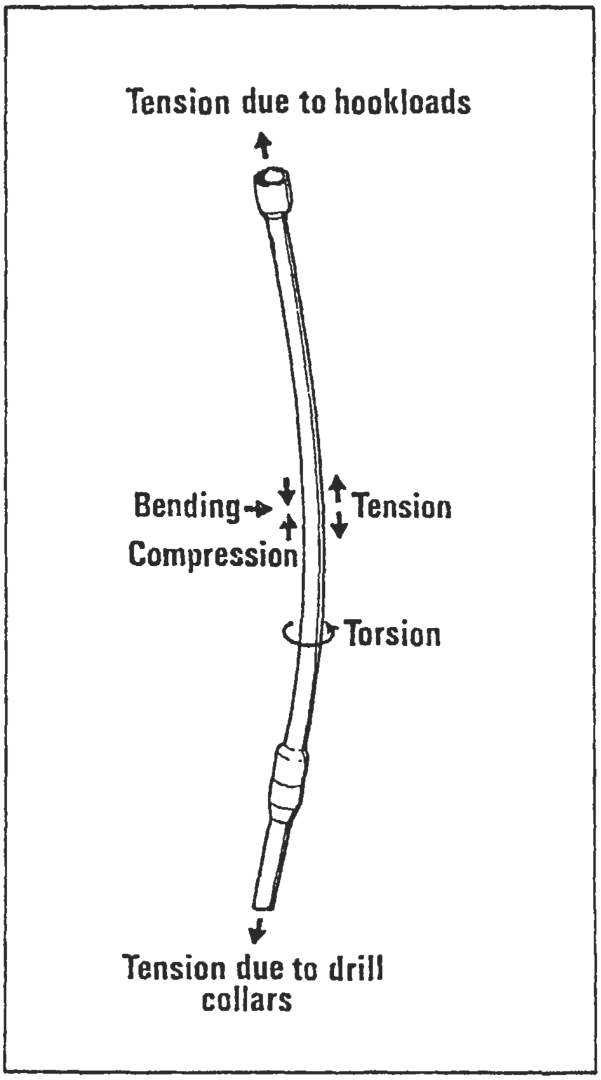

Figure 4-478. Stress loading on drillstem. *(From Ref. [218].)*

tool-joint outer surface and keeping the deviation angle (dogleg angle) low can help in reducing wear on the drillstring (Figure 4-478).

As the drilling progresses and the drillstring is rotated, the drillstring experiences four basic stresses. The four main stresses are tension, compression, bending and torsional stresses (Figure 4-478); a fifth stress, not shown in Figure 4-478, is caused by the vibration of the drillstring. The tension load is caused by the hook load (weight of the pipe) and the weight of the drill collars at the bottom of the assembly. Part of the drillstring is under compression and the rest is under tension. The point at which the string changes from tension to compression is called the neutral point. The location of the neutral point on the drillstring depends on the number of drill collars used and the weight of the drilling fluid (the weight adds buoyancy to the drillstring). Drilling with drillpipes under compression leads to drillpipe failure. Therefore, the drillstring is designed with enough drill collar weight to locate the neutral point in the drill collar section.

Bending stresses are caused by drilling deviated holes. Once bent, drill pipes remain weak. The fourth stress is the torsional stress. The drillstring experiences this stress when the bit is cutting into the formation. Torsional yield strength, therefore, has to be considered when designing the drillstring. Finally, there is a fifth stress caused by vibration of the drillstring. It has been discovered that definite benefits can result from damping severe, downhole vibrations. Therefore, the use of vibration dampers or shock subs above the bit in rough drilling areas is strongly recommended.

Once the wear of the drillstring and a better control of stress concentration are achieved, it is possible to reduce corrosion-related failure of the drillstem.

In the above discussion only the drillstem was discussed. However, when designing other systems in drilling operation, such as drilling fluid systems and casing design to name a few, the engineer must also consider corrosion prevention.

Environmental Control

Environmental control involves reducing the corrosivity of the drilling fluids. Controlling the factors influencing the corrosion rate is a way to reduce the corrosivity of the drilling fluid. These factors are temperature, pressure, velocity and corrodent concentration. Chemical treatment is also used in reducing the corrosivity of the drilling fluid. Corrosion inhibitors, neutralizers, scavengers and scale inhibitors can be used to minimize the corrosivity of the drilling fluids.

Temperature and Pressure. Temperature of the drilling fluid can be reduced by using cooling towers when drilling through high-temperature zones. Geothermal drilling is a good example of this situation. A degasser unit can be added to the circulatory system to separate the contaminating gas or gases from the drilling fluid. These units can be added to the surface part of the circulatory system.

Velocity. The velocity of the drilling fluid is a major problem in air and gas drilling operations (see section titled "Air and Gas Drilling"). High gas velocity (i.e., 8,000 ft/min) is maintained during dry gas drilling. The high annular velocities are necessary for dry-gas drilling fluid to function properly. Therefore, alternative drilling fluids should be considered if corrosion problems are severe. Stable foam can perform effectively at reduced annular velocity such as 2,000 ft/min or lower.

Corrodent Concentration.

Salt Concentration. The most cost-effective method of reducing salt concentration levels is to dilute water-base drilling fluid with freshwater. Care must be taken to make sure that the makeup water is compatible with the system. The water must not contain high concentrations of undesirable corrodents.

Gas Concentration. The principal source of oxygen and some carbon dioxide contamination is the surface part of the circulatory system. Care must be taken to reduce unnecessary aeration of the drilling fluid. Makeup water should be added to the system at the mud flow-line. This will increase the water temperature by the warm, returning, drilling fluid. The increase in water temperature reduces the solubility of gases. Influx of formation fluids can also introduce contaminating gases such as carbon dioxide and hydrogen sulfide. Increasing the mud weight to the level to prevent influx can minimize this type of contamination. As mentioned earlier, a degasser unit can be used to degas the drilling fluid.

Corrosion Inhibitors. An inhibitor is any substance that retards or slows down a chemical reaction. Thus, when added to the environment in small concentrations, inhibitors reduce the rate, or prevent the attack by the environment on the metal. The basic mechanisms by which inhibitors function are as follows [209]:

1. They adsorb onto the corroding material as a thin film.
2. They induce formation of a thick corrosion product, which form a passive layer.
3. They change the characteristics of the environment either by producing protective precipitates or by removing or inactivating an aggressive constituent of the environment.

To be used effectively, the inhibitor must be compatible with the expected environment and also be economical, while contributing the greatest desired effect. Inhibitors can be classed into two main categories, inorganic and organic inhibitors.

Inorganic Inhibitors. Inorganic inhibitors are generally crystalline salts such as sodium chromate, sodium silicate and sodium phosphates, among others. These salts form ions in solution. The positively charged cations (Na^+) and the negatively charged anions (CrO_4^{2-}, SiO_3^{2-}, PO_4^{2-}). The negatively charged anions cause the decrease in the corrosion rate.

There are basically three main types of inorganic inhibitors used in industry: anodic passivating inhibitors, cathodic inhibitors and cathodic precipitators.

Anodic Passivating Inhibitors. There are basically two types of anodic passivating inhibitors. Oxidizing anions, such as chromate, nitrite and nitrate, are capable of passivating the steel in the absence of oxygen. The second type is nonoxidizing ions, such as phosphates, tungstate and molybdate, which require the presence of oxygen to passivate the steel. When used in sufficient quantities, the inhibitors can be very effective in controlling the corrosion problem. However, if used in insufficient concentrations, they may cause pitting and can increase the rate of corrosion. Due to these possibilities, they are sometimes referred to as *dangerous inhibitors*.

Cathodic Inhibitors. Cathodic inhibitors retard the corrosion rate by affecting the reaction at the cathode, as the name suggests. Cathodic inhibitors can be divided into several categories according to the way they achieve inhibition by cathodic polarization. The three main types of these inhibitors are cathodic poisons, cathodic precipitates and oxygen scavengers. Cathodic poisons and cathodic precipitaters are not used in drilling operations. Therefore, the discussion will only focus on oxygen scavengers.

Oxygen Scavengers. When corrosion of steel occurs at pH levels of above 6, it is usually due to the presence of dissolved oxygen. Dissolved oxygen depolarizes the cathodic reaction and, therefore, increases corrosion. Oxygen scavengers achieve corrosion inhibition by preventing cathodic depolarization. Organic corrosion inhibitors are usually added with oxygen scavengers when oxygen contamination is suspected. This is because organic inhibitors alone retard general corrosion, but may not prevent pitting due to presence of oxygen.

There are several oxygen scavengers commercially available, such as hydrazine and sulfites, with the most cost-effective being sulfites. Sulfites react with oxygen in the following manner:

$$2RSO_3 + O_2 \rightarrow 2RSO_4$$

The most commonly used sulfite is sodium sulfite Na_2SO_3:

$$2Na_2SO_3 + O_2 \rightarrow 2Na_2SO_4$$

It is usually a good practice to add a catalyst to the system. Catalysts increase the rate of oxygen scavenging. The most commonly used catalysts are the transition elements in their II oxidation state. Some good examples are Co^{2+}, Mn^{2+}, Fe^{2+} and Ni^{2+}. Some other commercially available oxygen scavengers are ammonium bisulfite, various organic amine sulfites and a combination of amine-ammonium bisulfite with an organometallic catalyst component.

Organic Inhibitors. These organic compounds affect the entire surface of a corroding metal when present in sufficient concentrations. Figure 4-430 shows the effect of organic inhibitor concentrations on the rate of corrosion. The corrosion inhibition increases as the inhibitor concentration increases, a result of adsorption of organic inhibitors onto the metal surface. A thin film only a few molecules thick is adsorbed according to the ionic charge of the inhibitor molecule and the charge onto the metal surface. Cationic inhibitors (positively charged), such as amines, or anionic inhibitors (negatively charged), such as sulfonates, are adsorbed selectively depending on the charge, positive or negative. The molecules of these inhibitors are unbalanced with one portion, the tail end, being oil soluble or hydrophobic (water repelling). The other end is the polar head or hydrophilic (water loving). As only a part of these inhibitor molecules is polar, they are said to be "semipolar" (Figure 4-479). These molecules adsorb over the metal surface as these polar heads attach to the metal surface. The oil-soluble, hydrocarbon tail attracts some oil or hydrocarbon constituents flowing past the metal [210].

The oil enmeshes in the tail, as shown in Figure 4-480, and provides a mechanical barrier to attack of the aqueous corrodents on the base metal. The oily film also increases the resistance to corrosion current flow and, thus, stifles the rate of corrosion. An advantage of using organic film-forming inhibitors

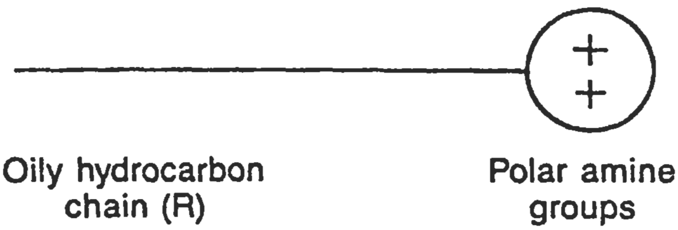

Figure 4-479. A semipolar molecule. *(From Ref. [201].)*

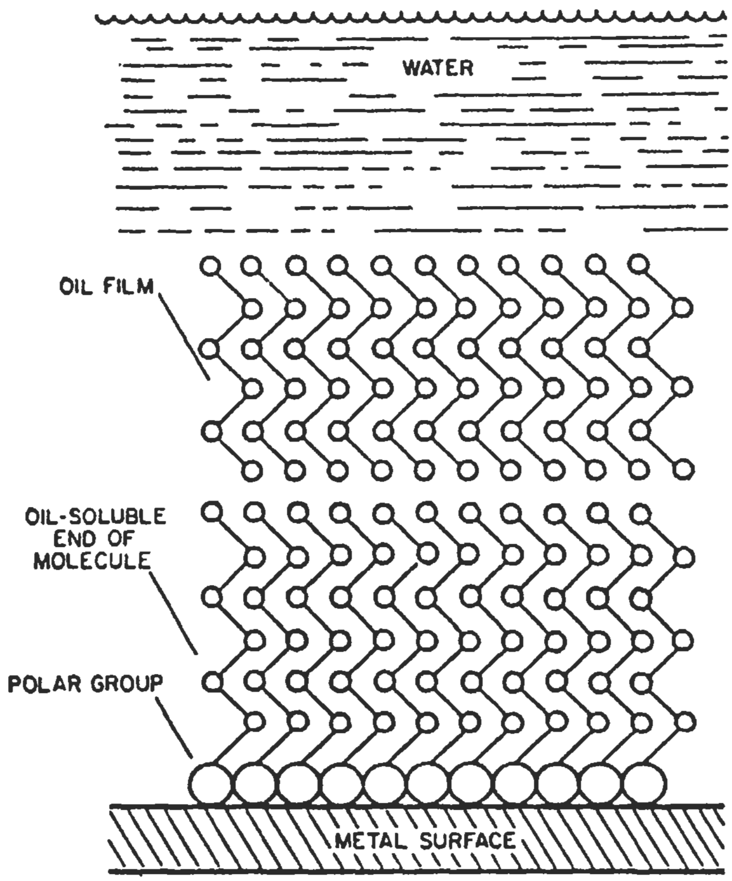

Figure 4-480. Schematic of film-forming inhibitor. *(From Ref. [210].)*

compared to applied coatings is that the inhibitor films are self-healing, as they are continuously added. However, coatings and linings are prone to mechanical damage, which requires shutdowns for maintenance and repair. A drawback of using this type of inhibitor is that if insufficient concentration is used, it may result in accelerated corrosion.

The factors contributing to effectiveness of organic inhibitors are:

1. the molecular size
2. aromaticity and/or conjugated bonding
3. bonding strength to the metal substrata
4. type and number of bonding atoms or group in the molecule
5. ability of the layer to become compact, or crosslink

Some examples of organic anionic inhibitors are sodium phosphates, thioureas and sodium MBT phosphionates (mercaptobenzothiazole). Some examples of organic cationic inhibitors are amines, amides, quaternary ammonium salts, and imidazoline.

Selection of Corrosion Inhibitors. When selecting inhibitors for use, it is always important to consider the following useful suggestions:

- Consult an outside consultant specializing in the use of inhibitors.
- Review available literature on the use of the inhibitor in question for a particular environment.
- Conduct in-house inhibitor tests to determine which inhibitor and at what concentration to use. The tests must be conducted following the recommended practices of NACE or ASTM. ASTM (American Society for Testing and Materials) and NACE (National Association of Corrosion Engineers) publish standards for such tests.
- The inhibitors must be readily available, and cost effective.
- Consideration should be given to the inhibitor's effects on drilling fluid properties, other than corrosivity.

Inhibitor Efficiency. The value of corrosion inhibitors can be compared on the basis of inhibitor efficiency. Inhibitor efficiency indicates the percentage that corrosion is lowered in the presence of the inhibitor as compared to that in its absence. Inhibitor efficiency can be calculated by using the formula

$$E = \left(\frac{R_0 - R_i}{R_0}\right)100 \tag{4-368}$$

where E = inhibitor efficiency
R_0 = rate of corrosion in the absence of an inhibitor
R_i = rate of corrosion in the presence of an inhibitor

Application of Corrosion Inhibitors. There are basically two main techniques used to apply corrosion inhibitors in drilling operations. In the first method inhibitors are added to the drilling fluid system either by mixing the additives through the rig's chemical hopper or through additions into the mud pit. The treatment can be achieved in two ways, batch treatment or continuous treatment. In some cases it may be necessary to use both types of treatment simultaneously. The second technique of applying is directly coating the corrosion inhibitors on the drillpipe.

Batch Treatment. This treatment is accomplished by pumping manufacturer's recommended volume (with concentration up to 10,000 ppm or according to the manufacturer's recommendations for prevailing conditions) to "batch" down the drillpipe initially. Once the film is formed, the inhibitor concentration can be lowered for batch treatment at regular predetermined intervals.

Continuous Treatment. Continuous treatment involves introducing a corrosion inhibitor on a regular basis to maintain the specified concentrations of inhibitors in the system. Depending on the prevailing conditions and manufacturer's recommendation, the concentration may vary from a few parts per million to 50 ppm or more.

Direct Treatment. Although batch and continuous treatment are both quite effective, there is a problem with inhibitor waste. When corrosion inhibitors contact the circulating drilling fluid, they are likely to coat the solids in the fluid system (cuttings or other solids). By applying the corrosion inhibitor directly to drillstem components before they are run in the hole, the corrosion inhibitor is the first thing that contacts the exposed metal surface. There are two methods for this type of corrosion inhibitor treatment.

Spray or Painting Application. The corrosion inhibitor is either sprayed (Figure 4-481) or painted (by paint brush) on the drillstem components, before they are run in the hole. The application process also takes place while tripping into the hole. Spray application systems can be mounted on brackets just above and around a bell nipple. A 1-in. circular pipe with one closed end and the other connected to an air-powered spray-application system is used. Three spray nozzles pointing inward and slightly downward into the wellhead are used to deliver the spray. The spray system is activated by a foot pedal situated near the driller's console as the drillstem is lowered into the borehole. Batch treatment for the inside of the drillstem and general conditioning should also be carried out at regular intervals (i.e., every 5 to 10 stands) [211,212].

Dipping Application. Treatment by dipping consists of dipping a joint of drillpipe into an "extra mouse hole" (Figure 4-482). The extra mouse hole may be located half way between the V-door and the rig's "working" mouse hole on the rig floor. The mouse hole can be constructed from a large-diameter casing (i.e., $9\frac{5}{8}$ in. diameter), cut to about 28 ft in length with the bottom sealed closed. The hole is filled with corrosion inhibitor and drained out into storage containers through a bottom valve (Figure 4-482) when not in use. The exposed part of the drillstem component is treated by inhibitor coating using a regular paint brush.

Whatever method of inhibitor application is used, care must be taken to maintain optimum concentration of the corrosion inhibitor. In the event of addition of makeup water or untreated reserve fluids to maintain other fluid properties, care must be taken to ensure the addition of the correct amounts of corrosion inhibitors.

Inhibitor Concentration. Corrosion inhibitors are commercially available in solid or liquid form. Liquid inhibitors are usually preferred as they are easier to transport, measure and use. Corrosion inhibitors are dissolved in appropriate solvents and mixed with property enhancers to achieve the desired properties. The prepared mixture of liquid inhibitors is sold by the gallon. The amount of the inhibitor present in the mixture is expressed as percent active. For example, a gallon of inhibitor which is 25% active contains 25% by weight of inhibitor.

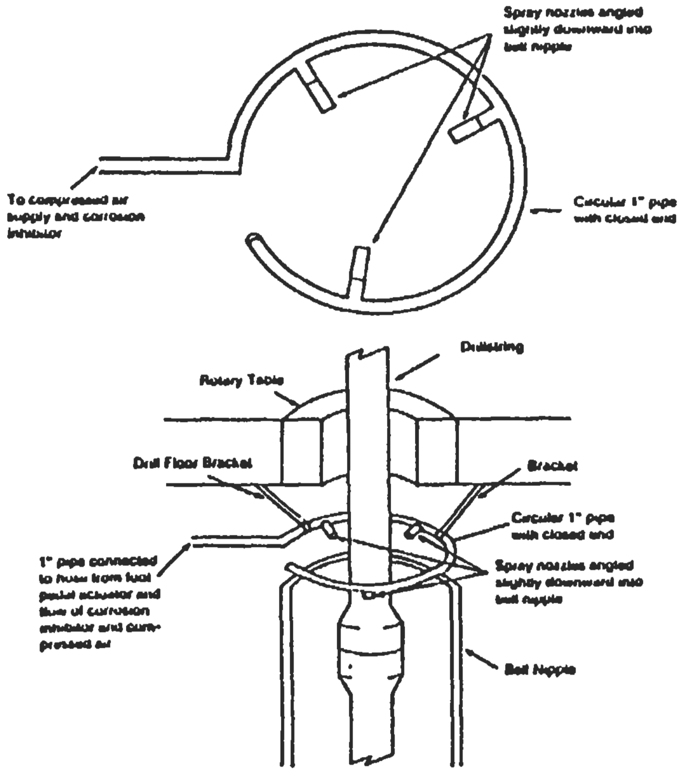

Figure 4-481. Inhibitor spraying equipment. *(From Ref. [212].)*

Inhibitor concentrations are generally expressed as parts per million (ppm). Solid inhibitors are expressed on a weight basis, such as pounds or kilograms of inhibitor per million pounds or kilograms of fluid. Liquid inhibitors are expressed in volumes used, such as liters of inhibitor per million liters of fluid. To find the quantity of inhibitor required for a given system, the following formula can be used:

$$Q = (V \times 10^{-6}) \times \text{ppm} \tag{4-369}$$

where Q = quantity of inhibitor required
V = amount of fluid to be inhibited
ppm = concentration of inhibitor in parts per million

The quantity of inhibitor Q is always in the same units as those used for the amount of fluid to be inhibited V.

Neutralizers. To maintain pH levels in alkaline regions, it is necessary to neutralize the acidic components of the corrosive medium. Water-soluble, alkaline materials, such as sodium hydroxide (NaOH), ammonium hydroxide (NH_4OH) and calcium hydroxide [$Ca(OH)_2$], can be used to obtain pH levels around 9.6. However, care must be taken to avoid reaching pH levels that are

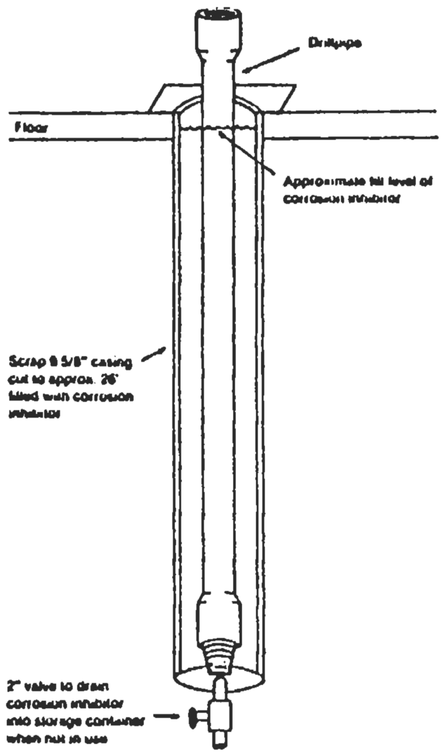

Figure 4-482. Drillpipe dipping treatment. *(From Ref. [212].)*

high enough to cause scale formation of calcium and magnesium salts. The neutralizing materials can be added to the drilling fluid circulation system directly or to the mud pits.

Scale Inhibitors. When scaling conditions exist, scale inhibitors can be used to control the scaling tendencies, and keep metal surfaces free of scale deposits. Scale inhibitors are chemicals that interrupt and deform the normal crystalline growth pattern of carbonate scales. The three most commonly used classes of scale-inhibiting chemicals used in drilling fluid are [191,197]:

1. amino-phosphonates
2. phosphate esters of amino-alcohols
3. sodium polyacrylate polymers

These chemicals are generally marketed as water solutions (20 to 30% active). Alcohols are usually added to lower the freezing point and keep the inhibitor chemical in solution.

Hydrogen Sulfide Scavengers. Hydrogen sulfide (H_2S) can be neutralized by using sodium hydroxide ($NaOH$) and calcium hydroxide [$Ca(OH)_2$] [213,214]. The reaction is

$$H_2S + OH^- \rightarrow H_2O + HS^-$$

$$HS^- + OH^- \rightarrow H_2O + S^{2-}$$

The neutralization is essentially complete at pH levels higher than 9.5; however, if the pH levels drop below 9.5, H_2S is released. As H_2S neutralization may reverse by change in pH, neutralization does not provide the degree of security required in H_2S control. Thus, special scavenging compounds are used to remove the H_2S from the drilling fluids. H_2S scavenging is a form of irreversible treatment at operating conditions. There are two types of scavengers available to the drilling industry, zinc-base scavengers and iron-base scavengers.

Soluble sulfides (i.e., H_2S, HS^- and S^{2-}, with sulfur at minus two oxidation state) are chemically very reactive. The two general types of soluble-sulfide reactions may be identified as precipitation reaction (type A) and redox reaction (type B).

In type A reaction soluble sulfide ions combine with metal ions to form a precipitate of insoluble metal sulfide. Sulfur's oxidation state of minus two does not change in this reaction. The reaction is

$$Zn^{2+} + S^{2-} \rightarrow ZnS \downarrow$$

This reaction occurs over a wide range of drilling fluid conditions.

Type B (redox) reactions are more complex. Sulfide in this reaction is converted into some other oxidation state of sulfur. For example, sulfides can be converted to a zero oxidation state of elemental sulfur by oxygen:

$$O_2 + 2H_2S \leftrightarrows 2S^0 + 2H_2O$$

In this way sulfides may disappear from circulating mud when muds contact air during circulation.

Zinc-Base Scavengers. There are two types of zinc-base scavengers available. These are:

1. Slightly water-soluble inorganic compounds, zinc carbonate being the most common. Zinc carbonate is only slightly soluble in the middle range of the pH scale (i.e., 8 through 11). Commercial compounds of zinc carbonate are not the same as the mineral smithsonite, $ZnCO_3$. The manufactured commercial compound has an approximate formula $3Zn(OH)_2 2ZnCO_3$. This commercial-grade compound contains 55 to 58 wt% zinc and about 20 wt% carbonate. The type A reaction that occurs between the zinc ion and sulfate ion is

   $$Zn^{2+} + S^{2-} \rightarrow ZnS \downarrow$$

   ZnS is stable and will not revert to H_2S unless the pH level drops below 3. It is highly unlikely for drilling muds to drop to pH levels below 3; generally the drilling mud pH is maintained around pH of 9.5. The slightly soluble, zinc-base scavenger can dissolve completely at high pH levels above pH of 11. This is because at high pH levels zincate ions are formed as high concentration of hydroxal ions (OH^-) combine with the zinc ions as shown below.

   $$Zn^{2+} + 3OH^- \rightarrow Zn(OH)^-_3$$

   $$Zn(OH)^-_3 + OH^- \rightarrow Zn(OH)_4^{2-}$$

This results in high concentrations of zincate ions that are effective scavengers. However, high concentrations of zincate can lead to problems related to mud performance.

2. The second type of zinc-base scavenger, utilizes highly water-soluble, organic, zinc-chelate compounds. The metal ions are bonded with organic compounds, forming "metal chelates" that are highly soluble in water. This high solubility enables their use in clear drilling fluids; whereas, a less-soluble scavenger can settle out and become ineffective. Another advantage of using zinc chelates is that they are effective over a wide range of pH.

Iron-Base Scavengers. There is only one iron compound that is commercially available as a sulfide scavenger for drilling fluids. The product is a synthetic, high-surface-area magnetite, Fe_3O_4. Fe_3O_4 is often used in low-pH muds, where sulfides in the mud exist largely as acidic molecular H_2S. This is because reactions of Fe_3O_4 in normal alkaline-mud pH ranges proceed at a slow rate. Consequently, a soluble sulfide may coexist with unreacted Fe_3O_4 for undesirably long periods of time. The scavenging reaction of H_2S by Fe_3O_4 proceeds according to the following equation by type A reaction:

$$Fe_3O_4 + 6H_2S \rightarrow 3FeS_2 \downarrow + 4H_2O + 2H_2 \uparrow$$

The reaction efficiency and products formed depend on downhole variables. The most important variables are pH, temperature, reaction time, post-reaction aging time, and mud-shear conditions. An advantage of using Fe_3O_4 over other chemicals is that large quantities of insoluble material can be added without affecting the drilling fluid properties. Under optimum conditions with adequate Fe_3O_4 surface area exposed, the product may remove up to 2,000 mg/L sulfides for 1 lbm/bbl (2.85 kg/m^3) Fe_3O_4 treatment. Fe_3O_4 is often used as a pretreatment to reduce the threat to drilling tools and health of the drilling crew, which may result from a kick of gas containing high H_2S content. Although the chemical composition is the same as magnetite, Fe_3O_4, it is not very magnetic and, therefore, does not cling to drillpipe or casing.

Microbiocides. There are several microbiocides available commercially that can perform an effective function in controlling microbial activity. Some of these chemicals are inorganic, such as chlorine, sodium hypochlorite, calcium hypochlorite, hydrogen peroxide, chromates and compounds of mercury and silver. However, the organic chemicals find the highest use as microbiocides. Some examples of these organic compounds are peracetic acid, paraformaldehyde, polychlorophenols and quaternary ammonium derivatives, to name a few [208].

Microbiocides may be toxic to humans; therefore, care must be taken when used. When selecting the microbiocide, the field engineer can obtain pertinent information on chemicals from the service company providing the chemicals. The microbiocide selected must be compatible with the system in which it is being used. Some chemicals such as quaternary amines have dual functions; one as microbiocides and the other as film-forming corrosion inhibitors. Insufficient concentrations of this type of chemical may not be enough to coat the whole surface of metal and can cause pitting corrosion. The selection must also depend on chemicals that can produce the desired control in minimum time limits and minimum cost. Commercially available microbiocides generally contain one or more chemical compounds. The amount of chemical present that has the microbiocidal property is expressed as percent active ingredient.

Microbiocidal Treatment. Once the microbial activity reaches a high enough level to cause problems, physical and chemical methods may be used to control the problem. In drilling operations, the most effective method is chemical treatment–the use of microbiocides [184,208].

To select a microbiocide one should consider its compatibility with the system. Some chemicals may affect the performance of the drilling fluids and lead to other problems. For example, quats, amines, and chlorinated phenols may cause clay flocculation of the drilling fluid if present in sufficient quantities. Therefore, the effect of the microbiocide on the drilling fluid properties must be thoroughly tested and understood before the chemical is used.

Once the microbiocide is selected, a method of application should be considered. The chemical can be introduced to the system by either batch treatment, continuous treatment or by a combination of both. For batch treatment, NACE provides an equation given below. This equation can be used to determine the concentration of chemical at any time during the eight hour period. The equation is

$$C = 10.7 - 10.7e^{-2.3t} \tag{4-370}$$

where C = concentration of the microbiocide in lb/300 bbl
t = time in hr
$C^*(\frac{9}{52})$ = parts per million (ppm)

The frequency of the batch treatment, or slug treatment, depends on the actual field response. The continuous method of treatment is relatively more expensive and can be four times as expensive as the batch treatment.

This is one of the most important factors in alleviating microbial problems. A decrease in microbial activity after treatment indicates a positive response to the microbiocidal treatment. Continuous monitoring is imperative for effective control.

Oil Muds. Oil-base muds can be used effectively to minimize the corrosion-related problems in drilling operations. These fluids are composed of a continuous oil phase in which water is emulsified. The performance of oil-base mud is very competitive with that of water-base muds, and is superior under some adverse conditions. However, as pointed out earlier, they are only used in special cases due to their high costs and environmental restrictions. Nevertheless, they are the most effective method to avoid corrosion-related problems.

For the corrosion process to proceed, the corrosion cell must contain an anode, a cathode, an electrolyte and an electronic conductor. When a properly prepared and conditioned mud is used, it causes preferential oil wetting on the metal. As the metal is completely enveloped and wet by an oil environment that is electrically nonconductive, corrosion does not occur. This is because the electric circuit of the corrosion cell is interrupted by the absence of an electrolyte. Excess calcium hydroxide [$Ca(OH)_2$] is added as it reacts with hydrogen sulfide and carbon dioxide if they are present. The protective layer of oil film on the metal is not readily removed by the oil-wet solids as the fluid circulates through the hole.

Oil muds are relatively resistant to contaminants generally encountered during drilling.

$$Ca(OH)_2 + H_2S \rightarrow 2H_2O + CaS$$

$$Ca(OH)_2 + CO_2 \rightarrow H_2O + CaCO_3$$

However, these two reactions result in an increase in the total solid content of the mud, affecting the mud performance.

When using oil-base mud, care must be taken to prevent water becoming free. Water is always present in oil-base mud whether added intentionally through the surface equipment or from the drilled formations. API RP 7G recommends the following factors to be monitored when oil-base muds are in use [181]:

- *Electrical stability:* The test measures the voltage required to cause current flow between two electrodes immersed in an oil-base mud. The higher the voltage, the greater the stability and, consequently, the greater the drillpipe protection (see API RP 13B for more details [206]).
- *Alkalinity:* Acidic gases may contaminate the oil muds and lower the pH to an undesirable range.
- *Corrosion test coupons:* As mentioned before, these coupons provide very important information on the corrosiveness of the drilling fluids (see section on corrosion coupons in this text and API RP 13B for more details [206]).

Corrosion Barrier

Plastic-Coated Drillpipes. Corrosion can be prevented by separating the corrosive environment from the metal. Plastic coating provides this separation when it is undamaged. The drill-pipe can only be coated internally. The outside surface of the pipe experiences high wear with the hole wall. While internal coatings reduce internal corrosion of the drillpipe, the corrosive attack is transferred to the outside surface of the drillpipe. Such occurrences suggest that coating alone cannot control corrosion. Therefore, they must be used in conjunction with other control measures that reduce the overall corrosivity of the drilling fluid. One of the serious problems is that it is very difficult to obtain holiday-free coating. Holidays (discontinuities) are areas of damaged coating where the metal is exposed to corrosive environments. This can either be the result of poor bonding or of mechanical damage. Holidays can completely nullify the beneficial effects of a coating. Therefore, it is essential that the internally coated drillpipes be free of holidays [211].

Plastic coating of drillpipe has extended its service life greatly. The drawback of coatings is that they can be damaged. Coatings can be damaged locally when tools such as wirelines, hole deviation tools, etc., are run down the drill pipe. Overtorque and improper handling may also contribute to coating damage. Quite often, the damage is in or near the tool joint area where corrosion fatigue failures are more common.

There are several plastic coatings available commercially, some being more effective than others. However, the selected plastic coating must possess flexibility, resistance against impact, chemical attack, flow of galvanic currents and permeation by moisture. The coating must have good adhesion, cohesion and high-temperature serviceability. Plastic coatings are composed of three basic components: pigment, binder and solvent.

Pigments. Pigments introduce strength and color to the coating. They are designed to be corrosion and abrasion resistant. Pigments reduce water vapor transmission and prolong coating life. They are usually composed of inert inorganic compounds that interlock among themselves to provide strength to the coating. Often zinc may be added to the pigment for greater corrosion resistance. Other compounds that may be used in this way are red-lead, zinc-chromate and basic lead silicochromate.

Binders. Binders are organic resins that are composed of high-molecular-weight polymers that provide specific properties to the coatings. Epoxies and phenolics are good examples of these binders.

Solvents. Solvents are only present in the manufacturing stage of the coating process. They are not present in the finished product. As the coating is baked, the solvent evaporates, polymerization occurs and the resin adheres to the base surface.

Personnel Training

Most often corrosion failures can be traced back to a lack of understanding and neglect on the part of personnel. Therefore, the success of a corrosion control program is dependent on the knowledge and efficiency of the personnel involved. After all, they are the people who may carry out the corrosion control program. Therefore, it is essential that they understand the importance of their responsibilities and carry them out efficiently. Personnel involved in drilling operations must have a basic knowledge of principles of corrosion control. Thus, training programs for drilling personnel must be developed. The program must present current technology in a manner conducive to learning and offering a well-rounded education. Training benefits can include more productive employees, lesser down time and, perhaps, a better cooperation between personnel. In other words, more efficient operations can result from an effective training program.

Corrosion Control in Arctic and Geothermal Drilling

Arctic Drilling. Corrosion problems encountered in arctic area drilling are no different from problems faced in other areas of the world. It is a general misconception that during arctic drilling corrosion-related problems are either not very severe or totally absent due to low temperatures. Cool temperatures may slow down the corrosion process. However, they also increase the solubility of oxygen, carbon dioxide and hydrogen sulfide. Therefore, the net result can be an increase in the rate of corrosion. While cold temperatures may cause problems, the temperature fluctuation common in arctic environments can be a more severe source of corrosion-related problems [215].

The severity of weather in arctic regions, especially in winter months, presents difficulties. They can be in monitoring corrosion, inspection of equipment, mobility of personnel, transportation delays of materials, accurate operation of electronic and other instruments. The storage and use of chemicals used to combat corrosion is also a problem in these conditions. Chemicals packaged in warm and humid areas of the world may be difficult to add to the circulatory system until the condensed and frozen moisture in the bay is thawed. As mentioned earlier, transportation to and from the drilling site is very slow and depends on the weather conditions. "Three-day blows" and "whiteouts" are very common and result in transportation "blackout." Transportation "blackouts" are total cessation of transport activity. The distances between major distribution points and drilling sites are generally very large. Therefore, the deliveries can be further slowed down.

Summer months in these areas may not necessarily be any better. In view of these problems, it is very important to consider all aspects of operation before embarking on the project. Therefore, before starting a drilling project, corrosion control must also be considered. Steps should be taken to ensure running an effective corrosion control program.

Geothermal Drilling. Geothermal drilling presents one of the most challenging environments for corrosion control. During geothermal drilling the bottomhole temperatures can reach around 480°F (250°C). Therefore, drilling fluids and chemical additives, such as corrosion inhibitors, must withstand high temperatures (480°F) and pressures around 4,000 psi. High temperatures, coupled with corrosive fluids encountered in the formation, result in increased corrosion rates. Water and oil-based drilling muds can damage the producing zones. To minimize this formation damage, aerated muds and gaseous drilling fluids are used. However, these drilling fluids increase the potential for accelerated corrosion of drilling equipment. This is because of high concentrations of oxygen in fluids and increased fluid velocities over the metal surface. Another problem that can arise during drilling a geothermal well is lost circulation. Lost circulation can occur while drilling through highly fractured formations. This leads to a displacement influx of corrosive agents into the wellbore. It also results in loss of expensive drilling fluid and chemical additives, such as corrosion inhibitors. When fluid losses occur, aerated or gaseous drilling techniques can be used. With these techniques, hydrostatic head can be adjusted (i.e., lightened) to balance the formation pressure and, consequently, reduce fluid losses to the formation. Another method sometimes used is blind drilling or dry drilling. In blind drilling, the drilling fluid is slowly pumped down the drillpipe without it returning to the surface. In this procedure drill cuttings are expected to be carried with drilling fluids into the host circulation zone. However, it is important to pick up the drillstem at regular intervals of short distances. This will reduce the chance of drill cuttings stacking up or accumulating in the annular chamber space above the bit, leading to a stuck drillstem. Quite often these cuttings will plug the lost circulation zone, and the drilling fluid circulates back to the surface. This method of drilling is extremely expensive as the drilling rates are reduced and drilling fluids are lost. The technique does not provide any means of protecting the external surface of the drillpipe from corrosive environment. Therefore, if possible aerated mud or gaseous drilling should be considered over the technique of blind drilling. Nitrogen can be used to aerate the drilling fluid to reduce corrosion problems. It can also substitute for air in gaseous drilling to minimize corrosion problems if economically feasible [216].

Recommended Practices

Equipment failure due to corrosion is one problem that inevitably rears its ugly head during drilling operations. Corrosion control is an essential factor in any engineering design and must be considered as early as the initial phase of the operation. An effective and most cost-efficient corrosion control program is imperative for a successful drilling operation. Some recommended practices are as follows [202,203,217–222]:

General Operation

1. Utilize drillpipe corresponding to C-75 tubing or casing specification set in API Specification 5AC [178]; the specification calls for certain chemical requirements. Heat treatment by normalizing and tempering at a minimum 1,150°F (621°C) after cold working. Yield strength from 75,000 psi to a maximum of 90,000 psi. Some grade E can be used: however, all grade D drillpipe can be used.
2. Avoid the use of shrink-type tool joints on drillpipes and drill collars (connectors, in this case). Flash-welded tool joints for drillpipes and integral drill-collar joints should be used.
3. Reduce the stress levels in the pin by utilizing makeup torque on tool joints at the low end of the recommended torque range.
4. Use internally, plastic-coated drillpipe at all times. The coating must be holiday-free to be effective.
5. Consider the use of heavy-wall (thick wall) drillpipe in severely corrosive conditions. Heavy walls reduce stress levels and extend the service life of drill pipe.
6. Materials used for tool joints are generally 4137 or 4140 steel. They are heat treated by quenching and tempering at temperatures equal to or above 1,150°F to hardness of Rc 30 to Rc 37 and yield strength ranging from 120,000 psi to 135,000 psi. The above specifications do not meet the requirements corresponding to C-75 specifications. One solution is to use smaller, nonstandard bore and larger outside diameter to achieve heavy-wall effects.
7. Most drill collars are also made with materials used for tool joints. Here again, heavy walls are used to reduce stress levels.
8. Limit exposure time of the drillstem to the corrosive environment during drillstem tests. A maximum of one hour is recommended.
9. Sustained exposure of drillpipe to operating temperatures above 300°F (149°C) must be limited to a maximum of 10 hr.
10. Run enough drill collars to keep all drillpipe in tension in order to reduce wear and stress on tool joints.
11. Minimize opportunities of corrodent contamination of drilling fluids.

Transportation

1. Drillpipes and drill collars should be tied down with suitable chains at the bolsters when being transported.
2. Load with all couplings on the same end of the truck.
3. Use thread protectors on tool joints when moving or racking pipes.
4. Avoid chafing of tool-joint shoulders on adjacent joints.

5. Keep load binding chains tight at all times.
6. Avoid damaging coatings as well as the pipe itself.

Storage

1. To clean, wash down pipe in derrick with freshwater.
2. Clean the pin and box threads and shoulders thoroughly, and inspect for any damage.
3. Inspect each joint for any damage or crack, and inspect internal coating for holidays.
4. Apply appropriate film-forming inhibitors to reduce atmospheric corrosion during storage and transit.
5. Do not store pipe directly on ground, rails, steel or concrete floors. A minimum of 18 in. clearance should be present between the ground and the first tier of pipe.
6. To help in inspection and handling, do not stack pipes higher than five tiers at the rig.

Handling

1. Before unloading, make sure that the thread protectors are tightly in place.
2. Avoid rough handling that can damage the pipe in any way.
3. The pipe must not be dropped at any time.
4. Unload one joint at a time.
5. When rolling down skids, roll pipe parallel to the stack. Do not let the pipe gather any momentum and strike the ends. This can damage threads even when the protectors are in place.
6. Avoid creating knicks or notches on the drillpipe.

Chemical Additives

1. When high temperatures above 300°F are expected, do not use sulfur-containing compounds as drilling fluid additives. In general, avoid using chemical additives that can at operating temperatures.
2. Use only chemical additives that are compatible with drilling fluids in circulatory systems.
3. Avoid using copper-based compounds such as copper carbonate. Copper can "plate out" on steel and set up galvanic corrosion cells, resulting in accelerated corrosion of the steel.
4. Maintain the pH level of the drilling fluid around 9.5 to reduce corrosion. However, when aluminum drillpipes are in use, the pH must be maintained between 9.5 and 7.
5. Avoid using concentrations high enough to lead to the loss of other desirable characteristics of the drilling fluid.
6. Unless unavoidable, do not let salt contents exceed 180,000 ppm.
7. Caustics should be dissolved in water before being added. The mixture should be mixed well with mud guns in the pit prior to pumping into the hole.

Chemical Treatment Dosage

Chemical treatment of drilling fluids has to be carried out with great care. The chemicals used must be appropriate and in proper quantities. If the chemicals are not compatible with the environment or if their concentrations are not sufficient, they can actually accelerate corrosion. The concentrations, the type and frequency of the treatment will vary with different environments and desired level of corrosion control. However, it should be noted that excesses of certain chemicals may alter the desirable characteristics of the drilling fluids. Most commercially available inhibitors are mixtures of various chemicals. The concentration of active ingredients varies. Before use, the manufacturers and suppliers of these chemicals must be consulted. Recommended procedures, method of treatment and dosage should be followed. Some general guidelines are presented in this section for treatment. These recommendations are by no means absolute. The treatment must be designed for prevailing conditions.

Oxygen Scavengers. Oxygen scavengers such as sodium sulfite should be added to the mud system, premixed in water solution with recommended catalyst. The solution should be injected into the pump-suction line. It can also be added to the circulatory system through a submerged drip line. The drip line is usually a rubber hose with one end connected to the chemical barrel and the other end 1 or 2 ft below the mud surface in the suction pit near the pump suction. The scavenger solution must not be added to the system through the mud hopper. In fact, the solution must not be exposed to air longer than necessary.

Initial treatment should range from about 300 mg/L of scavenger, and injection rates should be as high as the pump will allow. Treatment has to be adjusted to achieve from 100 to 300 mg/L sulfite residual at the flowline.

Organic Inhibitors. During drilling operations organic inhibitors can be used in two ways. They can be sprayed or brushed on directly to the external surface of the drillstem. Secondly, are "batch" treatment down the drillstem at regular intervals. For direct treatment, a mixture is prepared by mixing film-forming amine and diesel oil in a 1:1 ratio. "Batch" treatment is carried out by pumping 3 to 4 gal of corrosion inhibitor down the drillstem. Frequency of "batch" treatment should be every 5 to 10 stands while tripping into the hole.

Hydrogen Sulfide Scavengers. Hydrogen sulfide scavengers are very effective in controlling hydrogen sulfide, H_2S, contamination. The most common H_2S scavengers used in drilling operations are zinc carbonate ($ZnCO_3$) and iron oxide (Fe_3O_4). Zinc carbonate and iron oxides can be used either individually or in combination depending on environmental conditions. The concentration will, of course, vary according to the conditions. At least 5 lb of iron oxide per barrel should be added to the system. Large quantities of iron oxide in the system do not affect the drilling fluid properties. However, zinc carbonate in excess of 5 lb/bbl can affect the drilling fluid properties, especially when temperatures reach above 250°F (120°C). Therefore, when zinc-based scavengers are used, mud properties must be closely monitored for adverse effects. About 1 lb/bbl (2.85 kg/m^3) of zinc-based scavengers (10 to 20 wt% active) should be sufficient for 100 to 200 mg/L of sulfide contamination.

Scale Inhibitor. About 3 to 5 gal (20 to 30% active) of scale inhibitor per day should be sufficient for scale inhibition. The treatment can be reduced to 1 to 3 gal per day once the scale formation is under control. The chemical can be introduced to the system by injecting it into the pump suction line.

Neutralizers. Sodium hydroxide, calcium hydroxide and ammonium hydroxide are used to neutralize the acidic components of drilling fluids. Neutralizers are added to the system to maintain the desired pH levels. Normally, pH of around 9.5 is maintained during drilling operations.

Microbiocides. For a general guideline about 100 mg/L of the microbiocide with at least 50 wt% active ingredient should be added to the drilling fluid. Exact dosage and treatment frequency has to be estimated based on the severity of the problem.

ENVIRONMENTAL CONSIDERATIONS

Introduction

The key role of the engineer is the design of facilities and their operations. In designing a new operation or upgrading an old one, the engineer must be aware of environmental concerns such as air emissions, water discharges and hazardous waste generation. The engineer needs an in-depth understanding of the impact of the operation on the environment, and how considerations made prior to implementation of the project may minimize these impacts. The engineer must also be aware of the regulations involved so that designs are legally feasible and permitting is accomplished in a timely manner.

In drilling, planning should include environmental considerations. Environmental management at the wellsite involves thoughtful planning at the onset of exploration or development. In today's world of heightened environmental concern, a project may be postponed or terminated due to these issues. The prespud meeting in addition to discussion of well depth, casing points, and rig selection, should include topics pertinent to the environmental management of the drilling and completion operation. Regulating agencies are most concerned with these issues. A site will have particulars imminently apparent and those perhaps not so apparent except to groups exhibiting a certain interest. It is the role of the regulating agency to protect the public and public domain from detrimental effects caused by industrial operations. Compliance of the regulator's requirements is usually simple in nature. The requirements are documented and follow a particular order. Concerns disseminated by special interest groups are often unexpected. In an attempt to forego stalling of a project by these parties, public disclosure of a project should be made as early as possible, even if it is pending. A well-informed public disclosure statement released to the immediate community may in many instances preclude any future surprises.

In planning a drilling operation, the location and access are primarily keyed to environmental decisions. The access and location must be able to maintain the traffic load, and also mitigate any impact on local resources such as flora, fauna, cultural and aesthetic. In certain instances, the preparation of an environmental assessment followed by a environmental impact statement may be considered warranted due to proximate:

- wildlife refuges
- historic or cultural resources
- recreation areas
- land containing threatened or endangered species

In the United States drilling plans are submitted to the Bureau of Land Management or state oil and gas commission in the form of an APD (application for permit to drill) depending on ownership of mineral resources. Other countries have similar requirements. In addition to these agencies, other appropriate surface management agencies may have to be considered. These include, but are not limited to, the BLM, National Park Service, Tribal Authorities, State Environmental Program and County Environmental Program. In accessing the responsible authorities involved, the permittee should contact the primary responsible permitting agency and follow their guidelines for contact of interested parties.

In the permitting process, it is important to have the entire operation planned. In fact it is a necessary component of most APDs. In a federal APD, the 13-point surface use plan must address, prior to approval:

1. location of site and existing roads
2. planned access roads
3. location of existing wells
4. location of existing or proposed production facilities
5. location and type of water supply
6. source of construction materials
7. methods of handling waste disposal
8. ancillary facilities
9. wellsite layout
10. plans for restoration of the surface
11. surface ownership
12. other information such as proximity to water, inhabited dwellings, archeological, historic or cultural sites
13. certification of liability

Details of the actual drilling program are not considered in detail in an APD except for the casing and cementing program and how they are designed to protect any underground sources of drinking water (USDW). The bulk of the APD permit is designed to address the impact on surface resources and the mitigating procedures the operator plans to take to lessen these impacts [223].

Site Assessment and Construction

Access and Pad

The road to the wellpad should be constructed to prevent erosion and be of dimensions suitable for traffic. A 16-ft top-running width with a 35-ft bottom width has shown to sustain typical oilfield traffic. The road should be crowned to facilitate drainage and culverts placed at intersections of major runoff (see Figure 4-483). Before the pad and access is surveyed for final layout, preliminary routes should be studied and walked out. Trained personnel should note any significant attributes of the area such as archeological finds, plants and animals. As a result of this survey, the most favorable access is designated weighing economics with the environmental regulations.

The pad size and arrangement are functions of the drilling operation itself and primarily linked to:

- drilling fluid program
- periodic operations
- completions

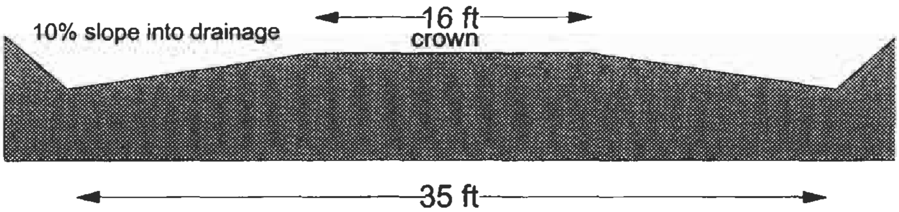

Figure 4-483. Typical oilfield access dimensions.

The rig selection will dictate the basic layout of the pad. Based on the necessary area needed to support its functions, ancillary equipment may be added in space conservative measures. In addition to the placement of various stationary rig site components, other operations such as logging, trucking and subsequent completion operations must be provided for. The most environmentally sensitive design will impact the least amount of area, and in that it will be the most economic. Potential pad sites and access routes should be laid out on a topographic map prior to the actual survey. At this time, construction costs can be estimated and compared. Figure 4-484 shows such a layout. The cost of building a location includes the cost of reclamation such as any remediation,

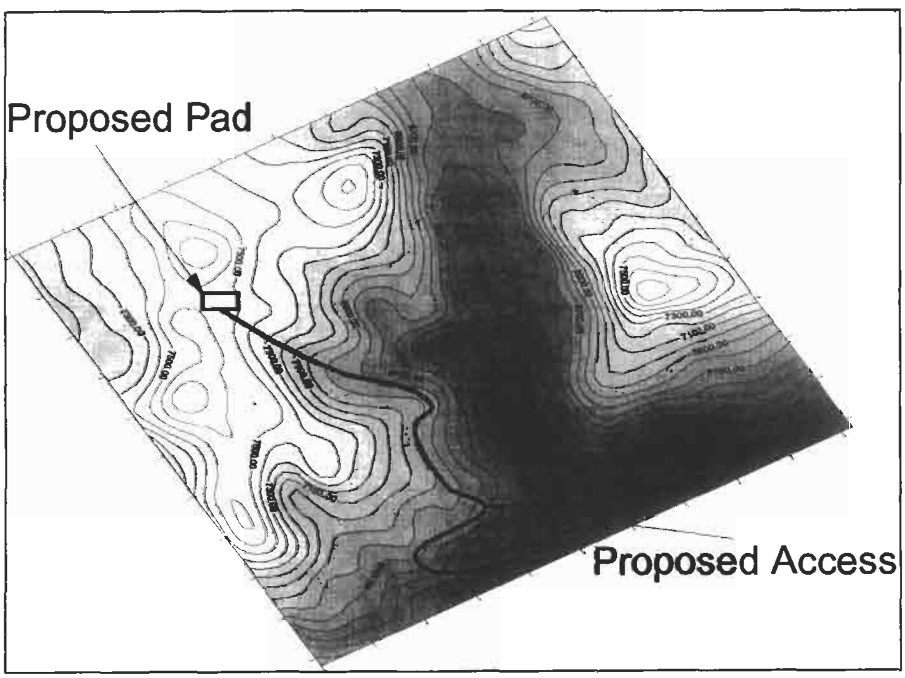

Figure 4-484. Topographic layout of proposed access and pad.

recontouring and reseeding of native plants. In the event of a producing well, only that area needed to support the production operations is left in place. The reserve pit and all outlying areas are reclaimed.

Rig Considerations

A rig layout diagram, provided from the contractor, will give the dimensions necessary to begin planning the drillsite. Figure 4-485 depicts a layout for a 10,000-ft mobile drilling rig. The main components to the rig are drive-in unit, substructure, mud pits, pumps, lightplant, catwalk, piperacks, lightplant and fuel supply. This type of rig, with a telescoping derrick, may be laid on less than 0.37 acre. The outstanding area is taken by the positioning of the deadmen. The deadmen, to which the guywires are connected, are specified through safety considerations for each rig model. Smaller mobile drilling units with less than 2,000-ft-depth capacity may be guyless and as such need less of an area on which to operate. A standard drilling rig with a 10,000-ft capacity will have a longer laydown side, as it is measured from the well center. Here, during rig up, the derrick is assembled on the ground and then lifted onto the A legs. The laydown side of the site must allow for this operation. As depicted in Figure 4-486, the standard rig exceeds the overall areal dimensions of the mobile rig at 0.41 acre with a laydown dimension of 130 ft as compared to 90 ft for the telescoping mast unit. A standard drilling rig with a 20,000-ft capacity may run a laydown of 185 ft, with all other dimensions proportionately increased.

With the basic rig layout defined, the ancillary equipment may be determined and laid out adjacent to the structures in place. Usually the access is determined where casing, mud and other equipment may be delivered without disturbance of the infrastructure. In the case where a crane (or forklift) is used, this may

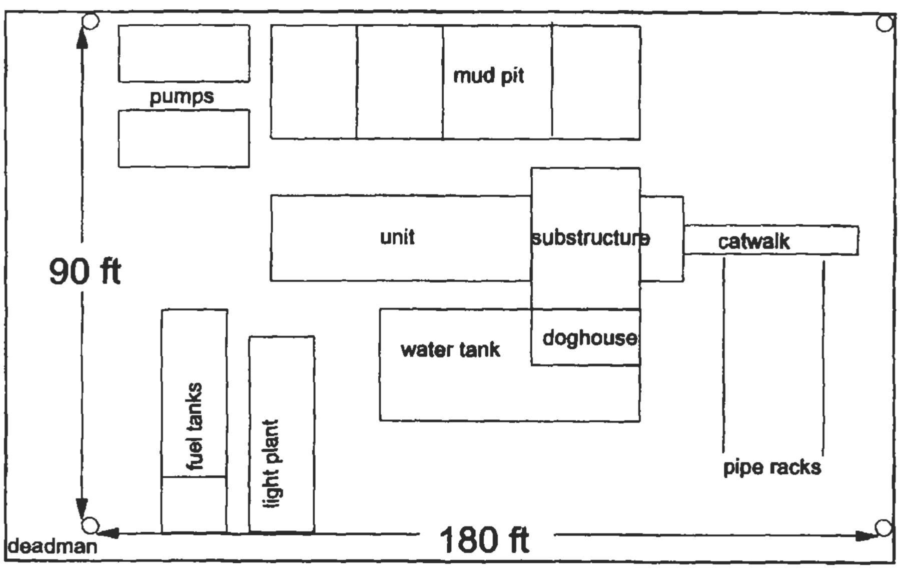

Figure 4-485. Basic mobile 10,000-ft drilling rig layout.

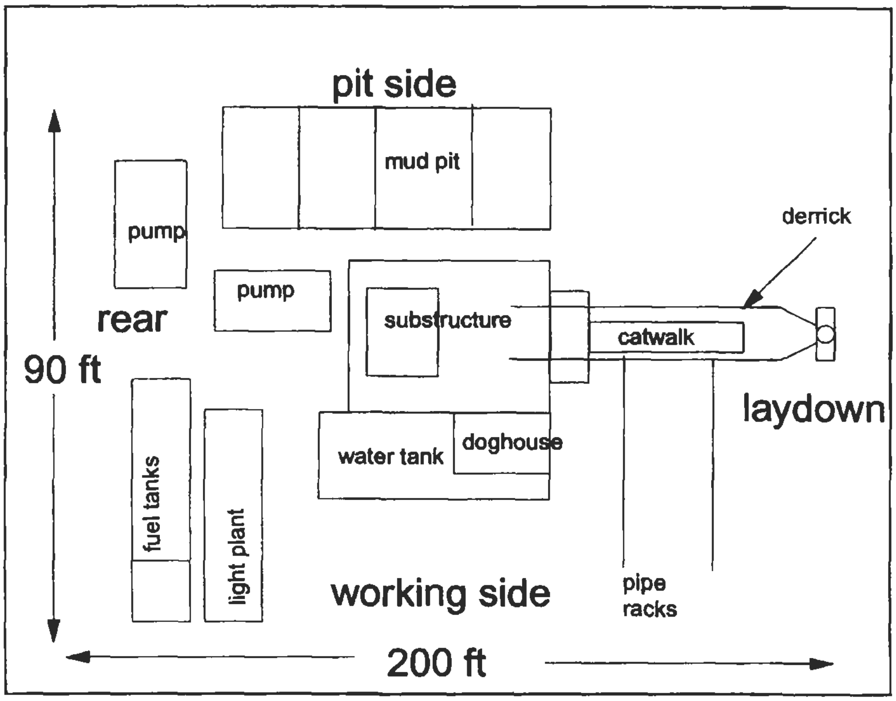

Figure 4-486. Basic layout for drilling rig with standard derrick.

mean a simple loop from the rear to the laydown on the working side of the location. A loop is the optimum arrangement whereby multiple-truck interference may be avoided. In the event a crane is not available, extra space is needed to accommodate the activity of gin trucks positioning the materials. Figure 4-487 shows an overall location view.

Drilling Fluid Considerations

The drilling fluid program will define to some extent a major portion of the pad including reserve pit, blow pit and equipment space. The program may include a closed system or a conventional one. The fluids can be oil-base mud, air, foam, water or other media. While the basic rig layout considers most mud drilling activities, it does not figure in mud storage, additional water storage, or air drilling systems. The overall layout may even be reduced in some cases. A closed mud system or air drilling eliminates the requirement for a large reserve pit.

Air Drilling. In the event of air drilling, the blooie line will exit from under the substructure and away from the rig. Depending on the nature of the payzone, the blooie line will extend different lengths from the rig. For example, a well, producing 20 MMcfd in addition to the injection of 2,000 cfm air, will require an extension of at least 150 ft, due to heat and dust accumulation. The blow pit with berm will often extend another 40 ft beyond that to include both the

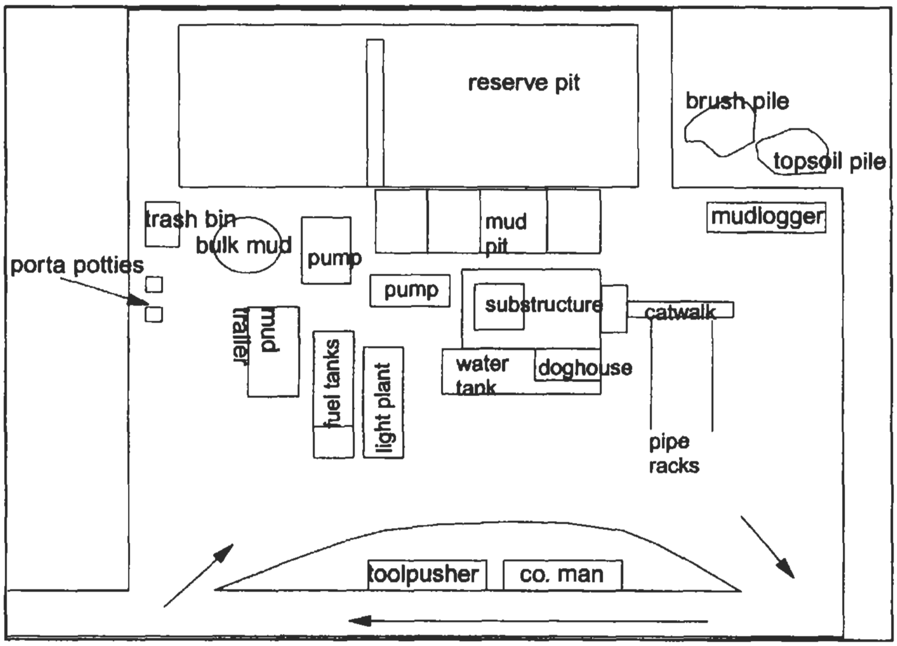

Figure 4-487. Overall location plan.

blow pit and berm. The blow pit is then connected to a reserve pit. This allows the transfer of any injected or produced fluid to a storage area away from the blow pit. The blow pit and berm should be designed with considerable contingency. During operations, the well may be producing oil and/or natural gas. In the case of oil, the well will have to be mudded up, once the well is under control. In the process of getting the well under control the oil may be sprayed a great distance if the blow pit and berm are not properly constructed. If natural gas is encountered, drilling is usually advanced with a flare in place. The pit and berm then must be able to protect the surrounding area from fire plus sustain the impact from a steady bombardment of drilling particles. 30 ft should be allowed from the end of the blooie line to the edge of the berm, with only 5 ft of depth needed at the pit sloping towards the reserve pit built to maintain a flowing velocity of 2 fps. The berm itself should be 20 ft high and composed primarily of 1 ft and larger diameter stones on the face and backed with a soil having poor permeability (see Figure 4-488). A lip should overhang from the top of the berm whereby the ejected materials are diverted back to the pit. The air compressors and boosters are typically located at the rear of the location, with the air piped from that point to the standpipe. This equipment should reside as close as possible to the standpipe to alleviate additional piping. Figure 4-489 shows an air drilling layout. Note that the location may be compacted somewhat due to the reduced reserve pit.

Mud Drilling. The conventional drillsite includes mud pits, mud cleaning equipment, water storage, mud storage, pumps and mixing facilities. Often the reserve pit associated with the conventional mud system is as large as the leveled

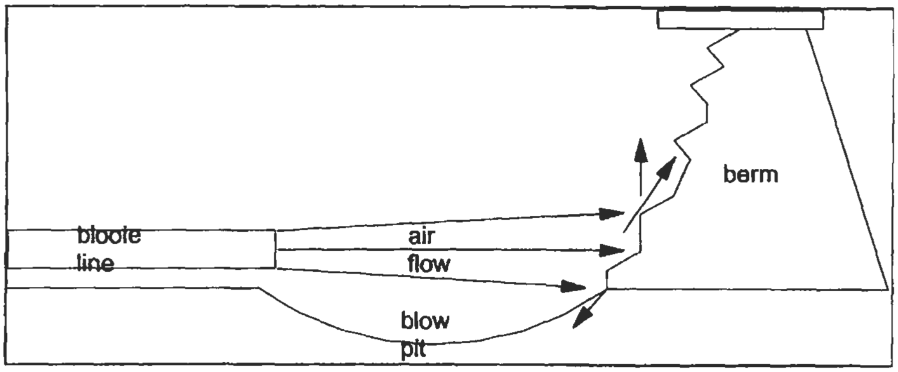

Figure 4-488. Side view of blow pit.

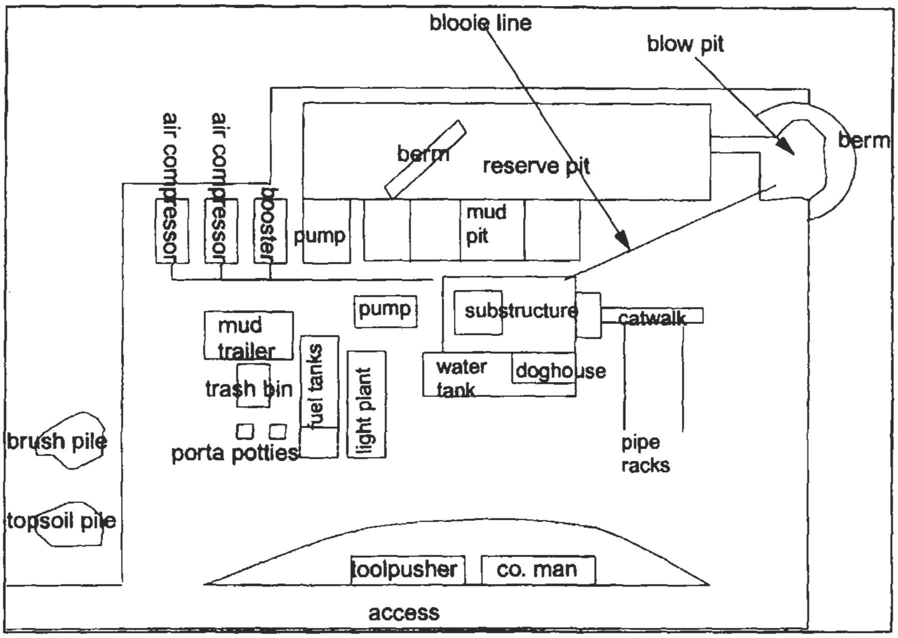

Figure 4-489. Air drilling pad.

location. A large reserve pit allows for contingency, but also increases cost of reclamation and construction. Generally a 3-ft freeboard is maintained for safety and may also be relied on for contingency. A good field estimate for a reserve pit size needed for a conventional operation is

$$V_r = 2(V_s + V_i + V_p) + 281D_d + 16.84MV + 3WL \tag{4-371}$$

$$WL = \frac{V_r}{H} \tag{4-372}$$

where V_r = volume of reserve pit in ft^3
V_s = volume of surface hole (ft^3)
V_i = volume of intermediate hole (ft^3)
V_p = volume of production hole (ft^3)
D_d = forcasted drilling time in days
MV = mud pit volume in bbl
L = pit length: 0.5–0.75 length of location in ft
W = pit width: 0.25–0.50 length of pit in ft
H = pit depth: $\leq$ 10 ft

Equation 4-371 assumes that for each operation a volume of cuttings is put into the reserve pit plus a mud volume equivalent to the circulation system. The additional volumes are attributed to mud dilution and maintenance. An additional contingency is added through 3 ft of freeboard. The reserve pit should be located in an area where capacity may be increased in an emergency or additional fluid trucked away.

Closed Mud System. The closed mud system presents today's solution to an environmentally sensitive drilling operation. The circulation system on the rig is fully self-supporting, requiring only the discharge of drill cuttings. On a simple system this may only necessitate the addition of a mud cleaner or centrifuge to the drilling rig's conventional mud system. A bid package to the drilling contractor may stipulate a closed system and the requirements, thus letting the burden of design fall to the contractor, although the liability still rests on the operator. With a closed system, additional area is sometimes needed within the basic layout for mud cleaning equipment and mud storage. A trench located at the pit side of the mud pits may be allowed then for cutting disposal. In the event of oil-base mud, the cuttings may be collected in a sloped container, where residual fluid is allowed to drain from the cuttings before disposal.

Periodic Operations

Periodic operations include cementing, running casing and logging. Room must be allocated for the storage of casing. Running 10,000 ft of range 3, $5\frac{1}{2}$-in casing requires a 40 × 115 ft area. The casing may be stacked, but never in excess of three layers and preferably only two. Cementing operations may include placement of bulk tanks in addition to pump trucks and bulk trailers. These are usually located near the water source and rig floor. Laying down drillpipe and logging both necessitate approximately 30 ft of space in front of the catwalk.

Completions

If a well proves productive, the ensuing completion operation may require an area in excess of the drilling area. This may mean allocations for frac tank placement, blenders, pump trucks, bulk trucks and nitrogen trucks. In today's economic climate, the operator should weigh the probability of success, Bayes theorem (Equation 4-373), with the cost of constructing and reclaiming an additional area needed for stimulation (Equation 4-374). Plans such as these should be considered and proposed in the APD. The operator may then construct the additional space without a permitting delay. Thus,

$$P_s = PR(E/R) = \frac{PR(E)PR(R/E)}{PR(E)PR(R/E) + PR(\text{not } E)PR(R/\text{not } E)} \tag{4-373}$$

$$P_s C_b \leq C_a \tag{4-374}$$

where P_s = probability of success
$PR(E/R)$ = probability of event E given information R
C_b = construction cost of additional space before drilling in \$
C_a = construction cost of additional space after drilling in \$

Pad Construction

Once access and size of pad have been determined, they may be physically flagged to further reduce the amount of dirt moved. Important features of the area should be noted at this time, including:

- depth of water table
- natural drainage patterns
- vegetation types and abundance
- surface water
- proximate structures

These features are quantified, as well as testing for contamination in both soil and water to prevent unnecessary litigation over previous pollution.

In construction of the pad, brush and trees are pushed to one area. Top soil is then removed from the site and stockpiled for respreading during subsequent seeding operations. The leveled pad is slightly crowned to move fluids collected on the pad to the perimeter where drainage ditches divert this fluid to the reserve pit. In the event a subsurface pit is not possible, the drainage will run to a small sump. This sump is used as a holding tank for pumping of collected fluids to an elevated reserve pit.

When possible, the reserve pit is placed on the low side of the location to reduce dirt removal. In this event, the pit wall should be keyed into the earth and the summation of forces and moments on the retaining wall calculated to prevent failure. Pits most often fail due to leaking liners undercutting the retaining wall and sliding out. A minimum horizontal-to-vertical slope of 2 is recommended for earthen dams [224]. The pit bottom should be soft filled to prevent liner tears. Other key factors of the location provided for the drainage of all precipitation away from the location. This prevents the operator from unduly managing it as a waste product. Any water landing on the location must be diverted to the reserve pit.

A new scheme for location management has developed whereby wastes are diverted to separate holding facilities according to the hazard imposed by the waste. Separate pits are created to hold rig washing and precipitation wastes, solid wastes and drilling fluids [225]. The waste is then reused, disposed on site, or hauled away for offsite treatment. The system reduces contamination of less hazardous materials with the more hazardous materials, thereby reducing disposal costs.

Environmental Concerns While in Operation

Drilling

A blowout primarily consisting of oil presents the greatest environmental hazard while drilling. During normal drilling situations, downhole drilling fluids are usually the greatest potential threat to the environment. In the case of oil-based mud, the cuttings also present a problem through adsorption of the diesel base. These cuttings are presently landfarmed, landspread, reinjected or thermally treated to drive off the hydrocarbon.

Most wells encounter shales when drilling. The oil-based muds are quite effective in reducing the swelling tendencies of the shale in addition to presenting less intrusive invasion characteristics to the reservoir. The oil-based muds may be sold back to the distributor, where they are recycled. An alternative to oil-based muds may be found in using a synthetic-based material [226]. Saline mud is also used to reduce the shale's swelling characteristics. Here, chlorides may be found to be within toxic levels in the drilling fluid, making its use less desirable. Since the swelling of the clay is attributed to the cation exchange capacity (CEC), the chloride levels may be reduced by replacement with another anion. Calculation of a shale's CEC estimates the amount of cations that can be added to the drilling mud to effectively reduce swelling tendencies. Generally, a multivalent cation is more effective in reducing hydration of the clay. An equilibrium equation is used to define the CEC of a shale (Equation 4-375). Figure 4-490 shows the CEC relationship between several multivalent cations and sodium on attapulgite clay [227]. While drilling, the drilled cutting's CEC is usually tested with methylene blue (Equation 4-376). Table 4-174 shows CEC of several clays encountered in drilling situations [228]. In reducing the amount of any one type cation or anion present in the drilling mud, the environmental risks are reduced. A mixed salt solution containing several salt combinations, each of which is below the toxic limits, may produce the desirous effects. Thus,

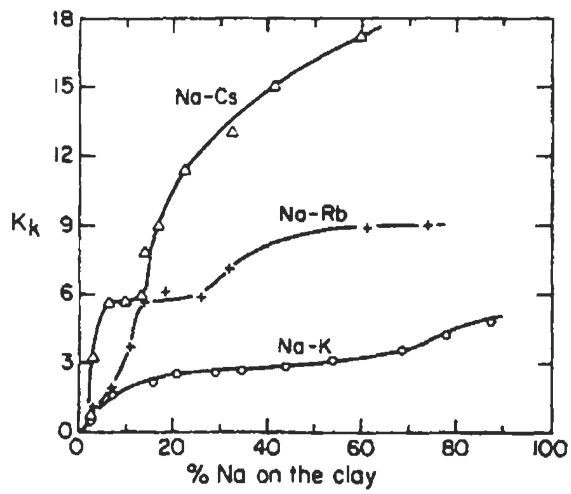

Figure 4-490. Selectivity number K_k vs. exchangeable ion composition on an attapulgite clay.

Table 4-174
Cation Exchange Capacity of Various Materials Encountered in Drilling [225].

Material	meq/100 g—moisture free
Wyoming Bentonite	75
Soft Shale	45
Kaolinite	10
Drilled Cuttings	8

$$K_k = \frac{(NaX)^c(M)}{(MX)(Na)^c} \tag{4-375}$$

where NaX = sodium on the shale in meq
M = multivalent cation in meq
c = charge of multivalent cation

$$MBC = \frac{MB}{M_{shale}} = CEC_{shale} \tag{4-376}$$

where MBC = methylene blue capacity in meq/100 g
MB = methylene blue in meq
M_{shale} = weight of shale in 100 g

There are a variety of chemicals that are toxic and used in the drilling fluid makeup. Chromates and asbestos were once commonly used and are now off the market. A mud inventory should be kept for all drilling additives. Included in the inventory are the material safety data sheets (MSDS) that describe each material's pertinent characteristics. The chemicals found on the MSDS sheet should be compared with the priority pollutants and any material should be eliminated if a match is found. The chemicals should also be checked on arrival for breakage and returned to the vendor if defective packaging is found. All mud additives should be housed in a dry area and properly cared for to prevent waste. Chemicals should always be mixed in packaged proportions. Wasted chemicals, ejected to the reserve pit by untrained personnel, can present future liabilities to the operator.

Rig Practice. All drilling fluids should be contained in the mud pits or reserve pit. The cellar should have a conduit linking it to the reserve pit such that accumulation of mud from connections does not spread over the location. In the case of toxic mud, a bell should be located beneath the rotary table to direct such fluids back into the drilling nipple or into the mud pit in the case of a rotating head. The use of a mud bucket is also recommended in situations where the pipe can not be *shook* dry.

Many operators are now requiring that absorbent or catch pans be placed beneath the drilling rig. Oil and grease leaks/spillage are common to the drilling rig operation. Even if the drilling rig is new, leaks and spills are inevitable. Fuel and oil racks should provide a spill-resistant pour device for direct placement of the lubricant/fuel into equipment. Oil changed on location must be caught in barrels and recycled by the contractor. Pipe dope should be environ-mentally sound and not a metal type. Thread cleaning on casing is now frequently done with machines that catch the solvent for reuse on subsequent operations.

Drilling contractors should be advised prior to the operations on what is trying to be accomplished environmentally on location. Contractors know that the standard practices of throwing dope buckets and everything else into the reserve pit is no longer acceptable, but an occasional drilling crew may not take the directives serious. Because of this, drilling contracts should line item the liabilities associated with imprudent practices.

Completions

Inherent to the completion operation are the stimulation fluids used to carry sand or otherwise enhance producing qualities of the well. These stimulation fluids may or may not be toxic in nature. The stimulation fluids, while sometimes batch mixed at the service company's facility, may also be mixed on location. The latter system is preferable unless the service company is willing to let from the operator any liability common to the fluid in the case of excess.

Currently many frac jobs are minifraced before the actual operation and the design criteria established before the actual frac job is accomplished. The method helps prevent the screening out of the well beforehand. A screened-out well causes potential problems not only in productivity but also in waste management. The unused mixed chemicals, such as KCl makeup water in frac tanks leftover due to the screenout, must be properly disposed. An alternative to this is the mixing of chemicals on the fly instead of preblending and stocking in frac tanks. Some chemicals, such as KCl in water, must be mixed well in advance on location to attain heightened concentrations. In these instances, a properly designed frac job, based on the minifrac, will allow for some certainty in getting the designed job away. Frac flow back may be introduced to the reserve pit after separation from any hydrocarbons. Current frac fluids are composed primarily of natural organics such as guar gum but may contain other components that may be harmful to the environment. Containment of flowback in a lined reserve pit prior to disposal is a prudent practice.

Acid jobs may also be designed according to prior investigation although most service companies will accept unused acid back into their facilities. Spent acid flow back may be introduced to the lined reserve pit with little consequence. Often the residual contains salts such as $CaCl_2$ previously introduced to the pit. Live acid is occasionally flowed backed to surface. This acid may be flowed to the lined pit given acceptance of the liner material for low pH. The buffer capacity of most drilling fluids is significant, thus it is able to assimilate the excess hydrogen ions introduced to the system. The buffer capacity of the fluid may be calculated from Equations 4-377 and 4-378 [229]. From a water analysis, pH and alkalinity are determined and the remaining parameters may then be calculated.

$$\beta = \frac{dC}{dpH} \tag{4-377}$$

$$\beta = 2.3\left[\frac{\alpha_1\left([\text{ALK}]-[\text{OH}^-]+[\text{H}^+]\right)\left([\text{H}^+]+\frac{(K_1'K_2')}{[\text{H}^+]}+4K_2'\right)}{K_1'\left(1+\frac{2K'2}{[\text{H}^+]}\right)}+[\text{H}^+]+[\text{OH}^-]\right] \tag{4-378}$$

$$[ALK] = [HCO_3^-] + 2[CO_3^{-2}] + [OH^-] - [H^+] \tag{4-379}$$

$$[Ct] = [H_2CO_3^*] + [HCO_3^-] + 2[CO_3^{-2}] \tag{4-380}$$

$$K_1' = \frac{\gamma[HCO_3^-](H^+)}{[H_2CO_3^*]} \tag{4-381}$$

$$K_2' = \frac{\gamma^4[CO_3^{-2}](H^+)}{\gamma[HCO_3^-]} \tag{4-382}$$

$$\alpha_1 = \frac{1}{1 + \frac{K_2'}{(H^+)} + \frac{(H^+)}{K_1'}} \tag{4-383}$$

$$\text{Log } \gamma = -AZ^2\left[\frac{\left[\sqrt{I}\right]}{\left[1+\sqrt{I}\right]}\right] \tag{4-384}$$

where β = buffer capacity, (equivalents/unit pH change)
C = equivalents of buffer available—assumed equal to [ALK]
[ALK] = equivalents of alkalinity
K = equilibrium coefficients
γ = the monovalent activity coefficient
I = the ionic strength
Z = the charge of the species of interest
A = $1.82 \times 10^6\ (DT)^{-1.5}$
D = the dielectric constant for water, 78.3 at 25°C
T = °K
( ) = activity of the ion
[ ] = concentration of the ion

The buffer capacity of the pit fluid is equal to the change in alkalinity of the system per unit change of pH. Figure 4-491 shows the buffer intensity (capacity) of a 0.1 M carbonate pit fluid. Calculating the initial buffer capacity of the pit fluid allows for prediction of the pH change upon introduction of live acid and also any addition of buffer, such as sodium bicarbonate, required to neutralize the excess hydrogen ions.

Care should be taken in every stimulation circumstance to allow fluids to drain to the reserve pit. In the completion operation it is exceedingly difficult to accomplish this due to traffic. Because of this, the service company should provide leak free hoses, lines, and connections. Upon completion of job, the hoses should be drained to a common area for holding subsequent to introduction to the reserve pit. Every precaution should be taken to prevent accumulation of fluids on the pad proper, thereby posing a potential risk to groundwater and runoff of location.

As with the drilling operation, the equipment on location providing the completion service may leak oil. The use of absorbents and catch pans is advised.

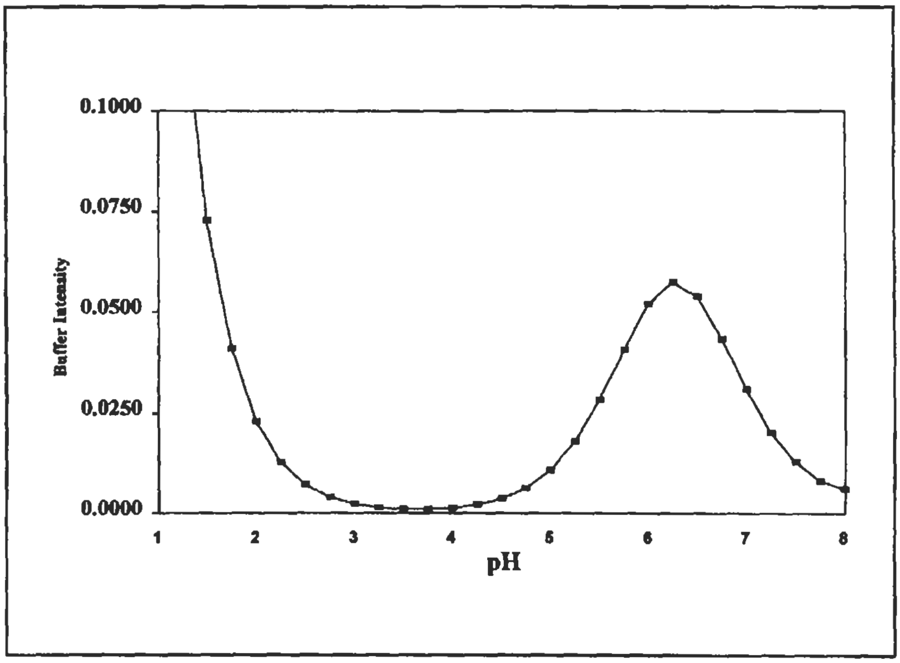

Figure 4-491. Nonideal buffer characteristics of a 0.10 M carbonate reserve pit fluid [228].

In the case of produced liquid hydrocarbons and other chemicals spilled during operations, subsequent remediation may be necessary. This section as well as Chapter 6, Environmental Considerations, details some remediation techniques currently employed.

Reclamation of the Drillsite

In the event of a dryhole, the reserve pit water usage should be maximized to prepare the mud spacers between plugs. Water in excess of this may be pumped into the hole, including solids. All USDW's must be protected in this event. Once the hole has been properly plugged and the drilling rig removed, the mousehole and rathole should be backed filled immediately to preclude any accidents. Trash is removed from the location and adjacent areas and is hauled to permitted facilities.

Reserve Pit Closure

The reserve pit commonly holds all fluids introduced to the wellbore during drilling and completion operations. This includes both the drilling and completion fluids in the event the well is stimulated for production and those cuttings produced during the drilling operation. The reserve pit, upon completion of the initial rig site activities must be reclaimed. Upon removal of the drilling rig, the reserve pit is fenced to prevent wildlife and livestock from watering. The fence is then removed upon initiation of reclaimation.

The fluids from the reserve pit may be hauled away from location for disposal, reclaimed insitu, or pumped into the wellbore given a dryhole. The operator of the wellsite is responsible for the transportation offsite of the drilling fluids. The fluids may be considered hazardous in nature due to the toxic characteristics of most drilling and completion fluids.

Evaporation. Evaporation of the water held in the pit is often the first step in reserve pit remediation, due to economic considerations over trucking and disposal. The evaporation may be mechanically driven or take place naturally. Natural evaporation is very effective in the semiarid regions. The Meyer Equation 4-385 as derived from Dalton's law may be used to estimate the local natural evaporation [224]. These are,

$$E = C(e_w - e_a)\psi \tag{4-385}$$

$$\psi = 1 + 0.1w \tag{4-386}$$

where E = evaporation rate (in/30 days)
C = empirical coefficient, equal to 15 for small shallow pools and 11 for large deep reservoirs
e_w = saturation vapor pressure corresponding to the monthly mean temperature of air for small bodies and the monthly mean temperature of water for reservoirs (inHg)
e_a = actual vapor pressure corresponding to the monthly mean temperature of air and relative humidity 30 ft above the body of water (inHg)
ψ = wind factor
w = monthly mean wind velocity measured at 30 ft above body of water (mi/hr)

Some mechanically driven systems include heated vessels or spraying of the water to enhance the natural evaporation rate. In heating, the energy needed to evaporate the water is equal to that needed to bring the water to the temperature of vaporization plus that energy required for the evaporation, where for constant volume this is

$$\Delta E = CpdT + \Delta H_{vap} \tag{4-387}$$

The heat capacity and ΔH_{vap} of pure water at 14.7 psia are commonly taken as 1 btu/lbm (°F) and 970 btu/lbm [230]. Ionic content in the pit fluid raises the energy necessary to evaporate the fluid. Figure 4-492 shows this relationship for brine water containing predominately NaCl.

In field evaporative units utilizing natural gas as the fuel source, the primary driving force is the heat supplied to the water. The theoretical evaporation rate for these units may be expressed as

$$\frac{H_c Q_g}{\Delta E \rho_w} = Q_{evap} \tag{4-388}$$

where H_c = natural gas heating value (btu/mcf)
Q_g = natural gas flow rate to burner (mcfpd)
ρ_w = density of water (ppg)
Q_{evap} = evaporation rate (gpd)

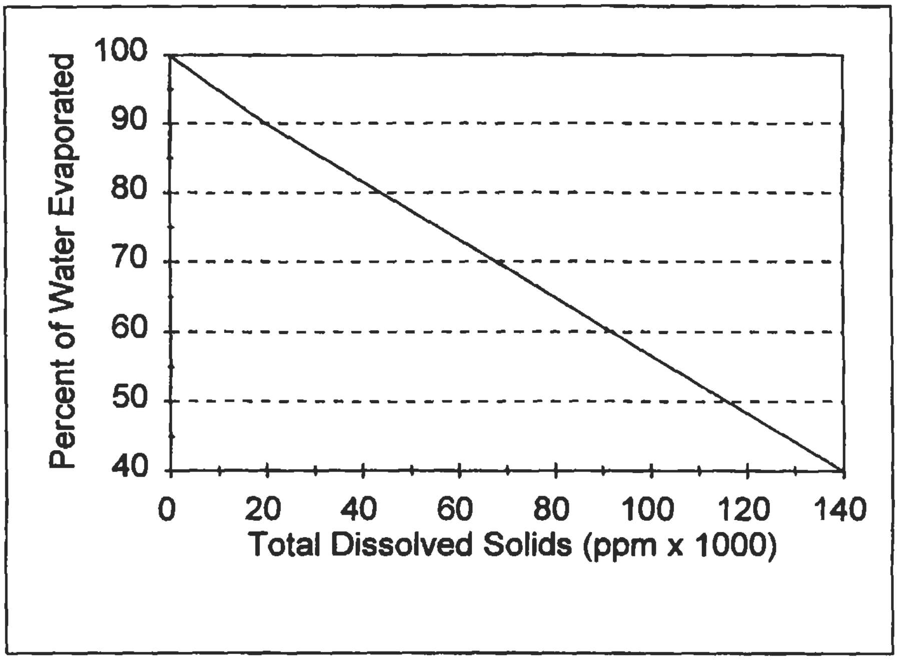

Figure 4-492. Maximum limit of evaporation as defined by TDS (NaCl) [228].

Mechanical efficiency may range from 25–75% of the theoretical evaporation rate. Efficiencies may be raised with the application of multieffect or vapor compression evaporators. The more complicated efficient systems can seldom be warranted due to the short service offered.

Spray systems rely on forming minuscule droplets of water and allowing the vaporization thereof while in suspension over the reserve pit. Allowance for wind carriage of the droplets beyond the pit must be made to prevent salting damage to the surrounding area. The shear force extended upon each droplet in combination with the relative humidity provide the driving force for the operation. Neglecting the shear component, driving force is actual and saturation vapor pressure differential. A derivation of Fick's law may be used to express the molar flux of water in air.

$$Na = \frac{DA}{1000RT}\frac{dV_p}{dx} \tag{4-389}$$

where Na = mols of water diffused to the air (mol/sec)
D = diffusivity of water in air (0.256 cm^2/sec–25°C, 1 atm)
A = the area perpendicular to the flux (cm^2)
V_p = vapor pressure of water in air (atm)
R = gas law constant (0.0821 atmL/mol°K)
x = the thickness of the film where dV_p exists (cm)

Inspection of Equation 4-389 shows that increasing the area of the active water surface will allow for greater evaporation rates. In the case of the spray systems, the evaporation from n bubbles is

$$Na = 0.01256 \frac{nD}{RT} \frac{dV_p}{dx} \tag{4-390}$$

Because of the difficulties in determining x, the thickness of the film between the two vapor pressures, an overall transfer coefficient is introduced. Based on the two film theory, the overall transfer coefficient is used. In the case of water evaporation, the gas film is the controlling mechanism and the resulting equation is

$$Na = \frac{Kga}{RT}(V_{Psat} - V_{Pact}) \tag{4-391}$$

where Kga = the overall mass transfer coefficient (t^{-1}).

Service companies offering evaporation services can supply the operator with values of Kga based on their experience in the area. The values of Kga may be used comparitively between all the systems for economic analysis.

Example: Using the Meyer Equation 4-385, the evaporation rate from a 5000 ft^2 pit is estimated. The average temperature in an area in the winter is 40°F, the corresponding saturated vapor pressure is 0.26 inHg with the actual average vapor pressure residing at 0.19 inHg. Wind velocity reaches a peak at 40 mph with a time weighted mean velocity of 5 mph, such that the evaporation rate may be estimated as

$$E = 15(0.26 - 0.19)(1 + .1(5)) = 1.58 \text{ in/mo or } 111 \text{ bbl/mo } (0.00083 \text{ molsec}^{-1})$$

Given this evaporation rate, the overall mass transfer coefficient may then be calculated from Equation 4-391,

$$Kga = \frac{NaRT}{(V_{Psat} - V_{Pact})} = 7.94 \text{ sec}^{-1}$$

Fixation of Reserve Pit Water and Solids. Another method of reclaiming the reserve pit involves combining water absorbing materials to the water and mud. Usually, the pit contents are pumped through tanks where the sorbant is combined with the pit fluid and solids. The mixture is dried and subsequently buried. Care must be taken with this method whereby any harmful contaminant is immobilized to prevent contamination to the surroundings. Studies have shown that for muds, once the majority of the water has been evaporated or pulled from the pit, the remainder may be solidified in order to comply with existing regulations. This may be done with cement, flyash, pozzalin, or any number of absorbents. Polymers have been developed to handle high pH, salt, and oil contents, where the previous mixtures have fell short. The mixture is then allowed to dry and the bulk mass is then buried. This method requires forethought on pit construction whereby complete mixing of the slurry is accomplished. If primarily bentonite and water are used, evidence has shown minor or no migration from the pit [235]. This is

$$W = \sum M_{a1\ldots n} R_{wi\ldots n} \tag{4-392}$$

where W = water available in pit (bbls)
M_a = Mass of absorbent (lbm)
R_w = water required by absorbent (bbl/lbm)

Even though materials such as bentonite will absorb tremendous amounts of water, they will not solidify to an extent that the pit may be reclaimed. In most instances, a dozer must be able to walk out to the center of the pit under a load of pushed dirt. In the event the pit materials are wet, the dozer may become mired and unable to complete the work. Thus it is often better to pick an sorbant that will harden sufficiently for this purpose.

Final Closure

Upon elimination of the fluids, the liner to the pit is folded over the residual solids in a way to prevent fluid migration. The liner is then buried inplace. The operator may choose to remove the liner contents completely to preclude any future contamination. In the case of a producing well, the location is reclaimed up to the deadmen. The adjacent areas are contoured to provide for drainage away from the production facilities. In the case of a dryhole, the entire location is reclaimed to the initial condition. All of the reclaimed area should be ripped to enhance soil conductivity. The top soil is then spread over the reclaimed area followed by seeding. Local seed mixtures are broadcast to quicken reintroduction of native plants.

Environmental Regulations

In the U.S., the Environmental Protection Agency sets policy concerning environmental regulations. The states are then allowed primacy over jurisdiction of environmental compliance if their regulations are at least as stringent as the federal regulations. Several bodies of government may be involved in environmental decisions concerning the wellsite (i.e., California). Common to most oil and gas producing states, the oil and gas division is allowed to administrate all matters concerning exploration and development matters. It is primarily their responsibility to protect the integrity of the state's land and water supplies from contamination due to oil and gas activity. The state oil and gas division's environmental regulation usually mirror closely those provided by the EPA.

Congress, in an attempt to promote mineral development in the United States, has exempted most hazardous wastes produced at the wellsite under the Resource Conservation and Recovery Act (RCRA) Subtitle C regulations. Hazardous wastes are listed due to inherent characteristics of:

- Toxicity,
- Ignitability,
- Corrosiveness,
- and Reactivity.

While a number of wastes produced at the wellsite are considered characteristic hazardous waste, some wastes fall under the nonhazardous description. The regulation of these fall under RCRA Subtitle D. Initially Subtitle "D" wastes were regulated to control dumping of domestic trash and city runoff. The EPA is considering promulgating regulation of certain oil and gas wastes under Subtitle "D" [231].

Under the RCRA exemption, wastes intrinsically associated with the exploration and development of oil and gas do not have to follow Subtitle C regulations for disposal. Under Subtitle C, hazardous wastes must follow strict guidelines for storage, treatment, and transportation and disposal. The cost of handling materials under the Subtitle C scenario is overwhelming. Under the exemption, the operator is allowed to dispose of wellsite waste in a prudent manner and is not obliged to use licensed hazardous waste transporters and licensed Treatment, Storage, and Disposal Facilities (TSDF).

Covered by the exemption are drilling fluids, cuttings, completion fluids, and rigwash. Not covered by the exemption are motor and chain oil wastes, thread cleaning solvents, painting waste, trash and unused completion fluids.

A waste product, whether exempt or not, should always be recycled if economically possible. Oil-based drilling mud typically is purchased back by the vendor for reuse. Unused chemicals are similarly taken back for resale. Arrangements should be made with the mud company for similar arrange for partial drums/sacks of chemical. Muds also may be used on more than one hole. With the advent of closed system drilling, the mud must be moved off location in the event of a producing well.

If a waste is generated that is a listed or characteristic, the operator must follow certain guidelines [232]. A listed hazardous waste (i.e., mercury, benzene) is considered hazardous if the concentrations in which they naturally occur are above certain limitations (40 CFR 261.31–261.33). The listed hazardous waste may not be diluted to achieve lesser concentrations and thus become nonhazardous. A characteristic hazardous waste (40 CFR 261.21–261.24) may be diluted to a nonhazardous status.

Most nonexempt nonacute hazardous waste generated on location is considered a small quantity. In this case, the waste may remain on location for 90 days. At that time, a Department of Transportation licensed motor carrier must transfer the waste to a EPA certified TSDF for disposal. Appropriate documentation and packaging must be conformed to. The operator continues to be liable for the waste as denoted by the cradle to grave concept [233].

Exempt wastes are usually disposed of on location following permission from the state oil and gas division. Liquid wastes, if not evaporated or fixated on location, are usually injected into Class II injection wells—refer to Chapter 6, Environmental Considerations. Solid wastes, if not acceptable to local landfills, are remediated onsite or buried in some instances. Table 4-175 shows exempt and nonexempt waste [234].

Table 4-175
Exempt and Nonexempt Oil and Gas Production Related Wastes

Exempt Wastes	Nonexempt Wastes
Produced water	Unused well completion/stimulation fluids
Drilling fluids	Radioactive tracer wastes
Drill cuttings	Painting wastes
Rigwash	Service company wastes such as empty drums, drum rinsate, vacuum truck rinsate, sandblast media, spent solvents, spilled chemicals, and waste acids
Well completion/stimulation fluids	Vacuum truck and drum rinsate containing nonexempt wastes

BS&W and other tank bottom material from storage facilities that hold product and exempt waste	Refinery wastes
Accumulated materials such as hydrocarbons, solids, sand, and emulsions from production separators, fluid treating vessels, and production impoundments	Liquid and solid wastes resulting from operations by tank bottom reclaimers
Pit sludges and contaminated bottoms from storage or disposal of exempt waste	Used lubrication oils
Workover wastes	Waste compressor oil, filters, and blowdown
Gas plant dehydration and sweetening waste	Used hydraulic oil
Cooling tower blowdown	Waste solvents
Spent filters, filter media, and backwash involved in the treatment of the product or an exempt waste	Waste in transportation pipeline related pits
Packer fluids	Caustic or acid cleaners
Produced sand	Boiler cleaning wastes
Pipe scale, hydrocarbon solids, hydrates, and other deposits removed from piping and equipment prior to transportation	Boiler refractory bricks
Hydrocarbon bearing soil	Boiler scrubbing fluids, sludges and ash
Pigging wastes	Incinerator ash
Wastes from subsurface gas storage and retrieval	Laboratory wastes
Liquid hydrocarbons removed from the production stream but not from oil refining	Pesticide wastes
Gases from the production stream, such as H_2S, CO_2, and volitized hydrocarbons	Gas plant cleaning wastes
Materials ejected from a producing well during blowdown	Drums, insulation, and miscellaneous solids
Waste crude from primary operations and production	
Light organics volatilized from exempt wastes in reserve pits, impoundments, or production equipment	
Constituents removed from produced water before it is injected or otherwise disposed of	

Generally, waste that *must* be produced to complete the work involved in the exploration and production of oil and gas is considered exempt, allowing the operation the option of disposing of such waste in a prudent matter. A nonexempt waste should be avoided at all costs if possible (i.e., radioactive tracer wastes). These wastes must be disposed of according to federal and state regulations. Because these regulations are becoming more complicated with time, the operator should consult the primary regulator in the event of questionable circumstances.

OFFSHORE OPERATIONS

The basic principles of rotary drilling defined for onshore operations are also applicable to offshore operations. The primary difference offshore is that a stable, self-contained platform must be provided for the drilling equipment. Communication with a well through possibly thousands of feet of water provides for mechanical as well as procedural differences, primarily in well control. Onshore technology can be applied to offshore operations in many instances on bottom-supported rigs, but the use of floating vessels has resulted in the development of new technology tailored to the offshore environment.

Drilling Vessels

Offshore drilling vessels are classified as either bottom-supported or floating-type vessels. Water depth is generally the governing factor as to which type of vessel is employed.

Bottom-supported vessels consist of drilling platforms, jackup rigs, and drilling barges. A drilling platform is not employed until reserves warranting field development is indicated. Platforms are currently being set in water depths up to 1,000 ft, and also serve as permanent production facilities.

A jackup rig (Figure 4-493) is used for drilling in water depths up to about 350 ft. The drilling platform on a jackup can be raised or lowered on the legs of the vessel to allow for use in various water depths. These rigs can also be used alongside production platforms for developmental drilling. Jackup rigs thus provide a mobile, bottom-supported drilling vessel suitable for use in relatively shallow water.

Drilling barges, commonly employed in inland waters and marshes, can be used where water depths do not exceed about 25 ft. These self-contained vessels provide the least expensive drilling vessel, but have limited applicability because of water depth limitations.

Drillships and semisubmersible rigs are the two types of floating drilling vessels.

A drillship (Figure 4-494) provides a drilling platform in waters up to 10,000 ft deep. The rig is kept on location by complex mooring and dynamic positioning systems requiring computer control.

A semisubmersible rig (Figure 4-495) can be used in water up to 6,000 ft deep, and provides a drilling platform generally more stable than does a drillship. As on a drillship, a semisubmersible rig is stabilized by a complex mooring and positioning system. Since a major portion of the vessel is submerged, wave action can be minimized. Mooring systems are specifically designed to resist surface forces, and ballast can be preferentially shifted within the vessel to provide stability during rough weather.

When choosing an offshore drillsite, the primary considerations are the location of shipping lanes, foundation stability (for bottom-supported vessels), and the possible presence of shallow gas. Seismic surveys generally provide

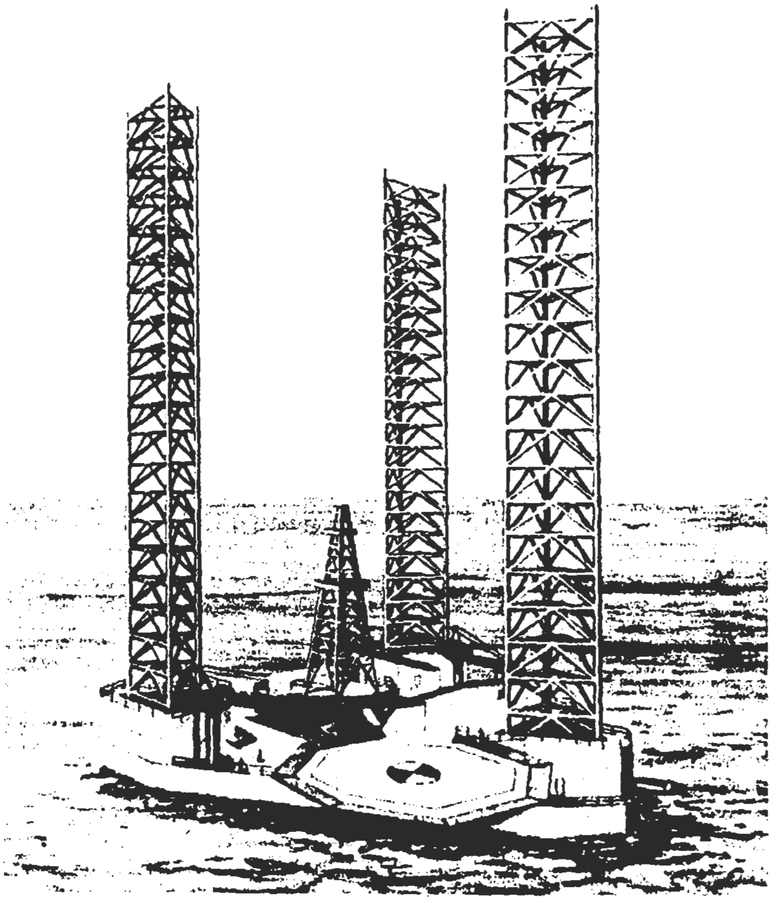

Figure 4-493. Jack-up drilling rig.

suitable information on foundation stability, and can indicate the presence of potentially dangerous shallow gas deposits.

Marine Riser Systems

The most fundamental difference found between onshore and offshore drilling occurs when the wellhead is located at the seafloor. This configuration makes communication with the well more complex. A marine riser provides communication and circulation capability between the surface and the seafloor, and is used at some point during most offshore drilling. The riser consists of large-diameter (17–20) in.) steel pipe joints of approximately 50-ft lengths, with quick-connect couplings. The riser can be connected at the seafloor to a wellhead or to a subsea blowout preventer stack. A diverter system is usually attached at the

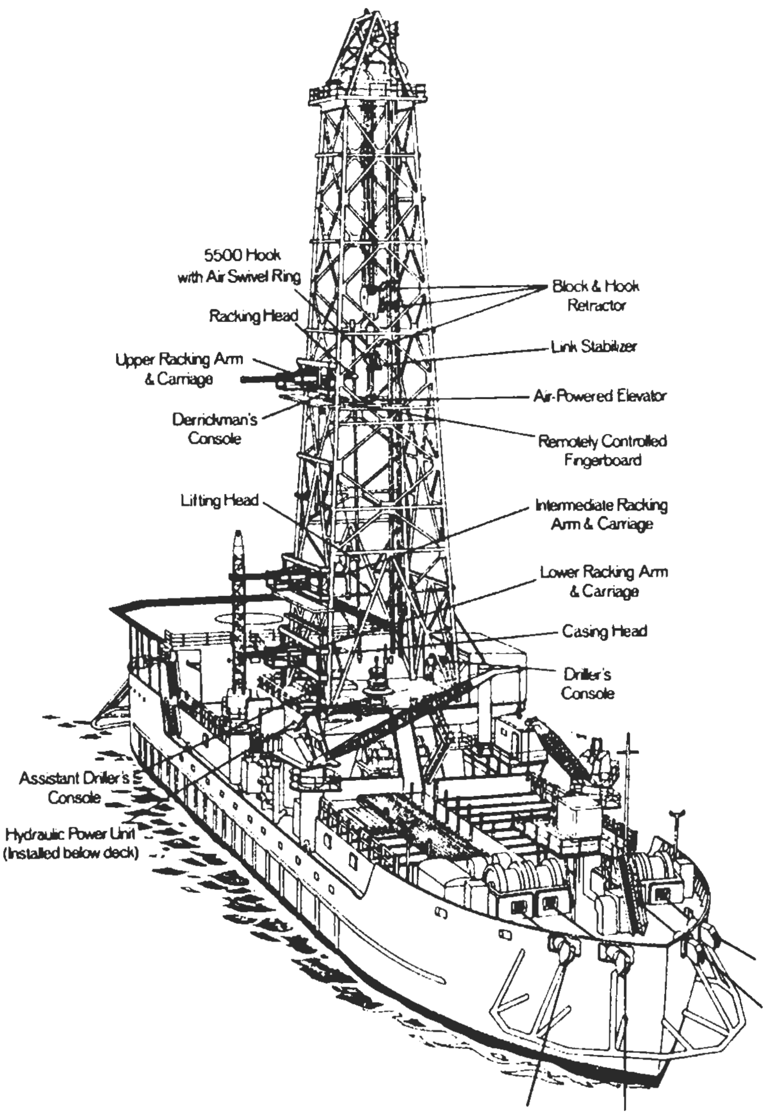

Figure 4-494. Drillship.

surface. Vertical vessel motion is tolerated by adding a telescoping (slip) joint at the surface. This joint will usually allow up to 30 ft of vertical vessel motion. Lateral motion is tolerated by use of ball joint connections at the seafloor and surface. The riser can be quickly detached from the wellhead or blowout preventer stack to facilitate vessel movement during adverse weather conditions. Figure 4-496 shows two typical drilling configurations with a riser in place, and Figure 4-497 shows in more detail a typical riser system that would be used while drilling the intermediate hole from a floating vessel.

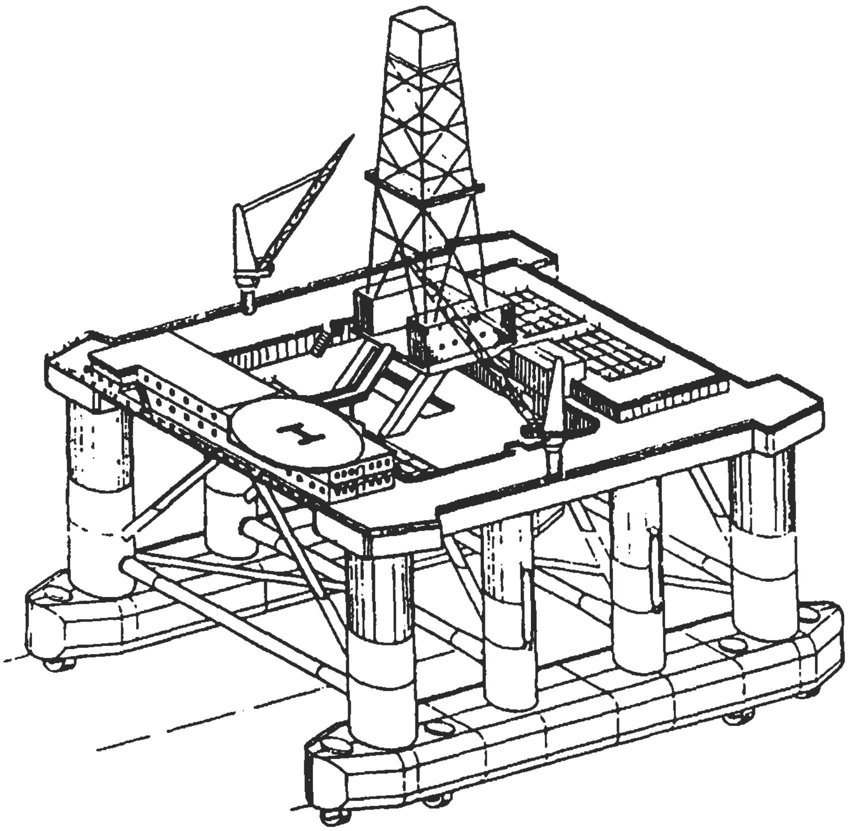

Figure 4-495. Semisubmersible drilling rig.

A marine riser must be held in tension to prevent the riser from collapsing under its own weight. This can be accomplished by adding buoyant material to the riser pipe, or by mechanical tensioning devices (Figures 4-498, 4-499). Tensioning devices are usually required in water depths greater than 250 ft.

Many operators drill the conductor hole (approximately 1,000 ft) without using a riser. When drilling with a floating vessel, the shallow portion of the surface hole is considered expendable, and can easily be abandoned in case of shallow gas flow, deviation difficulties, or other shallow-hole problems. Drilling is accomplished by circulating returns to the sea floor. Once the conductor hole is drilled, the marine riser can be attached and used to drill the remainder of the well. Problems of lost circulation can be avoided through use of riserless systems. Another advantage with riserless drilling is that the well can be abandoned quickly if shallow gas is encountered. Most operators prefer to vent the gas at the seafloor rather than bringing it to the surface. The disadvantage of the riserless system is in the lack of well control alternatives. The dynamic

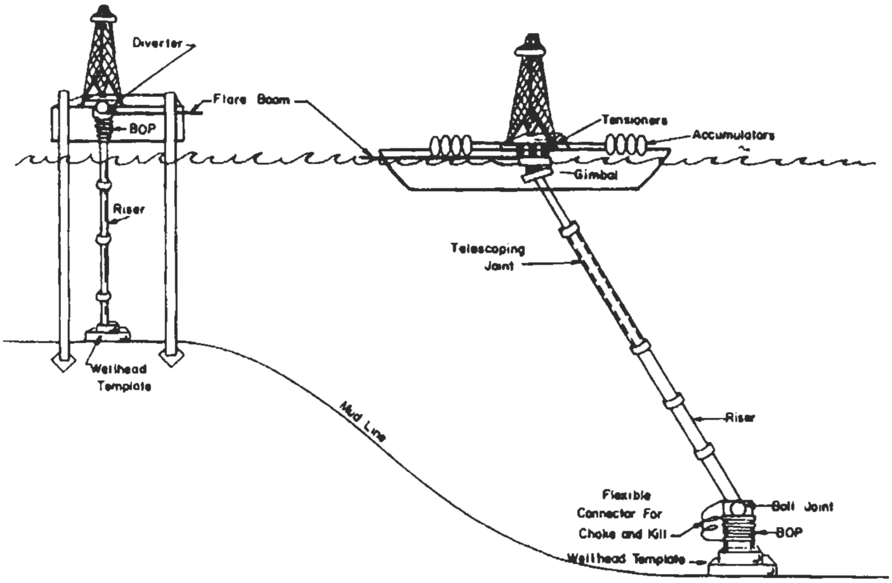

Figure 4-496. Marine riser systems.

kill, if attempted rapidly upon kick recognition, is the only recourse in attempting to control the well.

Casing Programs

Casing programs offshore typically consist of 30-in. structural, 20-in. conductor, $13\frac{3}{8}$-in. surface, and $9\frac{5}{8}$-in. intermediate casing strings. Because seawater is in perfect communication with the sediments, pore pressure is increased by increasing water depth, and the margin between fracture pressure and pore pressure is lessened. Also, abnormal pore pressure is found in many offshore geological environments, forcing fairly stringent casing designs to be used. Several intermediate casing strings or liners are often required to reach the target depth. Most operators prefer to set 7-in. production casing to facilitate larger production tubing and quicker payout.

Well Control

Offshore, well control equipment and associated operations present some differences from that seen and used onshore. In some instances onshore equipment can be employed, but the offshore environment generally dictates a modification of equipment and procedures. There are several different well configurations used offshore, often on the same well at different drilling intervals, and each configuration has specific well control procedures that should be followed. A well may be equipped with a surface blowout preventer stack; a subsea blowout preventer stack, riser and diverter system; a riser and diverter system with no blowout preventer; a diverter only; or a riserless system with no well control equipment.

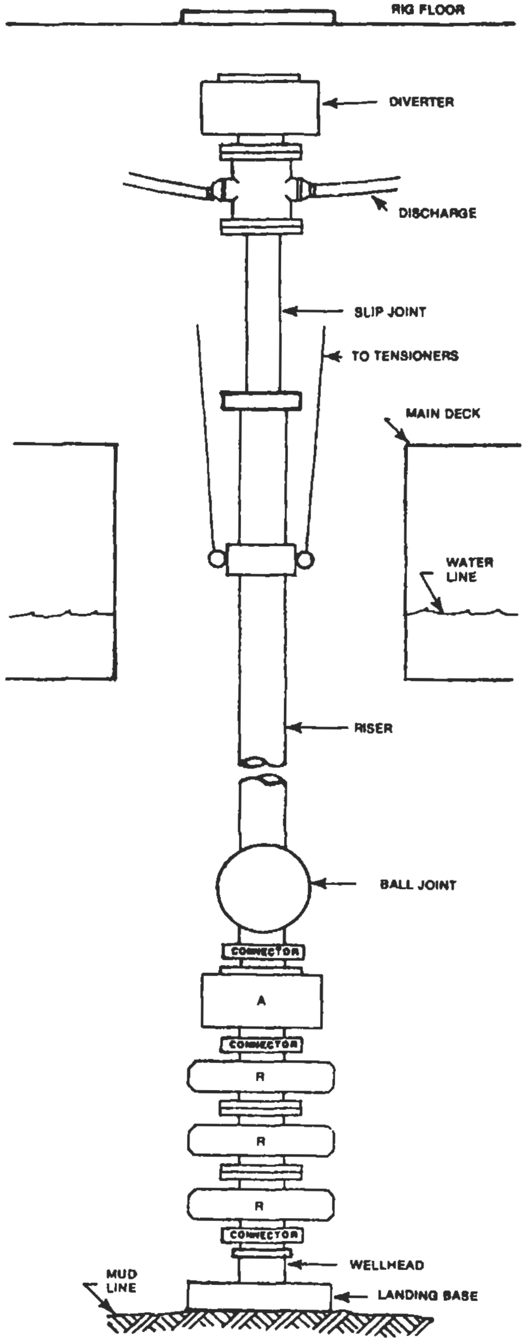

Figure 4-497. Marine riser for floating system.

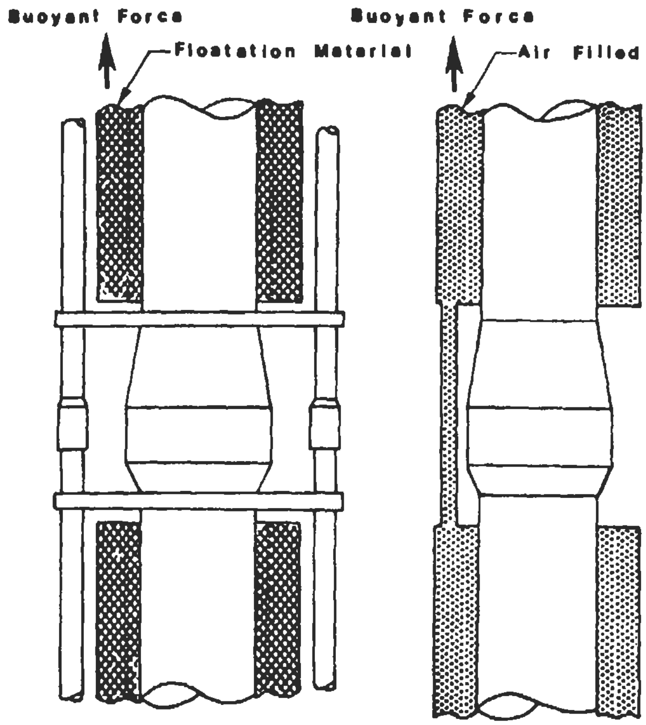

Figure 4-498. Buoyant riser tensioning device.

Surface Blowout Preventer Stacks

When a development platform has been set, wells can be drilled with a land-type well design. Conductor casing is set from above sea level, through the water, to as much as 1,500 ft subsea. The conductor provides for mud returns, and allows the blowout prevention system to exist in the surface environment. This well design is essentially identical to that onshore, as are the well control procedures that should be employed.

Subsea Blowout Preventer Stacks

Due to the nonpermanent nature of exploratory drilling offshore, the operator does not always have the luxury of placing the blowout preventers at the surface. By placing the blowout preventer stack at the seafloor and communicating with the well through a detachable marine riser, a well can be shut in at the seafloor. This action removes from the drilling vessel the responsibility of maintaining surface position at all times. Figure 4-500 shows a typical subsurface blowout preventer stack. Note the presence of choke and kill lines extending from the surface to the stack. These lines allow fluids to be pumped out of the well, bypassing the marine riser, and are the source of the procedural differences between onshore and offshore well control operations.

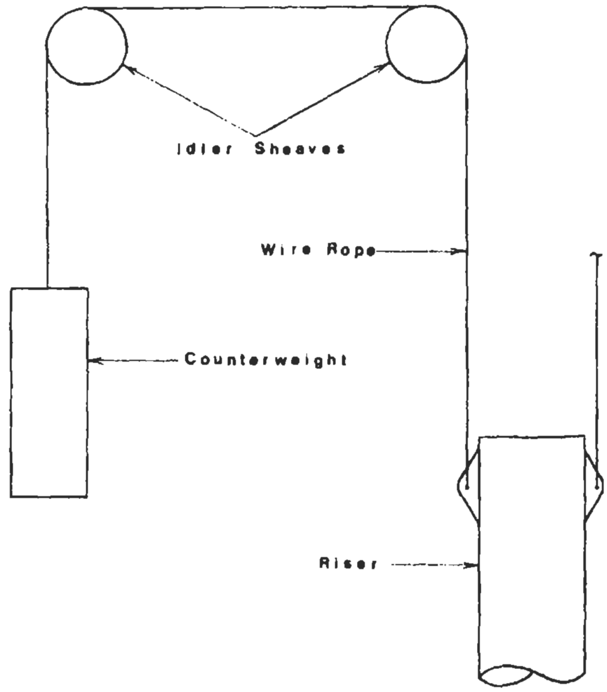

Figure 4-499. Mechanical riser tensioning device.

Well Control Procedures

There are several mechanical differences between onshore and offshore equipment that lead to procedural differences in well control. When a well is shut in by a subsea blowout preventer, fluids must be circulated down the drillstring, up the casing annulus, and up the choke line to the surface. The marine riser/drillstring annulus is thus bypassed. Because the choke line is of small diameter (3-4 in.) and, depending on water depth, may be very long, frictional pressure drop in the choke line may be very large. When preplanning for well control operations, the kill rate circulating pressure must be obtained by circulating the well through the choke line. Neglecting choke line friction in calculations can cause excessive backpressure to be placed on the casing shoe. Choke line friction is considered when calculating the pump pressure startup schedule. If the circulating pressure has been measured by circulating through the choke line, the remainder of the calculations are preformed as for a surface stack. Choke line friction can be reduced by using larger choke lines, or by circulating the well through both the choke and kill lines.

When the well is circulated through the choke line, a rapid loss in hydrostatic pressure is seen when the kick fluid begins to enter the choke line. Hydrostatic pressure is lost because low density gas is displacing the drilling mud from the small volume of choke line. Small kick volumes can result in long columns of gas in the choke line. Surface choke response must be rapid enough to prevent new kick fluid from entering the well due to the reduction in bottomhole

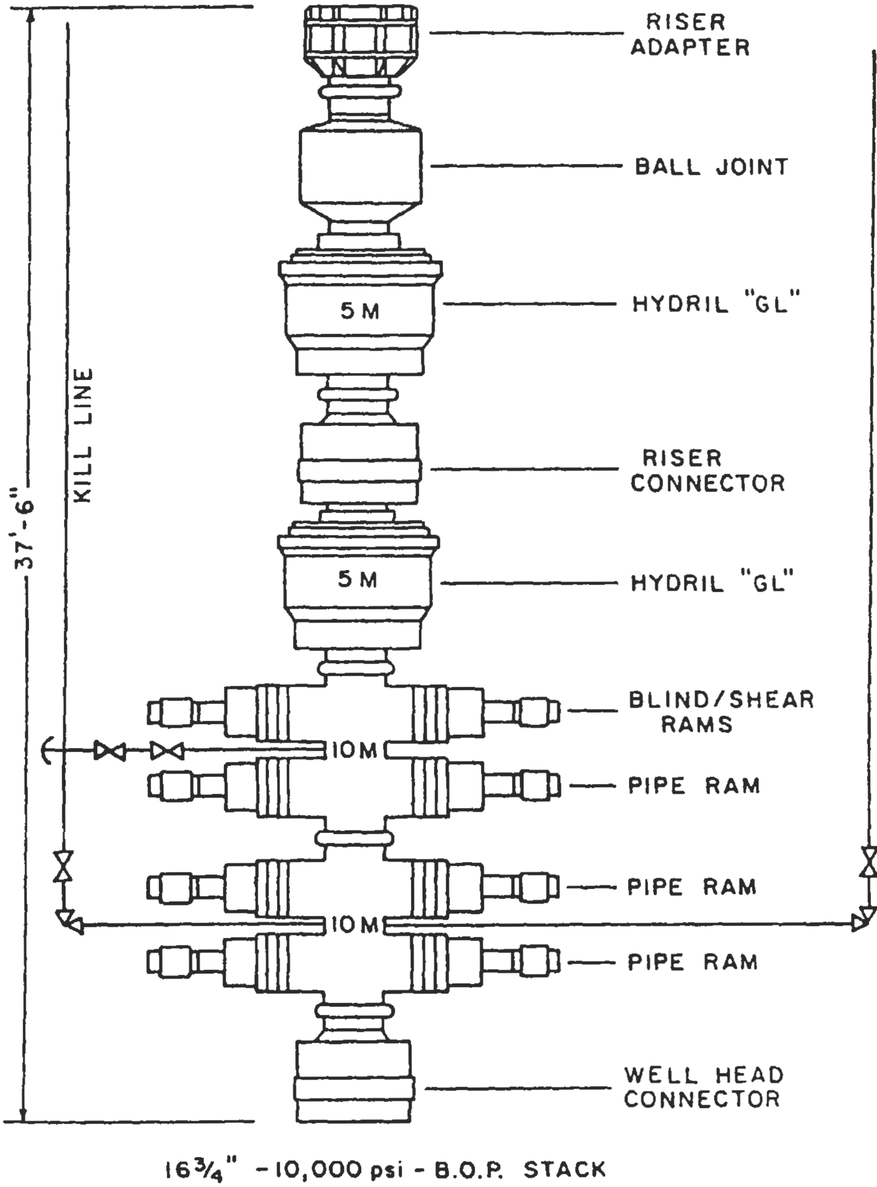

Figure 4-500. Subsea blowout preventer stack.

pressure. Also, when kill fluid enters the choke line and begins to displace the kick fluid, hydrostatic pressure will increase rapidly, thus increasing bottomhole pressure. Slow response by the choke operator can fracture the casing shoe. A choke operator is faced with an unusually demanding job when pumping out a kick through a subsurface blowout preventer.

In deep water, it is common to find kick fluids trapped above the blowout preventer stack when the well is shut in. These fluids must be vented through the diverter system.

Diverter Systems

It is common in many offshore areas to encounter a shallow gas hazard. Quite often, these hazards can be spotted on seismic, and a surface location is chosen to avoid the hazard. However, there is always a risk of encountering a shallow gas flow with insufficient casing in the well to allow a shut-in. In this instance a diverter system is called on as a safety measure. The ideal function of the diverter system is to allow the well to flow and subside by natural means. In many cases the diverter system simply provides enough time to evacuate the rig.

Figure 4-501 shows a typical diverter system setup. The components of the system are the annular preventer, vent lines, the control system, and the conductor or structural casing.

The major operational consideration when using a diverter system is to be certain that the valves on the vent lines are fully open before the annular preventers are closed. When drilling the conductor or surface hole, it is generally accepted that shutting in the well will cause a subsurface blowout that is likely to broach to the surface. This cannot be tolerated when drilling from a bottom-supported vessel. Experience has shown that the diverter operations can also cause a subsurface blowout. Large-diameter vent lines free from turns or bends generally lower the backpressure placed on the well by the diverter system, therefore reducing the risk of a subsurface blowout. The lack of bends in the lines greatly reduces the erosion potential of the flow. Sufficient conductor casing

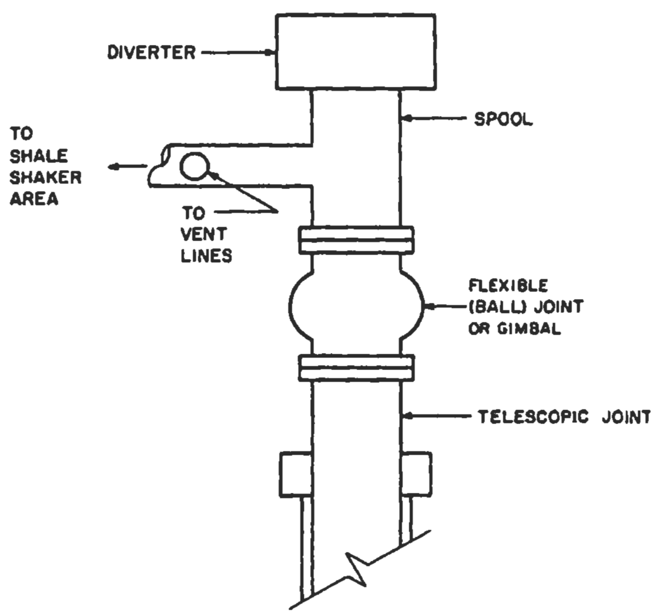

Figure 4-501. Diverter system.

depth is also very important in preventing a subsurface blowout. Conductor depths as much as 1,500 ft are sometimes required to prevent subsurface blowouts.

The Minerals Management Service requires a minimum 6-in.-diameter diverter vent line, but many operators are now using as large as 12-in.-diameter vent lines.

While a bottom-supported vessel must divert when shallow gas is encountered, a floating vessel has the additional option of simply abandoning the well. This option has led to the use of riserless systems when drilling the surface hole. However, a dynamic kill provides the only means of controlling the well. A dynamic kill makes use of annular friction as well as a heavier mud to hold backpressure on the formation. If very short wellbores are involved, the dynamic kill rates are usually to large to be practical. A well being drilled with a riserless system is very likely to be lost if shallow gas is encountered.

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Appendix

Units and Conversions

Table A-1
Alphabetical List of Units
(symbols of SI units given in parentheses)

To Convert From	To	Multiply By**	
abampere	ampere (A)	1.0*	E+01
abcoulomb	coulomb (C)	1.0*	E+01
abfarad	farad (F)	1.0*	E+09
abhenry	henry (H)	1.0*	E−09
abmho	siemens (S)	1.0*	E+09
abohm	ohm (Ω)	1.0*	E−09
abvolt	volt (V)	1.0*	E−08
acre-foot (U.S. survey)[1]	meter^3 (m^3)	1.233 489	E+03
acre (U.S. survey)[1]	meter^2 (m^2)	4.046 873	E+03
ampere hour	coulomb (C)	3.6*	E+03
are	meter^2 (m^2)	1.0*	E+02
angstrom	meter (m)	1.0*	E−10
astronomical unit	meter (m)	1.495 979	E+11
atmosphere (standard)	pascal (Pa)	1.013 250*	E+05
atmosphere (technical = 1 kgf/cm^2)	pascal (Pa)	9.806 650*	E+04
bar	pascal (Pa)	1.0*	E+05
barn	meter^2 (m^2)	1.0*	E−28
barrel (for petroleum, 42 gal)	meter^3 (m^3)	1.589 873	E−01
board foot	meter^3 (m^3)	2.359 737	E−03
British thermal unit (International Table)[2]	joule (J)	1.055 056	E+03
British thermal unit (mean)	joule (J)	1.055 87	E+03
British thermal unit (thermochemical)	joule (J)	1.054 350	E+03
British thermal unit (39°F)	joule (J)	1.059 67	E+03
British thermal unit (59°F)	joule (J)	1.054 80	E+03
British thermal unit (60°F)	joule (J)	1.054 68	E+03
Btu (International Table)-ft/(hr-ft^2-°F) (thermal conductivity)	watt per meter kelvin [W/(m·K)]	1.730 735	E+00
Btu (thermochemical)-ft/(hr-ft^2-°F) (thermal conductivity)	watt per meter kelvin [W/(m·K)]	1.729 577	E+00
Btu (International Table)-in./(hr-ft^2-°F) (thermal conductivity)	watt per meter kelvin [W/(m·K)]	1.442 279	E−01
Btu (thermochemical)-in./(hr-ft^2-°F) (thermal conductivity)	watt per meter kelvin [W/(m·K)]	1.441 314	E−01

Btu (International Table)-in./(s-ft²-°F) (thermal conductivity)	watt per meter kelvin [W/(m·K)]	5.192 204	E + 02
Btu (thermochemical)-in./(s-ft²-°F) (thermal conductivity)	watt per meter kelvin [W/(m·K)]	5.188 732	E + 02
Btu (International Table)/hr	watt (W)	2.930 711	E − 01
Btu (thermochemical)/hr	watt (W)	2.928 751	E − 01
Btu (thermochemical)/min	watt (W)	1.757 250	E + 01
Btu (thermochemical)/s	watt (W)	1.054 350	E + 03
Btu (International Table)/ft²	joule per meter² (J/m²)	1.135 653	E + 04
Btu (thermochemical)/ft²	joule per meter² (J/m²)	1.134 893	E + 04
Btu (thermochemical)/(ft²-hr)	watt per meter² (W/m²)	3.152 481	E + 00
Btu (thermochemical)/(ft²-min)	watt per meter² (W/m²)	1.891 489	E + 02
Btu (thermochemical)/(ft²-s)	watt per meter² (W/m²)	1.134 893	E + 04
Btu (thermochemical)/(in.²-s)	watt per meter² (W/m²)	1.634 246	E + 06
Btu (International Table)/(hr-ft²-°F) (thermal conductance)	watt per meter² kelvin [W/(m²·K)]	5.678 263	E + 00
Btu (thermochemical)/(hr-ft²-°F) (thermal conductance)	watt per meter² kelvin [W/(m²·K)]	5.674 466	E + 00
Btu (International Table)/(s-ft²-°F)	watt per meter² kelvin [W/(m²·K)]	2.044 175	E + 04
Btu (thermochemical)/(s-ft²-°F)	watt per meter² kelvin [W/(m²·K)]	2.042 808	E + 04
Btu (International Table)/lbm	joule per kilogram (J/kg)	2.326*	E + 03
Btu (thermochemical)/lbm	joule per kilogram (J/kg)	2.324 444	E + 03
Btu (International Table)/(lbm-°F) (heat capacity)	joule per kilogram kelvin [J/(kg·K)]	4.186 8*	E + 03
Btu (thermochemical)/(lbm-°F) (heat capacity)	joule per kilogram kelvin [J/(kg·K)]	4.184 000	E + 03

**See footnote on Page 13.

(1) Since 1893 the U.S. basis of length measurement has been derived from metric standards. In 1959 a small refinement was made in the definition of the yard to resolve discrepancies both in this country and abroad, which changed its length from 3600/3937 m to 0.9144 m exactly. This resulted in the new value being shorter by two parts in a million. At the same time it was decided that any data in feet derived from and published as a result of geodetic surveys within the U.S. would remain with the old standard (1 ft = 1200/3937 m) until further decision. This foot is named the U.S. survey foot. As a result, all U.S. land measurements in U.S. customary units will relate to the meter by the old standard. All the conversion factors in these tables for units referenced to this footnote are based on the U.S. survey foot, rather than the international foot. Conversion factors for the land measure given below may be determined from the following relationships:

1 league = 3 miles (exactly)
1 rod = 16½ ft (exactly)
1 chain = 66 ft (exactly)
1 section = 1 sq mile
1 township = 36 sq miles

(2) This value was adopted in 1956. Some of the older International Tables use the value 1.055 04 E + 03. The exact conversion factor is 1.055 055 852 62* E + 03.

Table A-1
(continued)

To Convert From	To	Multiply By**	
bushel (U.S.)	meter^3 (m^3)	3.523 907	E−02
caliber (inch)	meter (m)	2.54*	E−02
calorie (International Table)	joule (J)	4.186 8*	E+00
calorie (mean)	joule (J)	4.190 02	E+00
calorie (thermochemical)	joule (J)	4.184*	E+00
calorie (15°C)	joule (J)	4.185 80	E+00
calorie (20°C)	joule (J)	4.181 90	E+00
calorie (kilogram, International Table)	joule (J)	4.186 8*	E+03
calorie (kilogram, mean)	joule (J)	4.190 02	E+03
calorie (kilogram, thermochemical)	joule (J)	4.184*	E+03
cal (thermochemical)/cm^2	joule per meter^2 (J/m^2)	4.184*	E+04
cal (International Table)/g	joule per kilogram (J/kg)	4.186*	E+03
cal (thermochemical)/g	joule per kilogram (J/kg)	4.184*	E+03
cal (International Table)/(g·°C)	joule per kilogram kelvin [J/(kg·K)]	4.186 8*	E+03
cal (thermochemical)/(g·°C)	joule per kilogram kelvin [J/(kg·K)]	4.184*	E+03
cal (thermochemical)/min	watt (W)	6.973 333	E−02
cal (thermochemical)/s	watt (W)	4.184*	E+00
cal (thermochemical)/(cm^2·min)	watt per meter^2 (W/m^2)	6.973 333	E+02
cal (thermochemical)/(cm^2·s)	watt per meter^2 (W/m^2)	4.184*	E+04
cal (thermochemical)/(cm·s·°C)	watt per meter kelvin [W/(m·K)]	4.184*	E+02
capture unit (c.u. = 10^{-3} cm^{-1})	per meter (m^{-1})	1.0*	E−01
carat (metric)	kilogram (kg)	2.0*	E−04
centimeter of mercury (0°C)	pascal (Pa)	1.333 22	E+03
centimeter of water (4°C)	pascal (Pa)	9.806 38	E+01
centipoise	pascal second (Pa·s)	1.0*	E−03
centistokes	meter^2 per second (m^2/s)	1.0*	E−06
circular mil	meter^2 (m^2)	5.067 075	E−10
clo	kelvin meter^2 per watt [(K·m^2)/W]	2.003 712	E−01
cup	meter^3 (m^3)	2.365 882	E−04
curie	becquerel (Bq)	3.7*	E+10
cycle per second	hertz (Hz)	1.0*	E+00
day (mean solar)	second (s)	8.640 000	E+04
day (sidereal)	second (s)	8.616 409	E+04

degree (angle)	radian (rad)	1.745 329 E−02
degree Celsius	kelvin (K)	$T_K = T_{°C} + 273.15$
degree centigrade (see degree Celsius)		
degree Fahrenheit	degree Celsius	$T_{°C} = (T_{°F} - 32)/1.8$
degree Fahrenheit	kelvin (K)	$T_K = (T_{°F} + 459.67)/1.8$
degree Rankine	kelvin (K)	$T_K = T_{°R}/1.8$
°F·hr·ft²/Btu (International Table) (thermal resistance)	kelvin meter² per watt [(K·m²)/W]	1.781 102 E−01
°F·hr·ft²/Btu (thermochemical) (thermal resistance)	kelvin meter² per watt [(K·m²)/W]	1.762 250 E−01
denier	kilogram per meter (kg/m)	1.111 111 E−07
dyne	newton (N)	1.0* E−05
dyne-cm	newton meter (N·m)	1.0* E−07
dyne/cm²	pascal (Pa)	1.0* E−01
electronvolt	joule (J)	1.602 19 E−19
EMU of capacitance	farad (F)	1.0* E+09
EMU of current	ampere (A)	1.0* E+01
EMU of electric potential	volt (V)	1.0* E−08
EMU of inductance	henry (H)	1.0* E−09
EMU of resistance	ohm (Ω)	1.0* E−09
ESU of capacitance	farad (F)	1.112 650 E−12
ESU of current	ampere (A)	3.335 6 E−10
ESU of electric potential	volt (V)	2.997 9 E+02
ESU of inductance	henry (H)	8.987 554 E+11
ESU of resistance	ohm (Ω)	8.987 554 E+11
erg	joule (J)	1.0* E−07
erg/cm²·s	watt per meter² (W/m²)	1.0* E−03
erg/s	watt (W)	1.0* E−07
faraday (based on carbon-12)	coulomb (C)	9.648 70 E+04
faraday (chemical)	coulomb (C)	9.649 57 E+04
faraday (physical)	coulomb (C)	9.652 19 E+04
fathom	meter (m)	1.828 8 E+00
fermi (femtometer)	meter (m)	1.0* E−15
fluid ounce (U.S.)	meter³ (m³)	2.957 353 E−05
foot	meter (m)	3.048* E−01
foot (U.S. survey)(1)	meter (m)	3.048 006 E−01

Table A-1
(continued)

To Convert From	To	Multiply By**
foot of water (39.2°F)	pascal (Pa)	2.988 98 E+03
sq ft	meter^2 (m^2)	9.290 304* E−02
ft^2/hr (thermal diffusivity)	meter^2 per second (m^2/s)	2.580 640* E−05
ft^2/s	meter^2 per second (m^2/s)	9.290 304* E−02
cu ft (volume; section modulus)	meter^3 (m^3)	2.831 685 E−02
ft^3/min	meter^3 per second (m^3/s)	4.719 474 E−04
ft^3/s	meter^3 per second (m^3/s)	2.831 685 E−02
ft^4 (moment of section)(3)	meter^4 (m^4)	8.630 975 E−03
ft/hr	meter per second (m/s)	8.466 667 E−05
ft/min	meter per second (m/s)	5.080* E−03
ft/s	meter per second (m/s)	3.048* E−01
ft/s^2	meter per second^2 (m/s^2)	3.048* E−01
footcandle	lux (lx)	1.076 391 E+01
footlambert	candela per meter^2 (cd/m^2)	3.426 259 E+00
ft-lbf	joule (J)	1.355 818 E+00
ft-lbf/hr	watt (W)	3.766 161 E−04
ft-lbf/min	watt (W)	2.259 697 E−02
ft-lbf/s	watt (W)	1.355 818 E+00
ft-poundal	joule (J)	4.214 011 E−02
free fall, standard (*g*)	meter per second^2 (m/s^2)	9.806 650* E+00
cm/s^2	meter per second^2 (m/s^2)	1.0* E−02
gallon (Canadian liquid)	meter^3 (m^3)	4.546 090 E−03
gallon (U.K. liquid)	meter^3 (m^3)	4.546 092 E−03
gallon (U.S. dry)	meter^3 (m^3)	4.404 884 E−03
gallon (U.S. liquid)	meter^3 (m^3)	3.785 412 E−03
gal (U.S. liquid)/day	meter^3 per second (m^3/s)	4.381 264 E−08
gal (U.S. liquid)/min	meter^3 per second (m^3/s)	6.309 020 E−05
gal (U.S. liquid)/hp-hr (SFC, specific fuel consumption)	meter^3 per joule (m^3/J)	1.410 089 E−09
gamma (magnetic field strength)	ampere per meter (A/m)	7.957 747 E−04
gamma (magnetic flux density)	tesla (T)	1.0* E−09
gauss	tesla (T)	1.0* E−04
gilbert	ampere (A)	7.957 747 E−01

gill (U.K.)	meter^3 (m^3)	1.420 654	E − 04
gill (U.S.)	meter^3 (m^3)	1.182 941	E − 04
grad	degree (angular)	9.0*	E − 01
grad	radian (rad)	1.570 796	E − 02
grain (1/7000 lbm avoirdupois)	kilogram (kg)	6.479 891*	E − 05
grain (lbm avoirdupois/7000)/gal (U.S. liquid)	kilogram per meter^3 (kg/m^3)	1.711 806	E − 02
gram	kilogram (kg)	1.0*	E − 03
g/cm^3	kilogram per meter^3 (kg/m^3)	1.0*	E + 03
gram-force/cm^2	pascal (Pa)	9.806 650*	E + 01
hectare	meter^2 (m^2)	1.0*	E + 04
horsepower (550 ft-lbf/s)	watt (W)	7.456 999	E + 02
horsepower (boiler)	watt (W)	9.809 50	E + 03
horsepower (electric)	watt (W)	7.460*	E + 02
horsepower (metric)	watt (W)	7.354 99	E + 02
horsepower (water)	watt (W)	7.460 43	E + 02
horsepower (U.K.)	watt (W)	7.457 0	E + 02
hour (mean solar)	second (s)	3.600 000	E + 03
hour (sidereal)	second (s)	3.590 170	E + 03
hundredweight (long)	kilogram (kg)	5.080 235	E + 01
hundredweight (short)	kilogram (kg)	4.535 924	E + 01
inch	meter (m)	2.54*	E − 02
inch of mercury (32°F)	pascal (Pa)	3.386 38	E + 03
inch of mercury (60°F)	pascal (Pa)	3.376 85	E + 03
inch of water (39.2°F)	pascal (Pa)	2.490 82	E + 02
inch of water (60°F)	pascal (Pa)	2.488 4	E + 02
sq in.	meter^2 (m^2)	6.451 6*	E − 04
cu in. (volume; section modulus)[(4)]	meter^3 (m^3)	1.638 706	E − 05
in.^{3}/min	meter^3 per second (m^3/s)	2.731 177	E − 07
in.^{4} (moment of section)[(3)]	meter^4 (m^4)	4.162 314	E − 07
in./s	meter per second (m/s)	2.54*	E − 02
in./s^2	meter per second^2 (m/s^2)	2.54*	E − 02
kayser	1 per meter (1/m)	1.0*	E + 02
kelvin	degree Celsius	$T_{°C} = T_K - 273.15$	

[(3)] This sometimes is called the moment of inertia of a plane section about a specified axis.
[(4)] The exact conversion factor is 1.638 706 4*E − 05.

Table A-1
(continued)

To Convert From	To	Multiply By**
kilocalorie (International Table)	joule (J)	4.186 8* E+03
kilocalorie (mean)	joule (J)	4.190 02 E+03
kilocalorie (thermochemical)	joule (J)	4.184* E+03
kilocalorie (thermochemical)/min	watt (W)	6.973 333 E+01
kilocalorie (thermochemical)/s	watt (W)	4.184* E+03
kilogram-force (kgf)	newton (N)	9.806 65* E+00
kgf·m	newton meter (N·m)	9.806 65* E+00
kgf·s^2/m (mass)	kilogram (kg)	9.806 65* E+00
kgf/cm^2	pascal (Pa)	9.806 65* E+04
kgf/m^2	pascal (Pa)	9.806 65* E+00
kgf/mm^2	pascal (Pa)	9.806 65* E+06
km/h	meter per second (m/s)	2.777 778 E−01
kilopond	newton (N)	9.806 65* E+00
kilowatthour (kW-hr)	joule (J)	3.6* E+06
kip (1000 lbf)	newton (N)	4.448 222 E+03
kip/in.^{2} (ksi)	pascal (Pa)	6.894 757 E+06
knot (international)	meter per second (m/s)	5.144 444 E−01
lambert	candela per meter^2 (cd/m^2)	1/π* E+04
lambert	candela per meter^2 (cd/m^2)	3.183 099 E+03
langley	joule per meter^2 (J/m^2)	4.184* E+04
league	meter (m)	(see Footnote 1)
light year	meter (m)	9.460 55 E+15
liter[(5)]	meter^3 (m^3)	1.0* E−03
maxwell	weber (Wb)	1.0* E−08
mho	siemens (S)	1.0* E+00
microinch	meter (m)	2.54* E−08
microsecond/foot (μs/ft)	microsecond/meter (μs/m)	3.280 840 E+00
micron	meter (m)	1.0* E−06
mil	meter (m)	2.54* E−05
mile (international)	meter (m)	1.609 344* E+03
mile (statute)	meter (m)	1.609 3 E+03
mile (U.S. survey)[(1)]	meter (m)	1.609 347 E+03
mile (international nautical)	meter (m)	1.852* E+03

mile (U.K. nautical)	meter (m)	1.853 184*	E+03
mile (U.S. nautical)	meter (m)	1.852*	E+03
sq mile (international)	meter^2 (m^2)	2.589 988	E+06
sq mile (U.S. survey)[1]	meter^2 (m^2)	2.589 998	E+06
mile/hr (international)	meter per second (m/s)	4.470 4*	E−01
mile/hr (international)	kilometer per hour (km/h)	1.609 344*	E+00
mile/min (international)	meter per second (m/s)	2.682 24*	E+01
mile/s (international)	meter per second (m/s)	1.609 344*	E+03
millibar	pascal (Pa)	1.0*	E+02
millimeter of mercury (0°C)	pascal (Pa)	1.333 22	E+02
minute (angle)	radian (rad)	2.908 882	E−04
minute (mean solar)	second (s)	6.0*	E+01
minute (sidereal)	second (s)	5.983 617	E+01
month (mean calendar)	second (s)	2.628 000	E+06
oersted	ampere per meter (A/m)	7.957 747	E+01
ohm centimeter	ohm meter (Ω·m)	1.0*	E−02
ohm circular-mil per ft	ohm millimeter^2 per meter [(Ω·mm^2)m]	1.662 426	E−03
ounce (avoirdupois)	kilogram (kg)	2.834 952	E−02
ounce (troy or apothecary)	kilogram (kg)	3.110 348	E−02
ounce (U.K. fluid)	meter^3 (m^3)	2.841 307	E−05
ounce (U.S. fluid)	meter^3 (m^3)	2.957 353	E−05
ounce-force	newton (N)	2.780 139	E−01
ozf·in.	newton meter (N·m)	7.061 552	E−03
oz (avoirdupois)/gal (U.K. liquid)	kilogram per meter^3 (kg/m^3)	6.236 021	E+00
oz (avoirdupois)/gal (U.S. liquid)	kilogram per meter^3 (kg/m^3)	7.489 152	E+00
oz (avoirdupois)/in.^{3}	kilogram per meter^3 (kg/m^3)	1.729 994	E+03
oz (avoirdupois)/ft^2	kilogram per meter^2 (kg/m^2)	3.051 517	E−01
oz (avoirdupois)/yd^2	kilogram per meter^2 (kg/m^2)	3.390 575	E−02
parsec	meter (m)	3.085 678	E+16
peck (U.S.)	meter^3 (m^3)	8.809 768	E−03
pennyweight	kilogram (kg)	1.555 174	E−03
perm (°C)[6]	kilogram per pascal second meter^2 [kg/(Pa·s·m^2)]	5.721 35	E−11

[5]In 1964 the General Conference on Weights and Measures adopted the name liter as a special name for the cubic decimeter. Prior to this decision the liter differed slightly (previous value, 1.000 028 dm^3) and in expression of precision volume measurement this fact must be kept in mind.
[6]Not the same as reservoir "perm."

Table A-1
(continued)

To Convert From	To	Multiply By**
perm (23°C)[6]	kilogram per pascal second meter² [kg/(Pa·s·m²)]	5.745 25 E − 11
perm·in. (0°C)[7]	kilogram per pascal second meter [kg/(Pa·s·m)]	1.453 22 E − 12
perm·in. (23°C)[7]	kilogram per pascal second meter [km/(Pa·s·m)]	1.459 29 E − 12
phot	lumen per meter² (lm/m²)	1.0* E + 04
pica (printer's)	meter (m)	4.217 518 E − 03
pint (U.S. dry)	meter³ (m³)	5.506 105 E − 04
pint (U.S. liquid)	meter³ (m³)	4.731 765 E − 04
point (printer's)	meter (m)	3.514 598* E − 04
poise (absolute viscosity)	pascal second (Pa·s)	1.0* E − 01
pound (lbm avoirdupois)[8]	kilogram (kg)	4.535 924 E − 01
pound (troy or apothecary)	kilogram (kg)	3.732 417 E − 01
lbm-ft² (moment of inertia)	kilogram meter² (kg·m²)	4.214 011 E − 02
lbm-in.² (moment of inertia)	kilogram meter² (kg·m²)	2.926 397 E − 04
lbm/ft-hr	pascal second (Pa·s)	4.133 789 E − 04
lbm/ft-s	pascal second (Pa·s)	1.488 164 E + 00
lbm/ft²	kilogram per meter² (kg/m²)	4.882 428 E + 00
lbm/ft³	kilogram per meter³ (kg/m³)	1.601 846 E + 01
lbm/gal (U.K. liquid)	kilogram per meter³ (kg/m³)	9.977 633 E + 01
lbm/gal (U.S. liquid)	kilogram per meter³ (kg/m³)	1.198 264 E + 02
lbm/hr	kilogram per second (kg/s)	1.259 979 E − 04
lbm/(hp · hr) (SFC, specific fuel consumption)	kilogram per joule (kg/J)	1.689 659 E − 07
lbm/in.³	kilogram per meter³ (kg/m³)	2.767 990 E + 04
lbm/min	kilogram per second (kg/s)	7.559 873 E − 03
lbm/s	kilogram per second (kg/s)	4.535 924 E − 01
lbm/yd³	kilogram per meter³ (kg/m³)	5.932 764 E − 01
poundal	newton (N)	1.382 550 E − 01
poundal/ft²	pascal (Pa)	1.488 164 E + 00
poundal-s/ft²	pascal second (Pa·s)	1.488 164 E + 00
pound-force (lbf)[9]	newton (N)	4.448 222 E + 00

lbf-ft[10]	newton meter (N·m)	1.355 818 E+00
lbf-ft/in.[11]	newton meter per meter [(N·m)/m)]	5.337 866 E+01
lbf-in.[11]	newton meter (N·m)	1.129 848 E−01
lbf-in./in.[11]	newton meter per meter [(N·m)/m]	4.448 222 E+00
lbf-s/ft^2	pascal second (Pa·s)	4.788 026 E+01
lbf/ft	newton per meter (N/m)	1.459 390 E+01
lbf/ft^2	pascal (Pa)	4.788 026 E+01
lbf/in.	newton per meter (N/m)	1.751 268 E+02
lbf/in.^{2} (psi)	pascal (Pa)	6.894 757 E+03
lbf/lbm (thrust/weight [mass] ratio)	newton per kilogram (N/kg)	9.806 650 E+00
quart (U.S. dry)	meter^3 (m^3)	1.101 221 E−03
quart (U.S. liquid)	meter^3 (m^3)	9.463 529 E−04
rad (radiation dose absorbed)	gray (Gy)	1.0* E−02
rhe	1 per pascal second [1/(Pa·s)]	1.0* E+01
rod	meter (m)	(see Footnote 1)
roentgen	coulomb per kilogram (C/kg)	2.58 E−04
second (angle)	radian (rad)	4.848 137 E−06
second (sidereal)	second (s)	9.972 696 E−01
section	meter^2 (m^2)	(see Footnote 1)
shake	second (s)	1.000 000* E−08
slug	kilogram (kg)	1.459 390 E+01
slug/(ft-s)	pascal second (Pa·s)	4.788 026 E+01
slug/ft^3	kilogram per meter^3 (kg/m^3)	5.153 788 E+02
statampere	ampere (A)	3.335 640 E−10
statcoulomb	coulomb (C)	3.335 640 E−10
statfarad	farad (F)	1.112 650 E−12
stathenry	henry (H)	8.987 554 E+11
statmho	siemens (S)	1.112 650 E−12
statohm	ohm (Ω)	8.987 554 E+11
statvolt	volt (V)	2.997 925 E+02
stere	meter^3 (m^3)	1.0* E+00

[7] Not the same dimensions as "millidarcy-foot."
[8] The exact conversion factor is 4.535 923 7*E−01.
[9] The exact conversion factor is 4.448 221 615 260 5*E+00.
[10] Torque unit; see text discussion of "Torque and Bending Moment."
[11] Torque divided by length; see text discussion of "Torque and Bending Moment."

Table A-1
(continued)

To Convert From	To	Multiply By**	
stilb	candela per meter^2 (cd/m^2)	1.0*	E+04
stokes (kinematic viscosity)	meter^2 per second (m^2/s)	1.0*	E−04
tablespoon	meter^3 (m^3)	1.478 676	E−05
teaspoon	meter^3 (m^3)	4.928 922	E−06
tex	kilogram per meter (kg/m)	1.0*	E−06
therm	joule (J)	1.055 056	E+08
ton (assay)	kilogram (kg)	2.916 667	E−02
ton (long, 2,240 lbm)	kilogram (kg)	1.016 047	E+03
ton (metric)	kilogram (kg)	1.0*	E+03
ton (nuclear equivalent of TNT)	joule (J)	4.184	E+09[12]
ton (refrigeration)	watt (W)	3.516 800	E+03
ton (register)	meter^3 (m^3)	2.831 685	E+00
ton (short, 2000 lbm)	kilogram (kg)	9.071 847	E+02
ton (long)/yd^3	kilogram per meter^3 (kg/m^3)	1.328 939	E+03
ton (short)/hr	kilogram per second (kg/s)	2.519 958	E−01
ton-force (2000 lbf)	newton (N)	8.896 444	E+03
tonne	kilogram (kg)	1.0*	E+03
torr (mm Hg, 0°C)	pascal (Pa)	1.333 22	E+02
township	meter^2 (m^2)	(see Footnote 1)	
unit pole	weber (Wb)	1.256 637	E−07
watthour (W-hr)	joule (J)	3.60*	E+03
W-s	joule (J)	1.0*	E+00
W/cm^2	watt per meter^2 (W/m^2)	1.0*	E+04
W/in.^{2}	watt per meter^2 (W/m^2)	1.550 003	E+03
yard	meter (m)	9.144*	E−01
yd^2	meter^2 (m^2)	8.361 274	E−01
yd^3	meter^3 (m^3)	7.645 549	E−01
yd^3/min	meter^3 per second (m^3/s)	1.274 258	E−02
year (calendar)	second (s)	3.153 600	E+07
year (sidereal)	second (s)	3.155 815	E+07
year (tropical)	second (s)	3.155 693	E+07

[12]Defined (not measured) value.

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Table A-2
Conversion Factors for the Vara*

Location	Value of Vara in Inches	Conversion Factor, Varas to Meters	
Argentina, Paraguay	34.12	8.666	E−01
Cadiz, Chile, Peru	33.37	8.476	E−01
California,			
except San Francisco	33.3720	8.476 49	E−01
San Francisco	33.0	8.38	E−01
Central America	33.87	8.603	E−01
Colombia	31.5	8.00	E−01
Honduras	33.0	8.38	E−01
Mexico		8.380	E−01
Portugal, Brazil	43.0	1.09	E+00
Spain, Cuba, Venezuela, Philippine Islands	33.38**	8.479	E−01
Texas			
Jan. 26, 1801, to Jan. 27, 1838	32.8748	8.350 20	E−01
Jan. 27, 1838 to June 17, 1919, for surveys of state land made for Land Office	33-1/3	8.466 667	E−01
Jan. 27, 1838 to June 17, 1919, on private surveys (unless changed to 33-1/3 in. by custom arising to dignity of law and overcoming former law)	32.8748	8.350 20	E−01
June 17, 1919, to present	33-1/3	8.466 667	E−01

* McElwee, P. G., *The Texas Vara*, available from commissioner, General Land Office, State of Texas, Auston, Texas (April 30, 1940).

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Table A-3
"Memory Jogger"—Metric Units

Customary Unit	"BallPark" Metric Values; (Do *Not* Use As Conversion Factors)	
acre	4000	square meters
	0.4	hectare
barrel	0.16	cubic meter
British thermal unit	1000	joules
British thermal unit per pound-mass	2300	joules per kilogram
	2.3	kilojoules per kilogram
calorie	4	joules
centipoise	1*	millipascal-second
centistokes	1*	square millimeter per second
darcy	1	square micrometer
degree Fahrenheit (temperature *difference*)	0.5	kelvin
dyne per centimeter	1*	millinewton per meter
foot	30	centimeters
	0.3	meter
cubic foot (cu ft)	0.03	cubic meter
cubic foot per pound-mass (ft^3/lbm)	0.06	cubic meter per kilogram
square foot (sq ft)	0.1	square meter
foot per minute	0.3	meter per minute
	5	millimeters per second
foot-pound-force	1.4	joules
foot-pound-force per minute	0.02	watt
foot-pound-force per second	1.4	watts
horsepower	750	watts (¾ kilowatt)
horsepower, boiler	10	kilowatts
inch	2.5	centimeters
kilowatthour	3.6*	megajoules
mile	1.6	kilometers
ounce (avoirdupois)	28	grams
ounce (fluid)	30	cubic centimeters
pound-force	4.5	newtons
pound-force per square inch (pressure, psi)	7	kilopascals
pound-mass	0.5	kilogram
pound-mass per cubic foot	16	kilograms per cubic meter
	260	hectares
section	2.6	million square meters
	2.6	square kilometers
ton, long (2240 pounds-mass)	1000	kilograms
ton, metric (tonne)	1000*	kilograms
ton, short	900	kilograms

*Exact equivalents

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Index

I

J

K

S

T

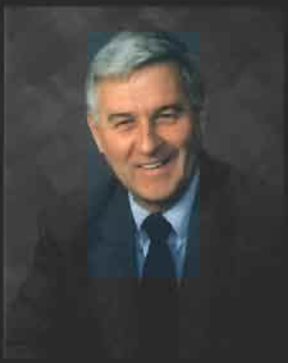
U

V

X

Y

Z



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CONTENTS

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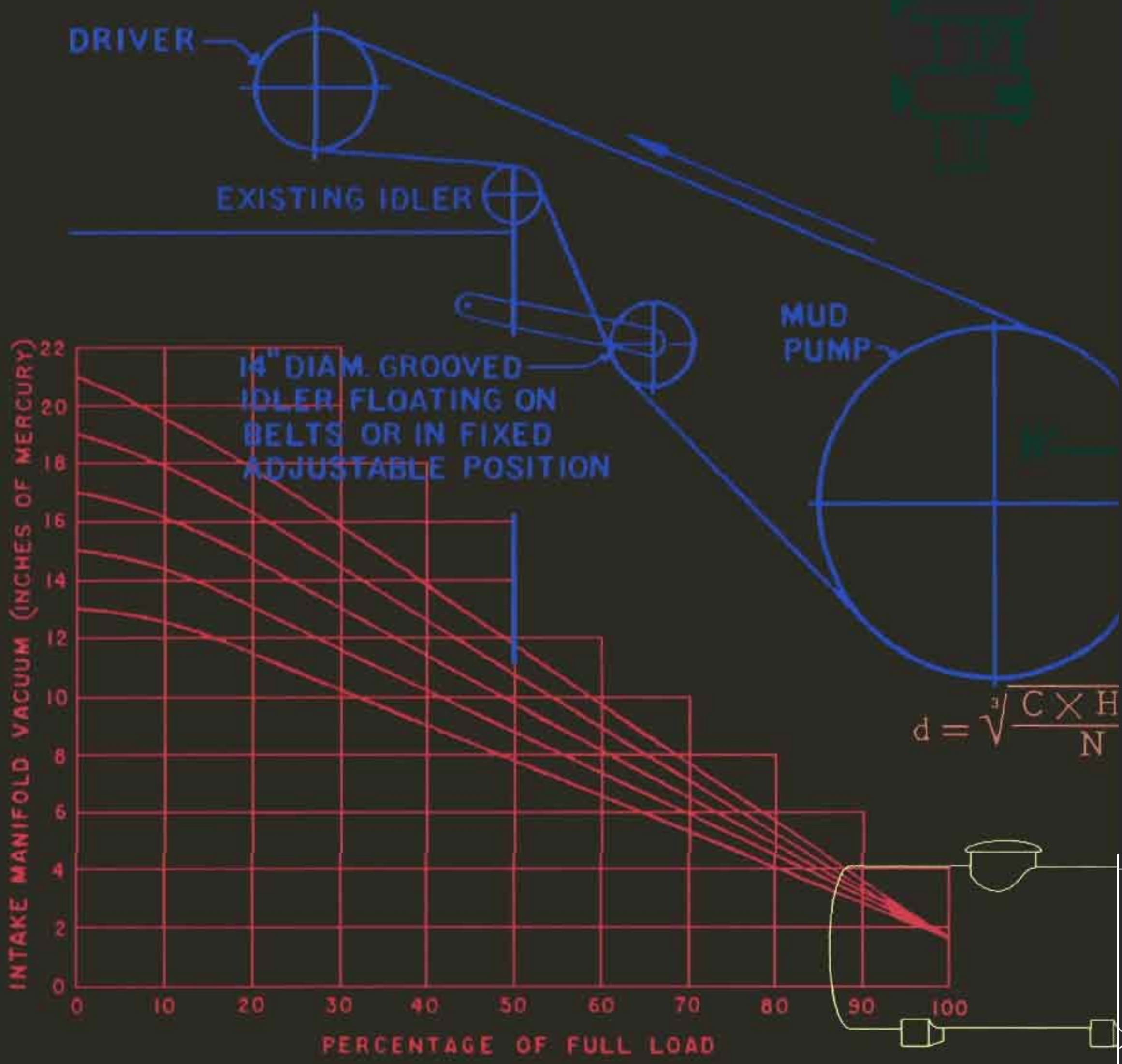

$$d = \sqrt[3]{\frac{C \times H}{N}}$$