Section 8
Machine Elements

BY

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Live Math

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8.1 MECHANISM

by Heard K. Baumeister, Amended by Staff

REFERENCES: Beggs, ''Mechanism,'' McGraw-Hill. Hrones and Nelson, ''Analysis of the Four Bag Linkage,'' Wiley. Jones, ''Ingenious Mechanisms for Designers and Inventors,'' 4 vols., Industrial Press. Moliam, ''The Design of Cam Mechanisms and Linkages,'' Elsevier. Chironis, ''Gear Design and Application,'' McGraw-Hill.

NOTE: The reader is referred to the current and near-past professional literature for extensive material on linkage mechanisms. The vast number of combinations thereof has led to the development of computer software programs to aid in the design of specific linkages.

Definition A **mechanism** is that part of a machine which contains two or more pieces so arranged that the motion of one compels the motion of the others, all in a fashion prescribed by the nature of the combination.

LINKAGES

Links may be of any form so long as they do not interfere with the desired motion. The simplest form is four bars *A, B, C,* and *D*, fastened together at their ends by cylindrical pins, and which are all movable in parallel planes. If the links are of different lengths and each is fixed in turn, there will be four possible combinations; but as two of these are similar there will be produced three mechanisms having distinctly different motions. Thus, in Fig. 8.1.1, if *D* is fixed *A* can rotate and *C* oscillate, giving the **beam-and-crank** mechanism, as used on side-wheel steamers. If *B* is fixed, the same motion will result; if *A* is fixed (Fig. 8.1.2), links *B* and *D* can rotate, giving the **drag-link** mechanism used to feather the floats on paddle wheels. Fixing link *C* (Fig. 8.1.3), *D* and *B* can only oscillate, and a **rocker** mechanism sometimes used in straight-line motions is produced. It is customary to call a rotating link a **crank;** an oscillating link a **lever,** or beam; and the connecting link a **connecting rod,** or **coupler**. Discrete points on the coupler, crank, or lever can be pressed into service to provide a desired motion. The fixed link is often enlarged and used as the supporting frame.

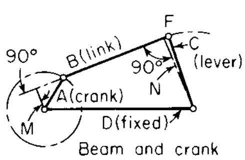


Fig. 8.1.1 Beam-and-crank mechanism.

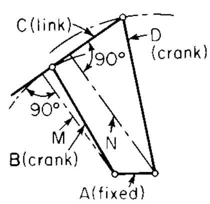


Fig. 8.1.2 Drag-link mechanism.

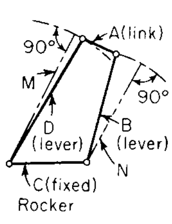
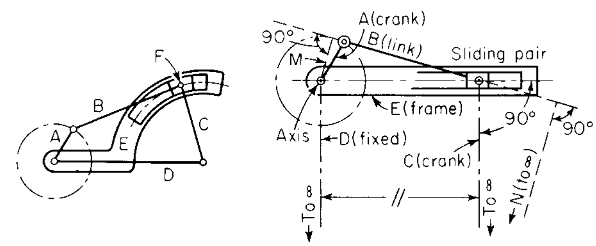


Fig. 8.1.3 Rocker mechanism.

If in the linkage (Fig. 8.1.1) the pin joint *F* is replaced by a slotted piece *E* (Fig. 8.1.4), no change will be produced in the resulting motion, and if the length of links *C* and *D* is made infinite, the slotted piece *E* will become straight and the motion of the slide will be that of pure translation, thus obtaining the engine, or **sliding-block, linkage** (Fig. 8.1.5).

If in the sliding-block linkage (Fig. 8.1.5) the long link *B* is fixed (Fig. 8.1.6), *A* will rotate and *E* will oscillate and the infinite links *C* and *D* may be indicated as shown. This gives the **swinging-block linkage.** When used as a quick-return motion the slotted piece and slide are usually interchanged (Fig. 8.1.7) which in no way changes the resulting motion. If the short link *A* is fixed (Fig. 8.1.8), *B* and *E* can both rotate,



Figs. 8.1.4 and 8.1.5 Sliding-block linkage.

and the mechanism known as the **turning-block linkage** is obtained. This is better known under the name of the **Whitworth quick-return motion,** and is generally constructed as in Fig. 8.1.9. The **ratio of time of advance to time of return** *H*/*K* of the two quick-return motions (Figs. 8.1.7 and 8.1.9) may be found by locating, in the case of the swinging block (Fig. 8.1.7), the two tangent points (*t*) and measuring the angles *H* and *K* made by the two positions of the crank *A*. If *H* and *K* are known, the axis of *E* may be located by laying off the angles *H* and *K* on the crank circle

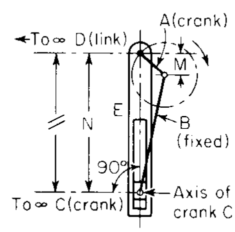


Fig. 8.1.6 Swinging-block linkage.

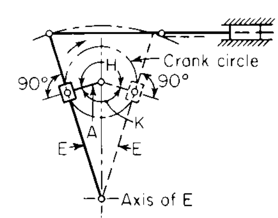


Fig. 8.1.7 Slow-advance, quick-return linkage.

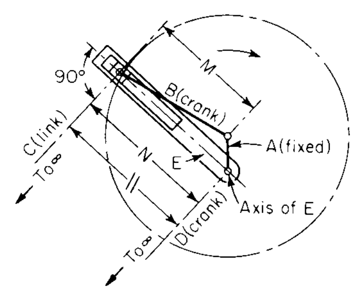


Fig. 8.1.8 Turning-block linkage.

and drawing the tangents *E*, their intersection giving the desired point. For the turning-block linkage (Fig. 8.1.9), determine the angles *H* and *K* made by the crank *B* when *E* is in the horizontal position; or, if the angles are known, the axis of *E* may be determined by drawing a horizontal line through the two crankpin positions (*S*) for the given angle, and the point where a line through the axis of *B* cuts *E* perpendicularly will be the axis of *E*.

Velocities of any two or more points on a link must fulfill the follow-

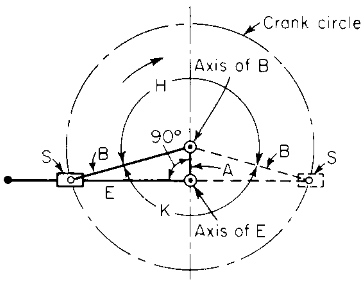


Fig. 8.1.9 Whitworth quick-return motion.

ing conditions (see Sec. 3). (1) Components along the link must be equal and in the same direction (Fig. 8.1.10): $V_a = V_b = V_c$. (2) Perpendiculars to V_A, V_B, V_C from the points *A, B, C* must intersect at a common point *d*, the **instant center** (or instantaneous axis). (3) The velocities of points *A, B,* and *C* are directly proportional to their distances from this center (Fig. 8.1.11): $V_A/a = V_B/b = V_C/c$. For a straight link the tips of

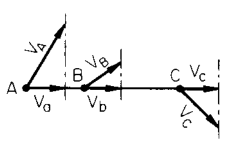


Fig. 8.1.10

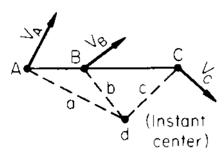


Fig. 8.1.11

the vectors representing the velocities of any number of points on the link will be on a straight line (Fig. 8.1.12); abc = a straight line. To find the velocity of any point when the velocity and direction of any two other points are known, condition 2 may be used, or a combination of conditions 1 and 3. The **linear velocity ratio** of any two points on a

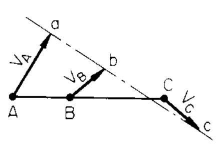


Fig. 8.1.12

linkage may be found by determining the distances *e* and *f* to the instant center (Fig. 8.1.13); then $V_c/V_b = e/f$. This may often be simplified by noting that a line drawn parallel to *e* and cutting *B* forms two similar triangles *efB* and *sAy*, which gives $V_c/V_b = e/f = s/A$. The **angular velocity ratio** for any position of two oscillating or rotating links *A* and *C* (Fig. 8.1.1), connected by a movable link *B*, may be determined by

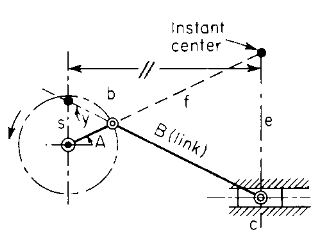


Fig. 8.1.13

scaling the length of the perpendiculars *M* and *N* from the axes of rotation to the centerline of the movable link. The angular velocity ratio is inversely proportional to these perpendiculars, or $O_C/O_A = M/N$. This method may be applied directly to a linkage having a sliding pair if the two infinite links are redrawn perpendicular to the sliding pair, as indicated in Fig. 8.1.14. *M* and *N* are shown also in Figs. 8.1.1, 8.1.2, 8.1.3, 8.1.5, 8.1.6, 8.1.8. In Fig. 8.1.5 one of the axes is at infinity; therefore, *N* is infinite, or the slide has pure translation.

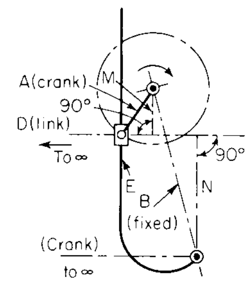


Fig. 8.1.14

Forces A mechanism must deliver as much work as it receives, neglecting friction; therefore, the force at any point *F* multiplied by the velocity V_F in the direction of the force at that point must equal the force at some other point *P* multiplied by the velocity V_P at that point; or the forces are inversely as their velocities and $F/P = V_P/V_F$. It is at times more convenient to equate the moments of the forces acting around each axis of rotation (sometimes using the instant center) to determine the force acting at some other point. In Fig. 8.1.15, $F \times a \times c/(b \times d) = P$.

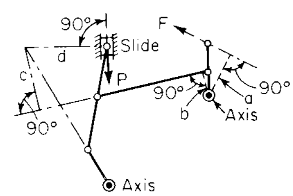


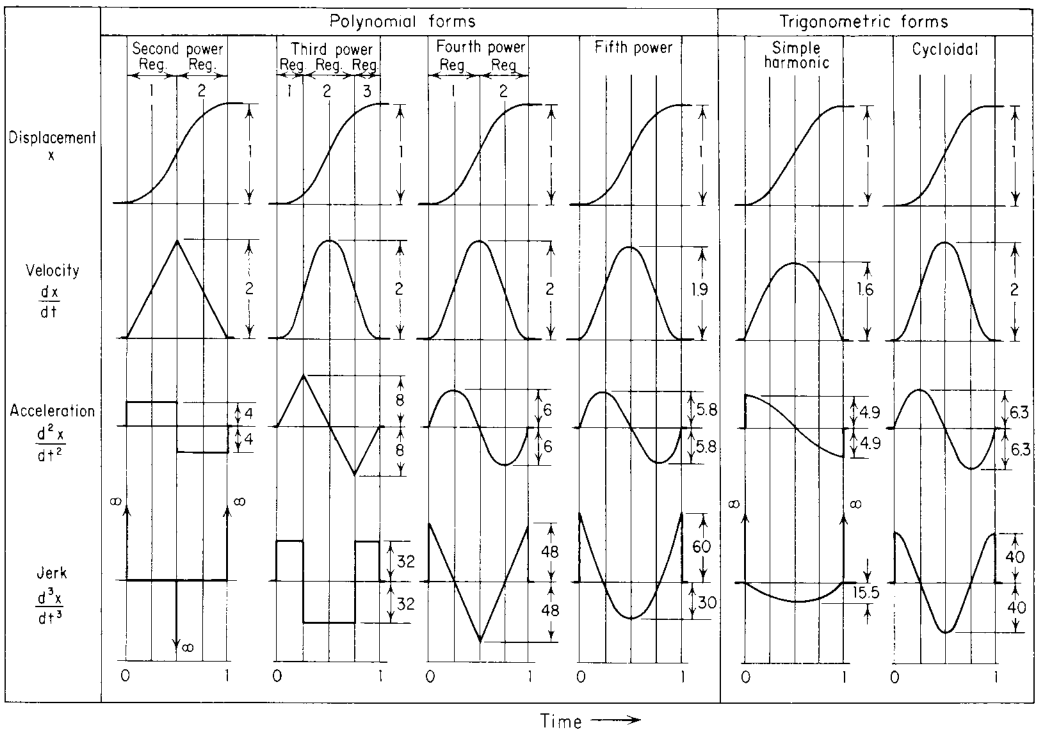
Fig. 8.1.15

CAMS

Cam Diagram A cam is usually a plate or cylinder which communicates motion to a follower as dictated by the geometry of its edge or of a groove cut in its surface. In the practical design of cams, the follower (1) must assume a definite series of positions while the driver occupies a corresponding series of positions or (2) must arrive at a definite location by the time the driver arrives at a particular position. The former design may be severely limited in speed because the interrelationship between the follower and cam positions may yield a follower displacement vs. time function that involves large values for the successive time derivatives, indicating large accelerations and forces, with concomitant large impacts and accompanying noise. The second design centers about finding that particular interrelationship between the follower and cam positions that results in the minimum forces and impacts so that the speed may be made quite large. In either case, the desired interrelationship must be put into hardware as discussed below. In the case of high-speed machines, small irregularities in the cam surface or geometry may be severely detrimental.

A stepwise displacement in time for the follower running on a cam driven at constant speed is, of course, impossible because the follower would require infinite velocities. A step in velocity for the follower would result in infinite accelerations; these in turn would bring into being forces that approach infinite magnitudes which would tend to destroy the machine. A step in acceleration causes a large jerk and large

Table 8.1.1 Displacement, Velocity, Acceleration, and Jerk for Some Cams



SOURCE: Adapted from Gutman, *Mach. Des.,* Mar. 1951.

shock waves to be transmitted and reflected throughout the parts that generate noise and would tend to limit the life of the machine. A step in jerk, the third derivative of the follower displacement with respect to time, seems altogether acceptable. In those designs requiring or exhibiting clearance between the follower and cam (usually at the bottom of the stroke), as gentle and slow a ramp portion as can be tolerated must be inserted on either side of the clearance region to limit the magnitude of the acceleration and jerk to a minimum. The tolerance on the clearance adjustment must be small enough to assure that the follower will be left behind and picked up gradually by the gentle ramp portions of the cam.

Table 8.1.1 shows the comparable and relative magnitudes of velocity, acceleration, and jerk for several high-speed cam, where the displacements are all taken as 1 at time 1 without any overshoot in any of the derivatives.

The three most common forms of motion used are uniform motion (Fig. 8.1.16), harmonic motion (Fig. 8.1.17), and uniformly accelerated and retarded motion (Fig. 8.1.18). In plotting the diagrams (Fig. 8.1.18) for this last motion, divide *ac* into an even number of equal parts and *bc* into the same number of parts with lengths increasing by a constant increment to a maximum and then decreasing by the same decrement, as, for example, 1, 3, 5, 5, 3, 1, or 1, 3, 5, 7, 9, 9, 7, 5, 3, 1. In order to prevent shock when the direction of motion changes, as at *a* and *b* in the uniform motion, the harmonic motion may be used; if the cam is to be operated at high speed, the uniformly accelerated and retarded motion should preferably be employed; in either case there is a very gradual change of velocity.

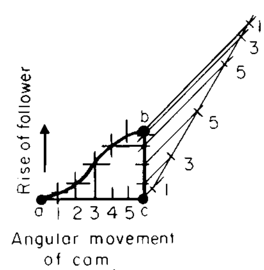


Fig. 8.1.18

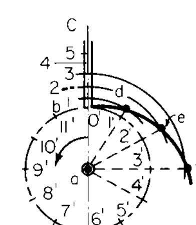


Fig. 8.1.19

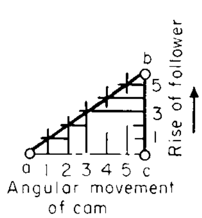


Fig. 8.1.16

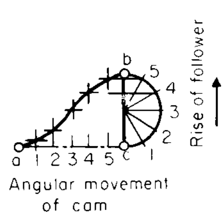


Fig. 8.1.17

Pitch Line The actual pitch line of a cam varies with the type of motion and with the position of the follower relative to the cam's axis. Most cams as ordinarily constructed are covered by the following four cases.

FOLLOWER ON LINE OF AXIS. (Fig. 8.1.19). To draw the pitch line, subdivide the motion *bc* of the follower in the manner indicated in Figs. 8.1.16, 8.1.17, and 8.1.18. Draw a circle with a radius equal to the smallest radius of the cam *a*0 and subdivide it into angles 0*a*1′, 0*a*2′,

$0a3'$, etc., corresponding with angular displacements of the cam for positions 1, 2, 3, etc., of the follower. With *a* as a center and radii *a*1, *a*2, *a*3, etc., strike arcs cutting radial lines at *d, e, f,* etc. Draw a smooth curve through points *d, e, f,* etc.

OFFSET FOLLOWER (Fig. 8.1.20). Divide *bc* as indicated in Figs. 8.1.16, 8.1.17, and 8.1.18. Draw a circle of radius *ac* (highest point of rise of follower) and one tangent to *cb* produced. Divide the outer circle into parts $1'$, $2'$, $3'$, etc., corresponding with the angular displacement of

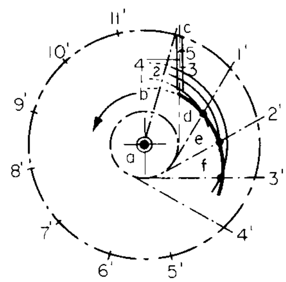


Fig. 8.1.20

the cam for positions 1, 2, 3, etc., of the follower, and draw tangents from points $1'$, $2'$, $3'$, etc., to the small circle. With *a* as a center and radii *a*1, *a*2, *a*3, etc., strike arcs cutting tangents at *d, e, f,* etc. Draw a smooth curve through *d, e, f,* etc.

ROCKER FOLLOWER (Fig. 8.1.21). Divide the stroke of the slide *S* in the manner indicated in Figs. 8.1.16, 8.1.17, and 8.1.18, and transfer these points to the arc *bc* as points 1, 2, 3, etc. Draw a circle of radius *ak* and divide it into parts $1'$, $2'$, $3'$, etc., corresponding with angular dis-

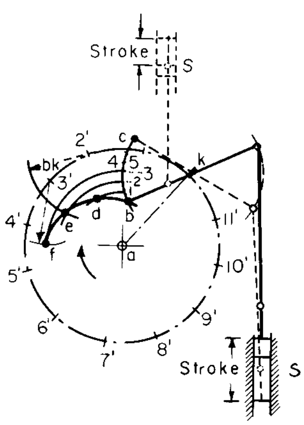


Fig. 8.1.21

placements of the cam for positions 1, 2, 3, etc., of the follower. With *k*, $1'$, $2'$, $3'$, etc., as centers and radius *bk*, strike arcs *kb*, $1'd$, $2'e$, $3'f$, etc., cutting at *bdef* arcs struck with *a* as a center and radii *ab, a*1, *a*2, *a*3, etc. Draw a smooth curve through *b, d, e, f,* etc.

CYLINDRICAL CAM (Fig. 8.1.22). In this type of cam, more than one complete turn may be obtained, provided in all cases the follower returns to its starting point. Draw rectangle *wxyz* (Fig. 8.1.22) representing the development of cylindrical surface of the cam. Subdivide the desired motion of the follower *bc* horizontally in the manner indicated in Figs. 8.1.16, 8.1.17, and 8.1.18, and plot the corresponding angular displacement $1'$, $2'$, $3'$, etc., of the cam vertically; then through the intersection of lines from these points draw a smooth curve. This may best be shown by an example, assuming the following data for the diagram in Fig. 8.1.22: Total motion of follower = *bc*; circumference of cam = $2\pi r$. Follower moves harmonically 4 units to right in 0.6 turn, then rests (or ''dwells'') 0.4 turn, and finishes with uniform motion 6 units to right and 10 units to left in 2 turns.

Cam Design In the practical design of cams the following points must be noted. If only a small force is to be transmitted, sliding contact may be used, otherwise **rolling contact**. For the latter the pitch line must

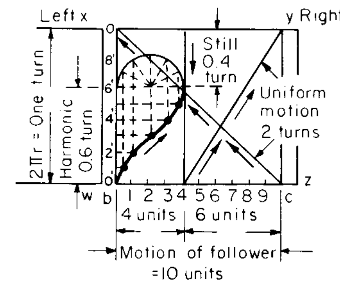


Fig. 8.1.22 Cylindrical cam.

be corrected in order to get the true slope of the cam. An approximate construction (Fig. 8.1.23) may be employed by using the pitch line as the center of a series of arcs the radii of which are equal to that of the follower roll to be used; then a smooth curve drawn tangent to the arcs will give the slope desired for a roll working on the periphery of the cam

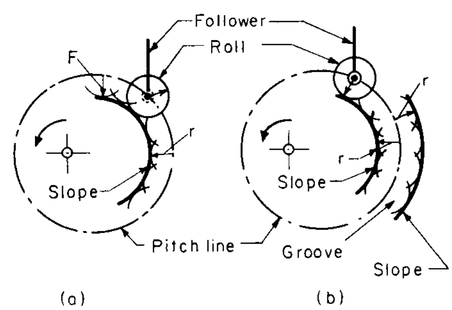


Fig. 8.1.23

(Fig. 8.1.23*a*) or in a groove (Fig. 8.1.23*b*). For plate cams the roll should be a small cylinder, as in Fig. 8.1.24*a*. In cylindrical cams it is usually sufficiently accurate to make the roll conical, as in Fig. 8.1.24*b*, in which case the taper of the roll produced should intersect the axis of the cam. If the pitch line *abc* is made too sharp (Fig. 8.1.25) the follower

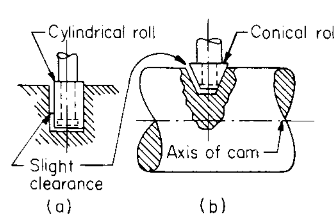


Fig. 8.1.24 Plate cam.

will not rise the full amount. In order to prevent this **loss of rise,** the pitch line should have a radius of curvature at all parts of not less than the roll's diameter plus ⅛ in. For the same rise of follower, *a*, the angular motion of the cam, *O*, the slope of the cam changes considerably, as indicated by the heavy lines *A, B,* and *C* (Fig. 8.1.26). Care should be taken to keep a moderate slope and thereby keep down the side thrust on the follower, but this should not be carried too far, as the cam would become too large and the friction increase.

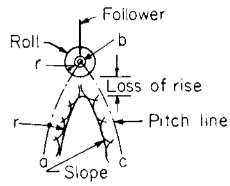


Fig. 8.1.25

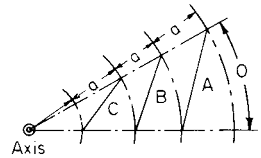


Fig. 8.1.26

ROLLING SURFACES

In order to connect two shafts so that they shall have a definite angular velocity ratio, rolling surfaces are often used; and in order to have no slipping between the surfaces they must fulfill the following two conditions: the line of centers must pass through the point of contact, and the arcs of contact must be of equal length. The angular velocities, expressed usually in r/min, will be inversely proportional to the radii: $N/n = r/R$. The two surfaces most commonly used in practice, and the only ones having a constant angular velocity ratio, are cylinders where the shafts are parallel, and cones where the shafts (projected) intersect at an angle. In either case there are two possible directions of rotation, depending upon whether the surfaces roll in opposite directions (external contact) or in the same direction (internal contact). In Fig. 8.1.27, $R = nc/(N + n)$ and $r = Nc/(N + n)$; in Fig. 8.1.28, $R = nc/(N - n)$ and $r = Nc/(N - n)$. In Fig. 8.1.29, $\tan B = \sin A/(n/N + \cos A)$ and $\tan C = \sin A/(N/n + \cos A)$; in Fig. 8.1.30, $\tan B = \sin A/(N/n - \cos A)$, and $\tan C = \sin A/(n/N - \cos A)$. With the above values for the angles B and C, and the length d or e of one of the cones, R and r may be calculated.

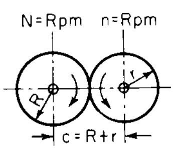


Fig. 8.1.27

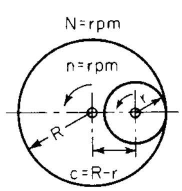


Fig. 8.1.28

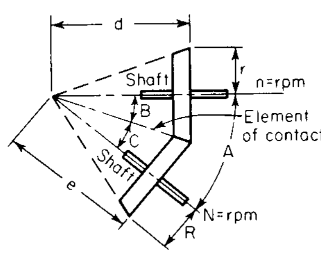


Fig. 8.1.29

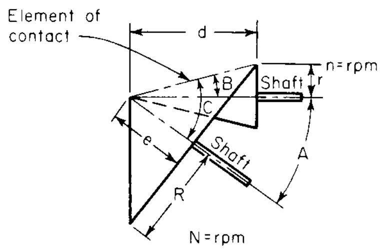


Fig. 8.1.30

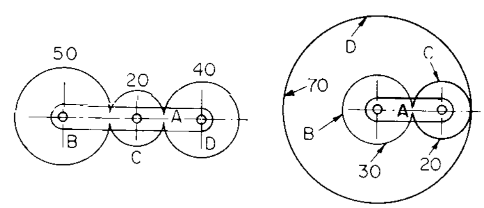
The natural limitations of **rolling without slip,** with the use of pure rolling surfaces limited to the transmission of very small amounts of torque, led historically to the alteration of the geometric surfaces to include teeth and tooth spaces, i.e., **toothed wheels,** or simply **gears**. Modern gear tooth systems are described in greater detail in Sec. 8.3. This brief discussion is limited to the kinematic considerations of some common gear combinations.

EPICYCLIC TRAINS

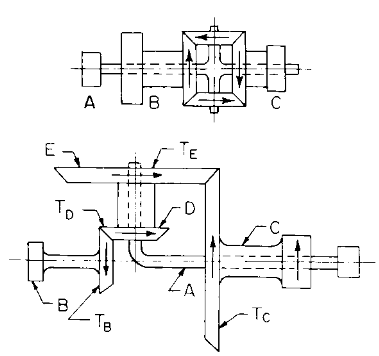
Epicyclic trains are combinations of gears in which some of or all the gears have a motion compounded of rotation about an axis and a translation or revolution of that axis. The gears are usually connected by a link called an arm, which often rotates about the axis of the first gear. Such trains may be calculated by first considering all gears locked and the arm turned; then the arm locked and the gears rotated. The algebraic sum of the separate motions will give the desired result. The following examples and method of tabulation will illustrate this. The figures on each gear refer to the number of teeth for that gear.

	A	*B*	*C*	*D*
Gear locked, Fig. 8.1.31	$+1$	$+1$	$+1$	$+1$
Arm locked, Fig. 8.1.31	0	-1	$+1 \times 50/20$	$-1 \times 50/20 \times 20/40$
Addition, Fig. 8.1.31	$+1$	0	$+3\frac{1}{2}$	$-\frac{1}{4}$
Gears locked, Fig. 8.1.32	$+1$	$+1$	$+1$	$+1$
Arm locked, Fig. 8.1.32	0	-1	$+1 \times 30/20$	$+1 \times 30/20 \times 20/70$
Addition, Fig. 8.1.32	$+1$	0	$+2\frac{1}{2}$	$+1\frac{3}{7}$

In Figs. 8.1.31 and 8.1.32 lock the gears and turn the arm *A* right-handed through 1 revolution (+ 1); then lock the arm and turn the gear *B* back to where it started (− 1); gears *C* and *D* will have rotated the amount indicated in the tabulation. Then the algebraic sum will give the relative turns of each gear. That is, in Fig. 8.1.31, for one turn of the arm, *B* does not move and *C* turns in the same direction $3\frac{1}{2}$ r, and *D* in the opposite direction $\frac{1}{4}$ r; whereas in Fig. 8.1.32, for one turn of the arm, *B* does not turn, but *C* and *D* turn in the same direction as the arm, respectively, $2\frac{1}{2}$ and $1\frac{3}{7}$ r. (Note: The arm in the above case was turned + 1 for convenience, but any other value might be used.)



Figs. 8.1.31 and 8.1.32 Epicyclic trains.



Figs. 8.1.33 and 8.1.34 Bevel epicyclic trains.

Bevel epicyclic trains are epicyclic trains containing bevel gears and may be calculated by the preceding method, but it is usually simpler to use the general formula which applies to all cases of epicyclic trains:

$$\frac{\text{Turns of } C \text{ relative to arm}}{\text{Turns of } B \text{ relative to arm}} = \frac{\text{absolute turns of } C - \text{turns of arm}}{\text{absolute turns of } B - \text{turns of arm}}$$

The left-hand term gives the value of the train and can always be expressed in terms of the number of teeth (T) on the gears. Care must be used, however, to express it as either plus (+) or minus (−), depending upon whether the gears turn in the same or opposite directions.

$$\frac{\text{Relative turns of } C}{\text{Relative turns of } B} = \frac{C - A}{B - A} = -1 \qquad \text{(in Fig. 8.1.33)}$$

$$= +\frac{T_E}{T_C} \times \frac{T_B}{T_D} \qquad \text{(Fig. 8.1.34)}$$

HOISTING MECHANISMS

Pulley Block (Fig. 8.1.35) Given the weight W to be raised, the force F necessary is $F = V_W W/V_F = W/n$ = load/number of ropes, V_W and V_F being the respective velocities of W and F.

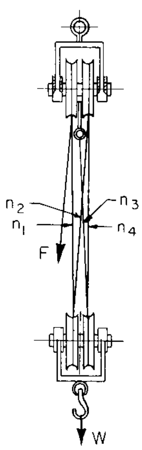


Fig. 8.1.35 Pulley block.

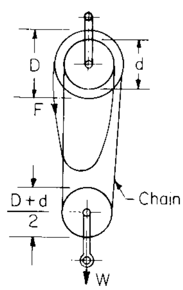


Fig. 8.1.36 Differential chain block.

Differential Chain Block (Fig. 8.1.36)

$$F = V_W W/V_F = W(D - d)/(2D)$$

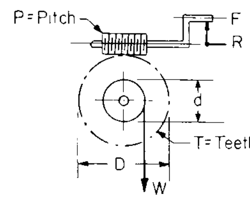


Fig. 8.1.37 Worm and worm wheel.

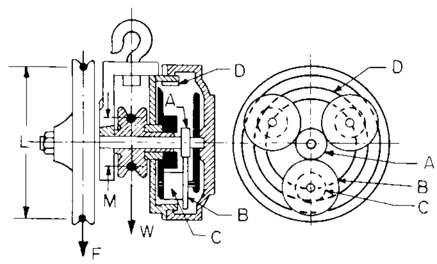


Fig. 8.1.38 Triplex chain block.

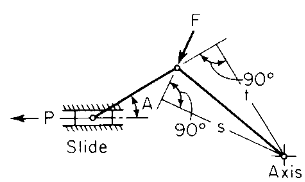


Fig. 8.1.39 Toggle joint.

Worm and Wheel (Fig. 8.1.37) $F = \pi d(n/T)W/(2\pi R) = WP(d/D)/(2\pi R)$, where n = number of threads, single, double, triple, etc.

Triplex Chain Block (Fig. 8.1.38) This geared hoist makes use of the epicyclic train. $W = FL/\{M[1 + (T_D/T_C) \times (T_B/T_A)]\}$, where T = number of teeth on gears.

Toggle Joint (Fig. 8.1.39) $P = Fs(\cos A)/t$.

8.2 MACHINE ELEMENTS

by Antonio F. Baldo

REFERENCES: American National Standards Institute (ANSI) Standards. International Organization for Standardization (ISO) Standards. Morden, ''Industrial Fasteners Handbook,'' Trade and Technical Press. Parmley, ''Standard Handbook of Fastening and Joining,'' McGraw-Hill. Bickford, ''An Introduction to the Design and Behavior of Bolted Joints,'' Marcel Dekker. Maleev, ''Machine Design,'' International Textbook. Shigley, ''Mechanical Engineering Design,'' McGraw-Hill. *Machine Design* magazine, Penton/IPC. ANSI/Rubber Manufacturers Assn. (ANSI/RMA) Standards. ''Handbook of Power Transmission Flat Belting,'' Goodyear Rubber Products Co. ''Industrial V-Belting,'' Goodyear Rubber Products Co. Carlson, ''Spring Designer's Handbook,'' Marcel Dekker. American Chain Assn., ''Chains for Power Transmission and Material Handling—Design and Applications Handbook,'' Marcel Dekker. ''Power Transmission Handbook,'' DAYCO. ''Wire Rope User's Manual,'' American Iron and Steel Institute. Blake, ''Threaded Fasteners—Materials and Design,'' Marcel Dekker.

NOTE. At this writing, conversion to metric hardware and machine elements continues. SI units are introduced as appropriate, but the bulk of the material is still presented in the form in which the designer or reader will find it available.

SCREW FASTENINGS

At present there exist two major standards for screw threads, namely Unified inch screw threads and metric screw threads. Both systems enjoy a wide application globally, but movement toward a greater use of the metric system continues.

Unified Inch Screw Threads (or Unified Screw Threads)

The Unified Thread Standard originated by an accord of screw thread standardization committees of Canada, the United Kingdom, and the United States in 1984. The Unified Screw-Thread Standard was published by ANSI as American Unified and American Screw Thread Publication B1.1-1974, revised in 1982 and then again in 1989. Revisions did not tamper with the basic 1974 thread forms. In conjunction with Technical Committee No. 1 of the ISO, the Unified Standard was adopted as an ISO Inch Screw Standard (ISO 5864-1978).

Of the numerous and different screw thread forms, those of greatest consequence are

UN—unified (no mandatory radiused root)
UNR—unified (mandatory radiused root; minimum 0.108 = p)
UNJ—unified (mandatory larger radiused root; recommended 0.150 = p)
M—metric (inherently designed and manufactured with radiused root; has 0.125 = p)
MJ—metric (mandatory larger radiused root; recommended 0.150 = p)

The basic American screw thread profile was standardized in 1974, and it now carries the UN designations (UN = unified). ANSI publishes these standards and all subsequent revisions. At intervals these standards are published with a ''reaffirmation date'' (that is, R1988). In 1969 an *international* basic thread profile standard was established, and it is designated as M. The ISO publishes these standards with yearly updates. The UN and M profiles are the same, but **UN screws** are manufactured to **inch** dimensions while **M screws** are manufactured to **metric** dimensions.

The **metric system** has only the two thread forms: **M**, standard for commercial uses, and **MJ**, standard for aerospace use and for aerospace-quality commercial use.

Certain groups of diameter and pitch combinations have evolved over time to become those most used commercially. Such groups are called **thread series.** Currently there are 11 UN series for inch products and 13 M series for metric products.

The Unified standard comprises the following two parts:

1. **Diameter-pitch combinations.** (See Tables 8.2.1 to 8.2.5.)
   a. UN inch series:

Coarse	UNC or UNRC
Fine	UNF or UNRF
Extra-fine	UNEF or UNREF
Constant-pitch	UN or UNR

   b. Metric series:

Coarse	M
Fine	M

   NOTE: Radiused roots apply only to external threads. The preponderance of important commercial use leans to UNC, UNF, 8UN (eight-threaded), and metric coarse M. Aerospace and aerospace-quality applications use UNJ and MJ.

2. **Tolerance classes.** The amounts of tolerance and allowance distinguish one thread class from another. Classes are designated by one of three numbers (1, 2, 3), and either letter A for external threads or letter B for internal threads. Tolerance decreases as class number increases. Allowance is specified only for classes 1A and 2A. Tolerances are based on engagement length equal to nominal diameter. 1A/1B—liberal tolerance and allowance required to permit easy assembly even with dirty or nicked threads. 2A/2B—most commonly used for general applications, including production of bolts, screws, nuts, and similar threaded fasteners. Permits external threads to be plated. 3A/3B—for closeness of fit and/or accuracy of thread applications where zero allowance is needed. 2AG—allowance for rapid assembly where high-temperature expansion prevails or where lubrication problems are important.

   Unified screw threads are designated by a set of numbers and letter symbols signifying, in sequence, the nominal size, threads per inch, thread series, tolerance class, hand (only for left hand), and in some instances in parentheses a Thread Acceptability System Requirement of ANSI B1.3.

   EXAMPLE. ¼-20 UNC-2A-LH (21), or optionally 0.250-20 UNC-2A-LH (21), where ¼ = nominal size (fractional diameter, in, or screw number, with decimal equivalent of either being optional); 20 = number of threads per inch, n; UNC = thread form and series; 2A = tolerance class; LH = left hand (no symbol required for right hand); (21) = thread gaging system per ANSI B1.3.

3. **Load considerations**
   a. *Static loading.* Only a slight increase in tensile strength in a screw fastener is realized with an increase in *root* rounding radius, because minor diameter (hence cross-sectional area at the root) growth is small. Thus the basic tensile stress area formula is used in stress calculations for all thread forms. See Tables 8.2.2, 8.2.3, and 8.2.4. The designer should take into account such factors as stress concentration as applicable.
   b. *Dynamic loading.* Few mechanical joints can remain absolutely free of some form of fluctuating stress, vibration, stress reversal, or impact. Metal-to-metal joints of very high-modulus materials or non-elastic-gasketed high-modulus joints plus preloading at assembly (preload to be greater than highest peak of the external fluctuating load) can realize absolute static conditions inside the screw fastener. For ordinary-modulus joints and elastic-gasketed joints, a fraction of the external fluctuating load will be transmitted to the interior of the screw fastener. Thus the fastener must be designed for fatigue according to a **static plus fluctuating load** model. See discussion under ''Strength'' later.

   Since **fatigue failures** generally occur at locations of high **stress concentration,** screw fasteners are especially vulnerable because of the abrupt change between head and body, notchlike conditions at the thread roots, surface scratches due to manufacturing, etc. The highest stress concentrations occur at the thread roots. The stress concentration factor can be very large for nonrounded roots, amounting to about 6 for sharp or flat roots, to less than 3 for UNJ and MJ threads which are generously rounded. This can effectively double the fatigue life. UNJ and MJ threads are especially well suited for dynamic loading conditions.

Screw Thread Profile

Basic Profile The basic profiles of UN and UNR are the same, and these in turn are identical to those of ISO metric threads. Basic thread shape (60° thread angle) and basic dimensions (major, pitch, and minor diameters; thread height; crest, and root flats) are defined. See Fig. 8.2.1.

Design Profile Design profiles define the maximum material (no allowance) for external and internal threads, and they are derived from the basic profile. UN threads (external) may have either flat or rounded crests and roots. UNR threads (external) must have rounded roots, but may have flat or rounded crests. UN threads (internal) *must* have rounded roots. Any rounding must clear the basic flat roots or crests.

Basic major diameter	Largest diameter of basic screw thread.
Basic minor diameter	Smallest diameter of basic screw thread.
Basic pitch diameter	Diameter to imaginary lines through thread profile and parallel to axis so that thread and groove widths are equal. These three definitions apply to both external and internal threads.
Maximum diameters (external threads)	Basic diameters minus allowance.
Minimum diameters (internal threads)	Basic diameters.
Pitch	$1/n$ (n = number of threads per inch).
Tolerance	Inward variation tolerated on maximum diameters of external threads and outward variation tolerated on minimum diameters of internal threads.

Metric Screw Threads

Metric screw thread standardization has been under the aegis of the International Organization for Standardization (ISO). The ISO basic profile is essentially the same as the Unified screw thread basic form,

Table 8.2.1 Standard Series Threads (UN/UNR)*

Nominal size, in		Basic major diameter, in	Threads per inch											Nominal size, in
			Series with graded pitches			Series with constant pitches								
Primary	Secondary		Coarse UNC	Fine UNF	Extra-fine UNEF	4UN	6UN	8UN	12UN	16UN	20UN	28UN	32UN	
0		0.0600	—	80	—	—	—	—	—	—	—	—	—	0
	1	0.0730	64	72	—	—	—	—	—	—	—	—	—	1
2		0.0860	56	64	—	—	—	—	—	—	—	—	—	2
	3	0.0990	48	56	—	—	—	—	—	—	—	—	—	3
4		0.1120	40	48	—	—	—	—	—	—	—	—	—	4
5		0.1250	40	44	—	—	—	—	—	—	—	—	—	5
6		0.1380	32	40	—	—	—	—	—	—	—	—	UNC	6
8		0.1640	32	36	—	—	—	—	—	—	—	—	UNC	8
10		0.1900	24	32	—	—	—	—	—	—	—	—	UNF	10
	12	0.2160	24	28	32	—	—	—	—	—	—	UNF	UNEF	12
1/4		0.2500	20	28	32	—	—	—	—	—	UNC	UNF	UNEF	1/4
5/16		0.3125	18	24	32	—	—	—	—	—	20	28	UNEF	5/16
3/8		0.3750	16	24	32	—	—	—	—	UNC	20	28	UNEF	3/8
7/16		0.4375	14	20	28	—	—	—	—	16	UNF	UNEF	32	7/16
1/2		0.5000	13	20	28	—	—	—	—	16	UNF	UNEF	32	1/2
9/16		0.5625	12	18	24	—	—	—	UNC	16	20	28	32	9/16
5/8		0.6250	11	18	24	—	—	—	12	16	20	28	32	5/8
	11/16	0.6875	—	—	24	—	—	—	12	16	20	28	32	11/16
3/4		0.7500	10	16	20	—	—	—	12	UNF	UNEF	28	32	3/4
	13/16	0.8125	—	—	20	—	—	—	12	16	UNEF	28	32	13/16
7/8		0.8750	9	14	20	—	—	—	12	16	UNEF	28	32	7/8
	15/16	0.9275	—	—	20	—	—	—	12	16	UNEF	28	32	15/16
1		1.0000	8	12	20	—	—	UNC	UNF	16	UNEF	28	32	1
	1 1/16	1.0625	—	—	18	—	—	8	12	16	20	28	—	1 1/16
1 1/8		1.1250	7	12	18	—	—	8	UNF	16	20	28	—	1 1/8
	1 3/16	1.1875	—	—	18	—	—	8	12	16	20	28	—	1 3/16
1 1/4		1.2500	7	12	18	—	—	8	UNF	16	20	28	—	1 1/4
	1 5/16	1.3125	—	—	18	—	—	8	12	16	20	28	—	1 5/16
1 3/8		1.3750	6	12	18	—	UNC	8	UNF	16	20	28	—	1 3/8
	1 7/16	1.4375	—	—	18	—	6	8	12	16	20	28	—	1 7/16
1 1/2		1.5000	6	12	18	—	UNC	8	UNF	16	20	28	—	1 1/2
1	1 9/16	1.5625	—	—	18	—	6	8	12	16	20	—	—	1 9/16
1 5/8		1.6250	—	—	18	—	6	8	12	16	20	—	—	1 5/8
	1 11/16	1.6875	—	—	18	—	6	8	12	16	20	—	—	1 11/16
1 3/4		1.7500	5	—	—	—	6	8	12	16	20	—	—	1 3/4
	1 13/16	1.8125	—	—	—	—	6	8	12	16	20	—	—	1 13/16
1 7/8		1.8750	—	—	—	—	6	8	12	16	20	—	—	1 7/8
	1 15/16	1.9375	—	—	—	—	6	8	12	16	20	—	—	1 15/16

2		2.0000	4½	—	—	—	6	8	12	16	20	—	—	2
	2⅛	2.1250	—	—	—	—	6	8	12	16	20	—	—	2⅛
2¼		2.2500	4½	—	—	—	6	8	12	16	20	—	—	2¼
	2⅜	2.3750	—	—	—	—	6	8	12	16	20	—	—	2⅜
2½		2.5000	4	—	—	UNC	6	8	12	16	20	—	—	2½
	2⅝	2.6250	—	—	—	4	6	8	12	16	20	—	—	2⅝
2¾		2.7500	4	—	—	UNC	6	8	12	16	20	—	—	2¾
	2⅞	2.8750	—	—	—	4	6	8	12	16	20	—	—	2⅞
3		3.0000	4	—	—	UNC	6	8	12	16	20	—	—	3
	3⅛	3.1250	—	—	—	4	6	8	12	16	—	—	—	3⅛
3¼		3.2500	4	—	—	UNC	6	8	12	16	—	—	—	3¼
	3⅜	3.3750	—	—	—	4	6	8	12	16	—	—	—	3⅜
3½		3.5000	4	—	—	UNC	6	8	12	16	—	—	—	3½
	3⅝	3.6250	—	—	—	4	6	8	12	16	—	—	—	3⅝
3¾		3.7500	4	—	—	UNC	6	8	12	16	—	—	—	3¾
	3⅞	3.8750	—	—	—	4	6	8	12	16	—	—	—	3⅞
4		4.0000	4	—	—	UNC	6	8	12	16	—	—	—	4
	4⅛	4.1250	—	—	—	4	6	8	12	16	—	—	—	4⅛
4¼		4.2500	—	—	—	4	6	8	12	16	—	—	—	4¼
	4⅜	4.3750	—	—	—	4	6	8	12	16	—	—	—	4⅜
4½		4.5000	—	—	—	4	6	8	12	16	—	—	—	4½
	4⅝	4.6250	—	—	—	4	6	8	12	16	—	—	—	4⅝
4¾		4.7500	—	—	—	4	6	8	12	16	—	—	—	4¾
	4⅞	4.8750	—	—	—	4	6	8	12	16	—	—	—	4⅞
5		5.0000	—	—	—	4	6	8	12	16	—	—	—	5
	5⅛	5.1250	—	—	—	4	6	8	12	16	—	—	—	5⅛
5¼		5.2500	—	—	—	4	6	8	12	16	—	—	—	5¼
	5⅜	5.3750	—	—	—	4	6	8	12	16	—	—	—	5⅜
5½		5.5000	—	—	—	4	6	8	12	16	—	—	—	5½
	5⅝	5.6250	—	—	—	4	6	8	12	16	—	—	—	5⅝
5¾		5.7500	—	—	—	4	6	8	12	16	—	—	—	5¾
	5⅞	5.8750	—	—	—	4	6	8	12	16	—	—	—	5⅞
6		6.0000	—	—	—	4	6	8	12	16	—	—	—	6

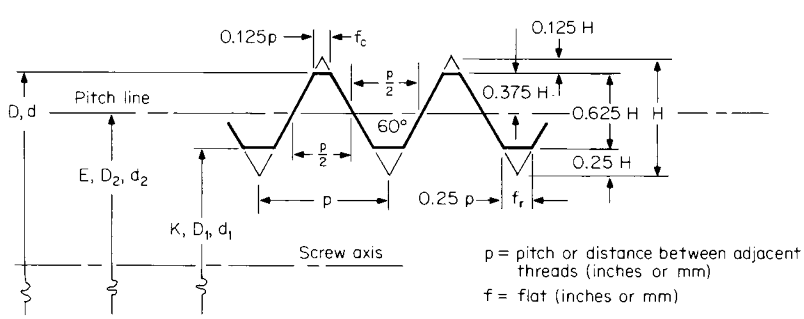
* Series designation shown indicates the UN thread form; however, the UNR thread form may be specified by substituting UNR in place of UN in all designations for external use only.

SOURCE: ANSI B1.1-1982; reaffirmed in 1989, reproduced by permission.

Table 8.2.2 Basic Dimensions for Coarse Thread Series (UNC/UNRC)

Nominal size, in	Basic major diameter D, in	Threads per inch n	Basic pitch diameter* E, in	UNR design minor diameter external† K_s, in	Basic minor diameter internal K, in	Section at minor diameter at $D - 2h_b$, in^2	Tensile stress area,‡ in^2
1 (0.073)§	0.0730	64	0.0629	0.0544	0.0561	0.00218	0.00263
2 (0.086)	0.0860	56	0.0744	0.0648	0.0667	0.00310	0.00370
3 (0.099)§	0.0990	48	0.0855	0.0741	0.0764	0.00406	0.00487
4 (0.112)	0.1120	40	0.0958	0.0822	0.0849	0.00496	0.00604
5 (0.125)	0.1250	40	0.1088	0.0952	0.0979	0.00672	0.00796
6 (0.138)	0.1380	32	0.1177	0.1008	0.1042	0.00745	0.00909
8 (0.164)	0.1640	32	0.1437	0.1268	0.1302	0.01196	0.0140
10 (0.190)	0.1900	24	0.1629	0.1404	0.1449	0.01450	0.0175
12 (0.216)§	0.2160	24	0.1889	0.1664	0.1709	0.0206	0.0242
¼	0.2500	20	0.2175	0.1905	0.1959	0.0269	0.0318
5⁄16	0.3125	18	0.2764	0.2464	0.2524	0.0454	0.0524
⅜	0.3750	16	0.3344	0.3005	0.3073	0.0678	0.0775
7⁄16	0.4375	14	0.3911	0.3525	0.3602	0.0933	0.1063
½	0.5000	13	0.4500	0.3334	0.4167	0.1257	0.1419
9⁄16	0.5625	12	0.5084	0.4633	0.4723	0.162	0.182
⅝	0.6250	11	0.5660	0.5168	0.5266	0.202	0.226
¾	0.7500	10	0.6850	0.6309	0.6417	0.302	0.334
⅞	0.8750	9	0.8028	0.7427	0.7547	0.419	0.462
1	1.0000	8	0.9188	0.8512	0.8647	0.551	0.606
1⅛	1.1250	7	1.0322	0.9549	0.9704	0.693	0.763
1¼	1.2500	7	1.1572	1.0799	1.0954	0.890	0.969
1⅜	1.3750	6	1.2667	1.1766	1.1946	1.054	1.155
1½	1.5000	6	1.3917	1.3016	1.3196	1.294	1.405
1¾	1.7500	5	1.6201	1.5119	1.5335	1.74	1.90
2	2.0000	4½	1.8557	1.7353	1.7594	2.30	2.50
2¼	2.2500	4½	2.1057	1.9853	2.0094	3.02	3.25
2½	2.5000	4	2.3376	2.2023	2.2294	3.72	4.00
2¾	2.7500	4	2.5876	2.4523	2.4794	4.62	4.93
3	3.0000	4	2.8376	2.7023	2.7294	5.62	5.97
3¼	3.2500	4	3.0876	2.9523	2.9794	6.72	7.10
3½	3.5000	4	3.3376	3.2023	3.2294	7.92	8.33
3¾	3.7500	4	3.5876	3.4523	3.4794	9.21	9.66
4	4.0000	4	3.8376	3.7023	3.7294	10.61	11.08

* British: effective diameter.
† See formula under definition of tensile stress area in Appendix B of ANSI B1.1-1987.
‡ Design form. See Fig. 2B in ANSI B1.1-1982 or Fig. 1 in 1989 revision.
§ Secondary sizes.
SOURCE: ANSI B1.1-1982, revised 1989; reproduced by permission.



Name	Symbol: Inch bolt or nut	Symbol: Metric bolt	Symbol: Metric nut
Major diameter	D	d	D
Pitch diam. (inch); effective diam. (metric)	E	d_2	D_2
Minor diameter	K	d_1	D_1

Fig. 8.2.1 Basic thread profile.

Table 8.2.3 Basic Dimensions for Fine Thread Series (UNF/UNRF)

Nominal size, in	Basic major diameter D, in	Threads per inch n	Basic pitch diameter* E, in	UNR design minor diameter external† K_s, in	Basic minor diameter internal K, in	Section at minor diameter at $D - 2h_b$, in^2	Tensile stress area,‡ in^2
0 (0.060)	0.0600	80	0.0519	0.0451	0.0465	0.00151	0.00180
1 (0.073)§	0.0730	72	0.0640	0.0565	0.0580	0.00237	0.00278
2 (0.086)	0.0860	64	0.0759	0.0674	0.0691	0.00339	0.00394
3 (0.099)§	0.0990	56	0.0874	0.0778	0.0797	0.00451	0.00523
4 (0.112)	0.1120	48	0.0985	0.0871	0.0894	0.00566	0.00661
5 (0.125)	0.1250	44	0.1102	0.0979	0.1004	0.00716	0.00830
6 (0.138)	0.1380	40	0.1218	0.1082	0.1109	0.00874	0.01015
8 (0.164)	0.1640	36	0.1460	0.1309	0.1339	0.01285	0.01474
10 (0.190)	0.1900	32	0.1697	0.1528	0.1562	0.0175	0.0200
12 (0.216)§	0.2160	28	0.1928	0.1734	0.1773	0.0226	0.0258
¼	0.2500	28	0.2268	0.2074	0.2113	0.0326	0.0364
5⁄16	0.3125	24	0.2854	0.2629	0.2674	0.0524	0.0580
⅜	0.3750	24	0.3479	0.3254	0.3299	0.0809	0.0878
7⁄16	0.4375	20	0.4050	0.3780	0.3834	0.1090	0.1187
½	0.5000	20	0.4675	0.4405	0.4459	0.1486	0.1599
9⁄16	0.5625	18	0.5264	0.4964	0.5024	0.189	0.203
⅝	0.6250	18	0.5889	0.5589	0.5649	0.240	0.256
¾	0.7500	16	0.7094	0.6763	0.6823	0.351	0.373
⅞	0.8750	14	0.8286	0.7900	0.7977	0.480	0.509
1	1.0000	12	0.9459	0.9001	0.9098	0.625	0.663
1⅛	1.1250	12	1.0709	1.0258	1.0348	0.812	0.856
1¼	1.2500	12	1.1959	1.1508	1.1598	1.024	1.073
1⅜	1.3750	12	1.3209	1.2758	1.2848	1.260	1.315
1½	1.5000	12	1.4459	1.4008	1.4098	1.521	1.581

* British: effective diameter.
† See formula under definition of tensile stress area in Appendix B of ANSI B1.1-1982.
‡ Design form. See Fig. 2B of ANSI B1.1-1982 or Fig. 1 in 1989 revision.
§ Secondary sizes.
SOURCE: ANSI B1.1-1982, revised 1989; reproduced by permission.

Table 8.2.4 Basic Dimensions for Extra-Fine Thread Series (UNEF/UNREF)

Nominal size, in		Basic major diameter D, in	Threads per inch n	Basic pitch diameter* E, in	UNR design minor diameter external† K_s, in	Basic minor diameter internal K, in	Section at minor diameter at $D - 2h_b$, in^2	Tensile stress area,‡ in^2
Primary	Secondary							
	12 (0.216)	0.2160	32	0.1957	0.1788	0.1822	0.0242	0.0270
¼		0.2500	32	0.2297	0.2128	0.2162	0.0344	0.0379
5⁄16		0.3125	32	0.2922	0.2753	0.2787	0.0581	0.0625
⅜		0.3750	32	0.3547	0.3378	0.3412	0.0878	0.0932
7⁄16		0.4375	28	0.4143	0.3949	0.3988	0.1201	0.1274
½		0.5000	28	0.4768	0.4573	0.4613	0.162	0.170
9⁄16		0.5625	24	0.5354	0.5129	0.5174	0.203	0.214
⅜		0.6250	24	0.5979	0.5754	0.5799	0.256	0.268
	11⁄16	0.6875	24	0.6604	0.6379	0.6424	0.315	0.329
¾		0.7500	20	0.7175	0.6905	0.6959	0.369	0.386
	13⁄16	0.8125	20	0.7800	0.7530	0.7584	0.439	0.458
⅞		0.8750	20	0.8425	0.8155	0.8209	0.515	0.536
	15⁄16	0.9375	20	0.9050	0.8780	0.8834	0.598	0.620
1		1.0000	20	0.9675	0.9405	0.9459	0.687	0.711
	1 1⁄16	1.0625	18	1.0264	0.9964	1.0024	0.770	0.799
1⅛		1.1250	18	1.0889	1.0589	1.0649	0.871	1.901
	1 3⁄16	1.1875	18	1.1514	1.1214	1.1274	0.977	1.009
1¼		1.2500	18	1.2139	1.1839	1.1899	1.090	1.123
	1 5⁄16	1.3125	18	1.2764	1.2464	1.2524	1.208	1.244
1⅜		1.3750	18	1.3389	1.3089	1.3149	1.333	1.370
	1 7⁄16	1.4375	18	1.4014	1.3714	1.3774	1.464	1.503
1½		1.5000	18	1.4639	1.4339	1.4399	1.60	1.64
	1 9⁄16	1.5625	18	1.5264	1.4964	1.5024	1.74	1.79
1⅝		1.6250	18	1.5889	1.5589	1.5649	1.89	1.94
	1 11⁄16	1.6875	18	1.6514	1.6214	1.6274	2.05	2.10

* British: effective diameter.
† Design form. See Fig. 2B in ANSI B1.1-1982 or Fig. 1 in 1989 revision.
‡ See formula under definition of tensile stress area in Appendix B in ANSI B1.1-1982.
SOURCE: ANSI B1.1-1982 revised 1989; reproduced by permission.

Table 8.2.5 ISO Metric Screw Thread Standard Series

Nominal size diam, mm			Pitches, mm														Nominal size diam, mm
Column*			Series with graded pitches		Series with constant pitches												
1	2	3	Coarse	Fine	6	4	3	2	1.5	1.25	1	0.75	0.5	0.35	0.25	0.2	
0.25			0.075	—	—	—	—	—	—	—	—	—	—	—	—	—	0.25
0.3			0.8	—	—	—	—	—	—	—	—	—	—	—	—	—	0.3
	0.35		0.09	—	—	—	—	—	—	—	—	—	—	—	—	—	0.35
0.4			0.1	—	—	—	—	—	—	—	—	—	—	—	—	—	0.4
	0.45		0.1	—	—	—	—	—	—	—	—	—	—	—	—	—	0.45
0.5			0.125	—	—	—	—	—	—	—	—	—	—	—	—	—	0.5
	0.55		0.125	—	—	—	—	—	—	—	—	—	—	—	—	—	0.55
0.6			0.15	—	—	—	—	—	—	—	—	—	—	—	—	—	0.6
	0.7		0.175	—	—	—	—	—	—	—	—	—	—	—	—	—	0.7
0.8			0.2	—	—	—	—	—	—	—	—	—	—	—	—	—	0.8
	0.9		0.225	—	—	—	—	—	—	—	—	—	—	—	—	—	0.9
1			0.25	—	—	—	—	—	—	—	—	—	—	—	—	0.2	1
	1.1		0.25	—	—	—	—	—	—	—	—	—	—	—	—	0.2	1.1
1.2			0.25	—	—	—	—	—	—	—	—	—	—	—	—	0.2	1.2
	1.4		0.3	—	—	—	—	—	—	—	—	—	—	—	—	0.2	1.4
1.6			0.35	—	—	—	—	—	—	—	—	—	—	—	—	0.2	1.6
	1.8		0.35	—	—	—	—	—	—	—	—	—	—	—	—	0.2	1.8
2			0.4	—	—	—	—	—	—	—	—	—	—	—	0.25	—	2
	2.5		0.45	—	—	—	—	—	—	—	—	—	—	—	0.25	—	2.2
2.5			0.45	—	—	—	—	—	—	—	—	—	—	0.35	—	—	2.5
3			0.5	—	—	—	—	—	—	—	—	—	—	0.35	—	—	3
	3.5		0.6	—	—	—	—	—	—	—	—	—	—	0.35	—	—	3.5
4			0.7	—	—	—	—	—	—	—	—	—	0.5	—	—	—	4
	4.5		0.75	—	—	—	—	—	—	—	—	—	0.5	—	—	—	4.5
5			0.8	—	—	—	—	—	—	—	—	—	0.5	—	—	—	5
		5.5	—	—	—	—	—	—	—	—	—	—	0.5	—	—	—	5.5
6			1	—	—	—	—	—	—	—	—	0.75	—	—	—	—	6
		7	1	—	—	—	—	—	—	—	—	0.75	—	—	—	—	7
8			1.25	1	—	—	—	—	—	—	1	0.75	—	—	—	—	8
		9	1.25	—	—	—	—	—	—	—	1	0.75	—	—	—	—	9
10			1.5	1.25	—	—	—	—	—	1.25	1	0.75	—	—	—	—	10
		11	1.5	—	—	—	—	—	—	—	1	0.75	—	—	—	—	11
12			1.75	1.25	—	—	—	—	1.5	1.25	1	—	—	—	—	—	12
	14		2	1.5	—	—	—	—	1.5	1.25†	1	—	—	—	—	—	14
		15	—	—	—	—	—	—	1.5	—	1	—	—	—	—	—	15
16			2	1.5	—	—	—	—	1.5	—	1	—	—	—	—	—	16
		17	—	—	—	—	—	—	1.5	—	1	—	—	—	—	—	17
	18		2.5	1.5	—	—	—	2	1.5	—	1	—	—	—	—	—	18
20			2.5	1.5	—	—	—	2	1.5	—	1	—	—	—	—	—	20
	22		2.5	1.5	—	—	—	2	1.5	—	1	—	—	—	—	—	22
24			3	2	—	—	—	2	1.5	—	1	—	—	—	—	—	24
		25	—	—	—	—	—	2	1.5	—	1	—	—	—	—	—	25
		26	—	—	—	—	—	—	1.5	—	1	—	—	—	—	—	26
	27		3	2	—	—	—	2	1.5	—	1	—	—	—	—	—	27
		28	—	—	—	—	—	2	1.5	—	1	—	—	—	—	—	28
30			3.5	2	—	—	(3)	2	1.5	—	1	—	—	—	—	—	30
		32	—	—	—	—	—	2	1.5	—	—	—	—	—	—	—	32
	33		3.5	2	—	—	(3)	2	1.5	—	—	—	—	—	—	—	33
		35‡	—	—	—	—	—	—	1.5	—	—	—	—	—	—	—	35‡
36			4	3	—	—	—	2	1.5	—	—	—	—	—	—	—	36
		38	—	—	—	—	—	—	1.5	—	—	—	—	—	—	—	38
	39		4	3	—	—	—	2	1.5	—	—	—	—	—	—	—	39
		40	—	—	—	—	3	2	1.5	—	—	—	—	—	—	—	40
42			4.5	3	—	4	3	2	1.5	—	—	—	—	—	—	—	42
	45		4.5	3	—	4	3	2	1.5	—	—	—	—	—	—	—	45

* Thread diameter should be selected from column 1, 2 or 3; with preference being given in that order.
† Pitch 1.25 mm in combination with diameter 14 mm has been included for spark plug applications.
‡ Diameter 35 mm has been included for bearing locknut applications.
NOTE: The use of pitches shown in parentheses should be avoided wherever possible. The pitches enclosed in the bold frame, together with the corresponding nominal diameters in columns 1 and 2, are those combinations which have been established by ISO Recommendations as a selected ''coarse'' and ''fine'' series for commercial fasteners. Sizes 0.25 mm through 1.4 mm are covered in ISO Recommendation R68 and, except for the 0.25-mm size, in ANSI B1.10.
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Table 8.2.6 Limiting Dimensions of Standard Series Threads for Commercial Screws, Bolts, and Nuts

Nominal size diam, mm	Pitch p, mm	Basic thread designation	External thread (bolt), mm: Tol. class	Allowance	Major diameter Max	Major diameter Min	Pitch diameter Max	Pitch diameter Min	Pitch diameter Tol.	Minor diameter Max*	Minor diameter Min†	Internal thread (nut), mm: Tol. class	Minor diameter Min	Minor diameter Max	Pitch diameter Min	Pitch diameter Max	Pitch diameter Tol.	Major diam, min
1.6	0.35	M1.6	6g	0.019	1.581	1.496	1.354	1.291	0.063	1.151	1.063	6H	1.221	1.321	1.373	1.458	0.085	1.600
1.8	0.35	M1.8	6g	0.019	1.781	1.696	1.554	1.491	0.063	1.351	1.263	6H	1.421	1.521	1.573	1.568	0.085	1.800
2	0.4	M2	6g	0.019	1.981	1.886	1.721	1.654	0.067	1.490	1.394	6H	1.567	1.679	1.740	1.830	0.090	2.000
2.2	0.45	M2.2	6g	0.020	2.180	2.080	1.888	1.817	0.071	1.628	1.525	6H	1.713	1.838	1.908	2.000	0.095	2.200
2.5	0.45	M2.5	6g	0.020	2.480	2.380	2.188	2.117	0.071	1.928	1.825	6H	2.013	2.138	2.208	2.303	0.095	2.500
3	0.5	M3	6g	0.020	2.980	2.874	2.655	2.580	0.075	2.367	2.256	6H	2.459	2.599	2.675	2.775	0.100	3.000
3.5	0.6	M3.5	6g	0.021	3.479	3.354	3.089	3.004	0.085	2.742	2.614	6H	2.850	3.010	3.110	3.222	0.112	3.500
4	0.7	M4	6g	0.022	3.978	3.838	3.523	3.433	0.090	3.119	2.979	6H	3.242	3.422	3.545	3.663	0.118	4.000
4.5	0.75	M4.5	6g	0.022	4.478	4.338	3.991	3.901	0.090	3.558	3.414	6H	3.688	3.878	4.013	4.131	0.118	4.500
5	0.8	M5	6g	0.024	4.976	4.826	4.456	4.361	0.095	3.994	3.841	6H	4.134	4.334	4.480	4.605	0.125	5.000
6	1	M6	6g	0.026	5.974	5.794	5.324	5.212	0.112	4.747	4.563	6H	4.917	5.153	5.350	5.500	0.150	6.000
7	1	M7	6g	0.026	6.974	6.794	6.234	6.212	0.112	5.747	5.563	6H	5.917	6.153	6.350	6.500	0.150	7.000
8	1.25	M8	6g	0.028	7.972	7.760	7.160	7.042	0.118	6.439	6.231	6H	6.647	6.912	7.188	7.348	0.160	8.000
	1	M8 × 1	6g	0.026	7.974	7.794	7.324	7.212	0.112	6.747	6.563	6H	6.918	7.154	7.350	7.500	0.150	8.000
10	1.5	M10	6g	0.032	9.968	9.732	8.994	8.862	0.132	8.127	7.879	6H	8.376	8.676	9.026	9.206	0.180	10.000
	1.25	M10 × 1.25	6g	0.028	9.972	9.760	9.160	9.042	0.118	8.439	8.231	6H	8.646	8.911	9.188	9.348	0.160	10.000
12	1.75	M12	6g	0.034	11.966	11.701	10.829	10.679	0.150	9.819	9.543	6H	10.106	10.441	10.863	11.063	0.200	12.000
	1.25	M12 × 1.25	6g	0.028	11.972	11.760	11.160	11.028	0.118	10.439	10.217	6H	10.646	10.911	11.188	11.368	0.180	12.000
14	2	M14	6g	0.038	13.962	13.682	12.663	12.503	0.160	11.508	11.204	6H	11.835	12.210	12.701	12.913	0.212	14.000
	1.5	M14 × 1.5	6g	0.032	13.968	13.732	12.994	12.854	0.140	12.127	11.879	6H	12.376	12.676	13.026	13.216	0.190	14.000
16	2	M16	6g	0.038	15.962	15.682	14.663	14.503	0.160	13.508	13.204	6H	13.385	14.210	14.701	14.913	0.212	16.000
	1.5	M16 × 1.5	6g	0.032	15.968	15.732	14.994	14.854	0.140	14.127	13.879	6H	14.376	14.676	15.026	15.216	0.190	16.000
18	2.5	M18	6g	0.038	17.958	17.623	16.334	16.164	0.170	14.891	14.541	6H	15.294	15.744	16.376	16.600	0.224	18.000
	1.5	M18 × 1.5	6g	0.032	17.968	17.732	16.994	15.854	0.140	16.127	15.879	6H	16.376	16.676	17.026	17.216	0.190	18.000
20	2.5	M20	6g	0.042	19.958	19.623	18.334	18.164	0.170	16.891	16.541	6H	17.294	17.744	18.376	18.600	0.224	20.000
	1.5	M20 × 1.5	6g	0.032	19.968	19.732	18.994	18.854	0.140	18.127	17.879	6H	18.376	18.676	19.026	19.216	0.190	20.000
22	2.5	M22	6g	0.042	21.958	21.623	20.334	20.164	0.170	18.891	18.541	6H	19.294	19.744	20.376	20.600	0.224	22.000
	1.5	M22 × 1.5	6g	0.032	21.968	21.732	20.994	20.854	0.140	20.127	19.879	6H	20.376	20.676	21.026	21.216	0.190	22.000
24	3	M24	6g	0.048	23.952	23.577	22.003	21.803	0.200	20.271	19.855	6H	20.752	21.252	22.051	22.316	0.265	24.000
	2	M24 × 2	6g	0.038	23.962	23.682	22.663	22.493	0.170	21.508	21.194	6H	21.835	22.210	22.701	22.925	0.224	24.000
27	3	M27	6g	0.048	26.952	26.577	25.003	24.803	0.200	23.271	22.855	6H	23.752	24.252	25.051	25.316	0.265	27.000
	2	M27 × 2	6g	0.038	26.962	26.682	25.663	25.493	0.170	24.508	24.194	6H	24.835	25.210	25.701	25.925	0.224	27.000
30	3.5	M30	6g	0.053	29.947	29.522	27.674	27.462	0.212	25.653	25.189	6H	26.211	26.771	27.727	28.007	0.280	30.000
	2	M30 × 2	6g	0.038	29.962	29.682	28.663	28.493	0.170	27.508	27.194	6H	27.835	28.210	28.701	28.925	0.224	30.000
33	3.5	M33	6g	0.053	32.947	32.522	30.674	30.462	0.212	28.653	28.189	6H	29.211	29.771	30.727	31.007	0.280	33.000
	2	M33 × 2	6g	0.038	32.962	32.682	31.663	31.493	0.170	30.508	30.194	6H	30.835	31.210	31.701	31.925	0.224	33.000
36	4	M36	6g	0.060	35.940	35.465	33.342	33.118	0.224	31.033	30.521	6H	31.670	32.270	33.402	33.702	0.300	36.000
	3	M36 × 3	6g	0.048	35.952	35.577	34.003	33.803	0.200	32.271	31.855	6H	32.752	33.252	34.051	34.316	0.265	36.000
39	4	M39	6g	0.060	38.940	38.465	36.342	36.118	0.224	34.033	33.521	6H	34.670	35.270	36.402	36.702	0.300	39.000
	3	M39 × 3	6g	0.048	38.952	38.577	37.003	36.803	0.200	35.271	34.855	6H	35.752	36.252	37.051	37.316	0.265	39.000

* Design form, see Figs. 2 and 5 of ANSI B1.13M-1979 (or Figs. 1 and 4 in 1983 revision).
† Required for high-strength applications where rounded root is specified.
SOURCE: [Appeared in ASME/SAE Interpretive document, Metric Screw Threads, B1.13 (Nov. 3, 1966), pp. 9, 10.] ISO 261-1973, reproduced by permission.

and it is shown in Fig. 8.2.1. The ISO thread series (see Table 8.2.5) are those published in ISO 261-1973. Increased overseas business sparked U.S. interest in metric screw threads, and the ANSI, through its Special Committee to Study Development of an Optimum Metric Fastener System, in joint action with an ISO working group (ISO/TC 1/TC 2), established compromise recommendations regarding metric screw threads. The approved results appear in ANSI B1.13-1979 (Table 8.2.6). This ANSI metric thread series is essentially a selected subset (boxed-in portion of Table 8.2.5) of the larger ISO 261-1973 set. The M profiles of tolerance class 6H/6g are intended for metric applications where inch class 2A/2B has been used.

Metric Tolerance Classes for Threads Tolerance classes are a selected combination of tolerance grades and tolerance positions applied to length-of-engagement groups.

Tolerance grades are indicated as numbers for crest diameters of nut and bolt and for pitch diameters of nut and bolt. Tolerance is the acceptable variation permitted on any such diameter.

Tolerance positions are indicated as letters, and are allowances (fundamental deviations) as dictated by field usage or conditions. Capital letters are used for internal threads (nut) and lower case for external threads (bolt).

There are three established groups of length of thread engagement, S (short), N (normal), and L (long), for various diameter-pitch combinations. Normal length of thread engagement is calculated from the formula $N = 4.5pd^{0.2}$, where p is pitch and d is the smallest nominal size within each of a series of groupings of nominal sizes.

In conformance with coating (or plating) requirements and demands of ease of assembly, the following tolerance positions have been established:

Bolt	*Nut*	
e		Large allowance
g	G	Small allowance
h	H	No allowance

See Table 8.2.7 for preferred tolerance classes.

ISO metric screw threads are designated by a set of number and letter symbols signifying, in sequence, metric symbol, nominal size, × (symbol), pitch, tolerance grade (on pitch diameter), tolerance position (for pitch diameter), tolerance grade (on crest diameter), and tolerance position (for crest diameter).

EXAMPLE. M6 × 0.75-5g6g, where M = metric symbol; 6 = nominal size, × = symbol; 0.75 = pitch-axial distance of adjacent threads measured between corresponding thread points (millimeters); 5 = tolerance grade (on pitch diameter); g = tolerance position (for pitch diameter); 6 = tolerance grade (on crest diameter); g = tolerance position (for crest diameter).

Power Transmission Screw Threads: Forms and Proportions

The **Acme** thread appears in four series [ANSI B1.8-1973 (revised 1988) and B1.5-1977]. Generalized dimensions for the series are given in Table 8.2.8.

The 29° general-purpose thread (Fig. 8.2.2) is used for all Acme thread applications outside of special design cases.

The 29° stub thread (Fig. 8.2.3) is used for heavy-loading designs and where space constraints or economic factors make a shallow thread advantageous.

The 60° stub thread (Fig. 8.2.4) finds special applications in the machine-tool industry.

The 10° modified square thread (Fig. 8.2.5) is, for all practical purposes, equivalent to a ''square'' thread.

For selected Acme diameter-pitch combinations, see Table 8.2.9.

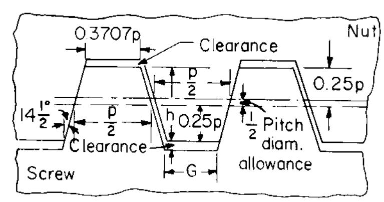


Fig. 8.2.2 29° Acme general-purpose thread.

Table 8.2.7 Preferred Tolerance Classes

	Length of engagement														
	External threads (bolts)									Internal threads (nuts)					
	Tolerance position e (large allowance)			Tolerance position g (small allowance)			Tolerance position h (no allowance)			Tolerance position G (small allowance)			Tolerance position H (no allowance)		
Quality	Group S	Group N	Group L	Group S	Group N	Group L	Group S	Group N	Group L	Group S	Group N	Group L	Group S	Group N	Group L
Fine							3h4h	4h	4h5h				4H	5H	6H
Medium		6e	7e6e	5g6g	6g	7g6g	5h6h	6h	7h6h	5G	6G	7G	5H	6H	7H
Coarse					8g	9g8g					7G	8G		7H	8H

NOTE: Fine quality applies to precision threads where little variation in fit character is permissible. Coarse quality applies to those threads which present manufacturing difficulties, such as the threading of hot-rolled bars or tapping deep blind holes.
SOURCE: ISO 261-1973, reproduced by permission.

Table 8.2.8 Acme Thread Series
(D = outside diam, p = pitch. All dimensions in inches.)
(See Figs. 8.2.2 to 8.2.5.)

	Thread dimensions			
Symbols	29° general purpose	29° stub	60° stub	10° modified
t = thickness of thread	$0.5p$	$0.5p$	$0.5p$	$0.5p$
R = basic depth of thread	$0.5p$	$0.3p$	$0.433p$	$0.5p$*
F = basic width of flat	$0.3707p$	$0.4224p$	$0.250p$	$0.4563p$†
G = (see Figs. 8.2.2, 8.2.3, 8.2.4)	$F - (0.52 \times \text{clearance})$	$F - (0.52 \times \text{clearance})$	$0.227p$	$F - (0.17 \times \text{clearance})$
E = basic pitch diam	$D - 0.5p$	$D - 0.3p$	$D - 0.433p$	$D - 0.5p$
K = basic minor diam	$D - p$	$D - 0.6p$	$D - 0.866p$	$D - p$
Range of threads, per inch	1–16	2–16	4–16	

* A clearance of at least 0.010 in is added to h on threads of 10-pitch and coarser, and 0.005 in on finer pitches, to produce extra depth, thus avoiding interference with threads of mating parts of a minor or major diameters.
† Measured at crest of screw thread.

Table 8.2.9 Acme Thread Diameter-Pitch Combinations
(See Figs. 8.2.2 to 8.2.5.)

Size	Threads per inch	Size	Threads per inch	Size	Threads per inch	Size	Threads per inch	Size	Threads per inch
¼	16	⅝	8	1¼	5	2¼	3	4	2
5⁄16	14	¾	6	1⅜	4	2½	3	4½	2
⅜	12	⅞	6	1½	4	2¾	3	5	2
7⁄16	12	1	5	1¾	4	3	2		
½	10	1⅛	5	2	4	3½	2		

Three classes (2G, 3G, 4G) of general-purpose threads have clearances on all diameters for free movement. A fourth class (5G) of general-purpose threads has no allowance or clearance on the pitch diameter for purposes of minimum end play or backlash.

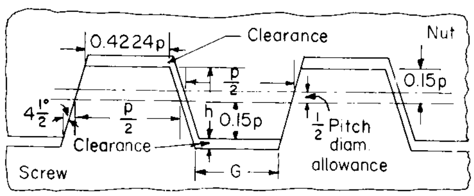


Fig. 8.2.3 29° stub Acme thread.

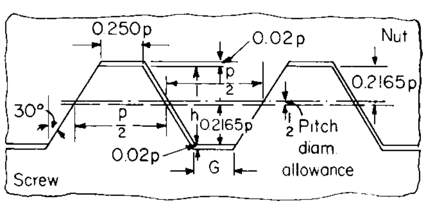


Fig. 8.2.4 60° stub Acme thread.

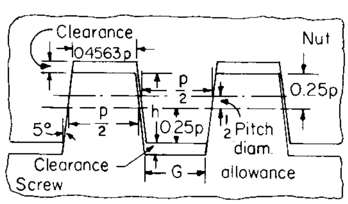


Fig. 8.2.5 10° modified square thread.

High-Strength Bolting Screw Threads

High-strength bolting applications include pressure vessels, steel pipe flanges, fittings, valves, and other services. They can be used for either hot or cold surfaces where high tensile stresses are produced when the joints are made up. For sizes 1 in and smaller, the ANSI coarse-thread series is used. For larger sizes, the ANSI 8-pitch thread series is used (see Table 8.2.10).

Screw Threads for Pipes

American National Standard Taper Pipe Thread (ANSI/ASME B1.20.1-1983) This thread is shown in Fig. 8.2.6. It is made to the following specifications: The taper is 1 in 16 or 0.75 in/ft. The basic length of the external taper thread is determined by $L_2 = p(0.8D + 6.8)$, where D is the basic outside diameter of the pipe (see Table 8.2.11). Thread designation and notation is written as: nominal size, number of threads per inch, thread series. For example: ⅜-18 NPT, ⅛-27 NPSC, ½-14 NPTR, ⅛-27 NPSM, ⅛-27 NPSL, 1-11.5 NPSH, where N = National (American) Standard, T = taper, C = coupling, S = straight, M = mechanical, L = locknut, H = hose coupling, and R = rail fittings. Where pressure-tight joints are required, it is intended that taper pipe threads be made up wrench-tight with a sealant. Descriptions of thread series include: NPSM = free-fitting mechanical joints for fixtures, NPSL = loose-fitting mechanical joints with locknuts, NPSH = loose-fitting mechanical joints for hose coupling.

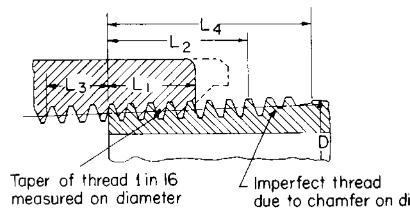


Fig. 8.2.6 American National Standard taper pipe threads.

American National Standard Straight Pipe Thread (ANSI/ASME B1.20.1-1983) This thread can be used to advantage for the following: (1) pressure-tight joints with sealer; (2) pressuretight joints without sealer for drain plugs, filler plugs, etc.; (3) free-fitting mechanical joints for fixtures; (4) loose-fitting mechanical joints with locknuts; and (5) loose-fitting mechanical joints for hose couplings. Dimensions are shown in Table 8.2.12.

American National Standard Dry-Seal Pipe Threads (ANSI B1.20.3-1976 (inch), ANSI B1.20.4-1976 (metric translation) Thread designation and notation include nominal size, number of threads per inch, thread series, class. For example, ⅛-27 NPTF-1, ⅛-27 NPTF-2, ⅛-27 PTF-SAE short, ⅛-27 NPSI, where N = National (American) standard, P = pipe, T = taper, S = straight, F = fuel and oil, I = intermediate. NPTF has two classes: class 1 = specific inspection of root and crest truncation *not* required; class 2 = specific inspection of root and crest truncation *is* required. The series includes: NPTF for all types of service; PTF-SAE short where clearance is not sufficient for full thread length as NPTF; NPSF, nontapered, economical to produce, and used with soft or ductile materials; NPSI nontapered, thick sections with little expansion.

Dry-seal pipe threads resemble tapered pipe threads except the form is truncated (see Fig. 8.2.7), and $L_4 = L_2 + 1$ (see Fig. 8.2.6). Although these threads are designed for nonlubricated joints, as in automobile work, under certain conditions a lubricant is used to prevent galling. Table 8.2.13 lists truncation values.

Tap drill sizes for tapered and straight pipe threads are listed in Table 8.2.14.

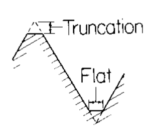


Fig. 8.2.7 American National Standard dry-seal pipe thread.

Wrench bolt heads, nuts, and wrench openings have been standardized (ANSI 18.2-1972). Wrench openings are given in Table 8.2.15; bolt head and nut dimensions are in Table 8.2.16.

Machine Screws

Machine screws are defined according to head types as follows:

Flat Head This screw has a flat surface for the top of the head with a

Table 8.2.10 Screw Threads for High-Strength Bolting
(All dimensions in inches)

Size	Threads per inch	Allowance (minus)	Major diam	Major diam tolerance	Max pitch diam*	Max pitch diam tolerance	Minor diam max	Nut max minor diam	Nut max minor diam tolerance	Nut max pitch diam*	Nut max pitch diam tolerance
¼	20	0.0010	0.2490	0.0072	0.2165	0.0026	0.1877	0.2060	0.0101	0.2211	0.0036
5⁄16	18	0.0011	0.3114	0.0082	0.2753	0.0030	0.2432	0.2630	0.0106	0.2805	0.0041
⅜	16	0.0013	0.3737	0.0090	0.3331	0.0032	0.2990	0.3184	0.0111	0.3389	0.0045
7⁄16	14	0.0013	0.4362	0.0098	0.3898	0.0036	0.3486	0.3721	0.0119	0.3960	0.0049
½	13	0.0015	0.4985	0.0104	0.4485	0.0037	0.4041	0.4290	0.0123	0.4552	0.0052
9⁄16	12	0.0016	0.5609	0.0112	0.5068	0.0040	0.4587	0.4850	0.0127	0.5140	0.0056
⅝	11	0.0017	0.6233	0.0118	0.5643	0.0042	0.5118	0.5397	0.0131	0.5719	0.0059
¾	10	0.0019	0.7481	0.0128	0.6831	0.0045	0.6254	0.6553	0.0136	0.6914	0.0064
⅞	9	0.0021	0.8729	0.0140	0.8007	0.0049	0.7366	0.7689	0.0142	0.8098	0.0070
1	8	0.0022	0.9978	0.0152	0.9166	0.0054	0.8444	0.8795	0.0148	0.9264	0.0076
1⅛	8	0.0024	1.1226	0.0152	1.0414	0.0055	0.9692	1.0045	0.0148	1.0517	0.0079
1¼	8	0.0025	1.2475	0.0152	1.1663	0.0058	1.0941	1.1295	0.0148	1.1771	0.0083
1⅜	8	0.0025	1.3725	0.0152	1.2913	0.0061	1.2191	1.2545	0.0148	1.3024	0.0086
1½	8	0.0027	1.4973	0.0152	1.4161	0.0063	1.3439	1.3795	0.0148	1.4278	0.0090
1⅝	8	0.0028	1.6222	0.0152	1.5410	0.0065	1.4688	1.5045	0.0148	1.5531	0.0093
1¾	8	0.0029	1.7471	0.0152	1.6659	0.0068	1.5937	1.6295	0.0148	1.6785	0.0097
1⅞	8	0.0030	1.8720	0.0152	1.7908	0.0070	1.7186	1.7545	0.0148	1.8038	0.0100
2	8	0.0031	1.9969	0.0152	1.9157	0.0073	1.8435	1.8795	0.0148	1.9294	0.0104
2⅛	8	0.0032	2.1218	0.0152	2.0406	0.0075	1.9682	2.0045	0.0148	2.0545	0.0107
2¼	8	0.0033	2.2467	0.0152	2.1655	0.0077	2.0933	2.1295	0.0148	2.1798	0.0110
2½	8	0.0035	2.4965	0.0152	2.4153	0.0082	2.3431	2.3795	0.0148	2.4305	0.0117
2¾	8	0.0037	2.7463	0.0152	2.6651	0.0087	2.5929	2.6295	0.0148	2.6812	0.0124
3	8	0.0038	2.9962	0.0152	2.9150	0.0092	2.8428	2.8795	0.0148	2.9318	0.0130
3¼	8	0.0039	3.2461	0.0152	3.1649	0.0093	3.0927	3.1295	0.0148	3.1820	0.0132
3½	8	0.0040	3.4960	0.0152	3.4148	0.0093	3.3426	3.3795	0.0148	3.4321	0.0133

The Unified form of thread shall be used. Pitch diameter tolerances include errors of lead and angle.
* The maximum pitch diameters of screws are smaller than the minimum pitch diameters of nuts by these amounts.

Table 8.2.11 ANSI Taper Pipe Thread
(All dimensions in inches)
(See Fig. 8.2.6.)

Nominal pipe size	OD of pipe	Threads per inch	Pitch of thread	Hand-tight engagement length L_1	Effective thread external length L_2	Wrench makeup length for internal thread length L_3	Overall length external thread L_4
1⁄16	0.3125	27	0.03704	0.160	0.2611	0.1111	0.3896
⅛	0.405	27	0.03704	0.180	0.2639	0.1111	0.3924
¼	0.540	18	0.05556	0.200	0.4018	0.1667	0.5946
⅜	0.675	18	0.05556	0.240	0.4078	0.1667	0.6006
½	0.840	14	0.07143	0.320	0.5337	0.2143	0.7815
¾	1.050	14	0.07143	0.339	0.5457	0.2143	0.7935
1	1.315	11½	0.08696	0.400	0.6828	0.2609	0.9845
1¼	1.660	11½	0.08696	0.420	0.7068	0.2609	1.0085
1½	1.900	11½	0.08696	0.420	0.7235	0.2609	1.0252
2	2.375	11½	0.08696	0.436	0.7565	0.2609	1.0582
2½	2.875	8	0.12500	0.682	1.1375	0.2500	1.5712
3	3.500	8	0.12500	0.766	1.2000	0.2500	1.6337
3½	4.000	8	0.12500	0.821	1.2500	0.2500	1.6837
4	4.500	8	0.12500	0.844	1.3000	0.2500	1.7337
5	5.563	8	0.12500	0.937	1.4063	0.2500	1.8400
6	6.625	8	0.12500	0.958	1.5125	0.2500	1.9462
8	8.625	8	0.12500	1.063	1.7125	0.2500	2.1462
10	10.750	8	0.12500	1.210	1.9250	0.2500	2.3587
12	12.750	8	0.12500	1.360	2.1250	0.2500	2.5587
14 OD	14.000	8	0.12500	1.562	2.2500	0.2500	2.6837
16 OD	16.000	8	0.12500	1.812	2.4500	0.2500	2.8837
18 OD	18.000	8	0.12500	2.000	2.6500	0.2500	3.0837
20 OD	20.000	8	0.12500	2.125	2.8500	0.2500	3.2837
24 OD	24.000	8	0.12500	2.375	3.2500	0.2500	3.6837

Table 8.2.12 ANSI Straight Pipe Threads
(All dimensions in inches)

		Pressure-tight with seals		Pressure-tight without seals		Free-fitting (NPSM)				Loose-fitting (NPSL)			
						External		Internal		External		Internal	
Nominal pipe size (1)	Threads per inch (2)	Pitch diam, max (3)	Minor diam, min (4)	Pitch diam, max (5)	Minor diam, min (6)	Pitch diam, max (7)	Major diam, max (8)	Pitch diam, max (9)	Minor diam, min (10)	Pitch diam, max (11)	Major diam, max (12)	Pitch diam, max (13)	Minor diam, min (14)
⅛	27	0.3782	0.342	0.3736	0.3415	0.3748	0.399	0.3783	0.350	0.3840	0.409	0.3989	0.362
¼	18	0.4951	0.440	0.4916	0.4435	0.4899	0.527	0.4951	0.453	0.5038	0.541	0.5125	0.470
⅜	18	0.6322	0.577	0.6270	0.5789	0.6270	0.664	0.6322	0.590	0.6409	0.678	0.6496	0.607
½	14	0.7851	0.715	0.7784	0.7150	0.7784	0.826	0.7851	0.731	0.7963	0.844	0.8075	0.753
¾	14	0.9956	0.925	0.9889	0.9255	0.9889	1.036	0.9956	0.941	1.0067	1.054	1.0179	0.964
1	11½	1.2468	1.161	1.2386	1.1621	1.2386	1.296	1.2468	1.181	1.2604	1.318	1.2739	1.208
1¼	11½	1.5915	1.506	—	—	1.5834	1.641	1.5916	1.526	1.6051	1.663	1.6187	1.553
1½	11½	1.8305	1.745	—	—	1.8223	1.880	1.8305	1.764	1.8441	1.902	1.8576	1.792
2	11½	2.3044	2.219	—	—	2.2963	2.354	2.3044	2.238	2.3180	2.376	2.3315	2.265
2½	8	2.7739	2.650	—	—	2.7622	2.846	2.7739	2.679	2.7934	2.877	2.8129	2.718
3	8	3.4002	3.277	—	—	3.3885	3.472	3.4002	3.305	3.4198	3.503	3.4393	3.344
3½	8	3.9005	3.777	—	—	3.8888	3.972	3.9005	3.806	3.9201	4.003	3.9396	3.845
4	8	4.3988	4.275	—	—	4.3871	4.470	4.3988	4.304	4.4184	4.502	4.4379	4.343
5	8	—	—	—	—	5.4493	5.533	5.4610	5.366	5.4805	5.564	5.5001	5.405
6	8	—	—	—	—	6.5060	6.589	6.5177	6.423	6.5372	6.620	6.5567	6.462
8	8	—	—	—	—	—	—	—	—	8.5313	8.615	8.5508	8.456
10	8	—	—	—	—	—	—	—	—	10.6522	10.735	10.6717	10.577
12	8	—	—	—	—	—	—	—	—	12.6491	12.732	12.6686	12.574

countersink angle of 82°. It is standard for machine screws, cap screws, and wood screws.

Round Head This screw has a semielliptical head and is standard for machine screws, cap screws, and wood screws except that for the cap screw it is called *button head.*

Fillister Head This screw has a rounded surface for the top of the head, the remainder being cylindrical. The head is standard for machine screws and cap screws.

Oval Head This screw has a rounded surface for the top of the head and a countersink angle of 82°. It is standard for machine screws and wood screws.

Hexagon Head This screw has a hexagonal head for use with external wrenches. It is standard for machine screws.

Socket Head This screw has an internal hexagonal socket in the head for internal wrenching. It is standard for cap screws.

These screw heads are shown in Fig. 8.2.8; pertinent dimensions are in Table 8.2.17. There are many more machine screw head shapes available to the designer for special purposes, and many are found in the literature. In addition, lots of different screw head configurations have been developed to render fasteners ''tamperproof''; these, too, are found in manufacturers' catalogs or the trade literature.

Eyebolts

Eyebolts are classified as rivet, nut, or screw, and can be had on a swivel. See Fig. 8.2.9 and Table 8.2.18. The safe working load may be obtained for each application by applying an appropriate factor of safety.

Driving recesses come in many forms and types and can be found in company catalogs. Figure 8.2.10 shows a representative set.

Setscrews are used for fastening collars, sheaves, gears, etc. to shafts to prevent relative rotation or translation. They are available in a variety of head and point styles, as shown in Fig. 8.2.11. A complete tabulation of dimensions is found in ANSI/ASME B18.3-1982 (R86), ANSI 18.6.2-1977 (R93), and ANSI 18.6.3-1977 (R91). Holding power for various sizes is given in Table 8.2.19.

Locking Fasteners

Locking fasteners are used to prevent loosening of a threaded fastener in service and are available in a wide variety differing vastly in design, performance, and function. Since each has special features which may make it of particular value in the solution of a given machine problem, it is important that great care be exercised in the selection of a particular

Table 8.2.13 ANSI Dry-Seal Pipe Threads*
(See Fig. 8.2.7.)

Threads per inch n	Truncation, in		Width of flat	
	Min	Max	Min	Max
27 Crest	$0.047p$	$0.094p$	$0.054p$	$0.108p$
Root	$0.094p$	$0.140p$	$0.108p$	$0.162p$
18 C	$0.047p$	$0.078p$	$0.054p$	$0.090p$
R	$0.078p$	$0.109p$	$0.090p$	$0.126p$
14 C	$0.036p$	$0.060p$	$0.042p$	$0.070p$
R	$0.060p$	$0.085p$	$0.070p$	$0.098p$
11½ C	$0.040p$	$0.060p$	$0.046p$	$0.069p$
R	$0.060p$	$0.090p$	$0.069p$	$0.103p$
8 C	$0.042p$	$0.055p$	$0.048p$	$0.064p$
R	$0.055p$	$0.076p$	$0.064p$	$0.088p$

* The *truncation* and *width-of-flat* proportions listed above are also valid in the metric system.

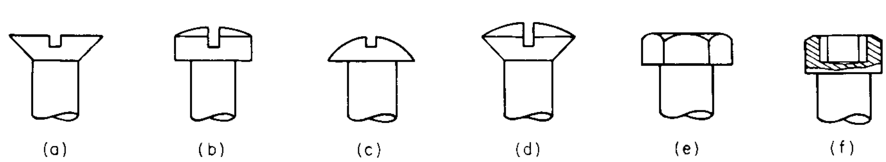


Fig. 8.2.8 Machine screw heads. (*a*) Flat; (*b*) fillister; (*c*) round; (*d*) oval; (*e*) hexagonal; (*f*) socket.

Table 8.2.14 Suggested Tap Drill Sizes for Internal Pipe Threads

		Taper pipe thread						Straight pipe thread			
		Minor diameter at distance		Drill for use without reamer		Drill for use with reamer		Minor diameter			
Size	Probable drill oversize cut (mean)	L_1 from large end	$L_1 + L_3$ from large end	Theoretical drill size*	Suggested drill size†	Theoretical drill size‡	Suggested drill size†	NPSF	NPSI	Theoretical drill size§	Suggested drill size†
	1	2	3	4	5	6	7	8	9	10	11
						Inch					
1/16–27	0.0038	0.2443	0.2374	0.2405	"C" (0.242)	0.2336	"A" (0.234)	0.2482	0.2505	0.2444	"D" (0.246)
1/32–27	0.0044	0.3367	0.3298	0.3323	"Q" (0.332)	0.3254	21/64 (0.328)	0.3406	0.3429	0.3362	"R" (0.339)
1/4–18	0.0047	0.4362	0.4258	0.4315	7/17 (0.438)	0.4211	27/64 (0.422)	0.4422	0.4457	0.4375	7/16 (0.438)
3/8–18	0.0049	0.5708	0.5604	0.5659	9/16 (0.562)	0.5555	9/16 (0.563)	0.5776	0.5811	0.5727	27/64 (0.578)
1/2–14	0.0051	0.7034	0.6901	0.6983	45/64 (0.703)	0.6850	11/16 (0.688)	0.7133	0.7180	0.7082	45/64 (0.703)
3/4–14	0.0060	0.9127	0.8993	0.9067	29/32 (0.906)	0.8933	57/64 (0.891)	0.9238	0.9283	0.9178	59/64 (0.922)
1–11 1/2	0.0080	1.1470	1.1307	1.1390	1 9/64 (1.141)	1.1227	1 1/8 (1.125)	1.1600	1.1655	1.1520	1 5/32 (1.156)
1 1/4–11 1/2	0.0100	1.4905	1.4742	1.4805	1 31/64 (1.484)	1.4642	1 15/32 (1.469)				
1 1/2–11 1/2	0.0120	1.7295	1.7132	1.7175	1 23/32 (1.719)	1.7012	1 45/64 (1.703)				
2–11 1/2	0.0160	2.2024	2.1861	2.1864	2 3/16 (2.188)	2.1701	2 11/64 (2.172)				
2 1/2–8	0.0180	2.6234	2.6000	2.6054	2 39/64 (2.609)	2.5820	2 37/64 (2.578)				
3–8	0.0200	3.2445	3.2211	3.2245	3 15/64 (3.234)	3.2011	3 13/64 (3.203)				
						Metric					
1/16–27	0.097	6.206	6.029	6.109	6.1	5.932	6.0	6.304	6.363	6.207	6.2
1/8–27	0.112	8.551	8.363	8.438	8.4	8.251	8.2	8.651	8.710	8.539	8.5
1/4–18	0.119	11.080	10.816	10.961	11.0	10.697	10.8	11.232	11.321	11.113	11.0
3/8–18	0.124	14.499	14.235	14.375	14.5	14.111	14.0	14.671	14.760	14.547	14.5
1/2–14	0.130	17.867	17.529	17.737	17.5	17.399	17.5	18.118	18.237	17.988	18.0
3/4–14	0.152	23.182	22.842	23.030	23.0	22.690	23.0	23.465	23.579	23.212	23.0
1–11 1/2	0.203	29.134	28.720	28.931	29.0	28.517	28.5	29.464	29.604	29.261	29.0
1 1/4–11 1/2	0.254	37.859	37.444	37.605	37.5	37.190	37.0				
1 1/2–11 1/2	0.305	43.929	43.514	43.624	43.5	43.209	43.5				
2–11 1/2	0.406	55.941	55.527	55.535	56.0	55.121	55.0				
2 1/2–8	0.457	66.634	66.029	66.177	66.0	65.572	65.0				
3–8	0.508	82.410	81.815	81.902	82.0	81.307	81.0				

* Column 4 values equal column 2 values minus column 1 values.
† Some drill sizes listed may not be standard drills, and in some cases, standard metric drill sizes may be closer to the theoretical inch drill size and standard inch drill sizes may be closer to the theoretical metric drill size.
‡ Column 6 values equal column 3 values minus column 1 values.
§ Column 10 values equal column 8 values minus column 1 values.
SOURCE: ANSI B1.20.3-1976 and ANSI B1.20.4-1976, reproduced by permission.

Table 8.2.15 Wrench Bolt Heads, Nuts, and Wrench Openings
(All dimensions in inches)

Basic or max width across flats, bolt heads, and nuts	Wrench openings Max	Wrench openings Min	Basic or max width across flats, bolt heads, and nuts	Wrench openings Max	Wrench openings Min	Basic or max width across flats, bolt heads, and nuts	Wrench openings Max	Wrench openings Min	Basic or max width across flats, bolt heads, and nuts	Wrench openings Max	Wrench openings Min
5/32	0.163	0.158	13/16	0.826	0.818	1 13/16	1.835	1.822	3	3.035	3.016
3/16	0.195	0.190	7/8	0.888	0.880	1 7/8	1.898	1.885	3 1/8	3.162	3.142
1/4	0.257	0.252	15/16	0.953	0.944	2	2.025	2.011	3 3/8	3.414	3.393
5/16	0.322	0.316	1	1.015	1.006	2 1/16	2.088	2.074	3 1/2	3.540	3.518
11/32	0.353	0.347	1 1/16	1.077	1.068	2 3/16	2.225	2.200	3 3/4	3.793	3.770
3/8	0.384	0.378	1 1/8	1.142	1.132	2 1/4	2.277	2.262	3 7/8	3.918	3.895
7/16	0.446	0.440	1 1/4	1.267	1.257	2 3/8	2.404	2.388	4 1/8	4.172	4.147
1/2	0.510	0.504	1 5/16	1.331	1.320	2 7/16	2.466	2.450	4 1/4	4.297	4.272
9/16	0.573	0.566	1 3/8	1.394	1.383	2 9/16	2.593	2.576	4 1/2	4.550	4.524
19/32	0.605	0.598	1 7/16	1.457	1.446	2 5/8	2.656	2.639	4 5/8	4.676	4.649
5/8	0.636	0.629	1 1/2	1.520	1.508	2 3/4	2.783	2.766	5	5.055	5.026
11/16	0.699	0.692	1 5/8	1.646	1.634	2 13/16	2.845	2.827	5 3/8	5.434	5.403
3/4	0.763	0.755	1 11/16	1.708	1.696	2 15/16	2.973	2.954	5 3/4	5.813	5.780
25/32	0.794	0.786							6 1/8	6.192	6.157

Wrenches shall be marked with the "nominal size of wrench" which is equal to the basic or maximum width across flats of the corresponding bolt head or nut.
Allowance (min clearance) between maximum width across flats of nut or bolt head and jaws of wrench equals $1.005W + 0.001$. Tolerance on wrench opening equals plus $0.005W + 0.004$ from minimum (W equals nominal size of wrench).

Table 8.2.16 Width Across Flats of Bolt Heads and Nuts
(All dimensions in inches)

Nominal size or basic major diam of thread	Dimensions of regular bolt heads unfinished, square, and hexagon		Dimensions of heavy bolt heads unfinished, square, and hexagon		Dimensions of cap-screw heads hexagon		Dimensions of setscrew heads		Dimensions of regular nuts and regular jam nuts, unfinished, square, and hexagon (jam nuts, hexagon only)		Dimensions of machine-screw and stove-bolt nuts, square and hexagon		Dimensions of heavy nuts and heavy jam nuts, unfinished, square, and hexagon (jam nuts, hexagon only)	
	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
No. 0	—	—	—	—	—	—	—	—	—	—	0.1562	0.150		
No. 1	—	—	—	—	—	—	—	—	—	—	0.1562	0.150		
No. 2	—	—	—	—	—	—	—	—	—	—	0.1875	0.180		
No. 3	—	—	—	—	—	—	—	—	—	—	0.1875	0.180		
No. 4	—	—	—	—	—	—	—	—	—	—	0.2500	0.241		
No. 5	—	—	—	—	—	—	—	—	—	—	0.3125	0.302		
No. 6	—	—	—	—	—	—	—	—	—	—	0.3125	0.302		
No. 8	—	—	—	—	—	—	—	—	—	—	0.3438	0.332		
No. 10	—	—	—	—	—	—	—	—	—	—	0.3750	0.362		
No. 12	—	—	—	—	—	—	—	—	—	—	0.4375	0.423		
¼	0.3750	0.362	—	—	0.4375	0.428	0.2500	0.241	0.4375	0.425	0.4375	0.423	0.5000	0.488
5⁄16	0.5000	0.484	—	—	0.5000	0.489	0.3125	0.302	0.5625	0.547	0.5625	0.545	0.5938	0.578
⅜	0.5625	0.544	—	—	0.5625	0.551	0.3750	0.362	0.6250	0.606	0.6250	0.607	0.6875	0.669
7⁄16	0.6250	0.603	—	—	0.6250	0.612	0.4375	0.423	0.7500	0.728	—	—	0.7812	0.759
½	0.7500	0.725	0.8750	0.850	0.7500	0.736	0.5000	0.484	0.8125	0.788	—	—	0.8750	0.850
9⁄16	0.8750	0.847	0.9375	0.909	0.8125	0.798	0.5625	0.545	0.8750	0.847	—	—	0.9375	0.909
⅝	0.9375	0.906	1.0625	1.031	0.8750	0.860	0.6250	0.606	1.0000	0.969	—	—	1.0625	1.031
¾	1.1250	1.088	1.2500	1.212	1.0000	0.983	0.7500	0.729	1.1250	1.088	—	—	1.2500	1.212
⅞	1.3125	1.269	1.4375	1.394	1.1250	1.106	0.8750	0.852	1.3125	1.269	—	—	1.4375	1.394
1	1.5000	1.450	1.6250	1.575	1.3125	1.292	1.0000	0.974	1.5000	1.450	—	—	1.6250	1.575
1⅛	1.6875	1.631	1.8125	1.756	1.5000	1.477	1.1250	1.096	1.6875	1.631	—	—	1.8125	1.756
1¼	1.8750	1.812	2.0000	1.938	1.6875	1.663	1.2500	1.219	1.8750	1.812	—	—	2.0000	1.938
1⅜	2.0625	1.994	2.1857	2.119	—	—	1.3750	1.342	2.0625	1.994	—	—	2.1875	2.119
1½	2.2500	2.175	2.3750	2.300	—	—	1.5000	1.464	2.2500	2.175	—	—	2.3750	2.300
1⅝	2.4375	2.356	2.5625	2.481	—	—	—	—	2.4375	2.356	—	—	2.5625	2.481
1¾	2.6250	2.538	2.7500	2.662	—	—	—	—	2.6250	2.538	—	—	2.7500	2.662
1⅞	2.8125	2.719	2.9375	2.844	—	—	—	—	2.8125	2.719	—	—	2.9375	2.844
2	3.0000	2.900	3.1250	3.025	—	—	—	—	3.0000	2.900	—	—	3.1250	3.025
2¼	3.3750	3.262	3.5000	3.388	—	—	—	—	3.3750	3.262	—	—	3.5000	3.388
2½	3.7500	3.625	3.8750	3.750	—	—	—	—	3.7500	3.625	—	—	3.8750	3.750
2¾	4.1250	3.988	4.2500	4.112	—	—	—	—	4.1250	3.988	—	—	4.2500	4.112
3	4.5000	4.350	4.6250	4.475	—	—	—	—	4.5000	4.350	—	—	4.6250	4.475
3¼	—	—	—	—	—	—	—	—	—	—	—	—	5.0000	4.838
3½	—	—	—	—	—	—	—	—	—	—	—	—	5.3750	5.200
3¾	—	—	—	—	—	—	—	—	—	—	—	—	5.7500	5.562
4	—	—	—	—	—	—	—	—	—	—	—	—	6.1250	5.925

Regular bolt heads are for general use. Unfinished bolt heads are not finished on any surface. Semifinished bolt heads are finished under head.
Regular nuts are for general use. Semifinished nuts are finished on bearing surface and threaded. Unfinished nuts are not finished on any surface but are threaded.

design in order that its properties may be fully utilized. These fasteners may be divided into six groups, as follows: seating lock, spring stop nut, interference, wedge, blind, and quick-release. The **seating-lock type** locks only when firmly seated and is therefore free-running on the bolt. The **spring stop-nut type** of fastener functions by a spring action clamping down upon the bolt. The **prevailing torque type** locks by elastic or plastic flow of a portion of the fastener material. A recent development employs an adhesive coating applied to the threads. The **wedge type** locks by relative wedging of either elements or nut and bolt. The **blind type** usually utilizes spring action of the fastener, and the **quick-release type** utilizes a quarter-turn release device. An example of each is shown in Fig. 8.2.12.

One such specification developed for prevailing torque fasteners by the Industrial Fasteners Institute is based on locking torque and may form a precedent for other types of fasteners as well.

Coach and lag screws find application in wood, or in masonry with an expansion anchor. Figure 8.2.13*a* shows two types, and Table 8.2.20 lists pertinent dimensions.

Wood screws [ANSI B18.22.1-1975 (R81)] are made in lengths from ¼ to 5 in for steel and from ¼ to 3½ in for brass screws, increasing by ⅛ in up to 1 in, by ¼ in up to 3 in, and by ½ in up to 5 in. Sizes are given in Table 8.2.21. Screws are made with flat, round, or oval heads. Figure 8.2.13*b* shows several heads.

Washers [ANSI B18.22.1-1975 (R81)] for bolts and lag screws, either round or square, are made to the dimensions given in Table 8.2.22. For other types of washers, see Fig. 8.2.14*a* and *b*.

Self-tapping screws are available in three types. **Thread-forming** tapping screws plastically displace material adjacent to the pilot hole. **Thread-cutting** tapping screws have cutting edges and chip cavities (flutes) and form a mating thread by removing material adjacent to the pilot hole. Thread-cutting screws are generally used to join thicker and harder materials and require a lower driving torque than thread-forming screws. **Metallic drive** screws are forced into the material by pressure and are intended for making permanent fastenings. These three types are further classified on the basis of thread and point form as shown in Table 8.2.23. In addition to these body forms, a number of different

Table 8.2.17 Head Diameters (Maximum), In

Machine screws						
Nominal size	Screw diam	Flat head	Round head	Fillister head	Oval head	Hexagonal head across flats
2	0.086	0.172	0.162	0.140	0.172	0.125
3	0.099	0.199	0.187	0.161	0.199	0.187
4	0.112	0.225	0.211	0.183	0.225	0.187
5	0.125	0.252	0.236	0.205	0.252	0.187
6	0.138	0.279	0.260	0.226	0.279	0.250
8	0.164	0.332	0.309	0.270	0.332	0.250
10	0.190	0.385	0.359	0.313	0.385	0.312
12	0.216	0.438	0.408	0.357	0.438	0.312
¼	0.250	0.507	0.472	0.414	0.507	0.375
5⁄16	0.3125	0.636	0.591	0.519	0.636	0.500
⅜	0.375	0.762	0.708	0.622	0.762	0.562
Cap screws						
Nominal size	Screw diam	Flat head	Button head	Fillister head	Socket head	
¼	0.250	½	7⁄16	⅜	⅜	
5⁄16	0.3125	⅝	9⁄16	7⁄16	7⁄16	
⅜	0.375	¾	⅝	9⁄16	9⁄16	
7⁄16	0.4375	13⁄16	¾	⅝	⅝	
½	0.500	⅞	13⁄16	¾	¾	
9⁄16	0.5625	1	15⁄16	13⁄16	13⁄16	
⅝	0.625	1⅛	1	⅞	⅞	
¾	0.750	1⅜	1¼	1	1	
⅞	—	—	—	1⅛	1⅛	
1	—	—	—	15⁄16	15⁄16	

head types are available. Basic dimensional data are given in Table 8.2.24.

Carriage bolts have been standardized in ANSI B18.5-1971, revised 1990. They come in styles shown in Fig. 8.2.15. The range of bolt diameters is no. 10 (= 0.19 in) to 1 in, no. 10 to ¾ in, no. 10 to ½ in, and no. 10 to ¾ in, respectively.

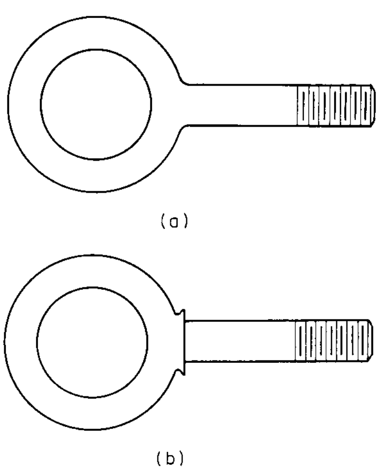


Fig. 8.2.9 Eyebolts.

Materials, Strength, and Service Adaptability of Bolts and Screws Materials

Table 8.2.25 shows the relationship between selected metric bolt classes and SAE and ASTM grades. The first number of a metric bolt class equals the minimum tensile strength (ultimate) in megapascals (MPa) divided by 100, and the second number is the approximate ratio between minimum yield and minimum ultimate strengths.

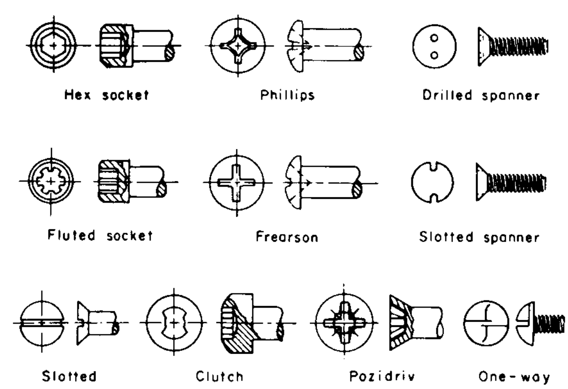


Fig. 8.2.10 Driving recesses. *(Adapted, with permission, from Machine Design.)*

EXAMPLE. Class 5.8 has a minimum ultimate strength of approximately 500 MPa and a minimum yield strength approximately 80 percent of minimum ultimate strength.

Strength The fillet between head and body, the thread runout point, and the first thread to engage the nut all create stress concentrations causing local stresses much greater than the average tensile stress in the bolt body. The complexity of the stress patterns renders ineffective the ordinary design calculations based on yield or ultimate stresses. Bolt strengths are therefore determined by laboratory tests on bolt-nut assemblies and published as **proof loads.** Fastener manufacturers are required to periodically repeat such tests to ensure that their products meet the original standards.

In order that a bolted joint remain firmly clamped while carrying its external load P, the bolt must be tightened first with sufficient torque to induce an initial tensile preload F_i. The total load F_B experienced by the bolt is then $F_B = F_i + \varepsilon P$. The fractional multiplier ε is given by $\varepsilon =$

Table 8.2.18 Regular Nut Eyebolts—Selected Sizes
(Thomas Laughlin Co., Portland, Me.)
(All dimensions in inches)
(See Fig. 8.2.9.)

Diam and shank length	Thread length	Eye dimension ID	Eye dimension OD	Approx breaking strength, lb	Diam and shank length	Thread length	Eye dimension ID	Eye dimension OD	Approx breaking strength, lb
¼ × 2	1½	½	1	2,200	¾ × 6	3	1½	3	23,400
5⁄16 × 2¼	1½	⅝	1¼	3,600	¾ × 10	3	1½	3	23,400
⅜ × 4½	2½	¾	1½	5,200	¾ × 15	5	1½	3	23,400
½ × 3¼	1½	1	2	9,800	⅞ × 8	4	1¾	3½	32,400
½ × 6	3	1	2	9,800	1 × 6	3	2	4	42,400
½ × 10	3	1	2	9,800	1 × 9	4	2	4	42,400
⅝ × 4	2	1¼	2½	15,800	1 × 18	7	2	4	42,400
⅝ × 6	3	1¼	2½	15,800	1¼ × 8	4	2½	5	67,800
⅝ × 10	3	1¼	2½	15,800	1¼ × 20	6	2½	5	67,800

Table 8.2.19 Cup-Point Setscrew Holding Power

Nominal screw size	Seating torque, lb·in	Axial holding power, lb
No. 0	0.5	50
No. 1	1.5	65
No. 2	1.5	85
No. 3	5	120
No. 4	5	160
No. 5	9	200
No. 6	9	250
No. 8	20	385
No. 10	33	540
¼ in	87	1,000
5⁄16 in	165	1,500
⅜ in	290	2,000
7⁄16 in	430	2,500
½ in	620	3,000
9⁄16 in	620	3,500
⅝ in	1,225	4,000
¾ in	2,125	5,000
⅞ in	5,000	6,000
1 in	7,000	7,000

NOTES: 1. Torsional holding power in inch-pounds is equal to one-half of the axial holding power times the shaft diameter in inches.
2. Experimental data were obtained by seating an alloy-steel cup-point setscrew against a steel shaft with a hardness of Rockwell C 15. Screw threads were class 3A, tapped holes were class 2B. Holding power was defined as the minimum load necessary to produce 0.01 in of relative movement between the shaft and the collar.
3. Cone points will develop a slightly greater holding power; flat, dog, and oval points, slightly less.
4. Shaft hardness should be at least 10 Rockwell C points less than the setscrew point.
5. Holding power is proportional to seating torque. Torsional holding power is increased about 6% by use of a flat on the shaft.
6. Data by F. R. Kull, Fasteners Book Issue, *Mach. Des.*, Mar. 11, 1965.

$K_B/(K_B + K_M)$, where K_B = elastic constant of the bolt and $1/K_M = 1/K_N + 1/K_W + 1/K_G + 1/K_J$. K_N = elastic constant of the nut; K_W = elastic constant of the washer; K_G = elastic constant of the gasket; K_J = elastic constant of the clamped surfaces or joint.

By manipulation, the fractional multiplier can be written $\varepsilon = 1/(1 + K_M/K_B)$. When K_M/K_B approaches 0, $\varepsilon \rightarrow 1$. When K_M/K_B approaches infinity, $\varepsilon \rightarrow 0$. Generally, K_N, K_W, and K_J are much stiffer than K_B, while K_G can vary from very soft to very stiff. In a metal-to-metal joint, K_G is effectively infinity, which causes K_M to approach infinity and ε to approach 0. On the other hand, for a very soft gasketed joint, $K_M \rightarrow 0$ and $\varepsilon \rightarrow 1$. For a metal-to-metal joint, then, $F_B = F_i + 0 \times P = F_i$; thus no fluctuating load component enters the bolt. In that case, the bolt remains at *static force* F_i at all times, and the static design will suffice. For a very soft gasketed joint, $F_B = F_i + 1 \times P = F_i + P$, which means that if P is a dynamically fluctuating load, it will be superimposed onto the static value of F_i. Accordingly, one must use the *fatigue design* for the bolt. Of course, for conditions between $\varepsilon = 0$ and $\varepsilon = 1$, the load within the bolt body is $F_B = F_i + \varepsilon P$, and again the *fatigue design* must be used.

In general, one wants as much preload as a bolt and joint will tolerate, without damaging the clamped parts, encouraging stress corrosion, or reducing fatigue life. For ungasketed, unpressurized joints under static loads using high-quality bolt materials, such as SAE 3 or better, the preload should be about 90 percent of proof load.

The **proof strength** is the stress obtained by dividing **proof load** by **stress area.** Stress area is somewhat larger than the root area and can be found in thread tables, or calculated approximately from a diameter which is the mean of the root and pitch diameters.

Initial sizing of bolts can be made by calculating area = (% × proof load)/(proof strength). See Table 8.2.26 for typical physical properties.

NOTE. In European practice, proof stress of a given grade is independent of diameter and is accomplished by varying chemical composition with diameters.

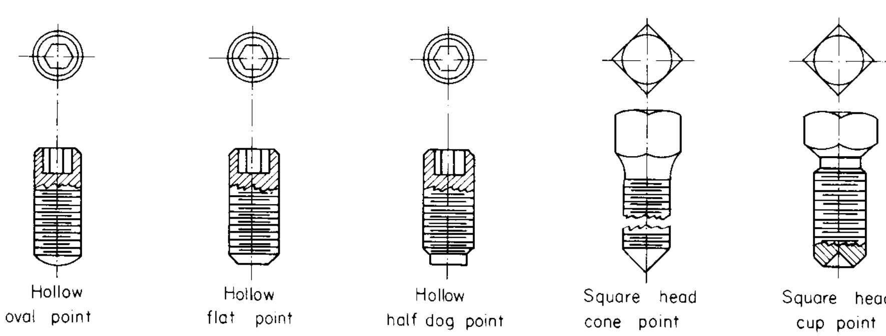


Fig. 8.2.11 Setscrews.

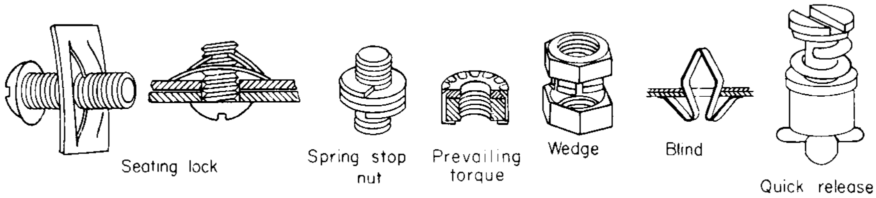


Fig. 8.2.12 Locking fasteners.

General Notes on the Design of Bolted Joints

Bolts subjected to shock and sudden change in load are found to be more serviceable when the unthreaded portion of the bolt is turned down or drilled to the area of the root of the thread. The drilled bolt is stronger in torsion than the turned-down bolt.

When a **number of bolts** are **employed in fastening** together two parts of a machine, such as a cylinder and cylinder head, the load carried by each bolt depends on its relative tightness, the tighter bolts carrying the greater loads. When the conditions of assembly result in differences in tightness, lower working stresses must be used in designing the bolts than otherwise are necessary. On the other hand, it may be desirable to have the bolts the weakest part of the machine, since their breakage from overload in the machine may result in a minimum replacement cost. In such cases, the breaking load of the bolts may well be equal to the load which causes the weakest member of the machine connected to be stressed up to the elastic limit.

Table 8.2.20 Coach and Lag Screws

Diam of screw, in	¼	5⁄16	⅜	7⁄16	½	⅝	¾	⅞	1
No. of threads per inch	10	9	7	7	6	5	4½	4	3½
Across flats of hexagon and square heads, in	⅜	15⁄32	9⁄16	21⁄32	¾	15⁄16	1⅛	15⁄16	1½
Thickness of hexagon and square heads, in	3⁄16	¼	5⁄16	⅜	7⁄16	17⁄32	⅝	¾	⅞

Length of threads for screws of all diameters

Length of screw, in	1½	2	2½	3	3½	4	4½
Length of thread, in	To head	1½	2	2¼	2½	3	3½
Length of screw, in	5	5½	6	7	8	9	10–12
Length of thread, in	4	4	4½	5	6	6	7

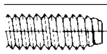
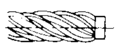
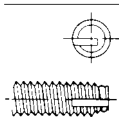
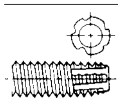
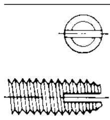
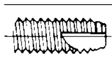
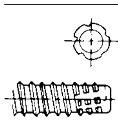
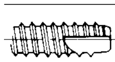
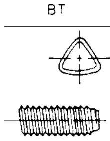
Table 8.2.21 American National Standard Wood Screws

Number	0	1	2	3	4	5	6	7	8
Threads per inch	32	28	26	24	22	20	18	16	15
Diameter, in	0.060	0.073	0.086	0.099	0.112	0.125	0.138	0.151	0.164
Number	9	10	11	12	14	16	18	20	24
Threads per inch	14	13	12	11	10	9	8	8	7
Diameter, in	0.177	0.190	0.203	0.216	0.242	0.268	0.294	0.320	0.372

Table 8.2.22 Dimensions of Steel Washers, in

	Plain washer			Lock washer		
Bolt size	Hole diam	OD	Thickness	Hole diam	OD	Thickness
3⁄16	¼	9⁄16	3⁄64	0.194	0.337	0.047
¼	5⁄16	¾	1⁄16	0.255	0.493	0.062
5⁄16	⅜	⅞	1⁄16	0.319	0.591	0.078
⅜	7⁄16	1	5⁄64	0.382	0.688	0.094
7⁄16	½	1¼	5⁄64	0.446	0.781	0.109
½	9⁄16	1⅜	3⁄32	0.509	0.879	0.125
9⁄16	⅝	1½	3⁄32	0.573	0.979	0.141
⅝	11⁄16	1¾	⅛	0.636	1.086	0.156
¾	13⁄16	2	⅛	0.763	1.279	0.188
⅞	15⁄16	2¼	5⁄32	0.890	1.474	0.219
1	11⁄16	2½	5⁄32	1.017	1.672	0.250
1⅛	1¼	2¾	5⁄32	1.144	1.865	0.281
1¼	1⅜	3	5⁄32	1.271	2.058	0.312
1⅜	1½	3¼	11⁄64	1.398	2.253	0.344
1½	1⅝	3½	11⁄64	1.525	2.446	0.375
1⅝	1¾	3¾	11⁄64			
1¾	1⅞	4	11⁄64			
1⅞	2	4¼	11⁄64			
2	2⅛	4½	11⁄64			
2¼	2⅜	4¾	3⁄16			
2½	2⅝	5	7⁄32			

Table 8.2.23 Tapping Screw Forms

ASA type and thread form	Description and recommendations
 AB	Spaced thread, with gimlet point, designed for use in sheet metal, resin-impregnated plywood, wood, and asbestos compositions. Used in pierced or punched holes where a sharp point for starting is needed.
 B	Type B is a blunt-point spaced-thread screw, used in heavy-gage sheet-metal and nonferrous castings.
 BP	Same as type B, but has a 45 deg included angle unthreaded cone point. Used for locating and aligning holes or piercing soft materials.
 C	Blunt point with threads approximating machine-screw threads. For applications where a machine-screw thread is preferable to the spaced-thread form. Unlike thread-cutting screws, type C makes a chip-free assembly.
 U	Multiple-threaded drive screw with steep helix angle and a blunt, unthreaded starting pilot. Intended for making permanent fastenings in metals and plastics. Hammered or mechanically forced into work. Should not be used in materials less than one screw diameter thick.
 D	Blunt point with single narrow flute and threads approximating machine-screw threads. Flute is designed to produce a cutting edge which is radial to screw center. For low-strength metals and plastics; for high-strength brittle metals; and for rethreading clogged pretapped holes.
 F	Approximate machine-screw thread and blunt point.
 G	Approximate machine-screw thread with single through slot which forms two cutting edges. For low-strength metals and plastics.
 T	Same as type D with single wide flute for more chip clearance.
 BF	Spaced thread with blunt point and five evenly spaced cutting grooves and chip cavities. Wall thickness should be 1½ times major diameter of screw. Reduces stripping in brittle plastics and die castings.
 BT	Same as type BF except for single wide flute which provides room for twisted, curly chips.
 TT	Thread-rolling screws roll-form clean, screw threads. The plastic movement of the material it is driven into locks it in place. The Taptite form is shown here.

SOURCE: *Mach. Des.*, Mar. 11, 1965.

Bolts screwed up tight have an initial stress due to the tightening (preload) before any external load is applied to the machine member. The initial tensile load due to screwing up for a tight joint varies approximately as the diameter of the bolt, and may be estimated at 16,000 lb/in of diameter. The actual value depends upon the applied torque and the efficiency of the screw threads. Applying this rule to bolts of 1-in diam or less results in excessively high stresses, thus demonstrating why bolts of small diameter frequently fail during assembly. It is advisable to use as large-diameter bolts as possible in pressure-tight joints requiring high tightening loads.

In pressure-tight joints without a gasket the force on the bolt under load is essentially never greater than the initial tightening load. When a gasket is used, the total bolt force is approximately equal to the initial tightening load plus the external load. In the first case, deviations from the rule are a result of elastic behavior of the joint faces without a gasket, and inelastic behavior of the gasket in the latter case. The fol-

Table 8.2.24 Self-Tapping Screws

Screw size	Basic major diam, in	Threads per inch: AB	B, BP	C	D, F, G, T	BF, BT	Type U: Max outside diam, in	Number of thread starts
00	—	—	—	—	—	—	0.060	6
0	0.060	40	48	—	—	48	0.075	6
1	0.073	32	42	—	—	42		
2	0.086	32	32	56 and 64	56 and 64	32	0.100	8
3	0.099	28	28	48 and 56	48 and 56	28		
4	0.112	24	24	40 and 48	40 and 48	24	0.116	7
5	0.125	20	20	40 and 44	40 and 44	20		
6	0.138	18	20	32 and 40	32 and 40	20	0.140	7
7	0.151	16	19	—	—	19	0.154	8
8	0.164	15	18	32 and 36	32 and 36	18	0.167	8
10	0.190	12	16	24 and 32	24 and 32	16	0.182	8
12	0.216	11	14	24 and 28	24 and 28	14	0.212	8
14	0.242	10	—	—	—	—	0.242	9
¼	0.250	—	14	20 and 28	20 and 28	14		
16	0.268	10						
18	0.294	9						
5⁄16	0.3125	—	12	18 and 24	18 and 24	12	0.315	14
20	0.320	9						
24	0.372	9						
⅜	0.375	—	12	16 and 24	16 and 24	12	0.378	12
7⁄16	0.4375	—	10	—	—	10		
½	0.500	—	10	—	—	10		

lowing generalization will serve as a guide. If the bolt is more yielding than the connecting members, it should be designed simply to resist the initial tension or the external load, whichever is greater. If the probable yielding of the bolt is 50 to 100 percent of that of the connected members, take the resultant bolt load as the initial tension plus one-half the external load. If the yielding of the connected members is probably four to five times that of the bolt (as when certain packings are used), take the resultant bolt load as the initial tension plus three-fourths the external load.

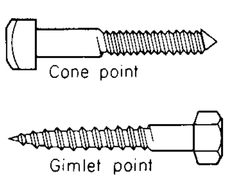


Fig. 8.2.13*a* Coach and lag screws.

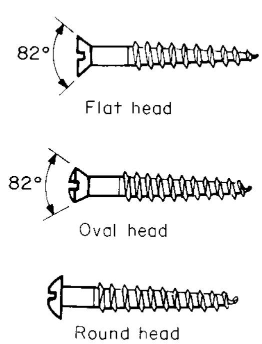


Fig. 8.2.13*b* Wood screws.

Table 8.2.25 ISO Metric Fastener Materials

Metric bolt	Metric nut class normally used	Roughly equivalent U.S. bolt materials: SAE J429 grades	ASTM grades
4.6	4 or 5	1	A193, B8; A307, grade A
4.8	4 or 5	2	
5.8	5	2	
8.8	8	5	A325, A449
9.9	9	5+	A193, B7 and B16
10.9	10 or 12	8	A490; A354, grade 8D
12.9	10 or 12		A540; B21 through B24

SOURCE: Bickford, ''An Introduction to the Design and Behavior of Bolted Joints,'' Marcel Dekker, 1981; reproduced by permission. See Appendix G for additional metric materials.

In cases where bolts are subjected to cyclic loading, an increase in the initial tightening load decreases the operating stress range. In certain applications it is customary to fix the tightening load as a fraction of the yield-point load of the bolt.

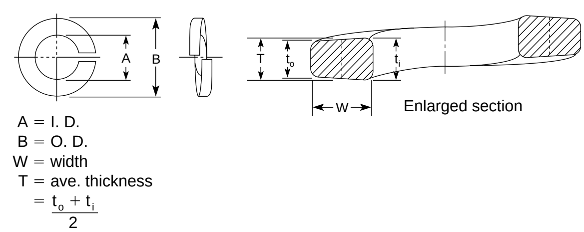


Fig. 8.2.14*a* Regular helical spring lock washer.

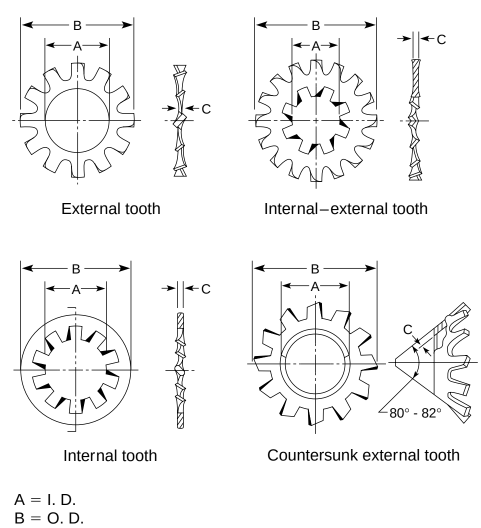


Fig. 8.2.14*b* Toothed lock washers.

Table 8.2.26*a* Specifications and Identification Markings for Bolts, Screws, Studs, Sems,[a] and U Bolts[b]
(Multiply the strengths in kpsi by 6.89 to get the strength in MPa.)

SAE grade	ASTM grade	Metric[c] grade	Nominal diameter, in	Proof strength, kpsi	Tensile strength, kpsi	Yield[d] strength, kpsi	Core hardness, Rockwell min/max	Products[e]
1	A307	4.6	¼ thru 1½	33	60	36	B70/B100	B, Sc, St
2		5.8	¼ thru ¾	55	74	57	B80/B100	B, Sc, St
		4.6	Over ¾ thru 1½	33	60	36	B70/B100	B, Sc, St
4		8.9	¼ thru 1½	65[f]	115	100	C22/C32	St
5	A449 or A325 type 1	8.8	¼ thru 1	85	120	92	C25/C34	B, Sc, St
		7.8	Over 1 thru 1½	74	105	81	C19/C30	B, Sc, St
		8.6	Over 1½ to 3	55	90	58		B, Sc, St
5.1		8.8	No. 6 thru ⅝	85	120		C25/C40	Se
		8.8	No. 6 thru ½	85	120		C25/C40	B, Sc, St
5.2	A325 type 2	8.8	¼ thru 1	85	120	92	C26/C36	B, Sc
7[g]		10.9	¾ thru 1½	105	133	115	C28/C34	B, Sc
8	A354 Grade BD	10.9	¼ thru 1½	120	150	130	C33/C39	B, Sc, St
8.1		10.9	¼ thru 1½	120	150	130	C32/C38	St
8.2		10.9	¼ thru 1	120	150	130	C35/C42	B, Sc
	A574	12.9	0 thru ½	140	180	160	C39/C45	SHCS
		12.9	⅝ thru 1½	135	170	160	C37/C45	SHCS

NOTE: Company catalogs should be consulted regarding *proof loads.* However, approximate values of proof loads may be calculated from: proof load = proof strength × stress area.
[a] Sems are screw and washer assemblies.
[b] Compiled from ANSI/SAE J429j; ANSI B18.3.1-1978; and ASTM A307, A325, A354, A449, and A574.
[c] Metric grade is *xx.x* where *xx* is approximately 0.01 S_{ut} in MPa and *.x* is the ratio of the minimum S_y to S_{ut}.
[d] Yield strength is stress at which a permanent set of 0.2% of gage length occurs.
[e] B = bolt, Sc = Screws, St = studs, Se = sems, and SHCS = socket head cap screws.
[f] Entry appears to be in error but conforms to the standard, ANSI/SAE J429j.
[g] Grade 7 bolts and screws are roll-threaded after heat treatment.
SOURCE: Shigley and Mitchell, ''Mechanical Engineering Design,'' 4th ed., McGraw-Hill, 1983, by permission.

In order to avoid the possibility of bolt failure in pressure-tight joints and to obtain uniformity in bolt loads, some means of determining initial bolt load **(preload)** is desirable. Calibrated torque wrenches are available for this purpose, reading directly in inch-pounds or inch-ounces. Inaccuracies in initial bolt load are possible even when using a torque wrench, owing to variations in the coefficient of friction between the nut and the bolt and, further, between the nut or bolthead and the abutting surface.

An exact method to determine the preload in a bolt requires that the bolt elongation be measured. For a through bolt in which both ends are accessible, the elongation is measured, and the preload force P is obtained from the relationship

$$P = AEe \div l$$

where E = modulus of elasticity, l = original length, A = cross-sectional area, e = elongation. In cases where both ends of the bolt are not accessible, strain-gage techniques may be employed to determine the strain in the bolt, and thence the preload.

High-strength bolts designated ASTM A325 and A490 are almost exclusively employed in the assembly of structural steel members, but they are applied in mechanical assemblies such as flanged joints. The direct tension indicator (DTI) (Fig. 8.2.16), for use with high-strength bolts, allows **bolt preload** to be applied rapidly and simply. The device is a hardened washer with embossed protrusions (Fig. 8.2.16*a*). Tightening the bolt causes the protrusions to flatten and results in a decrease in the gap between washer and bolthead. The prescribed degree of bolt-tightening load, or preload, is obtained when the gap is reduced to a predetermined amount (Fig. 8.2.16*c*). A feeler gage of a given thickness is used to determine when the gap has been closed to the prescribed amount (Fig. 8.2.16*b* and *c*). With a paired bolt and DTI, the degree of gap closure is proportional to bolt preload. The system is reported to provide bolt-preload force accuracy within +15 percent of that prescribed (Fig. 8.2.16*d*). The devices are available in both inch and metric series and are covered under ASTM F959 and F959M.

Preload-indicating bolts and nuts provide visual assurance of preload in that tightening to the desired preload causes the wavy flange to flatten flush with the clamped assembly (Fig. 8.2.17).

In drilling and tapping cast iron for steel studs, it is necessary to tap to a depth equal to 1½ times the stud diameter so that the strength of the cast-iron threads in shear may equal the tensile strength of the stud. Drill sizes and depths of hole and thread are given in Table 8.2.27.

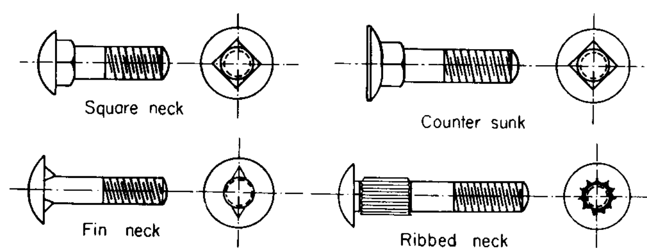


Fig. 8.2.15 Carriage bolts.

It is not good practice to drill holes to be tapped through the metal into pressure spaces, for even though the bolt fits tightly, leakage will result that is difficult to eliminate.

Screw thread inserts made of high-strength material (Fig. 8.2.18) are useful in many cases to provide increased thread strength and life. Soft or ductile materials tapped to receive thread inserts exhibit improved load-carrying capacity under static and dynamic loading conditions. Holes in which threads have been stripped or otherwise damaged can be restored through the use of thread inserts.

Holes for thread inserts are drilled oversize and specially tapped to receive the insert selected to mate with the threaded fastener used. The standard material for inserts is 18-8 stainless steel, but other materials are available, such as phosphor bronze and Inconel. Recommended insert lengths are given in Table 8.2.28.

Drill Sizes Unified thread taps are listed in Table 8.2.29.

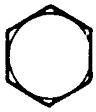
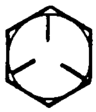
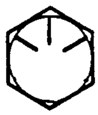
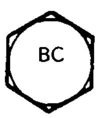
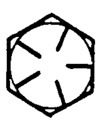
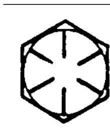
RIVET FASTENINGS

Forms and Proportion of Rivets The forms and proportions of small and large rivets have been standardized and conform to ANSI B18.1.1-1972 (R89) and B18.1.2-1972 (R89) (Figs. 8.2.19*a* and *b*).

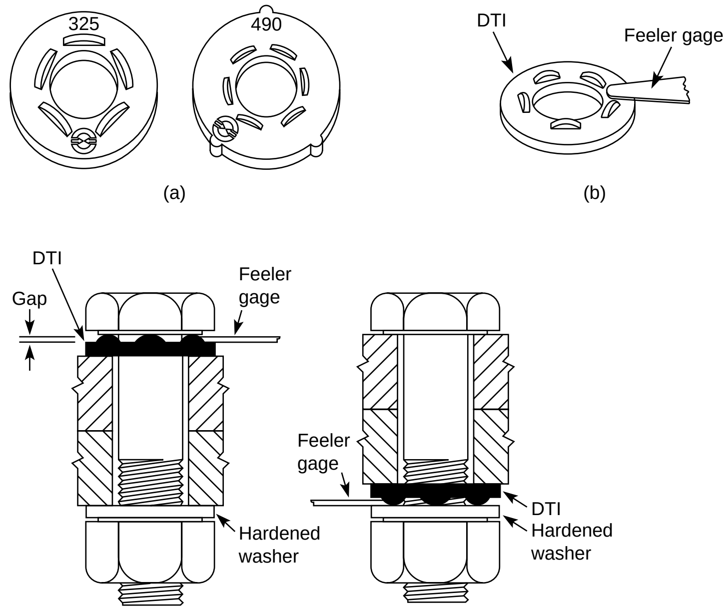
Materials Specifications for Rivets and Plates See Sec. 6 and 12.2.

Conventional signs to indicate the form of the head to be used and

Table 8.2.26*b* ASTM and SAE Grade Head Markings for Steel Bolts and Screws

Grade marking	Specification	Material
 No mark	SAE grade 1	Low- or medium-carbon steel
	ASTM A307	Low-carbon steel
	SAE grade 2	Low- or medium-carbon steel
	SAE grade 5	Medium-carbon steel, quenched and tempered
	ASTM A449	
	SAE grade 5.2	Low-carbon martensite steel, quenched and tempered
	ASTM A325 type 1	Medium-carbon steel, quenched and tempered; radial dashes optional
	ASTM A325 type 2	Low-carbon martensite steel, quenched and tempered
	ASTM A325 type 3	Atmospheric corrosion (weathering) steel, quenched and tempered
	ASTM A354 grade BC	Alloy steel, quenched and tempered
	SAE grade 7	Medium-carbon alloy steel, quenched and tempered, roll-threaded after heat treatment
	SAE grade 8	Medium-carbon alloy steel, quenched and tempered
	ASTM A354 grade BD	Alloy steel, quenched and tempered
	SAE grade 8.2	Low-carbon martensite steel, quenched and tempered
	ASTM A490 type 1	Alloy steel, quenched and tempered
	ASTM A490 type 3	Atmospheric corrosion (weathering) steel, quenched and tempered

SOURCE: ANSI B18.2.1–1981 (R92), Appendix III, p. 41. By permission.



DTI Gaps To Give Required Minimum Bolt Tension

DTI Fitting	325	490
Under bolt head Plain finish DTIs	0.015″ (.40 mm)	0.015″
Mechanically galvanized DTIs	0.005″ (.125 mm)	—
Epoxy coated on mechanically galvanized DTIs	0.005″ (.125 mm)	—
Under turned Element With hardened washers (plain finish)	0.005″ (.125 mm)	0.005″

With average gaps equal or less than above, bolt tensions will be greater than in adjacent listing

Minimum Bolt Tensions
In thousands of pounds (Kips)

Bolt dia.	A325	A490
1/2″	12	—
5/8″	19	—
3/4″	28	35
7/8″	39	49
1 ″	51	64
1 1/8″	56	80
1 1/4″	71	102
1 3/8″	85	121
1 1/2″	103	—

(d)

Fig. 8.2.16 Direct tension indicators; gap and tension data. *(Source: J & M Turner Inc.)*

whether the rivet is to be driven in the shop or the field at the time of erection are given in Fig. 8.2.20. **Rivet lengths** and **grips** are shown in Fig. 8.2.19*b*.

For **structural riveting,** see Sec. 12.2.

Punched vs. Drilled Plates Holes in plates forming parts of riveted structures are punched, punched and reamed, or drilled. Punching, while

Fig. 8.2.17 Load-indicating wavy-flange bolt (or nut).

cheaper, is objectionable. The holes in different plates cannot be spaced with sufficient accuracy to register perfectly on being assembled. If the hole is punched out, say 1⁄16 in smaller than is required and then reamed to size, the metal injury by cold flow during punching will be removed. Drilling, while more expensive, is more accurate and does not injure the metal.

Tubular Rivets

In tubular rivets, the end opposite the head is made with an axial hole (partway) to form a thin-walled, easily upsettable end. As the material at the edge of the rivet hole is rolled over against the surface of the joint, a **clinch** is formed (Fig. 8.2.21*a*).

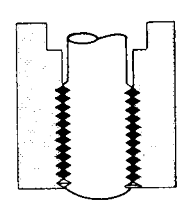


Fig. 8.2.18 Screw thread insert.

Two-part tubular rivets have a thin-walled head with attached thin-walled rivet body and a separate thin-walled expandable plug. The head-body is inserted through a hole in the joint from one side, and the plug from the other. By holding an anvil against the plug bottom and

Table 8.2.27 Depths to Drill and Tap Cast Iron for Studs

Diam of stud, in	¼	5/16	⅜	7/16	½	9/16	⅝	¾	⅞	1
Diam of drill, in	13/64	17/64	5/16	⅜	27/64	31/64	17/32	41/64	¾	55/64
Depth of thread, in	⅜	15/32	9/16	21/32	¾	27/32	15/16	1⅛	1 5/16	1½
Depth to drill, in	7/16	17/32	⅝	23/32	27/32	15/16	1 1/32	1¼	1 7/16	1⅝

Table 8.2.28 Screw-Thread Insert Lengths
(Heli-Coil Corp.)

Shear strength of parent material, lb/in²	Bolt material ultimate tensile strength, lb/in²				
	60,000	90,000	125,000	170,000	220,000
	Length in terms of nominal insert diameter				
15,000	1½	2	2½	3	
20,000	1	1½	2	2½	3
25,000	1	1½	2	2	2½
30,000	1	1	1½	2	2
40,000	1	1	1½	1½	2
50,000	1	1	1	1½	1½

hammering on the head, the plug is caused to expand within the head, thus locking both parts together (Fig. 8.2.21*a*).

Blind Rivets

Blind rivets are inserted and set all from one side of a structure. This is accomplished by mechanically expanding, through the use of the rivet's built-in mandrel, the back (blind side) of the rivet into a bulb or upset head after insertion. Blind rivets include the *pull type* and *drive-pin type*.

The pull-type rivet is available in two configurations: a self-plugging type and a pull-through type. In the self-plugging type, part of the mandrel remains permanently in the rivet body after setting, contributing additional shear strength to the fastener. In the pull-through type, the entire mandrel is pulled through, leaving the installed rivet empty.

In a drive-pin rivet, the rivet body is slotted. A pin is driven forward into the rivet, causing both flaring of the rivet body and upset of the blind side (Fig. 8.2.21*b*).

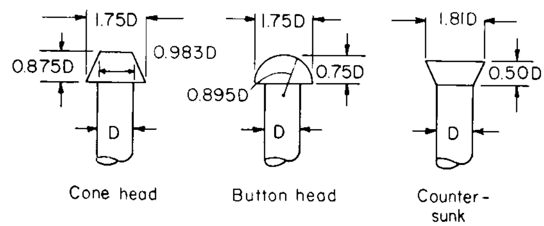


Fig. 8.2.19*a* Rivet heads.

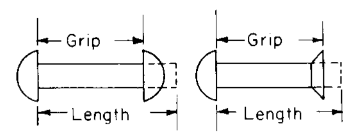


Fig. 8.2.19*b* Rivet length and grip.

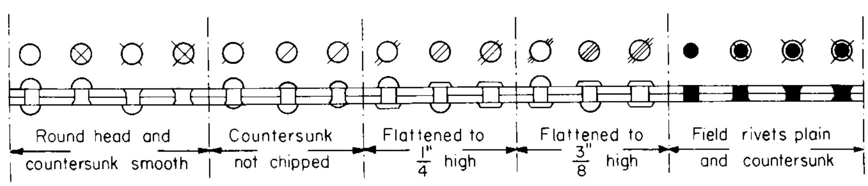


Fig. 8.2.20 Conventional signs for rivets.

Table 8.2.29 Tap-Drill Sizes for American National Standard Screw Threads
(The sizes listed are the commercial tap drills to produce approx 75% full thread)

Size	Coarse-thread series: Threads per inch	Coarse-thread series: Tap drill size	Fine-thread series: Threads per inch	Fine-thread series: Tap drill size	Size	Coarse-thread series: Threads per inch	Coarse-thread series: Tap drill size	Fine-thread series: Threads per inch	Fine-thread series: Tap drill size
No. 0	—	—	80	3/64	¾	10	21/32	16	11/16
No. 1	64	No. 53	72	No. 53	⅞	9	49/64	14	13/16
No. 2	56	No. 50	64	No. 50	1	8	⅞	14	15/16
No. 3	48	No. 47	56	No. 45	1⅛	7	63/64	12	1 3/64
No. 4	40	No. 43	48	No. 42	1¼	7	1 7/64	12	1 11/64
No. 5	40	No. 38	44	No. 37	1⅜	6	1 7/32	12	1 19/64
No. 6	32	No. 36	40	No. 33	1½	6	1 21/64	12	1 27/64
No. 8	32	No. 29	36	No. 29	1¾	5	1 35/64		
No. 10	24	No. 25	32	No. 21	2	4½	1 25/32		
No. 12	24	No. 16	28	No. 14	2¼	4½	2 1/32		
¼	20	No. 7	28	No. 3	2½	4	2¼		
5/16	18	F	24	I	2¾	4	2½		
⅜	16	5/16	24	Q	3	4	2¾		
7/16	14	U	20	25/64	3¼	4	3		
½	13	27/64	20	29/64	3½	4	3¼		
9/16	12	31/64	18	33/64	3¾	4	3½		
⅝	11	17/32	18	37/64	4	4	3¾		

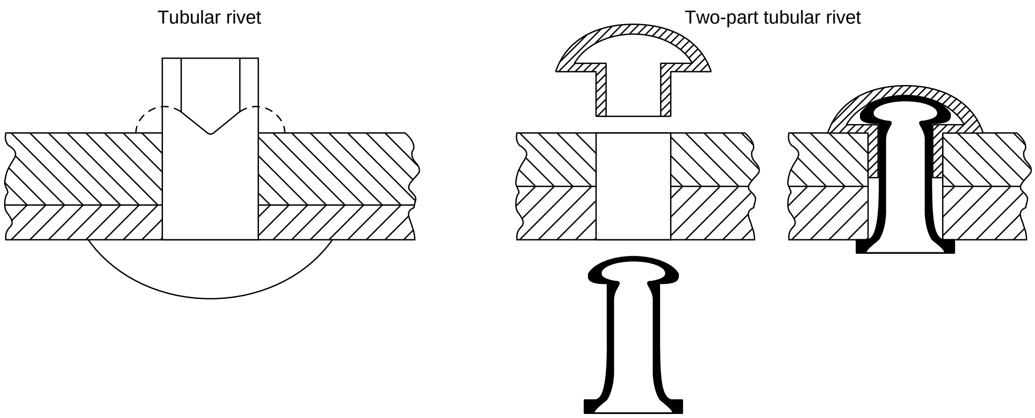


Fig. 8.2.21*a* Tubular rivets.

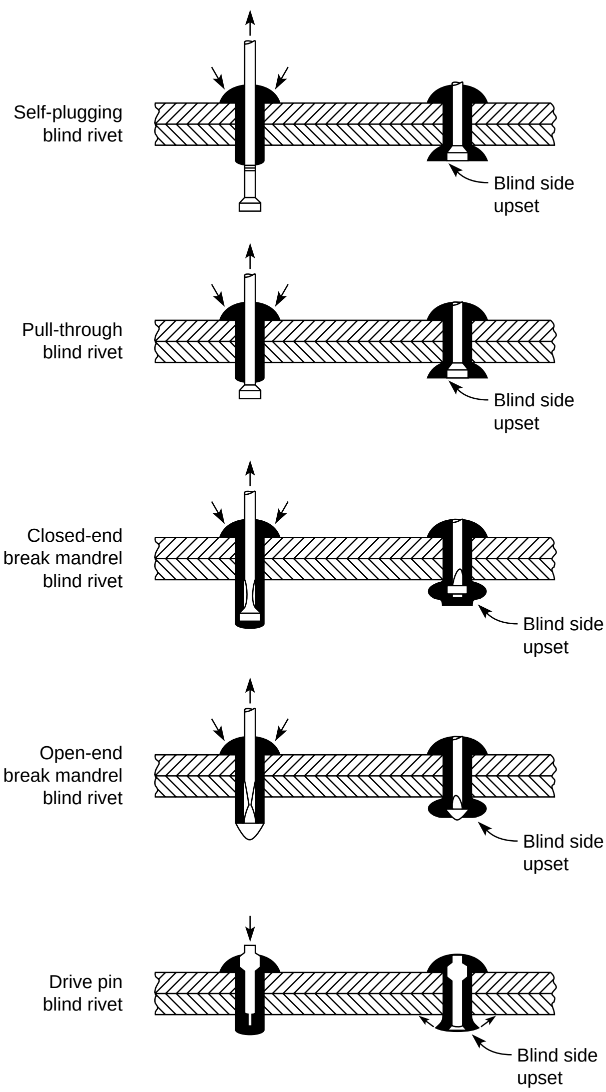


Fig. 8.2.21*b* Blind rivets.

KEYS, PINS, AND COTTERS

Keys and key seats have been standardized and are listed in ANSI B17.1-1967 (R89). Descriptions of the principal key types follow.

Woodruff keys [ANSI B17.2-1967 (R90)] are made to facilitate removal of pulleys from shafts. They should not be used as sliding keys. Cutters for milling out the key seats, as well as special machines for using the cutters, are to be had from the manufacturer. Where the hub of the gear or pulley is relatively long, two keys should be used. Slightly rounding the corners or ends of these keys will obviate any difficulty met with in removing pulleys from shafts. The key is shown in Fig. 8.2.22 and the dimensions in Table 8.2.30.

Square and flat plain taper keys have the same dimensions as gib-head keys (Table 8.2.31) up to the dotted line of Fig. 8.2.23. **Gib-head keys** (Fig. 8.2.23) are necessary when the smaller end is inaccessible for drifting out and the larger end is accessible. It can be used, with care, with all sizes of shafts. Its use is forbidden in certain jobs and places for safety reasons. Proportions are given in Table 8.2.31.

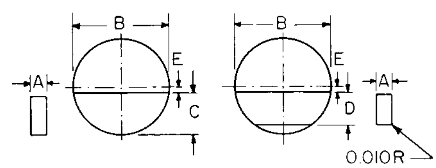


Fig. 8.2.22 Woodruff key.

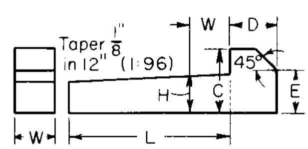


Fig. 8.2.23 Gib-head taper stock key.

The minimum stock length of keys is 4 times the key width, and maximum stock length of keys is 16 times the key width. The increments of increase of length are 2 times the width.

Sunk keys are made to the form and dimensions given in Fig. 8.2.24 and Table 8.2.32. These keys are adapted particularly to the case of fitting adjacent parts with neither end of the key accessible. **Feather keys** prevent parts from turning on a shaft while allowing them to move in a lengthwise direction. They are of the forms shown in Fig. 8.2.25 with dimensions as given in Table 8.2.32.

In **transmitting large torques,** it is customary to use two or more keys.

Table 8.2.30 Woodruff Key Dimensions [ANSI B17.2-1967 (R90)]
(All dimensions in inches)

Key no.	Nominal key size, *A* × *B*	Width of key *A*		Diam of key *B*		Height of key *C*		Height of key *D*		Distance below *E*
		Max	Min	Max	Min	Max	Min	Max	Min	
204	1/16 × 1/2	0.0635	0.0625	0.500	0.490	0.203	0.198	0.194	0.188	3/64
304	3/32 × 1/2	0.0948	0.0928	0.500	0.490	0.203	0.198	0.194	0.188	3/64
305	3/32 × 5/8	0.0948	0.0938	0.625	0.615	0.250	0.245	0.240	0.234	1/16
404	1/8 × 1/2	0.1260	0.1250	0.500	0.490	0.203	0.198	0.194	0.188	3/64
405	1/8 × 5/8	0.1260	0.1250	0.625	0.615	0.250	0.245	0.240	0.234	1/16
406	1/8 × 3/4	0.1260	0.1250	0.750	0.740	0.313	0.308	0.303	0.297	1/16
505	5/32 × 5/8	0.1573	0.1563	0.625	0.615	0.250	0.245	0.240	0.234	1/16
506	5/32 × 3/4	0.1573	0.1563	0.750	0.740	0.313	0.308	0.303	0.297	1/16
507	5/32 × 7/8	0.1573	0.1563	0.875	0.865	0.375	0.370	0.365	0.359	1/16
606	3/16 × 3/4	0.1885	0.1875	0.750	0.740	0.313	0.308	0.303	0.297	1/16
607	3/16 × 7/8	0.1885	0.1875	0.875	0.865	0.375	0.370	0.365	0.359	1/16
608	3/16 × 1	0.1885	0.1875	1.000	0.990	0.438	0.433	0.428	0.422	1/16
609	3/16 × 1 1/8	0.1885	0.1875	1.125	1.115	0.484	0.479	0.475	0.469	5/64
807	1/4 × 7/8	0.2510	0.2500	0.875	0.865	0.375	0.370	0.365	0.359	1/16
808	1/4 × 1	0.2510	0.2500	1.000	0.990	0.438	0.433	0.428	0.422	1/16
809	1/4 × 1 1/8	0.2510	0.2500	1.125	1.115	0.484	0.479	0.475	0.469	5/64
810	1/4 × 1 1/4	0.2510	0.2500	1.250	1.240	0.547	0.542	0.537	0.531	5/64
811	1/4 × 1 3/8	0.2510	0.2500	1.375	1.365	0.594	0.589	0.584	0.578	3/32
812	1/4 × 1 1/2	0.2510	0.2500	1.500	1.490	0.641	0.636	0.631	0.625	7/64
1008	5/16 × 1	0.3135	0.3125	1.000	0.990	0.438	0.433	0.428	0.422	1/16
1009	5/16 × 1 1/8	0.3135	0.3125	1.125	1.115	0.484	0.479	0.475	0.469	5/64
1010	5/16 × 1 1/4	0.3135	0.3125	1.250	1.240	0.547	0.542	0.537	0.531	5/64
1011	5/16 × 1 3/8	0.3135	0.3125	1.375	1.365	0.594	0.589	0.584	0.578	3/32
1012	5/16 × 1 1/2	0.3135	0.3125	1.500	1.490	0.641	0.636	0.631	0.625	7/64
1210	3/8 × 1 1/4	0.3760	0.3750	1.250	1.240	0.547	0.542	0.537	0.531	5/64
1211	3/8 × 1 3/8	0.3760	0.3750	1.375	1.365	0.594	0.589	0.584	0.578	3/32
1212	3/8 × 1 1/2	0.3760	0.3750	1.500	1.490	0.641	0.636	0.631	0.625	7/64

Numbers indicate the nominal key dimensions. The last two digits give the nominal diameter (*B*) in eighths of an inch, and the digits preceding the last two give the nominal width (*A*) in thirty-seconds of an inch. Thus, 204 indicates a key 2/32 × 4/8 or 1/16 × 1/2 in; 1210 indicates a key 12/32 × 10/8 or 3/8 × 1 1/4 in.

Another means for fastening gears, pulleys flanges, etc., to shafts is through the use of mating pairs of tapered sleeves known as **grip springs**. A set of sleeves is shown in Fig. 8.2.26. For further references see data issued by the Ringfeder Corp., Westwood, NJ.

Tapered pins (Fig. 8.2.27) can be used to transmit very small torques or for positioning. They should be fitted so that the parts are drawn together to prevent their working loose when the pin is driven home. Table 8.2.33 gives dimensions of Morse tapered pins.

The Groov-Pin Corp., New Jersey, has developed a special **grooved pin** (Fig. 8.2.28) which may be used instead of smooth taper pins in certain cases.

Straight pins, likewise, are used for transmission of light torques or for positioning. **Spring pins** have come into wide use recently. Two types shown in Figs. 8.2.29 and 8.2.30 deform elastically in the radial direction when driven; the resiliency of the pin material locks the pin in place. They can replace straight and taper pins and combine the advantages of both, i.e., simple tooling, ease of removal, reusability, ability to be driven from either side.

Table 8.2.31 Dimensions of Square and Flat Gib-Head Taper Stock Keys, in

Shaft diam	Square type: Key, Max width *W*	Square type: Key, Height at large end,† *H*	Square type: Gib head, Height *C*	Square type: Gib head, Length *D*	Square type: Gib head, Height edge of chamfer *E*	Flat type: Key, Max width *W*	Flat type: Key, Height at large end,† *H*	Flat type: Gib head, Height *C*	Flat type: Gib head, Length *D*	Flat type: Gib head, Height edge of chamfer *E*	Tolerance: On width (−)	Tolerance: On height (+)
1/2–9/16	1/8	1/8	1/4	7/32	5/32	1/8	3/32	3/16	1/8	1/8	0.0020	0.0020
5/8–7/8	3/16	3/16	5/16	9/32	7/32	3/16	1/8	1/4	3/16	5/32	0.0020	0.0020
15/16–1 1/4	1/4	1/4	7/16	11/32	11/32	1/4	3/16	5/16	1/4	3/16	0.0020	0.0020
1 5/16–1 3/8	5/16	5/16	9/16	13/32	13/32	5/16	1/4	3/8	5/16	1/4	0.0020	0.0020
1 7/16–1 3/4	3/8	3/8	11/16	15/32	15/32	3/8	1/4	7/16	3/8	5/16	0.0020	0.0020
1 13/16–2 1/4	1/2	1/2	7/8	19/32	5/8	1/2	3/8	5/8	1/2	7/16	0.0025	0.0025
2 5/16–2 3/4	5/8	5/8	1 1/16	23/32	3/4	5/8	7/16	3/4	5/8	1/2	0.0025	0.0025
2 7/8–3 1/4	3/4	3/4	1 1/4	7/8	7/8	3/4	1/2	7/8	3/4	5/8	0.0025	0.0025
3 3/8–3 3/4	7/8	7/8	1 1/2	1	1	7/8	5/8	1 1/16	7/8	3/4	0.0030	0.0030
3 7/8–4 1/2	1	1	1 3/4	1 3/16	1 3/16	1	3/4	1 1/4	1	13/16	0.0030	0.0030
4 3/4–5 1/2	1 1/4	1 1/4	2	1 7/16	1 7/16	1 1/4	7/8	1 1/2	1 1/4	1	0.0030	0.0030
5 3/4–6	1 1/2	1 1/2	2 1/2	1 3/4	1 3/4	1 1/2	1	1 3/4	1 1/2	1 1/4	0.0030	0.0030

* Stock keys are applicable to the general run of work and the tolerances have been set accordingly. They are not intended to cover the finer applications where a closer fit may be required.
† This height of the key is measured at the distance *W* equal to the width of the key, from the gib head.

Cotter pins (Fig. 8.2.31) are used to secure or lock nuts, clevises, etc. Driven into holes in the shaft, the eye prevents complete passage, and the split ends, deformed after insertion, prevent withdrawal.

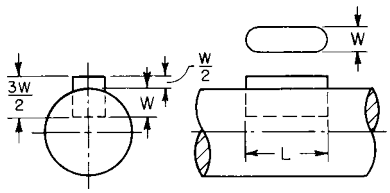


Fig. 8.2.24 Sunk key.

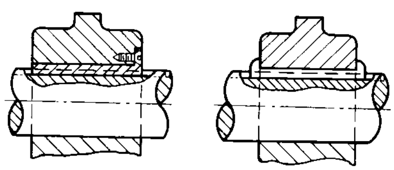


Fig. 8.2.25 Feather key.

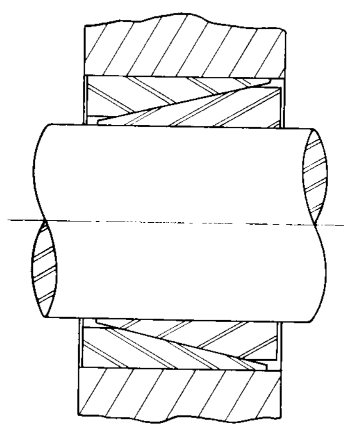


Fig. 8.2.26 Grip springs.

When two rods are to be joined so as to permit movement at the joint, a round pin is used in place of a cotter. In such cases, the proportions may be as shown in Fig. 8.2.32 (knuckle joint).

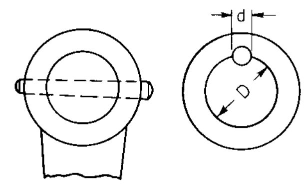


Fig. 8.2.27 Taper pins.

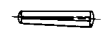


Fig. 8.2.28 Grooved pins.

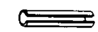


Fig. 8.2.29 Roll pin.

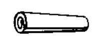


Fig. 8.2.30 Spiral pins.

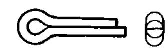


Fig. 8.2.31 Cotter pin.

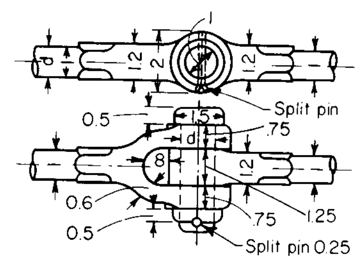


Fig. 8.2.32 Knuckle joint.

SPLINES

Involute spline proportions, dimensions, fits, and tolerances are given in detail in ANSI B92.1-1970. External and internal involute splines (Fig. 8.2.33) have the same general form as involute gear teeth, except that the teeth are one-half the depth of standard gear teeth and the pressure angle is 30°. The spline is designated by a fraction in which the numerator is the diametral pitch and the denominator is always twice the numerator.

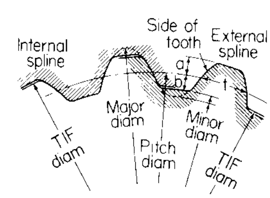


Fig. 8.2.33 Involute spline.

Table 8.2.32 Dimensions of Sunk Keys
(All dimensions in inches. Letters refer to Fig. 8.2.24)

Key no.	*L*	*W*	Key no.	*L*	*W*	Key no.	*L*	*W*	Key no.	*L*	*W*
1	½	1⁄16	13	1	3⁄16	22	1⅜	¼	54	2¼	¼
2	½	3⁄32	14	1	7⁄32	23	1⅜	5⁄16	55	2¼	5⁄16
3	½	⅛	15	1	¼	F	1⅜	⅜	56	2¼	⅜
4	⅝	3⁄32	B	1	5⁄16	24	1½	¼	57	2¼	7⁄16
5	⅝	⅛	16	1⅛	3⁄16	25	1½	5⁄16	58	2½	5⁄16
6	⅝	5⁄32	17	1⅛	7⁄32	G	1½	⅜	59	2½	⅜
7	¾	⅛	18	1⅛	¼	51	1¾	¼	60	2½	7⁄16
8	¾	5⁄32	C	1⅛	5⁄16	52	1¾	5⁄16	61	2½	½
9	¾	3⁄16	19	1¼	3⁄16	53	1¾	⅜	30	3	⅜
10	⅞	5⁄32	20	1¼	7⁄32	26	2	3⁄16	31	3	7⁄16
11	⅞	3⁄16	21	1¼	¼	27	2	¼	32	3	½
12	⅞	7⁄32	D	1¼	5⁄16	28	2	5⁄16	33	3	9⁄16
A	⅞	¼	E	1¼	⅜	29	2	⅜	34	3	⅝

Table 8.2.33 Morse Standard Taper Pins

(Taper, ⅛ in/ft. Lengths increase by ¼ in. Dimensions in inches)

Size no.	0	1	2	3	4	5	6	7	8	9	10
Diam at large end	0.156	0.172	0.193	0.219	0.250	0.289	0.341	0.409	0.492	0.591	0.706
Length	0.5–3	0.5–3	0.75–3.5	0.75–3.5	0.75–4	0.75–4	0.75–5	1–5	1.25–5	1.5–6	1.5–6

There are 17 series, as follows: 2.5/5, 3/6, 4/8, 5/10, 6/12, 8/16, 10/20, 12/24, 16/32, 20/40, 24/48, 32/64, 40/80, 48/96, 64/128, 80/160, 128/256. The number of teeth within each series varies from 6 to 50. Both a flat-root and a fillet-root type are provided. There are three **types of fits:** (1) **major diameter**—fit controlled by varying the major diameter of the external spline; (2) **sides of teeth**—fit controlled by varying tooth thickness and customarily used for fillet-root splines; (3) **minor diameter**—fit controlled by varying the minor diameter of the internal spline. Each type of fit is further divided into three classes: (*a*) **sliding**—clearance at all points; (*b*) **close**—close on either major diameter, sides of teeth, or minor diameter; (*c*) **press**—interference on either the major diameter, sides of teeth, or minor diameter. Important basic formulas for tooth proportions are:

D = pitch diam
N = number of teeth
P = diametral pitch
p = circular pitch
t = circular tooth thickness
a = addendum
b = dedendum
D_O = major diam
TIF = true involute form diam
D_R = minor diam

Flat and Fillet Roots

$$D = N/P$$
$$p = \pi/P$$
$$t = p/2$$
$$a = 0.5000/P$$
$$D_O\ (\text{external}) = \frac{N+1}{P}$$
$$\text{TIF (internal)} = \frac{N+1}{P}$$
$$D_R = \frac{N+1}{P} \qquad \text{(minor-diameter fits only)}$$
$$\text{TIF (external)} = \frac{N-1}{P}$$

$b = 0.600/P + 0.002$ (For major-diameter fits, the internal spline dedendum is the same as the addendum; for minor-diameter fits, the dedendum of the external spline is the same as the addendum.)

Fillet Root Only

½ through 12⁄24	16⁄32 through 48⁄96
$D_O\ (\text{internal}) = \frac{N+1.8}{P}$	$D_O\ (\text{internal}) = \frac{N+1.8}{P}$
$D_R\ (\text{external}) = \frac{N-1.8}{P}$	$D_R\ (\text{external}) = \frac{N-2}{P}$
b (internal) = $0.900/P$	b (internal) = $0.900/P$
b (external) = $0.900/P$	b (external) = $1.000/P$

Internal spline dimensions are basic while external spline dimensions are varied to control fit.

The advantages of involute splines are: (1) maximum strength at the minor diameter, (2) self-centering equalizes bearing and stresses among all teeth, and (3) ease of manufacture through the use of standard gear-cutting tools and methods.

The design of involute splines is critical in shear. The torque capacity may be determined by the formula $T = LD^2S_s/1.2732$, where L = spline length, D = pitch diam, S_s = allowable shear stress.

Parallel-side splines have been standardized by the SAE for 4, 6, 10, and 16 spline fittings. They are shown in Fig. 8.2.34; pertinent data are in Tables 8.2.34 and 8.2.35.

DRY AND VISCOUS COUPLINGS

A **coupling** makes a semipermanent connection between two shafts. They are of three main types: **rigid, flexible,** and **fluid.**

Rigid Couplings

Rigid couplings are used only on shafts which are perfectly aligned. The **flanged-face coupling** (Fig. 8.2.35) is the simplest of these. The flanges must be keyed to the shafts. The **keyless compression coupling** (Fig. 8.2.36) affords a simple means for connecting abutting shafts without the necessity of key seats on the shafts. When drawn over the slotted tapered sleeve the two flanges automatically center the shafts and provide sufficient contact pressure to transmit medium or light loads. **Ribbed-clamp couplings** (Fig. 8.2.37) are split longitudinally and are bored to the shaft diameter with a shim separating the two halves. It is necessary to key the shafts to the coupling.

Flexible Couplings

Flexible couplings are designed to connect shafts which are misaligned either laterally or angularly. A secondary benefit is the absorption of

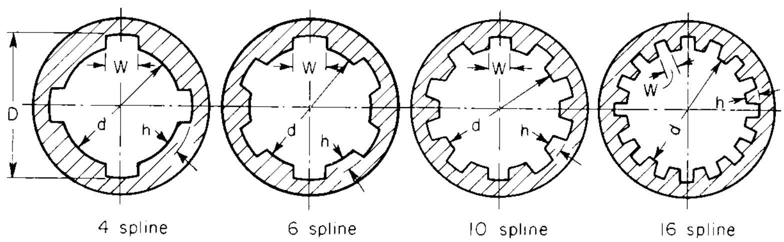


Fig. 8.2.34 Parallel-sided splines.

Table 8.2.34 Dimensions of Spline Fittings, in
(SAE Standard)

Nominal diam	4-spline for all fits		6-spline for all fits		10-spline for all fits		16-spline for all fits	
	D max*	*W* max†	*D* max*	*W* max†	*D* max*	*W* max†	*D* max*	*W* max†
¾	0.750	0.181	0.750	0.188	0.750	0.117		
⅞	0.875	0.211	0.875	0.219	0.875	0.137		
1	1.000	0.241	1.000	0.250	1.000	0.156		
1⅛	1.125	0.271	1.125	0.281	1.125	0.176		
1¼	1.250	0.301	1.250	0.313	1.250	0.195		
1⅜	1.375	0.331	1.375	0.344	1.375	0.215		
1½	1.500	0.361	1.500	0.375	1.500	0.234		
1⅝	1.625	0.391	1.625	0.406	1.625	0.254		
1¾	1.750	0.422	1.750	0.438	1.750	0.273		
2	2.000	0.482	2.000	0.500	2.000	0.312	2.000	0.196
2¼	2.250	0.542	2.250	0.563	2.250	0.351		
2½	2.500	0.602	2.500	0.625	2.500	0.390	2.500	0.245
3	3.000	0.723	3.000	0.750	3.000	0.468	3.000	0.294
3½	—	—	—	—	3.500	0.546	3.500	0.343
4	—	—	—	—	4.000	0.624	4.000	0.392
4½	—	—	—	—	4.500	0.702	4.500	0.441
5	—	—	—	—	5.000	0.780	5.000	0.490
5½	—	—	—	—	5.500	0.858	5.500	0.539
6	—	—	—	—	6.000	0.936	6.000	0.588

* Tolerance allowed of − 0.001 in for shafts ¾ to 1¾ in, inclusive; of − 0.002 for shafts 2 to 3 in, inclusive; − 0.003 in for shafts 3½ to 6 in, inclusive, for 4-, 6-, and 10-spline fittings; tolerance of − 0.003 in allowed for all sizes of 16-spline fittings.
† Tolerance allowed of − 0.002 in for shafts ¾ in to 1¾ in, inclusive; of − 0.003 in for shafts 2 to 6 in, inclusive, for 4-, 6-, and 10-spline fittings; tolerance of − 0.003 allowed for all sizes of 16-spline fittings.

Table 8.2.35 Spline Proportions

No. of splines	*W* for all fits	Permanent fit		To slide when not under load		To slide under load	
		h	*d*	*h*	*d*	*h*	*d*
4	0.241*D*	0.075*D*	0.850*D*	0.125*D*	0.750*D*		
6	0.250*D*	0.050*D*	0.900*D*	0.075*D*	0.850*D*	0.100*D*	0.800*D*
10	0.156*D*	0.045*D*	0.910*D*	0.070*D*	0.860*D*	0.095*D*	0.810*D*
16	0.098*D*	0.045*D*	0.910*D*	0.070*D*	0.860*D*	0.095*D*	0.810*D*

impacts due to fluctuations in shaft torque or angular speed. The **Oldham,** or **double-slider, coupling** (Fig. 8.2.38) may be used to connect shafts which have only lateral misalignment. The **''Fast'' flexible coupling** (Fig. 8.2.39) consists of two hubs each keyed to its respective shaft. Each hub has generated splines cut at the maximum possible

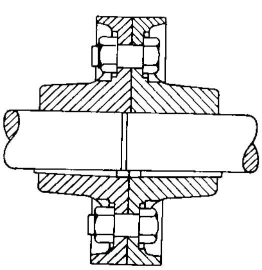


Fig. 8.2.35 Flanged face coupling.

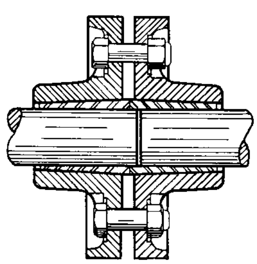


Fig. 8.2.36 Keyless compression coupling.

distance from the shaft end. Surrounding the hubs is a casing or sleeve which is split transversely and bolted by means of flanges. Each half of this sleeve has generated internal splines cut on its bore at the end opposite to the flange. These internal splines permit a definite error of alignment between the two shafts.

Another type, the Waldron coupling (Midland-Ross Corp.), is shown in Fig. 8.2.40.

The chain coupling shown in Fig. 8.2.41 uses silent chain, but standard roller chain can be used with the proper mating sprockets. Nylon links enveloping the sprockets are another variation of the chain coupling.

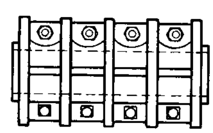


Fig. 8.2.37 Ribbed-clamp coupling.

Steelflex couplings (Fig. 8.2.42) are made with two grooved steel hubs keyed to their respective shafts. Connection between the two halves is secured by a specially tempered alloy-steel member called the ''grid.''

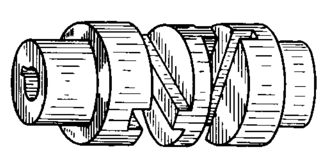


Fig. 8.2.38 Double-slider coupling.

In the rubber flexible coupling shown in Fig. 8.2.43, the torque is transmitted through a comparatively soft rubber section acting in shear. The type in Fig. 8.2.44 loads the intermediate rubber member in compression. Both types permit reasonable shaft misalignment and are recommended for light loads only.

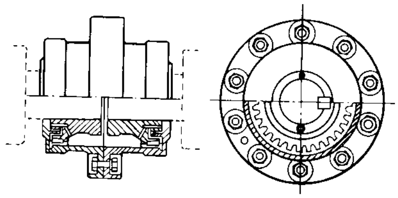


Fig. 8.2.39 ''Fast'' flexible coupling.

Universal joints are used to connect shafts with much larger values of misalignment than can be tolerated by the other types of flexible couplings. Shaft angles up to 30° may be used. The Hooke's-type joint (Fig. 8.2.45) suffers a loss in efficiency with increasing angle which

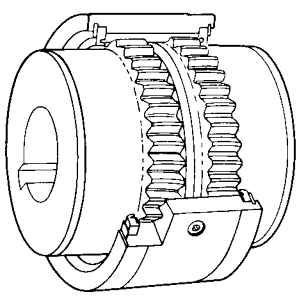


Fig. 8.2.40 Waldron coupling.

may be approximated for angles up to 15° by the following relation: efficiency = $100(1 - 0.003\theta)$, where θ is the angle between the shafts. The velocity ratio between input and output shafts with a single universal joint is equal to

$$\omega_2/\omega_1 = \cos\theta/1 - \sin^2\theta \sin^2(\alpha + 90°)$$

where ω_2 and ω_1 are the angular velocities of the driven and driving shafts respectively, θ is the angle between the shafts, and α is the angu-

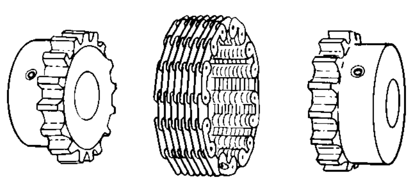


Fig. 8.2.41 Chain coupling.

lar displacement of the driving shaft from the position where the pins on the drive-shaft yoke lie in the plane of the two shafts. A velocity ratio of 1 may be obtained at any angle using two Hooke's-type joints and an intermediate shaft. The intermediate shaft must make equal angles with the main shafts, and the driving pins on the yokes attached to the intermediate shaft must be set parallel to each other.

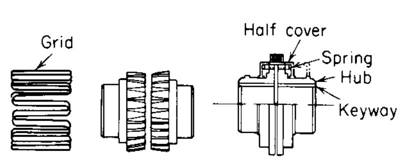


Fig. 8.2.42 Falk Steelflex coupling.

The Bendix-Weiss ''rolling-ball'' universal joint provides constant angular velocity. Torque is transmitted between two yokes through a set of four balls such that the centers of all four balls lie in a plane which bisects the angle between the shafts. Other variations of constant velocity universal joints are found in the Rzeppa, Tracta, and double Cardan types.

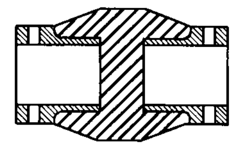


Fig. 8.2.43 Rubber flexible coupling, shear type.

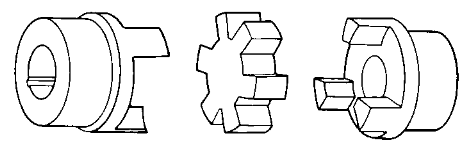


Fig. 8.2.44 Rubber flexible coupling, compression type.

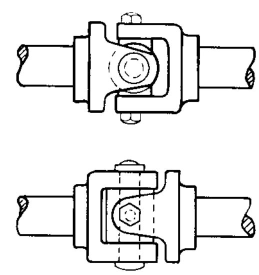


Fig. 8.2.45 Hooke's universal joint.

Fluid Couplings

(See also Sec. 11.)

Fluid couplings (Fig. 8.2.46) have two basic parts—the input member, or impeller, and the output member, or runner. There is no mechanical connection between the two shafts, power being transmitted by kinetic energy in the operating fluid. The impeller *B* is fastened to the flywheel *A* and turns at engine speed. As this speed increases, fluid within the impeller moves toward the outer periphery because of centrifugal force. The circular shape of the impeller directs the fluid toward the runner *C*, where its kinetic energy is absorbed as torque delivered by shaft *D*. The positive pressure behind the fluid causes flow to continue toward the hub and back through the impeller. The toroidal space in both the impeller and runner is divided into compartments by a series of flat radial vanes.

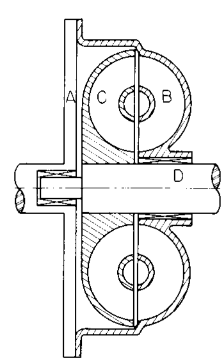


Fig. 8.2.46*a* Fluid coupling. (*A*) Flywheel; (*B*) impeller; (*C*) runner; (*D*) output shaft.

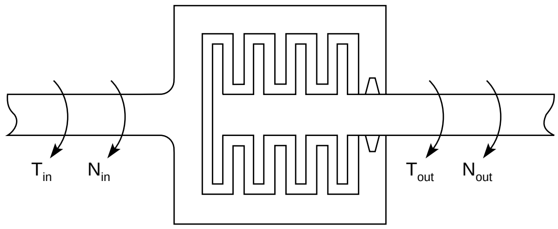


Fig. 8.2.46*b* Schematic of viscous coupling.

The torque capacity of a fluid coupling with a full-load slip of about 2.5 percent is $T = 0.09n^2D^5$, where n is the impeller speed, hundreds of r/min, and D is the outside diameter, ft. The output torque is equal to the input torque over the entire range of input-output speed ratios. Thus the prime mover can be operated at its most effective speed regardless of the speed of the output shaft. Other advantages are that the prime mover cannot be stalled by application of load and that there is no transmission of shock loads or torsional vibration between the connected shafts.

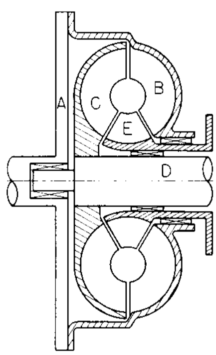


Fig. 8.2.47 Hydraulic torque converter. (*A*) Flywheel; (*B*) impeller; (*C*) runner; (*D*) output shaft; (*E*) reactor.

A **hydraulic torque converter** (Fig. 8.2.47) is similar in form to the hydraulic coupling, with the addition of a set of stationary guide vans, the reactor, interposed between the runner and the impeller. All blades in a converter have compound curvature. This curvature is designed to control the direction of fluid flow. Kinetic energy is therefore transferred as a result of both a scalar and vectorial change in fluid velocity. The blades are designed such that the fluid will be moving in a direction parallel to the blade surface at the entrance (Fig. 8.2.48) to each section. With a design having fixed blading, this can be true at only one value of runner and impeller velocity, called the design point. Several design modifications are possible to overcome this difficulty. The angle of the blades can be made adjustable, and the elements can be divided into sections operating independently of each other according to the load requirements. Other refinements include the addition of multiple stages in the runner and reactor stages as in steam reaction turbines (see Sec. 9). The advantages of a torque converter are the ability to multiply starting torque 5 to 6 times and to serve as a stepless transmission. As in the coupling, torque varies as the square of speed and the fifth power of diameter.

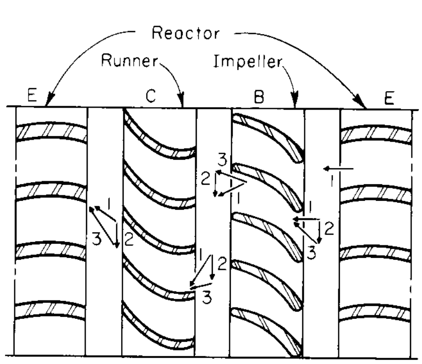


Fig. 8.2.48 Schematic of converter blading. (1) Absolute fluid velocity; (2) velocity vector—converter elements; (3) fluid velocity relative to converter elements.

Optimum efficiency (Fig. 8.2.49) over the range of input-output speed ratios is obtained by a combination converter coupling. When the output speed rises to the point where the torque multiplication factor is 1.0, the clutch point, the torque reaction on the reactor element reverses direction. If the reactor is mounted to freewheel in this opposite direction, the unit will act as a coupling over the higher speed ranges. An automatic friction clutch (see ''Clutches,'' below) set to engage at or near the clutch point will also eliminate the poor efficiency of the converter at high output speeds.

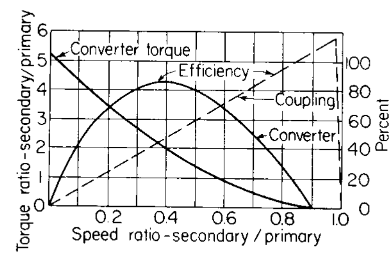


Fig. 8.2.49 Hydraulic coupling characteristic curves. *(Heldt, ''Torque Converters and Transmissions,'' Chilton.)*

Viscous couplings are becoming major players in mainstream front-wheel-drive applications and are already used in four-wheel-drive vehicles.

Torque transmission in a viscous coupling relies on shearing forces in an entrapped fluid between axially positioned disks rotating at different angular velocities (Fig. 8.2.46*b*), all encased in a lifetime leakproof housing. A hub carries the so-called inner disks while the housing carries the so-called outer disks. Silicone is the working fluid.

Operation of the coupling is in normal (slipping) mode when torque is being generated by viscous shear. However, prolonged slipping under severe starting conditions causes heat-up, which in turn causes the fluid, which has a high coefficient of thermal expansion, to expand considerably with increasing temperature. It then fills the entire available space, causing a rapid pressure increase, which in turn forces the disks together into metal-to-metal frictional contact. Torque transmission now increases substantially. This self-induced torque amplification is known as the **hump effect.** The point at which the hump occurs can be set by the design and coupling setup. Under extreme conditions, 100 percent lockup occurs, thus providing a self-protecting relief from overheating as fluid shear vanishes. This effect is especially useful in autos using viscous couplings in their limited-slip differentials, when one wheel is on low-friction surfaces such as ice. The viscous coupling transfers torque to the other gripping wheel. This effect is also useful when one is driving up slopes with uneven surface conditions, such as rain or snow, or on very rough surfaces. Such viscous coupling differentials have allowed a weight and cost reduction of about 60 percent. A fuller account can be found in Barlage, Viscous Couplings Enter Main Stream Vehicles, *Mech. Eng.,* Oct. 1993.

CLUTCHES

Clutches are couplings which permit the disengagement of the coupled shafts during rotation.

Positive clutches are designed to transmit torque without slip. The **jaw clutch** is the most common type of positive clutch. These are made with **square jaws** (Fig. 8.2.50) for driving in both directions or **spiral jaws** (Fig. 8.2.51) for unidirectional drive. Engagement speed should be limited to 10 r/min for square jaws and 150 r/min for spiral jaws. If disengagement under load is required, the jaws should be finish-machined and lubricated.

Friction clutches are designed to reduce coupling shock by slipping during the engagement period. They also serve as safety devices by slipping when the torque exceeds their maximum rating. They may be divided into two main groups, **axial** and **rim clutches,** according to the direction of contact pressure.

The **cone clutch** (Fig. 8.2.52) and the **disk clutch** (Fig. 8.2.53) are examples of axial clutches. The disk clutch may consist of either a

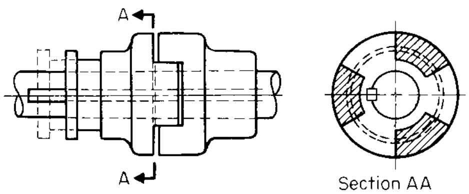


Fig. 8.2.50 Square-jaw clutch.

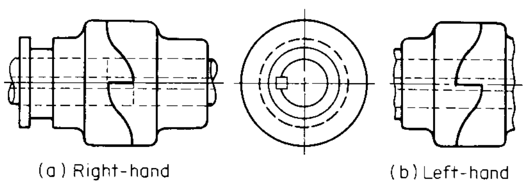


Fig. 8.2.51 Spiral-jaw clutch.

single plate or multiple disks. Table 8.2.36 lists typical friction materials and important design data. The torque capacity of a disk clutch is given by $T = 0.5ifF_aD_m$, where T is the torque, i the number of pairs of contact surfaces, f the applicable coefficient of friction, F_a the axial

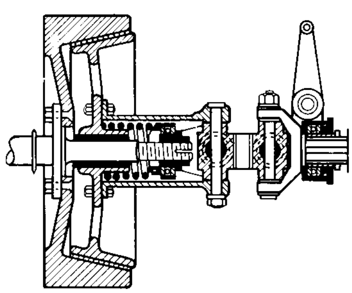


Fig. 8.2.52 Cone clutch.

engaging force, and D_m the mean diameter of the clutch facing. The spring forces holding a disk clutch in engagement are usually of relatively high value, as given by the allowable contact pressures. In order to lower the force required at the operating lever, elaborate linkages are required, usually having lever ratios in the range of 10 to 12. As these linkages must rotate with the clutch, they must be adequately balanced and the effect of centrifugal forces must be considered. Disk clutches are often run wet, either immersed in oil or in a spray. The advantages are reduced wear, smoother action, and lower operating temperatures. Disk clutches are often operated automatically by either air or hydraulic cylinders as, for examples, in automobile automatic transmissions.

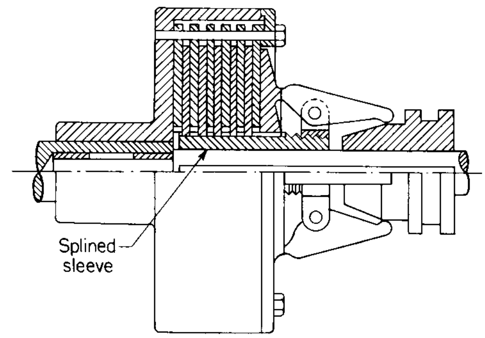


Fig. 8.2.53 Multidisk clutch.

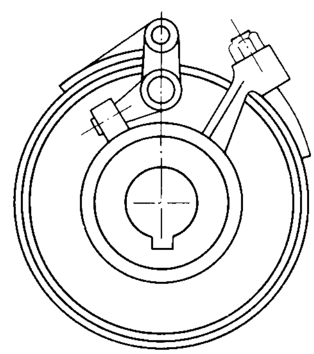


Fig. 8.2.54 Band clutch.

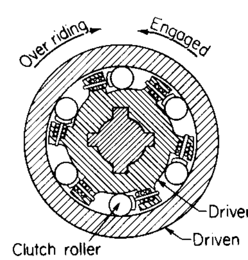


Fig. 8.2.55 Overrunning clutch.

Table 8.2.36 Friction Coefficients and Allowable Pressures

Materials in contact	Friction coefficient f: Dry	Greasy	Lubricated	Allowable pressure, lb/in²
Cast iron on cast iron	0.2–0.15	0.10–0.06	0.10–0.05	150–250
Bronze on cast iron	—	0.10–0.05	0.10–0.05	80–120
Steel on cast iron	0.30–0.20	0.12–0.07	0.10–0.06	120–200
Wood on cast iron	0.25–0.20	0.12–0.08	—	60–90
Fiber on metal	—	0.20–0.10	—	10–30
Cork on metal	0.35	0.30–0.25	0.25–0.22	8–15
Leather on metal	0.5–0.3	0.20–0.15	0.15–0.12	10–30
Wire asbestos on metal	0.5–0.35	0.30–0.25	0.25–0.20	40–80
Asbestos blocks on metal	0.48–0.40	0.30–0.25	—	40–160
Asbestos on metal, short action	—	—	0.25–0.20	200–300
Metal on cast iron, short action	—	—	0.10–0.05	200–300

SOURCE: Maleev, *Machine Design,* International Textbook, by permission.

Rim clutches may be subdivided into two groups: (1) those employing either a band or block (Fig. 8.2.54) in contact with the rim and (2) overrunning clutches (Fig. 8.2.55) employing the wedging action of a roller or sprag. Clutches in the second category will automatically engage in one direction and freewheel in the other.

HYDRAULIC POWER TRANSMISSION

Hydraulic power transmission systems comprise machinery and auxiliary components which function to generate, transmit, control, and utilize hydraulic power. The **working fluid,** a pressurized incompressible liquid, is usually either a petroleum base or a fire-resistant type. The latter are water and oil emulsions, glycol-water mixtures, or synthetic liquids such as silicones or phosphate esters.

Liquid is pressurized in a **pump** by virtue of its resistance to flow; the pressure difference between pump inlet and outlet results in flow. Most hydraulic applications employ positive-displacement pumps of the gear, vane, screw, or piston type; piston pumps are axial, radial, or reciprocating (see Sec. 14).

Power is transmitted from pump to controls and point of application through a combination of **conduit and fittings** appropriate to the particular application. Flow characteristics of hydraulic circuits take into account fluid properties, pressure drop, flow rate, and pressure-surging tendencies. Conduit systems must be designed to minimize changes in flow velocity, velocity distribution, and random fluid eddies, all of which dissipate energy and result in pressure drops in the circuit (see Sec. 3). Pipe, tubing, and flexible hose are used as hydraulic power conduits; suitable fittings are available for all types and for transition from one type to another.

Controls are generally interposed along the conduit between the pump and point of application (i.e., an actuator or motor), and act to control pressure, volume, or flow direction.

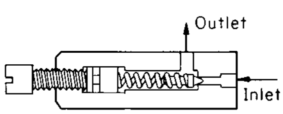


Fig. 8.2.56 Relief valve.

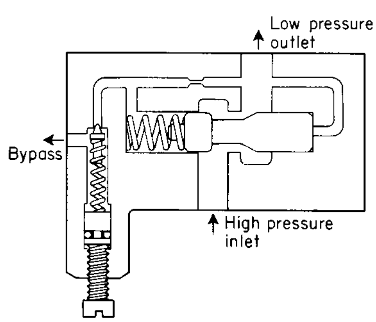


Fig. 8.2.57 Reducing valve.

Pressure control valves, of which an ordinary safety valve is a common type (normally closed), include relief and reducing valves and pressure switches (Figs. 8.2.56 and 8.2.57). Pressure valves, normally closed, can be used to control sequential operations in a hydraulic circuit. **Flow control valves** throttle flow to or bypass flow around the unit being controlled, resulting in pressure drop and temperature increase as pressure energy is dissipated. Figure 8.2.58 shows a simple needle valve with variable orifice usable as a flow control valve. **Directional control valves** serve primarily to direct hydraulic fluid to the point of application. Directional control valves with rotary and sliding spools are shown in Figs. 8.2.59 and 8.2.60.

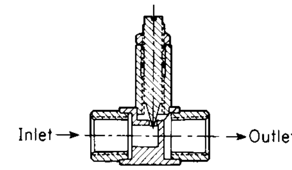


Fig. 8.2.58 Needle valve.

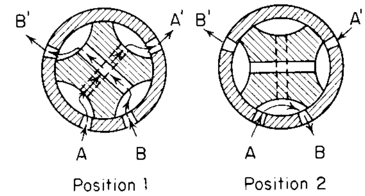


Fig. 8.2.59 Rotary-spool directional flow valve.

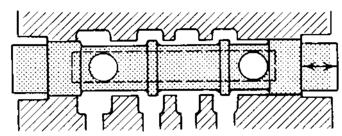


Fig. 8.2.60 Sliding-spool directional flow valve.

A poppet (valve) mechanism is shown in Fig. 8.2.61, a diaphragm valve in Fig. 8.2.62, and a shear valve in Fig. 8.2.63.

Accumulators are effectively ''hydraulic flywheels'' which store potential energy by accumulating a quantity of pressurized hydraulic fluid

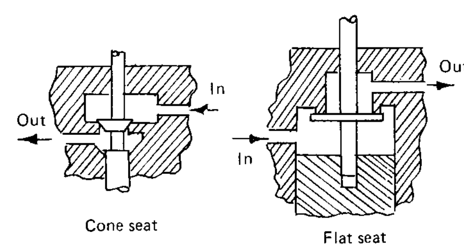


Fig. 8.2.61 Poppet valve.

in a suitable enclosed vessel. The bag type shown in Fig. 8.2.64 uses pressurized gas inside the bag working against the hydraulic fluid outside the bag. Figure 8.2.65 shows a piston accumulator.

Pressurized hydraulic fluid acting against an **actuator** or motor converts fluid pressure energy into mechanical energy. Motors providing

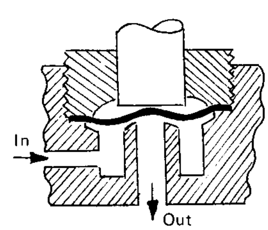


Fig. 8.2.62 Diaphragm valve.

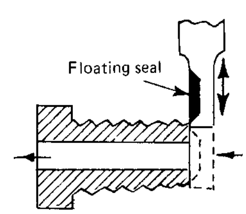


Fig. 8.2.63 Shear valve.

continuous rotation have operating characteristics closely related to their pump counterparts. A linear actuator, or cylinder (Fig. 8.2.66), provides straight-line reciprocating motion; a rotary actuator (Fig. 8.2.67) provides arcuate oscillatory motion. Figure 8.2.68 shows a one-shot booster (linear motion) which can be used to deliver sprays through a nozzle.

Hydraulic fluids (liquids and air) are conducted in **pipe, tubing,** or **flexible hose.** Hose is used when the lines must flex or in applications in which fixed, rigid conduit is unsuitable. Table 8.2.37 lists SAE standard hoses. Maximum recommended operating pressure for a broad range of industrial applications is approximately 25 percent of rated bursting pressure. Due consideration must be given to the operating-temperature range; most applications fall in the range from − 40 to 200°F (− 40 to 95°C). Higher operating temperatures can be accommodated with appropriate materials.

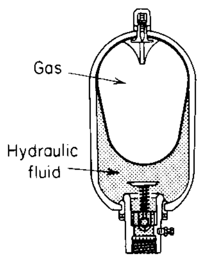


Fig. 8.2.64 Bag accumulator.

Hose fittings are of the screw-type or swaged, depending on the particular application and operating pressure and temperature. A broad variety of hose-end fittings is available from the industry.

Pipe has the advantage of being relatively cheap, is applied mainly in straight runs, and is usually of steel. Fittings for pipe are either standard pipe fittings for fairly low pressures or more elaborate ones suited to leakproof high-pressure operation.

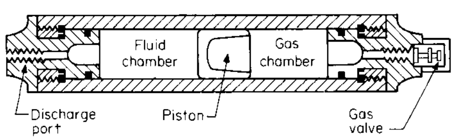


Fig. 8.2.65 Piston accumulator.

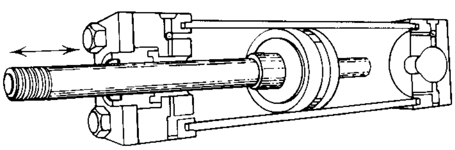


Fig. 8.2.66 Linear actuator or hydraulic cylinder.

Tubing is more easily bent into neat forms to fit between inlet and outlet connections. Steel and stainless-steel tubing is used for the highest-pressure applications; aluminum, plastic, and copper tubing is also used as appropriate for the operating conditions of pressure and temperature. Copper tubing hastens the oxidation of oil-base hydraulic fluids; accordingly, its use should be restricted either to air lines or with liquids which will not be affected by copper in the operating range.

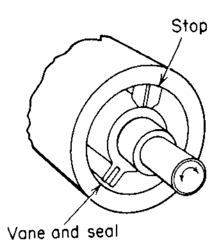


Fig. 8.2.67 Rotary actuator.

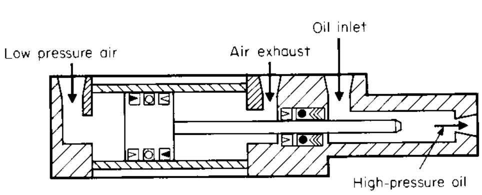


Fig. 8.2.68 One-shot booster.

Table 8.2.37 SAE Standard Hoses

100R1A	One-wire-braid reinforcement, synthetic rubber cover
100R1T	Same as R1A except with a thin, nonskive cover
100R2A	Two-wire-braid reinforcement, synthetic rubber cover
100R2B	Two spiral wire plus one wire-braid reinforcement, synthetic rubber cover
100R2AT	Same as R2A except with a thin, nonskive cover
100R2BT	Same as R2B except with a thin, nonskive cover
100R3	Two rayon-braid reinforcement, synthetic rubber cover
100R5	One textile braid plus one wire-braid reinforcement, textile braid cover
100R7	Thermoplastic tube, synthetic fiber reinforcement, thermoplastic cover (thermoplastic equivalent to SAE 100R1A)
100R8	Thermoplastic tube, synthetic fiber reinforcement, thermoplastic cover (thermoplastic equivalent to SAE 100R2A)
100R9	Four-ply, light-spiral-wire reinforcement, synthetic rubber cover
100R9T	Same as R9 except with a thin, nonskive cover
100R10	Four-ply, heavy-spiral-wire reinforcement, synthetic rubber cover
100R11	Six-ply, heavy-spiral-wire reinforcement, synthetic rubber cover

Tube fittings for permanent connections allow for brazed or welded joints. For temporary or separable applications, **flared** or **flareless fittings** are employed (Figs. 8.2.69 and 8.2.70). ANSI B116.1-1974 and B116.2-1974 pertain to tube fittings. The variety of fittings available is vast; the designer is advised to refer to manufacturers' literature for specifics.

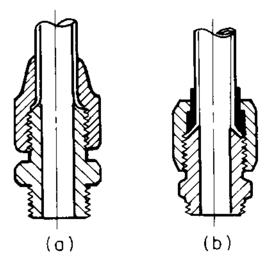


Fig. 8.2.69 Flared tube fittings. (*a*) A 45° flared fitting; (*b*) Triple-lok flared fitting. *(Parker-Hannafin Co.)*

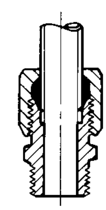


Fig. 8.2.70 Ferulok flareless tube fitting. *(Parker-Hannafin Co.)*

Parameters entering into the design of a hydraulic system are volume of flow per unit time, operating pressure and temperature, viscosity characteristics of the fluid within the operating range, and compatibility of the fluid/conduit material.

Flow velocity in suction lines is generally in the range of 1 to 5 ft/s (0.3 to 1.5 m/s); in discharge lines it ranges from 10 to 25 ft/s (3 to 8 m/s).

The pipe or tubing is under internal pressure. Selection of material and wall thickness follows from suitable equations (see Sec. 5). Safety factors range from 6 to 10 or higher, depending on the severity of the application (i.e., vibration, shock, pressure surges, possibility of physical abuse, etc.). JIC specifications provide a guide to the designer of hydraulic systems.

BRAKES

Brakes may be classified as: (1) rim type—internally expanding or externally contracting shoes, (2) band type, (3) cone type, (4) disk or axial type, (5) miscellaneous.

Rim Type—Internal Shoe(s) (Fig. 8.2.71)

$$F = \begin{cases} \dfrac{M_N - M\mu}{d} & \text{clockwise rotation} \\[2ex] \dfrac{M_N + M\mu}{d} & \text{counterclockwise rotation} \end{cases}$$

where

$$M\mu = \frac{\mu P_a Br}{\sin\theta_a}\int_{\theta_1}^{\theta_2}(\sin\theta)(r - d\cos\theta)\,d\theta$$

B = face width of frictional material; P_a = maximum pressure; θ_a = angle to point of maximum pressure (if $\theta_2 > 90°$; then $\theta_a = 90°$; $\theta_2 < 90°$, then $\theta_a = \theta_2$); μ = coefficient of friction; r = radius of drum; d = distance from drum center to brake pivot;

$$M_N = \frac{P_a Brd}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin^2\theta\,d\theta$$

and torque on drum is

$$T = \frac{\mu P_a Br^2(\cos\theta_1 - \cos\theta_2)}{\sin\theta_a}$$

Self-locking of the brake ($F = 0$) will occur for clockwise rotation when $M_N = M\mu$. This self-energizing phenomenon can be used to advantage without actual locking if μ is replaced by a larger value μ' so that $1.25\,\mu \le \mu' \le 1.50$, from which the pivot position a can be solved.

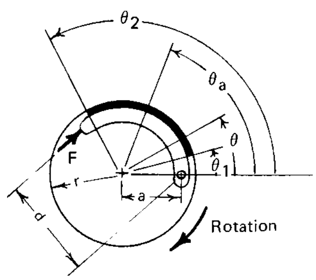


Fig. 8.2.71 Rim brake: internal friction shoe.

In automative use there are two shoes made to pivot in opposition, so that self-energization is present and can be used to great advantage (Fig. 8.2.72).

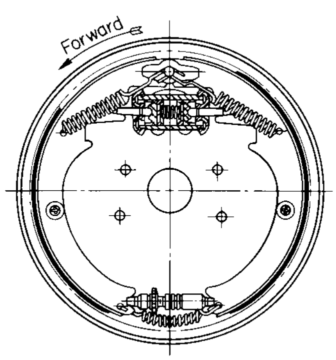


Fig. 8.2.72 Internal brake.

Rim Type—External Shoe(s) (Fig. 8.2.73)

The equations for M_N and $M\mu$ are the same as above:

$$F = \begin{cases} \dfrac{M_N + M\mu}{d} & \text{clockwise rotation} \\[2ex] \dfrac{M_N - M\mu}{d} & \text{counterclockwise rotation} \end{cases}$$

Self-locking ($F = 0$) can occur for counterclockwise rotation at $M_N = M\mu$.

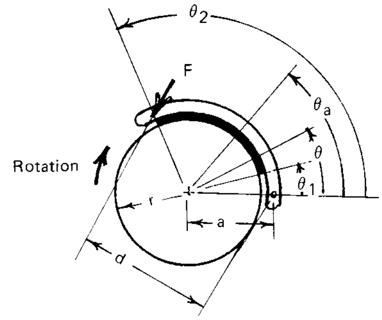


Fig. 8.2.73 Rim brake: external friction shoe.

Band Type (Fig. 8.2.74*a*, *b*, and *c*)

Flexible band brakes are used in power excavators and in hoisting. The bands are usually of an asbestos fabric, sometimes reinforced with copper wire and impregnated with asphalt.

In Fig. 8.2.74*a*, F = force at end of brake handle; P = tangential force at rim of wheel; f = coefficient of friction of materials in contact;

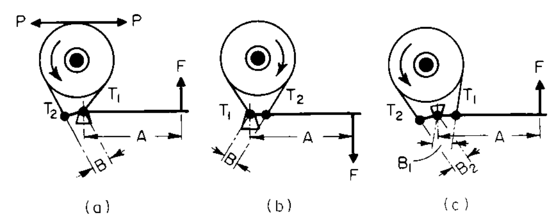


Fig. 8.2.74 Band brakes.

a = angle of wrap of band, deg; T_1 = total tension in band on tight side; T_2 = total tension in band on slack side. Then $T_1 - T_2 = P$ and $T_1/T_2 = 10^{0.0076fa} = 10^b$, where $b = 0.0076fa$. Also, $T_2 = P/(10^b - 1)$ and $T_1 = P \times 10^b/(10^b - 1)$. The values of $10^{0.0076fa}$ are given in Fig. 8.2.90 for a in radians.

For the arrangement shown in Fig. 8.2.74*a*,

$$FA = T_2B = PB/(10^b - 1)$$

and
$$F = PB/[A(10^b - 1)]$$

For the construction illustrated in Fig. 8.2.74*b*,

$$F = PB/\{A[10^b/(10^b - 1)]\}$$

For the differential brake shown in Fig. 8.2.74*c*,

$$F = (P/A)[(B_2 - 10^bB_1)/(10^b - 1)]$$

In this arrangement, the quantity 10^bB_1 must always be less than B_2, or the band will grip the wheel and the brake, or part of the mechanism to which it is attached, will rupture.

It is usual in practice to have the leverage ratio A/B for band brakes about 10 : 1.

If f for wood on iron is taken at 0.3 and the angle of wrap for the band is 270°, i.e., subtends three-fourths of the circumference, then $10^b = 4$ approx; the loads required for a given torque will be as follows for the cases just considered and for the leverage ratios stated above:

Band brake, Fig. 8.2.74*a*	$F = 0.033P$
Band brake, Fig. 8.2.74*b*	$F = 0.133P$
Band brake, Fig. 8.2.74*c*	$F = 0.016P$

In the case of Fig. 8.2.74*c*, the dimension B_2 must be greater than $B_1 \times 10^b$. Accordingly, B_1 is taken as ¼, A as 10, and, since $10^b = 4$, B_2 is taken as 1½.

The principal function of a brake is to absorb energy. This energy appears at the surface of the brake as heat, which must be carried away at a sufficiently rapid rate to prevent burning of the wooden blocks. Suitable proportions may be arrived at as follows:

Let p = unit pressure on brake surface, lb/in^2 = R (reaction against block)/area of block; v = velocity of brake rim surface, ft/s = $2\pi rn/60$, where n = speed of brake wheel, r/min. Then pv = work absorbed per in^2 of brake surface per second, and $pv \leq 1{,}000$ for intermittent applications of load with comparatively long periods of rest and poor means for carrying away heat (wooden blocks); $pv \leq 500$ for continuous application of load and poor means for carrying away heat (wooden blocks); $pv \leq 1{,}400$ for continuous application of load with effective means for carrying away heat (oil bath).

Cone Brake (see Fig. 8.2.75)

Uniform Wear

$$F = \frac{\pi P_a d}{2}(D - d)$$

$$T = \frac{\pi \mu P_a d}{8 \sin \alpha}(D^2 - d)$$

where P_a = maximum pressure occurring at $d/2$.

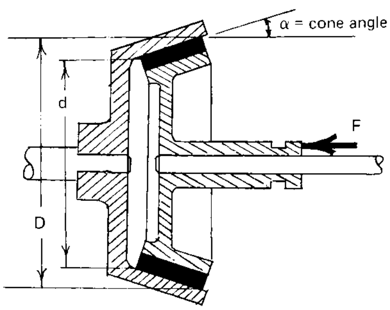


Fig. 8.2.75 Cone break.

Uniform Pressure

$$F = \frac{\pi P_a}{4}(D^2 - d^2)$$

$$T = \frac{\pi \mu P_a}{12 \sin \alpha}(D^3 - d^3) = \frac{F\mu}{3 \sin \alpha} \times \frac{D^3 - d^3}{D^2 - d^2}$$

Figure 8.2.76 shows a cone brake arrangement used for lowering heavy loads.

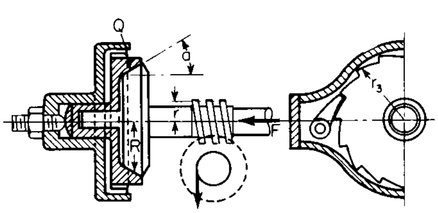


Fig. 8.2.76 Cone brake for lowering loads.

Disk Brakes (see Fig. 8.2.77)

Disk brakes are free from ''centrifugal'' effects, can have large frictional areas contained in small space, have better heat dissipation qualities than the rim type, and enjoy a favorable pressure distribution.

Uniform Wear

$$F = \frac{\pi P_a d}{2}(D - d)$$

$$T = \frac{F\mu}{4}(D + d)$$

Uniform Pressure

$$F = \frac{\pi P_a}{4}(D^2 - d^2)$$

$$T = \frac{F\mu}{3} \times \frac{D^3 - d^3}{D^2 - d^2}$$

These relations apply to a single surface of contact. For caliper disk, or multidisk brakes, the above relations are applied for each surface of contact.

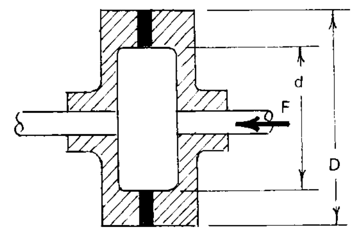


Fig. 8.2.77 Disk brake.

Selected friction materials and properties are listed in Table 8.2.38.

Frequently disk brakes are made as shown in Fig. 8.2.78. The pinion Q engages the gear in the drum (not shown). When the load is to be raised, power is applied through the gear and the connection between B and C is accomplished by the advancing of B along A and the clamping of the friction disks D and D and the ratchet wheel E. The reversal of the motor disconnects B and C. In lowering the load, only as much reversal of rotation of the gear is given as is needed to reduce the force in the friction disks so that the load may be lowered under control.

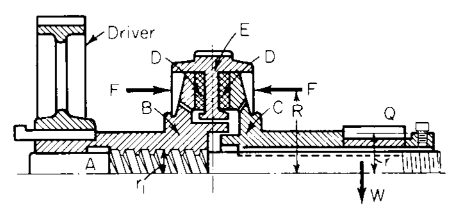


Fig. 8.2.78 Disk brake.

A **multidisk brake** is shown in Fig. 8.2.79. This type of construction results in an increase in the number of friction faces. The drum shaft is geared to the pinion A, while the motive power for driving comes through the gear G. In raising the load, direct connection is had between G, B, and A. In lowering, B moves relatively to G and forces the friction plates together, those plates fast to E being held stationary by the pawl on E. In the figure, there are three plates fast to E, one fast to G, and one fast to C.

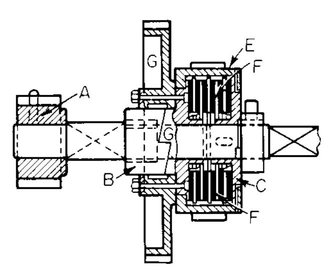


Fig. 8.2.79 Multidisk brake.

Eddy-current brakes (Fig. 8.2.80) are used with flywheels where quick braking is essential, and where large kinetic energy of the rotating

Table 8.2.38 Selected Friction Materials and Properties

Material	Opposing material	Friction coefficient			Max pressure		Max temperature	
		Dry	Wet	In oil	lb/in²	kPa	°F	°C
Sintered metal	Cast iron or steel	0.1–0.4	0.05–.01	0.05–0.08	150–250	1,000–1,720	450–1,250	232–677
Wood	Cast iron or steel	0.2–0.35	0.16	0.12–0.16	60–90	400–620	300	149
Leather	Cast iron or steel	0.3–0.5	0.12		10–40	70–280	200	93
Cork	Cast iron or steel	0.3–0.5	0.15–0.25	0.15–0.25	8–14	55–95	180	82
Felt	Cast iron or steel	0.22	0.18		5–10	35–70	280	138
Asbestos-woven	Cast iron or steel	0.3–0.6	0.1–0.2	0.08–0.10	50–100	350–700	400–500	204–260
Asbestos-molded	Cast iron or steel	0.2–0.5	0.08–0.12	0.06–0.09	50–150	350–1,000	400–500	204–260
Asbestos-impregnated	Cast iron or steel	0.32	0.12					
Cast iron	Cast iron	0.15–0.20	0.05	0.03–0.06	150–250	1,000–1,720	500	260
Cast iron	Steel			0.03–0.06	100–250	690–1,720	500	260
Graphite	Steel	0.25	0.05–0.1	0.12 (av)	300	2,100	370–540	188–282

masses precludes the use of block brakes due to excessive heating, as in reversible rolling mills. A number of poles *a* are electrically excited (north and south in turn) and create a magnetic flux which permeates the gap and the iron of the rim, causing eddy current. The flywheel energy

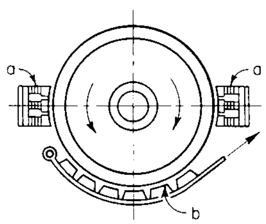


Fig. 8.2.80 Eddy-current brake.

is converted through these currents into heat. The hand brake *b* may be used for quicker stopping when the speed of the wheel is considerably decreased; i.e., when the eddy-current brake is inefficient. Two brakes are provided to avoid bending forces on the shaft.

Electric brakes are often used in cranes, bridges, turntables, and machine tools, where an automatic application of the brake is important as soon as power is cut off. The brake force is supplied by an adjustable spring which is counteracted by the force of a solenoid or a centrifugal thrustor. Interruption of current automatically applies the spring-activated brake shoes. Figures 8.2.81 and 8.2.82 show these types of electric brake.

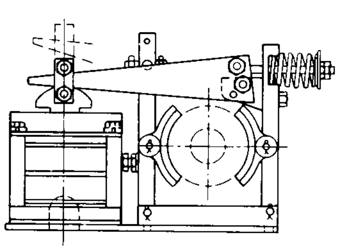


Fig. 8.2.81 Solenoid-type electric brake.

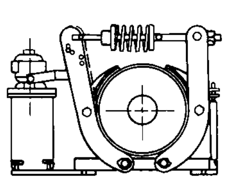


Fig. 8.2.82 Thrustor-type electric brake.

SHRINK, PRESS, DRIVE, AND RUNNING FITS

Inch Systems ANSI B4.1-1967 (R87) recommends preferred sizes, allowances, and tolerances for fits between plain cylindrical parts. Such fits include bearing, shrink and drive fits, etc. Terms used in describing fits are defined as follows: **Allowance:** minimum clearance (positive allowance) or maximum interference (negative allowance) between mating parts. **Tolerance:** total permissible variation of size. **Limits of size:** applicable maximum and minimum sizes. **Clearance fit:** one having limits of size so prescribed that a clearance always results when mating parts are assembled. **Interference fit:** In this case, limits are so prescribed that interference always results on assembly. **Transition fit:** This may have either a clearance or an interference on assembly. **Basic size:** one from which limits of size are derived by the application of allowances and tolerances. **Unilateral tolerance:** In this case a variation in size is permitted in only one direction from the basic size.

Fits are divided into the following general classifications: (1) running and sliding fits, (2) locational clearance fits, (3) transition fits, (4) locational interference fits, and (5) force or shrink fits.

1. **Running and sliding fits.** These are intended to provide similar running performance with suitable lubrication allowance throughout the range of sizes. These fits are further subdivided into the following classes:

Class RC1: close-sliding fits. Intended for accurate location of parts which must assemble without perceptible play.

Class RC2: sliding fits. Parts made with this fit move and turn easily but are not intended to run freely; also, in larger sizes they may seize under small temperature changes.

Class RC3: precision-running fits. These are intended for precision work at slow speeds and light journal pressures but are not suitable where appreciable temperature differences are encountered.

Class RC4: close-running fits. For running fits on accurate machinery with moderate surface speeds and journal pressures, where accurate location and minimum play is desired.

Classes RC5 and RC6: medium-running fits. For higher running speeds or heavy journal pressures.

Class RC7: free-running fits. For use where accuracy is not essential, or where large temperature variations are likely to be present, or both.

Classes RC8 and RC9: loose-running fits. For use with materials such as cold-rolled shafting or tubing made to commercial tolerances.

Limits of clearance given in ANSI B4.1-1967 (R87) for each of these classes are given in Table 8.2.39. Hole and shaft tolerances are listed on a unilateral tolerance basis in this reference to give the clearance limits of Table 8.2.39, the hole size being the basic size.

2. **Locational clearance fits.** These are intended for normally stationary parts which can, however, be freely assembled or disassembled. These are subdivided into various classes which run from snug fits for parts requiring accuracy of location, through medium clearance fits (spigots) to the looser fastener fits where freedom of assembly is of prime importance.

Table 8.2.39 Limits of Clearance for Running and Sliding Fits (Basic Hole)
(Limits are in thousandths of an inch on diameter)

Nominal size range, in	Class								
	RC1	RC2	RC3	RC4	RC5	RC6	RC7	RC8	RC9
0–0.12	0.1 0.45	0.1 0.55	0.3 0.95	0.3 1.3	0.6 1.6	0.6 2.2	1.0 2.6	2.5 5.1	4.0 8.1
0.12–0.24	1.5 0.5	0.15 0.65	0.4 1.2	0.4 1.6	0.8 2.0	0.8 2.7	1.2 3.1	2.8 5.8	4.5 9.0
0.24–0.40	0.2 0.6	0.2 0.85	0.5 1.5	0.5 2.0	1.0 2.5	1.0 3.3	1.6 3.9	3.0 6.6	5.0 10.7
0.40–0.71	0.25 0.75	0.25 0.95	0.6 1.7	0.6 2.3	1.2 2.9	1.2 3.8	2.0 4.6	3.5 7.9	6.0 12.8
0.71–1.19	0.3 0.95	0.3 1.2	0.8 2.1	0.8 2.8	1.6 3.6	1.6 4.8	2.5 5.7	4.5 10.0	7.0 15.5
1.19–1.97	0.4 1.1	0.4 1.4	1.0 2.6	1.0 3.6	2.0 4.6	2.0 6.1	3.0 7.1	5.0 11.5	8.0 18.0
1.97–3.15	0.4 1.2	0.4 1.6	1.2 3.1	1.2 4.2	2.5 5.5	2.5 7.3	4.0 8.8	6.0 13.5	9.0 20.5
3.15–4.73	0.5 1.5	0.5 2.0	1.4 3.7	1.4 5.0	3.0 6.6	3.0 8.7	5.0 10.7	7.0 15.5	10.0 24.0

3. **Transition fits.** These are for applications where accuracy of location is important, but a small amount of either clearance or interference is permissible.

4. **Locational interference fits.** Used where accuracy of location is of prime importance and for parts requiring rigidity and alignment with no special requirements for bore pressure.

Data on clearance limits, interference limits, and hole and shaft diameter tolerances for locational clearance fits, transition fits, and locational interference fits are given in ANSI B4.1-1967 (R87).

5. **Force or shrink fits.** These are characterized by approximately constant bore pressures throughout the range of sizes; interference varies almost directly as the diameter, and the differences between maximum and minimum values of interference are small. These are divided into the following classes:

Class FN1: light-drive fits. For applications requiring light assembly pressures (thin sections, long fits, cast-iron external members).

Class FN2: medium-drive fits. Suitable for ordinary steel parts or for shrink fits on light sections. These are about the tightest fits that can be used on high-grade cast-iron external members.

Class FN3: heavy-drive fits. For heavier steel parts or shrink fits in medium sections.

Classes FN4 and FN5: force fits. These are suitable for parts which can be highly stressed. Shrink fits are used instead of press fits in cases where the heavy pressing forces required for mounting are impractical.

In Table 8.2.40 are listed the limits of interference (maximum and minimum values) for the above classes of force or shrink fits for various diameters, as given in ANSI B4.1-1967 (R87). Hole and shaft tolerances to give these interference limits are also listed in this reference.

Metric System ANSI B4.2-1978 (R94) and ANSI B4.3-1978 (R94) define limits and fits for cylindrical parts, and provide tables listing preferred values.

The standard ANSI B4.2-1978 (R94) is essentially in accord with ISO R286.

ANSI B4.2 provides 22 basic deviations, each for the shaft (a to z plus js), and the hole (A to Z plus Js). International has 18 tolerance grades: IT 01, IT 0, and IT 1 through 16.

IT grades are roughly applied as follows: measuring tools, 01 to 7; fits, 5 to 11; material, 8 to 14; and large manufacturing tolerances, 12 to 16. See Table 8.2.42 for metric preferred fits.

Basic size—The basic size is the same for both members of a fit, and is the size to which limits or deviations are assigned. It is designated by 40 in 40H7.

Table 8.2.40 Limits of Interference for Force and Shrink Fits
(Limits are in thousandths of an inch on diameter)

Nominal size range, in	Class				
	FN 1	FN 2	FN 3	FN 4	FN 5
0.04–0.12	0.05 0.5	0.2 0.85		0.3 0.95	0.3 1.3
0.12–0.24	0.1 0.6	0.2 1.0		0.95 1.2	1.3 1.7
0.24–0.40	0.1 0.75	0.4 1.4		0.6 1.6	0.5 2.0
0.40–0.56	0.1 0.8	0.5 1.6		0.7 1.8	0.6 2.3
0.56–0.71	0.2 0.9	0.5 1.6		0.7 1.8	0.8 2.5
0.71–0.95	0.2 1.1	0.6 1.9		0.8 2.1	1.0 3.0
0.95–1.19	0.3 1.2	0.6 1.9	0.8 2.1	1.0 2.3	1.3 3.3
1.19–1.58	0.3 1.3	0.8 2.4	1.0 2.6	1.5 3.1	1.4 4.0
1.58–1.97	0.4 1.4	0.8 2.4	1.2 2.8	1.8 3.4	2.4 5.0
1.97–2.56	0.6 1.8	0.8 2.7	1.3 3.2	2.3 4.2	3.2 6.2
2.56–3.15	0.7 1.9	1.0 2.9	1.8 3.7	2.8 4.7	4.2 7.2
3.15–3.94	0.9 2.4	1.4 3.7	2.1 4.4	3.6 5.9	4.8 8.4
3.94–4.73	1.1 2.6	1.6 3.9	2.6 4.9	4.6 6.9	5.8 9.4
4.73–5.52	1.2 2.9	1.9 4.5	3.4 6.0	5.4 8.0	7.5 11.6
5.52–6.30	1.5 3.2	2.4 5.0	3.4 6.0	5.4 8.0	9.5 13.6
6.30–7.09	1.8 3.5	2.9 5.5	4.4 7.0	6.4 9.0	9.5 13.6

Deviation—The algebraic difference between a size and the corresponding basic size.

Upper deviation—The algebraic difference between the maximum limit of size and the corresponding basic size.

Lower deviation—The algebraic difference between the minimum limit of size and the corresponding basic size.

Fundamental deviation—That one of the two deviations closest to the basic size. It is designated by the letter H in 40H7.

Tolerance—The difference between the maximum and minimum size limits on a part.

International tolerance grade (IT)—A group of tolerances which vary depending on the basic size, but which provide the same relative level of accuracy within a grade. It is designated by 7 in 40H7 (IT 7).

Tolerance zone—A zone representing the tolerance and its position in relation to the basic size. The symbol consists of the fundamental deviation letter and the tolerance grade number (i.e., H7).

Hole basis—The system of fits where the minimum hole size is basic. The fundamental deviation for a hole basis system is H.

Shaft basis—Maximum shaft size is basic in this system. Fundamental deviation is h. NOTE: Capital letters refer to the hole and lowercase letters to the shaft.

Clearance fit—A fit in which there is clearance in the assembly for all tolerance conditions.

Interference fit—A fit in which there is interference for all tolerance conditions. Table 8.2.41 lists preferred metric sizes. Table 8.2.42 lists preferred tolerance zone combinations for clearance, transition and interference fits. Table 8.2.43 lists dimensions for the grades corresponding to preferred fits. Table 8.2.44*a* and *b* lists limits (numerical) of preferred hole-basis clearance, transitions, and interference fits.

Stresses Produced by Shrink or Press Fit

STEEL HUB ON STEEL SHAFT. The maximum equivalent stress, pounds per square inch, set up by a given press-fit allowance (in inches per inch of shaft diameter) is equal to $3x \times 10^7$, where x is the allowance per inch of shaft diameter (Baugher, *Trans. ASME*, 1931, p. 85). The press-fit pressures set up between a steel hub and shaft, for various ratios d/D between shaft and hub outside diameters, are given in Fig. 8.2.83. These curves are accurate to 5 percent even if the shaft is hollow, provided the inside shaft diameter is not over 25 percent of the outside. The equivalent stress given above is based on the maximum shear theory and is numerically equal to the radial-fit pressure added to the tangential tension in the hub. Where the shaft is hollow, with an inside diameter equal to more than about 25 percent of the outside diameter, the allowance in inches per inch to obtain an equivalent hub stress of 30,000 lb/in² may be determined by using Lamé's thick-cylinder formulas (*Jour. Appl.*

Table 8.2.41 Preferred Sizes (Metric)

Basic size, mm		Basic size, mm		Basic size, mm	
First choice	Second choice	First choice	Second choice	First choice	Second choice
1		10		100	
	1.1		11		110
1.2		12		120	
	1.4		14		140
1.6		16		160	
	1.8		18		180
2		20		200	
	2.2		22		220
2.5		25		250	
	2.8		28		280
3		30		300	
	3.5		35		350
4		40		400	
	4.5		45		450
5		50		500	
	5.5		55		550
6		60		600	
	7		70		700
8		80		800	
	9		90		900
				1000	

SOURCE: ANSI B 4.2-1978 (R94), reproduced by permission.

Table 8.2.42 Description of Preferred Fits (Metric)

ISO symbol: Hole basis	ISO symbol: Shaft basis	Description	
		Clearance fits	More clearance
H11/c11	C11/h11	*Loose-running* fit for wide commercial tolerances or allowances on external members.	
H9/d9	D9/h9	*Free-running* fit not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures.	
H8/f7	F8/h7	*Close-running* fit for running on accurate machines and for accurate location at moderate speeds and journal pressures.	
H7/g6	G7/h6	*Sliding fit* not intended to run freely, but to move and turn freely and locate accurately.	
H7/h6	H7/h6	*Locational clearance* fit provides snug fit for locating stationary parts; but can be freely assembled and disassembled.	
		Transition fits	
H7/k6	K7/h6	*Locational transition* fit for accurate location, a compromise between clearance and interference.	
H7/n6	N7/h6	*Locational transition* fit for more accurate location where greater interference is permissible.	
		Interference fits	More interference
H7/p6*	P7/h6	*Locational interference* fit for parts requiring rigidity and alignment with prime accuracy of location but without special bore pressure requirements.	
H7/s6	S7/h6	*Medium-drive* fit for ordinary steel parts or shrink fits on light sections, the tightest fit usable with cast iron.	
H7/u6	U7/h6	*Force* fit suitable for parts which can be highly stressed or for shrink fits where the heavy pressing forces required are impractical.	

* Transition fit for basic sizes in range from 0 through 3 mm.
SOURCE: ANSI B4.2-1978 (R94), reproduced by permission.

Table 8.2.43 International Tolerance Grades

Basic sizes: Over	Basic sizes: Up to and including	Tolerance grades, mm: IT6	IT7	IT8	IT9	IT10	IT11
0	3	0.006	0.010	0.014	0.025	0.040	0.060
3	6	0.008	0.012	0.018	0.030	0.048	0.075
6	10	0.009	0.015	0.022	0.036	0.058	0.090
10	18	0.011	0.018	0.027	0.043	0.070	0.110
18	30	0.013	0.021	0.033	0.052	0.084	0.130
30	50	0.016	0.025	0.039	0.062	0.100	0.160
50	80	0.019	0.030	0.046	0.074	0.120	0.190
80	120	0.022	0.035	0.054	0.087	0.140	0.220
120	180	0.025	0.040	0.063	0.100	0.160	0.250
180	250	0.029	0.046	0.072	0.115	0.185	0.290
250	315	0.032	0.052	0.081	0.130	0.210	0.320
315	400	0.036	0.057	0.089	0.140	0.230	0.360
400	500	0.040	0.063	0.097	0.155	0.250	0.400
500	630	0.044	0.070	0.110	0.175	0.280	0.440
630	800	0.050	0.080	0.125	0.200	0.320	0.500
800	1,000	0.056	0.090	0.140	0.230	0.360	0.560
1,000	1,250	0.066	0.105	0.165	0.260	0.420	0.660
1,250	1,600	0.078	0.125	0.195	0.310	0.500	0.780
1,600	2,000	0.092	0.150	0.230	0.370	0.600	0.920
2,000	2,500	0.110	0.175	0.280	0.440	0.700	1.100
2,500	3,150	0.135	0.210	0.330	0.540	0.860	1.350

SOURCE: ANSI B4.2-1978 (R94), reproduced by permission.

Mech., 1937, p. A-185). It should be noted that these curves hold only when the maximum equivalent stress is below the yield point; above the yield point, plastic flow occurs and the stresses are less than calculated.

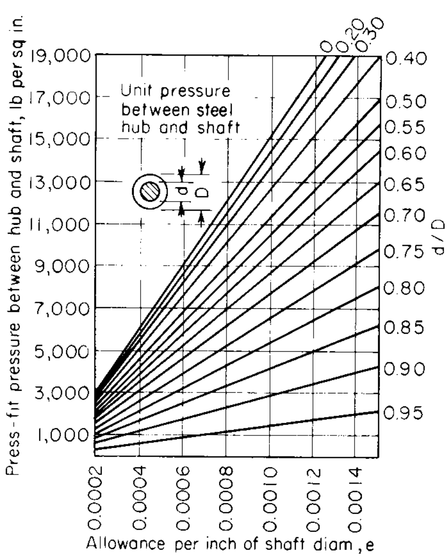


Fig. 8.2.83 Press-fit pressures between steel hub and shaft.

Cast-Iron Hub on Steel Shaft Where the shaft is solid, or hollow with an inside diameter not over 25 percent of the outside diameter, Fig. 8.2.84 may be used to determine maximum tensile stresses in the cast-iron hub, resulting from the press-fit allowance; for various ratios *d/D*, Fig. 8.2.85 gives the press-fit pressures. These curves are based on a modulus of elasticity of 30×10^6 lb/in^2 for steel and 15×10^6 for cast iron. For a hollow shaft with an inside diameter more than about ¼ the outside, the Lamé formulas may be used.

Pressure Required in Making Press Fits The force required to press a hub on the shaft is given by $\pi fpdl$, where l is length of fit, p the unit press-fit pressure between shaft and hub, f the coefficient of friction, and d the shaft diameter. Values of f varying from 0.03 to 0.33 have been reported, the lower values being due to yielding of the hub as a consequence of too high a fit allowance; the average is around 0.10 to 0.15. (For additional data see Horger and Nelson, ''Design Data and Methods,'' ASME, 1953, pp. 87–91.)

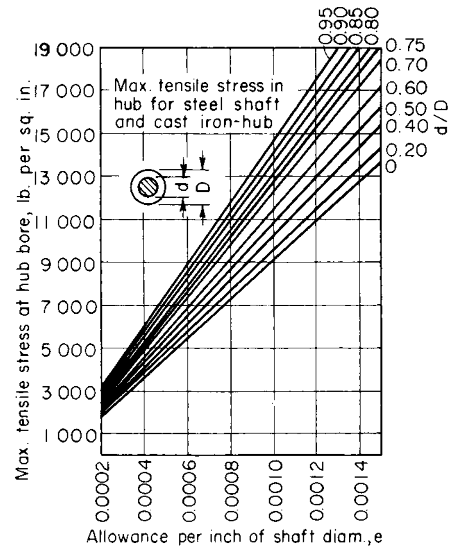


Fig. 8.2.84 Variation of tensile stress in cast-iron hub in press-fit allowance.

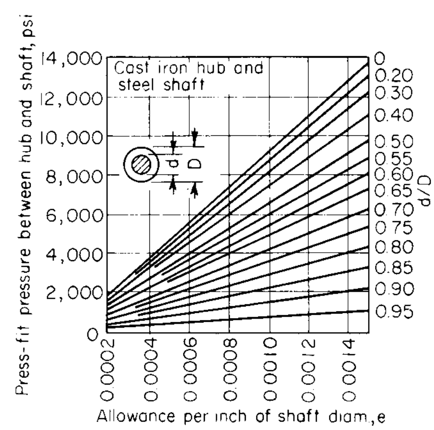


Fig. 8.2.85 Press-fit pressures between cast-iron hub and shaft.

Torsional Holding Ability The torque required to cause complete slippage of a press fit is given by $T = \frac{1}{2}\pi fpld^2$. Local slippage will usually occur near the end of the fit at much lower torques. If the torque is alternating, stress concentration and rubbing corrosion will occur at the hub face so that, eventually, fatigue failure may occur at considerably lower torques. Only in cases of static torque application is it justifiable to use ultimate torque as a basis for design.

A designer can often improve shrink-, press-, and slip-fit cylindrical assemblies with adhesives. When applied, adhesives can achieve high frictional force with attendant greater torque transmission without extra bulk, and thus augment or even replace press fits, compensate for differential thermal expansion, make fits with leakproof seals, eliminate backlash and clearance, etc.

The adhesives used are the anaerobic (see Sec. 6.8, ''Adhesives'') variety, such as Loctite products. Such adhesives destabilize and tend to harden when deprived of oxygen. Design suggestions on the use of such adhesives appear in industrial catalogs.

SHAFTS, AXLES, AND CRANKS

Most shafts are subject to combined bending and torsion, either of which may be steady or variable. Impact conditions, such as sudden starting and stopping, will cause momentary peak stresses greater than those related to the steady or variable portions of operation.

Design of shafts requires a theory of failure to express a stress in terms of loads and shaft dimensions, and an allowable stress as fixed by material strength and safety factor. Maximum shear theory of failure and distortion energy theory of failure are the two most commonly used in shaft design. Material strengths can be estimated from any one of several analytic representations of combined-load fatigue test data, starting from the linear (Soderberg, modified Goodman) which tend to give conservative designs to the nonlinear (Gerber parabolic, quadratic, Kececioglu, Bagci) which tend to give less conservative designs.

When linear representations of material strengths are used, and where both bending and torsion stresses have steady and variable components, the maximum shear theory and the distortion energy theory lead to somewhat similar formulations:

$$d = \left\{ \varepsilon \, \frac{n}{\pi} \left[\left(\frac{T_a}{S_{se}} + \frac{T_m}{S_{sy}} \right)^2 + \left(\frac{M_a}{S_{se}} + \frac{M_m}{S_{sy}} \right)^2 \right]^{1/2} \right\}^{1/3}$$

where $\varepsilon = 32$ (maximum shear theory) or 48 (distortion energy theory); d = shaft diameter; n = safety factor; T_a = amplitude torque = $(T_{\max} - T_{\min})/2$; T_m = mean torque = $(T_{\max} + T_{\min})/2$; M_a = amplitude bending moment = $(M_{\max} - M_{\min})/2$; M_m = mean bending moment = $(M_{\max} + M_{\min})/2$; S_{sy} = yield point in shear; S_{se} = completely corrected shear endurance limit = $S'_{se}k_a k_b k_c k_d / K_f$; S'_{se} = statistical average endurance limit of mirror finish, standard size, laboratory test specimen at standard room temperature; k_a = surface factor, a decimal to adjust S'_e for other than mirror finish; k_b = size factor, a decimal to adjust S'_e for other than standard test size; k_c = reliability factor, a decimal to adjust S'_{se} to other than its implied statistical average of 50 percent safe, 50 percent fail rate (50 percent reliability); k_d = temperature factor, decimal, to adjust S'_{se} to other than room temperature; $K_f = 1 + q(K_t - 1)$ = actual or fatigue stress concentration factor, a number greater than unity to adjust the nominal stress implied by T_a and M_a to a peak stress as induced by stress-raising conditions such as holes, fillets, keyways, press fits, etc. (K_f for T_a need not necessarily be the same as K_f for M_a); q = notch sensitivity; K_t = theoretical or geometric stress concentration factor.

For specific values of endurance limits and various factors the reader is referred to the technical literature (e.g., ASTM, NASA technical reports, ASME technical papers) or various books [e.g., ''Machinery's Handbook'' (Industrial Press, New York) and machine design textbooks].

If one allows for a variation of at least 15 percent, the following approximation is useful for the endurance limit in bending: $S'_e = 0.5S_{ut}$. This becomes for maximum shear theory $S'_{se} = 0.5(0.5S_{ut})$ and for distortion energy theory $S'_{se} = 0.577(0.5S_{ut})$.

Representative or approximate values for the various factors mentioned above were abstracted from Shigley, ''Mechanical Engineering Design,'' McGraw-Hill, and appear below with permission.

Surface Factor k_a

Surface condition	S_{ut}, kips 60	120	180	240
Polished	1.00	1.00	1.00	1.00
Ground	0.89	0.89	0.89	0.89
Machined or cold-drawn	0.84	0.71	0.66	0.63
Hot-rolled	0.70	0.50	0.39	0.31
As forged	0.54	0.36	0.27	0.20

Size Factor k_b

$$k_b = \begin{cases} 0.869d^{-0.097} & 0.3 \text{ in} < d \le 10 \text{ in} \\ 1 & d \le 0.3 \text{ in or } d \le 8 \text{ mm} \\ 1.189d^{-0.097} & 8 \text{ mm} < d < 250 \text{ mm} \end{cases}$$

Reliability Factor k_c

Reliability, %	k_c
50	1.00
90	0.89
95	0.87
99	0.81

Temperature Factor k_d

Temperature °F	°C	k_d
840	450	1.00
940	482	0.71
1,020	550	0.42

Notch Sensitivity q

S_{ut}, kips	Notch radius r, in 0.02	0.06	0.10	0.14
60	0.56	0.70	0.74	0.78
100	0.68	0.79	0.83	0.85
150	0.80	0.90	0.91	0.92
200	0.90	0.95	0.96	0.96

See Table 8.2.45 for fatigue stress concentration factors for plain press fits.

A general representation of material strengths (Marin, Design for Fatigue Loading, *Mach. Des.* **29,** no. 4, Feb. 21, 1957, pp. 128–131, and series of the same title) is given as

$$\left(\frac{S_a}{S_e}\right)^m + \left(\frac{KS_m}{S_{ut}}\right)^p = 1$$

where S_a = variable portion of material strength; S_m = mean portion of material strength; S_e = adjusted endurance limit; S_{ut} = ultimate strength. Table 8.2.46 lists the constants m, K, and P for various failure criteria.

For purposes of design, safety factors are introduced into the equation resulting in:

$$\left(\frac{\sigma'_{a,p}}{S_e/n_{se}}\right)^m + \left(\frac{K\,\sigma'_{m,p}}{S_{ut}/n_{ut}}\right)^P = 1$$

where

$$\sigma'_{a,p} = \frac{1}{\pi d^3}\sqrt{(32n_{Ma}M_a)^2 + 3(16n_{\tau a}T_a)^2}$$

$$\sigma'_{m,p} = \frac{1}{\pi d^3}\sqrt{(32n_{Mm}M_m)^2 + 3(16n_{\tau m}T_m)^2}$$

and n_{ij} = safety factor pertaining to a particular stress (that is, n_a = safety factor for amplitude shear stress).

Stiffness of shafting may become important where critical speeds, vibration, etc., may occur. Also, the lack of sufficient stiffness in shafts may give rise to bearing troubles. Critical speeds of shafts in torsion or bending and shaft deflections may be calculated using the methods of Sec. 5. For shafts of variable diameter see Spotts, ''Design of Machine Elements,'' Prentice-Hall. In order to avoid trouble where sleeve bearings are used, the angular deflections at the bearings in general must be kept within certain limits. One rule is to make the shaft deflection over the bearing width equal to a small fraction of the oil-film thickness. Note that since stiffness is proportional to the modulus of elasticity, alloy-steel shafts are no stiffer than carbon-steel shafts of the same diameter.

Crankshafts For calculating the torsional stiffness of crankshafts, the formulas given in Sec. 5 may be used.

Table 8.2.44a Limits of Fits
(Dimensions in mm)

		Preferred hole basis clearance fits														
		Loose-running			Free-running			Close-running			Sliding			Locational clearance		
Basic size		Hole H11	Shaft c11	Fit	Hole H9	Shaft d9	Fit	Hole H8	Shaft f7	Fit	Hole H7	g6	Fit	Hole H7	Shaft h6	Fit
1	max	1.060	0.940	0.180	1.025	0.980	0.070	1.014	0.994	0.030	1.010	0.998	0.018	1.010	1.000	0.016
	min	1.000	0.880	0.060	1.000	0.955	0.020	1.000	0.984	0.006	1.000	0.992	0.002	1.000	0.994	0.000
1.2	max	1.260	1.140	0.180	1.225	1.180	0.070	1.214	1.194	0.030	1.210	1.198	0.018	1.210	1.200	0.016
	min	1.200	1.080	0.060	1.200	1.155	0.020	1.200	1.184	0.006	1.200	1.192	0.002	1.200	1.194	0.000
1.6	max	1.660	1.540	1.180	1.625	1.580	0.070	1.614	1.594	0.030	1.610	1.598	0.018	1.610	1.600	0.016
	min	1.600	1.480	0.060	1.600	1.555	0.020	1.600	1.584	0.006	1.600	1.592	0.002	1.600	1.594	0.000
2	max	2.060	1.940	0.180	2.025	1.980	0.070	2.014	1.994	0.030	2.010	1.998	0.018	2.010	2.000	0.016
	min	2.000	1.880	0.060	2.000	1.955	0.020	2.000	1.984	0.006	2.000	1.992	0.002	2.000	1.994	0.000
2.5	max	2.560	2.440	0.180	2.525	2.480	0.070	2.514	2.494	0.030	2.510	2.498	0.018	2.510	2.500	0.016
	min	2.500	2.380	0.060	2.500	2.455	0.020	2.500	2.484	0.006	2.500	2.492	0.002	2.500	2.494	0.000
3	max	3.060	2.940	0.180	3.025	2.980	0.070	3.014	2.994	0.030	3.010	2.998	0.018	3.010	3.000	0.016
	min	3.000	2.880	0.060	3.000	2.955	0.020	3.000	2.984	0.006	3.000	2.992	0.002	3.000	2.994	0.000
4	max	4.075	3.930	0.220	4.030	3.970	0.090	4.018	3.990	0.040	4.012	3.996	0.024	4.012	3.000	0.020
	min	4.000	3.855	0.070	4.000	3.940	0.030	4.000	3.978	0.010	4.000	3.988	0.004	4.000	3.992	0.000
5	max	5.075	4.930	0.220	5.030	4.970	0.090	5.018	4.990	0.040	5.012	4.996	0.024	5.012	5.000	0.020
	min	5.000	4.855	0.070	5.000	4.940	0.030	5.000	4.978	0.010	5.000	4.988	0.004	5.000	4.992	0.000
6	max	6.075	5.930	0.220	6.030	5.970	0.090	6.018	5.990	0.040	6.012	5.996	0.024	6.012	6.000	0.020
	min	6.000	5.855	0.070	6.000	5.940	0.030	6.000	5.978	0.010	6.000	5.988	0.004	6.000	5.992	0.000
8	max	8.090	7.920	0.260	8.036	7.960	0.112	8.022	7.987	0.050	8.015	7.995	0.029	8.015	8.000	0.024
	min	8.000	7.830	0.080	8.000	7.924	0.040	8.000	7.972	0.013	8.000	7.986	0.005	8.000	7.991	0.000
10	max	10.090	9.920	0.260	10.036	9.960	0.112	10.022	9.987	0.050	10.015	0.995	0.029	10.015	10.000	0.024
	min	10.000	9.830	0.080	10.000	9.924	0.040	10.000	9.972	0.013	10.000	9.986	0.005	10.000	9.991	0.000
12	max	12.110	11.905	0.315	12.043	11.950	0.136	12.027	11.984	0.061	12.018	11.994	0.035	12.018	12.000	0.029
	min	12.000	11.795	0.095	12.000	11.907	0.050	12.000	11.966	0.016	12.000	11.983	0.006	12.000	11.989	0.000
16	max	16.110	15.905	0.315	16.043	15.950	0.136	16.027	15.984	0.061	16.018	15.994	0.035	16.018	16.000	0.029
	min	16.000	15.795	0.095	16.000	15.907	0.050	16.000	15.966	0.016	16.000	15.983	0.006	16.000	15.989	0.000
20	max	20.130	19.890	0.370	20.052	19.935	0.169	20.033	19.980	0.074	20.021	19.993	0.041	20.021	20.000	0.034
	min	20.000	19.760	0.110	20.000	19.883	0.065	20.000	19.959	0.020	20.000	19.980	0.007	20.000	19.987	0.000
25	max	25.130	24.890	0.370	25.052	24.935	0.169	25.033	24.980	0.074	25.021	24.993	0.041	25.021	25.000	0.034
	min	25.000	24.760	0.110	25.000	24.883	0.065	25.000	24.959	0.020	25.000	24.980	0.007	25.000	24.987	0.000
30	max	30.130	29.890	0.370	30.052	29.935	0.169	30.033	29.980	0.074	30.021	29.993	0.041	30.021	30.000	0.034
	min	30.000	29.760	0.110	30.000	29.883	0.065	30.000	29.959	0.020	30.000	29.980	0.007	30.000	29.987	0.000
40	max	40.160	39.880	0.440	40.062	39.920	0.204	40.039	39.975	0.089	40.025	39.991	0.050	40.025	40.000	0.041
	min	40.000	39.720	0.120	40.000	39.858	0.080	40.000	39.950	0.025	40.000	39.975	0.009	40.000	39.984	0.000
50	max	50.160	49.870	0.450	50.062	49.920	0.204	50.039	49.975	0.089	50.025	49.991	0.050	50.025	50.000	0.041
	min	50.000	49.710	0.130	50.000	49.858	0.080	50.000	49.950	0.025	50.000	49.975	0.009	50.000	49.984	0.000
60	max	60.190	59.860	0.520	60.074	59.900	0.248	60.046	59.970	0.106	60.030	59.990	0.059	60.030	60.000	0.049
	min	60.000	59.670	0.140	60.000	59.826	0.100	60.000	59.940	0.030	60.000	59.971	0.010	60.000	59.981	0.000
80	max	80.190	79.850	0.530	80.074	79.900	0.248	80.046	79.970	0.106	80.030	79.990	0.059	80.030	80.000	0.049
	min	80.000	79.660	0.150	80.000	79.826	0.100	80.000	79.940	0.030	80.000	70.971	0.010	80.000	79.981	0.000
100	max	100.220	99.830	0.610	100.087	99.880	0.294	100.054	99.964	0.125	100.035	99.988	0.069	100.035	100.000	0.057
	min	100.000	99.610	0.170	100.000	99.793	0.120	100.000	99.929	0.036	100.000	99.966	0.012	100.000	99.978	0.000
120	max	120.220	119.820	0.620	120.087	119.880	0.294	120.054	119.964	0.125	120.035	119.988	0.069	120.035	120.000	0.057
	min	120.000	119.600	0.180	120.000	119.793	0.120	120.000	119.929	0.036	120.000	119.966	0.012	120.000	119.978	0.000
160	max	160.250	159.790	0.710	160.100	159.855	0.345	160.063	159.957	0.146	160.040	159.986	0.079	160.040	160.000	0.065
	min	160.000	159.540	0.210	160.000	159.755	0.145	160.000	159.917	0.043	160.000	159.961	0.014	160.000	159.975	0.000

Table 8.2.44*b* Limits of Fits
(Dimensions in mm)

		Preferred hole basis transition and interference fits														
		Locational transition			Locational transition			Locational interference			Medium drive			Force		
Basic size		Hole H7	Shaft k6	Fit	Hole H7	Shaft n6	Fit	Hole H7	Shaft p6	Fit	Hole H7	Shaft s6	Fit	Hole H7	Shaft u6	Fit
1	max	1.010	1.006	0.010	1.010	1.010	0.006	1.010	1.012	0.004	1.010	1.020	−0.004	1.010	1.024	−0.008
	min	1.000	1.000	−0.006	1.000	1.004	−0.010	1.000	1.006	−0.012	1.000	1.014	−0.020	1.000	1.018	−0.024
1.2	max	1.210	1.206	0.010	1.210	1.210	0.006	1.210	1.212	0.004	1.210	1.220	−0.004	1.210	1.224	−0.008
	min	1.200	1.200	−0.006	1.200	1.204	−0.010	1.200	1.206	−0.012	1.200	1.214	−0.020	1.200	1.218	−0.024
1.6	max	1.610	1.606	0.010	1.610	1.610	0.006	1.610	1.612	0.004	1.610	1.620	−0.004	1.610	1.624	−0.008
	min	1.600	1.600	−0.006	1.600	1.604	−0.010	1.600	1.606	−0.012	1.600	1.614	−0.020	1.600	1.618	−0.024
2	max	2.010	2.006	0.010	2.010	2.010	0.006	2.010	2.012	0.004	2.010	2.020	−0.004	2.010	2.024	−0.008
	min	2.000	2.000	−0.006	2.000	2.004	−0.010	2.000	2.006	−0.012	2.000	2.014	−0.020	2.000	2.018	−0.024
2.5	max	2.510	2.506	0.010	2.510	2.510	0.006	2.510	2.512	0.004	2.510	2.520	−0.004	2.510	2.524	−0.008
	min	2.500	2.500	−0.006	2.500	2.504	−0.010	2.500	2.506	−0.012	2.500	2.514	−0.020	2.500	2.518	−0.024
3	max	3.010	3.006	0.010	3.010	3.010	0.006	3.010	3.012	0.004	3.010	3.020	−0.004	3.010	3.024	−0.008
	min	3.000	3.000	−0.006	3.000	3.004	−0.010	3.000	3.006	−0.012	3.000	3.014	−0.020	3.000	3.018	−0.024
4	max	4.012	4.009	0.011	4.012	4.016	0.004	4.012	4.020	0.000	4.012	4.027	−0.007	4.012	4.031	−0.011
	min	4.000	4.001	−0.009	4.000	4.008	−0.016	4.000	4.012	−0.020	4.000	4.019	−0.027	4.000	4.023	−0.031
5	max	5.012	5.009	0.011	5.012	5.016	0.004	5.012	5.020	0.000	5.012	5.027	−0.007	5.012	5.031	−0.011
	min	5.000	5.001	−0.009	5.000	5.008	−0.016	5.000	5.012	−0.020	5.000	5.019	−0.027	5.000	5.023	−0.031
6	max	6.012	6.009	0.011	6.012	6.016	0.004	6.012	6.020	0.000	6.012	6.027	−0.007	6.012	6.031	−0.011
	min	6.000	6.001	−0.009	6.000	6.008	−0.016	6.000	6.012	−0.020	6.000	6.019	−0.027	6.000	6.023	−0.031
8	max	8.015	8.010	0.014	8.015	8.019	0.005	8.015	8.024	0.000	8.015	8.032	−0.008	8.015	8.037	−0.013
	min	8.000	8.001	−0.010	8.000	8.010	−0.019	8.000	8.015	−0.024	8.000	8.023	−0.032	8.000	8.028	−0.037
10	max	10.015	10.010	0.014	10.015	10.019	0.005	10.015	10.024	0.000	10.015	10.032	−0.008	10.015	10.037	−0.013
	min	10.000	10.001	−0.010	10.000	10.010	−0.019	10.000	10.015	−0.024	10.000	10.023	−0.032	10.000	10.028	−0.037
12	max	12.018	12.012	0.017	12.018	12.023	0.006	12.018	12.029	0.000	12.018	12.039	−0.010	12.018	12.044	−0.015
	min	12.000	12.001	−0.012	12.000	12.012	−0.023	12.000	12.018	−0.029	12.000	12.028	−0.039	12.000	12.033	−0.044
16	max	16.018	16.012	0.017	16.018	16.023	0.006	16.018	16.029	0.000	16.018	16.039	−0.010	16.018	16.044	−0.015
	min	16.000	16.001	−0.012	16.000	16.012	−0.023	16.000	16.018	−0.029	16.000	16.028	−0.039	16.000	16.033	−0.044
20	max	20.021	20.015	0.019	20.021	20.028	0.006	20.021	20.035	−0.001	20.021	20.048	−0.014	20.021	20.054	−0.020
	min	20.000	20.002	−0.015	20.000	20.015	−0.028	20.000	20.022	−0.035	20.000	20.035	−0.048	20.000	20.041	−0.054
25	max	25.021	25.015	0.019	25.021	25.028	0.006	25.021	25.035	−0.001	25.021	25.048	−0.014	25.021	25.061	−0.027
	min	25.000	25.002	−0.015	25.000	25.015	−0.028	25.000	25.022	−0.035	25.000	25.035	−0.048	25.000	25.048	−0.061
30	max	30.021	30.015	0.019	30.021	30.028	0.006	30.021	30.035	−0.001	30.021	30.048	−0.014	30.021	30.061	−0.027
	min	30.000	30.002	−0.015	30.000	30.015	−0.028	30.000	30.022	−0.035	30.000	30.035	−0.048	30.000	30.048	−0.061
40	max	40.025	40.018	0.023	40.025	40.033	0.008	40.025	40.042	−0.001	40.025	40.059	−0.018	40.025	40.076	−0.035
	min	40.000	40.002	−0.018	40.000	40.017	−0.033	40.000	40.026	−0.042	40.000	40.043	−0.059	40.000	40.060	−0.076
50	max	50.025	50.018	0.023	50.025	50.033	0.008	50.025	50.042	−0.001	50.025	50.059	−0.018	50.025	50.086	−0.045
	min	50.000	50.002	−0.018	50.000	50.017	−0.033	50.000	50.026	−0.042	50.000	50.043	−0.059	50.000	50.070	−0.086
60	max	60.030	60.021	0.028	60.030	60.039	0.010	60.030	60.051	−0.002	60.030	60.072	−0.023	60.030	60.106	−0.057
	min	60.000	60.002	−0.021	60.000	60.020	−0.039	60.000	60.032	−0.051	60.000	60.053	−0.072	60.000	60.087	−0.106
80	max	80.030	80.021	0.028	80.030	80.039	0.010	80.030	80.051	−0.002	80.030	80.078	−0.029	80.030	80.121	−0.072
	min	80.000	80.002	−0.021	80.000	80.020	−0.039	80.000	80.032	−0.051	80.000	80.059	−0.078	80.000	80.102	−0.121
100	max	100.035	100.025	0.032	100.035	100.045	0.012	100.035	100.059	−0.002	100.035	100.093	−0.036	100.035	100.146	−0.089
	min	100.000	100.003	−0.025	100.000	100.023	−0.045	100.000	100.037	−0.059	100.000	100.071	−0.093	100.000	100.124	−0.146
120	max	120.035	120.025	0.032	120.035	120.045	0.012	120.035	120.059	−0.002	120.035	120.101	−0.044	120.035	120.166	−0.109
	min	120.000	120.003	−0.025	120.000	120.023	−0.045	120.000	120.037	−0.059	120.000	120.079	−0.101	120.000	120.144	−0.166
160	max	160.040	160.028	0.037	160.040	160.052	0.013	160.040	160.068	−0.003	160.040	160.125	−0.060	160.040	160.215	−0.150
	min	160.000	160.003	−0.028	160.000	160.027	−0.052	160.000	160.043	−0.068	160.000	160.100	−0.125	160.000	160.190	−0.215

SOURCE: ANSI B4.2-1978 (R94), reprinted by permission.

Table 8.2.45 K_f Values for Plain Press Fits
Obtained from fatigue tests in bending)

Shaft material	Shaft diam, in	Collar or hub material	K_f	Remarks
0.42% carbon steel	1⅝	0.42% carbon steel	2.0	No external reaction through collar
0.45% carbon axle steel	2	Ni-Cr-Mo steel (case-hardened)	2.3	No external reaction through collar
0.45% carbon axle steel	2	Ni-Cr-Mo steel (case-hardened)	2.9	External reaction taken through collar
Cr-Ni-Mo steel (310 Brinell)	2	Ni-Cr-Mo steel (case-hardened)	3.9	External reaction taken through collar
2.6% Ni steel (57,000 lb/in² fatigue limit)	2	Ni-Cr-Mo steel (case-hardened)	3.3–3.8	External reaction taken through collar
Same, heat-treated to 253 Brinell	2	Ni-Cr-Mo steel (case-hardened)	3.0	External reaction taken through collar

Marine-engine shafts and **diesel-engine crankshafts** should be designed not only for strength but for avoidance of critical speed. (See Applied Mechanics, *Trans. ASME,* **50,** no. 8, for methods of calculating critical speeds of diesel engines.)

Table 8.2.46 Constants for Use in $(S_a/S_e)^m + (KS_m/S_{ut})^P = 1$

Failure theory	K	P	m
Soderberg	S_{ut}/S_y	1	1
Bagci	S_{ut}/S_y	4	1
Modified Goodman	1	1	1
Gerber parabolic	1	2	1
Kececioglu	1	2	m†
Quadratic (elliptic)	1	2	2

$$\dagger m = \begin{cases} 0.8914 \text{ UNS G 10180 } H_B = 130 \\ 0.9266 \text{ UNS G 10380 } H_B = 164 \\ 1.0176 \text{ UNS G 41300 } H_B = 207 \\ 0.9685 \text{ UNS G 43400 } H_B = 233 \end{cases}$$

PULLEYS, SHEAVES, AND FLYWHEELS

Arms of pulleys, sheaves, and flywheels are subjected to stresses due to condition of founding, to details of construction (such as split or solid), and to conditions of service, which are difficult to analyze. For these reasons, no accurate stress relations can be established, and the following formulas must be understood to be only approximately correct. It has been established experimentally by Benjamin (*Am. Mach.,* Sept. 22, 1898) that thin-rim pulleys do not distribute equal loads to the several pulley arms. For these, it will be safe to assume the tangential force on the pulley rim as acting on half of the number of arms. Pulleys with comparatively thick rims, such as engine band wheels, have all the arms taking the load. Furthermore, while the stress action in the arms is similar to that in a beam fixed at both ends, the amount of restraint at the rim depending on the rim's elasticity, it may nevertheless be assumed for purposes of design that cantilever action is predominant. The bending moment at the hub in arms of thin-rim pulleys will be $M = PL/(\tfrac{1}{2}N)$, where M = bending moment, in·lb; P = tangential load on the rim, lb; L = length of the arm, in; and N = number of arms. For thick-rim pulleys and flywheels, $M = PL/N$.

For arms of elliptical section having a width of two times the thickness, where E = **width of arm section** at the rim, in, and s_t = intensity of tensile stress, lb/in²

$$E = \sqrt[3]{40PL/(s_tN)} \text{ (thin rim)} = \sqrt[3]{20PL/(s_tN)} \text{ (thick rim)}$$

For single-thickness belts, P may be taken as $50B$ lb and for double-thickness belts $P = 75B$ lb, where B is the width of pulley face, in. Then $E = k \times \sqrt[3]{BL/(s_tN)}$, where k has the following values: for thin rim, single belt, 13; thin rim, double belt, 15; thick rim, single belt, 10; thick rim, double belt, 12. For cast iron of good quality, s_t due to bending may be taken at 1,500 to 2,000. The arm section at the rim may be made from ⅔ to ¾ the dimensions at the hub.

For high-speed pulleys and flywheels, it becomes necessary to check the arm for tension due to rim expansion. It will be safe to assume that each arm is in tension due to one-half the centrifugal force of that portion of the rim which it supports. That is, $T = As_t = Wv^2/(2NgR)$, lb, where T = tension in arm, lb; N = number of arms; v = speed of rim, ft/s; R = radius of pulley, ft; A = area of arm section, in²; W = weight of pulley rim, lb; and s_t = intensity of tensile stress in arm section, lb/in², whence $s_t = WRn^2/(6{,}800NA)$, where n = r/min of pulley.

Arms of flywheels having heavy rims may be subjected to severe stress action due to the inertia of the rim at sudden load changes. There being no means of predicting the probable maximum to which the inertia may rise, it will be safe to make the arms equal in strength to ¾ of the shaft strength in torsion. Accordingly, for elliptical arm sections,

$$N \times 0.5E^3s_t = \tfrac{3}{4} \times 2s_sd^3 \qquad \text{or} \qquad E = 1.4d\sqrt[3]{s_s/(s_tN)}$$

For steel shafts with s_s = 8,000 and cast-iron arms with s = 1,500,

$$E = 2.4d/\sqrt[3]{N} = 1.3d \text{ (for 6 arms)} = 1.2d \text{ (for 8 arms)}$$

where $2E$ = width of elliptical arm section at hub, in (thickness = E), and d = shaft diameter, in.

Rims of belted pulleys cast whole may have the following proportions (see Fig. 8.2.86):

$$t_2 = \tfrac{3}{4}h + 0.005D \qquad t_1 = 2t_2 + C \qquad W = \tfrac{9}{8}B \text{ to } \tfrac{5}{4}B$$

where h = belt thickness, $C = \tfrac{1}{24}W$, and B = belt width, all in inches.

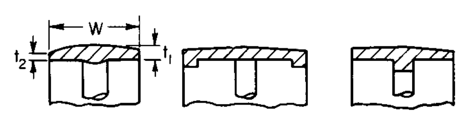


Fig. 8.2.86 Rims for belted pulleys.

Engine band wheels, flywheels, and pulleys run at **high speeds** are subjected to the following stress actions in the rim:

Considering the rim as a free ring, i.e., without arm restraint, and made of cast iron or steel, $s_t = v^2/10$ (approx), where s_t = intensity of tensile stress, lb/in², and v = rim speed, ft/s. For beam action between the arms of a solid rim, $M = Pl/12$ (approx), where M = bending moment in rim, in·lb; P = centrifugal force of that portion of rim between arms, lb, and l = length of rim between arms, in; from which $s_t = WR^2n^2/(450N^2Z)$, where W = weight of entire rim, lb; R = radius of wheel, ft; n = r/min of wheel; and Z = section modulus of rim section, in³. In case the rim section is of the forms shown in Fig. 8.2.86, care must be taken that the flanges do not reduce the section modulus from that of the rectangular section. For **split rims** fastened with bolts the stress analysis is as follows:

Let w = weight of rim portion, lb (with length L, in) lb; w_1 = weight of lug, lb; L_1 = lever arm of lug, in; and s_t = intensity of tensile stress lb/in² in rim section joining arm. Then $s_t = 0.00034n^2R(w_1L_1 + wL/2)/Z$, where n = r/min of wheel; R = wheel radius, ft; and Z = section modulus of rim section, in^3. The above equation gives the value of s_t for bending when the bolts are loose, which is the worst possible condition that may arise. On this basis of analysis, s_t should not be greater than 8,000 lb/in^2. The stress due to bending in addition to the stress due to rim expansion as analyzed previously will be the probable maximum intensity of stress for which the rim should be checked for strength. The flange bolts, because of their position, do not materially relieve the bending action. In case a tie rod leads from the flange to the hub, it will be *safe* to consider it as an additional factor of safety. When the tie rod is kept tight, it very materially strengthens the rim.

A more accurate method for calculating maximum stresses due to centrifugal force in flywheels with arms cast integral with the rim is given by Timoshenko, ‘‘Strength of Materials,’’ Pt. II, 1941, p. 98. More exact equations for calculating stresses in the arms of flywheels and pulleys due to a combination of belt pull, centrifugal force, and changes in velocity are given by Heusinger, *Forschung,* 1938, p. 197. In both treatments, shrinkage stresses in the arms due to casting are neglected.

Large flywheels for high rim speeds and severe working conditions (as for rolling-mill service) have been made from flat-rolled steel plates with holes bored for the shaft. A group of such plates may be welded together by circumferential welds to form a large flywheel. By this means, the welds do not carry direct centrifugal loads, but serve merely to hold the parts in position. Flywheels up to 15-ft diam for rolling-mill service have been constructed in this way.

BELT DRIVES

Flat-Belt Drives

The primary drawback of flat belts is their reliance on belt tension to produce frictional grip over the pulleys. Such high tension can shorten bearing life. Also, tracking may be a problem. However, flat belts, being thin, are not subject to centrifugal loads and so work well over small pulleys at high speeds in ranges exceeding 9,000 ft/min. In light service flat belts can make effective clutching drives. Flat-belt drives have efficiencies of about 90 percent, which compares favorably to geared drives. Flat belts are also quiet and can absorb torsional vibration readily.

Leather belting has an ultimate tensile strength ranging from 3,000 to 5,000 lb/in^2. Average values of breaking strength of good oak-tanned belting (determined by Benjamin) are as follows: single (double) in solid leather 900 (1,400); at riveted joint 600 (1,200); at laced joint 350 lb/in of width. Well-made cemented joints have strengths equal to the belt, leather-laced and riveted joints about one-third to two-thirds as strong, and wire-laced joints about 85 to 95 percent as strong.

Rubber belting is made from fabric or cord impregnated and bound together by vulcanized rubber compounds. The fabric or cord may be of cotton or rayon. Nylon cord and steel cord or cable are also available. Advantages are high tensile strength, strength to hold metal fasteners satisfactorily, and resistance to deterioration by moisture. The best rubber fabric construction for most types of service is made from hard or tight-woven fabric with a ‘‘skim coat’’ or thin layer of rubber between plies. The cord type of construction allows the use of smaller pulley diameters than the fabric type, and also develops less stretch in service. It must be used in the endless form, except in cases where the oil-field type of clamp may be used.

Initial tensions in rubber belts run from 15 to 25 lb/ply/in width. A common rule is to cut belts 1 percent less than the minimum tape-line measurement around the pulleys. For heavy loads, a 1½ percent allowance is usually required, although, because of shrinkage, less initial tension is required for wet or damp conditions. Initial tensions of 25 lb/ply/in may overload shafts or bearings. Maximum safe tight-side tensions for rubber belts are as follows:

Duck weight, oz	28	32	32.66	34.66	36
Tension, lb/ply/in width	25	28	30	32	35

Centrifugal forces at high speeds require higher tight-side tensions to carry rated horsepower.

Rubber belting may be bought in endless form or made endless in the field by means of a vulcanized splice produced by a portable electric vulcanizer. For endless belts the drive should provide take-up of 2 to 4 percent to allow for length variation as received and for stretch in service. The amount of take-up will vary with the type of belt used. For certain drives, it is possible to use endless belts with no provision for take-up, but this involves a heavier belt and a higher initial unit tension than would be the case otherwise. Ultimate tensile strength of rubber belting varies from 280 to 600 lb or more per inch width per ply. The weight varies from 0.02 to 0.03 or more lb/in width/ply. Belts with steel reinforcement are considerably heavier. For horsepower ratings of rubber belts, see Table 8.2.47*c*.

Arrangements for Belt Drives In belt drives, the centerline of the belt advancing on the pulley should lie in a plane passing through the midsection of the pulley at right angles to the shaft. Shafts inclined to each other require connections as shown in Fig. 8.2.87*a*. In case guide pulleys are needed their positions can be determined as shown in Fig. 8.2.87*a*, *b*, and *c*. In Fig. 8.2.87*d* the center circles of the two pulleys to be connected are set in correct relative position in two planes, a being the angle between the planes (= supplement of angle between shafts). If any two points as E and F are assumed on the line of intersection MN of the planes, and tangents EG, EH, FJ, and FK are drawn from them to the circles, the center circles of the guide pulleys must be so arranged that these tangents are also tangents to them, as shown. In other words, the middle planes of the guide pulleys must lie in the planes GEH and JFK.

Table 8.2.47*a* Service Factors *S*

Application	Squirrel-cage ac motor: Normal torque, line start	Squirrel-cage ac motor: High torque	Wound rotor ac motor (slip ring)	Single-phase capacitor motor	DC shunt-wound motor	Diesel engine, 4 or more cyl, above 700 r/min
Agitators	1.0–1.2	1.2–1.4	1.2			
Compressors	1.2–1.4		1.4	1.2	1.2	1.2
Belt conveyors (ore, coal, sand)	—	1.4	—	—	1.2	
Screw conveyors	—	1.8	—	—	1.6	
Crushing machinery	—	1.6	1.4	—	—	1.4–1.6
Fans, centrifugal	1.2	—	1.4	—	1.4	1.4
Fans, propeller	1.4	2.0	1.6	—	1.6	1.6
Generators and exciters	1.2	—	—	—	1.2	2.0
Line shafts	1.4	—	1.4	1.4	1.4	1.6
Machine tools	1.0–1.2	—	1.2–1.4	1.0	1.0–1.2	
Pumps, centrifugal	1.2	1.4	1.4	1.2	1.2	
Pumps, reciprocating	1.2–1.4	—	1.4–1.6	—	—	1.8–2.0

Table 8.2.47*b* Arc of Contact Factor *K*—Rubber Belts

Arc of contact, deg	140	160	180	200	220
Factor *K*	0.82	0.93	1.00	1.06	1.12

When these conditions are met, the belts will run in either direction on the pulleys.

To avoid the necessity of taking up the **slack** in belts which have become stretched and permanently lengthened, a **belt tightener** such as shown in Fig. 8.2.88 may be employed. It should be placed on the slack side of the belt and nearer the driving pulley than the driven pulley. Pivoted motor drives may also be used to maintain belt tightness with minimum initial tension.

Length of Belt for a Given Drive The length of an **open belt** for a given drive is equal to $L = 2C + 1.57(D + d) + (D - d)^2/(4C)$, where L = length of belt, in; D = diam of large pulley, in; d = diam of small pulley, in; and C = distance between pulley centers, in. Center distance C is given by $C = 0.25b + 0.25\sqrt{b^2 - 2(D - d)^2}$, where $b = L - 1.57(D + d)$. When a **crossed belt** is used, the length in $L = 2C + 1.57(D + d) + (D + d)^2/(4C)$.

Step or Cone Pulleys For belts operating on step pulleys, the pulley diameters must be such that the belt will fit over any pair with equal tightness. With **crossed belts,** it will be apparent from the equation for length of belt that the sum of the pulley diameters need only be constant in order that the belt may fit with equal tightness on each pair of pulleys. With open belts, the length is a function of both the sum and the difference of the pulley diameters; hence no direct solution of the problem is possible, but a graphical approach can be of use.

A graphical method devised by Smith (*Trans. ASME,* 10) is shown in Fig. 8.2.89. Let A and B be the centers of any pair of pulleys in the set, the diameters of which are known or assumed. Bisect AB in C, and draw CD at right angles to AB. Take CD = 0.314 times the center distance L, and draw a circle tangent to the belt line EF. The belt line of any other pair of pulleys in the set will then be tangent to this circle. If the angle

Table 8.2.47*c* Horsepower Ratings of Rubber Belts
(hp/in of belt width for 180° wrap)

		Belt speed, ft/min										
	Ply	500	1,000	1,500	2,000	2,500	3,000	4,000	5,000	6,000	7,000	8,000
32-oz fabric	3	0.7	1.4	2.1	2.7	3.3	3.9	4.9	5.6	6.0		
	4	0.9	1.9	2.8	3.6	4.4	5.2	6.5	7.4	7.9		
	5	1.2	2.3	3.4	4.5	5.5	6.5	8.1	9.2	9.8		
	6	1.4	2.8	4.1	5.4	6.6	7.8	9.6	11.0	11.7		
	7	1.6	3.2	4.7	6.2	7.7	9.0	11.2	12.8	13.6		
	8	1.8	3.6	5.3	7.0	8.7	10.2	12.7	14.6	15.5		
32-oz hard fabric	3	0.7	1.5	2.2	2.9	3.5	4.1	5.1	5.8	6.2	6.1	5.5
	4	1.0	2.0	3.0	3.9	4.7	5.5	6.8	7.8	8.3	8.1	7.3
	5	1.3	2.5	3.7	4.9	5.9	6.9	8.5	9.8	10.3	9.1	9.0
	6	1.5	3.0	4.5	5.9	7.1	8.3	10.2	11.7	12.3	12.1	10.7
	7	1.7	3.5	5.2	6.9	8.3	9.7	11.9	13.6	14.3	14.1	12.4
	8	1.9	4.0	5.9	7.9	9.5	11.1	13.6	15.5	16.3	16.0	14.1
	9	2.1	4.5	6.6	8.9	10.6	12.4	15.3	17.4	18.3	17.9	15.8
	10	2.3	5.0	7.3	9.8	11.7	13.7	17.0	19.3	20.3	19.8	17.5
No. 70 rayon cord	3	1.6	3.1	4.6	6.0	7.3	8.6	10.6	12.0	12.7	12.3	10.7
	4	2.1	4.1	6.1	8.0	9.8	11.5	14.5	16.6	17.8	17.8	16.4
	5	2.6	5.1	7.6	10.1	12.3	14.5	18.3	21.1	23.0	23.5	22.2
	6	3.1	6.2	9.2	12.1	14.8	17.5	22.1	25.7	28.1	28.9	27.9
	7	3.6	7.2	10.7	14.1	17.4	20.4	26.0	30.3	33.2	34.5	33.7
	8	4.1	8.2	12.2	16.2	19.9	23.4	29.8	34.8	38.4	40.0	39.4

Table 8.2.47*d* Minimum Pulley Diameters—Rubber Belts, in

		Belt speed, ft/min										
	Ply	500	1,000	1,500	2,000	2,500	3,000	4,000	5,000	6,000	7,000	8,000
32-oz fabric	3	4	4	4	4	5	5	5	6	6		
	4	4	5	6	6	7	7	8	9	10		
	5	6	7	9	10	10	11	12	13	14		
	6	9	10	11	13	14	14	16	18	19		
	7	13	14	16	17	18	19	21	22	24		
	8	18	19	21	22	23	24	25	27	29		
32-oz hard fabric	3	3	3	3	4	4	4	4	5	5	6	7
	4	4	4	5	5	6	6	7	7	8	9	12
	5	5	6	7	8	8	9	10	11	12	13	16
	6	6	8	10	11	11	12	13	15	16	18	21
	7	10	12	14	15	15	16	17	19	20	22	26
	8	14	16	17	18	19	20	21	23	24	27	31
	9	18	20	21	22	23	24	25	27	28	31	36
	10	22	24	25	26	27	28	29	31	33	35	41
No. 70 rayon cord	3	5	6	7	7	8	8	9	10	11	12	13
	4	7	8	9	9	10	11	12	12	14	15	17
	5	9	10	11	12	13	13	15	16	17	19	21
	6	13	14	15	16	16	17	18	19	21	23	25
	7	16	17	18	19	20	21	22	23	24	26	29
	8	19	20	22	23	23	24	25	26	28	30	33

EF makes with *AB* is greater than 18°, draw a tangent to the circle *D*, making an angle of 18° with *AB*; and from a center on *CD* distant 0.298*L* above *C*, draw an arc tangent to this 18° line. All belt lines with angles greater than 18° will be tangent to this last drawn arc.

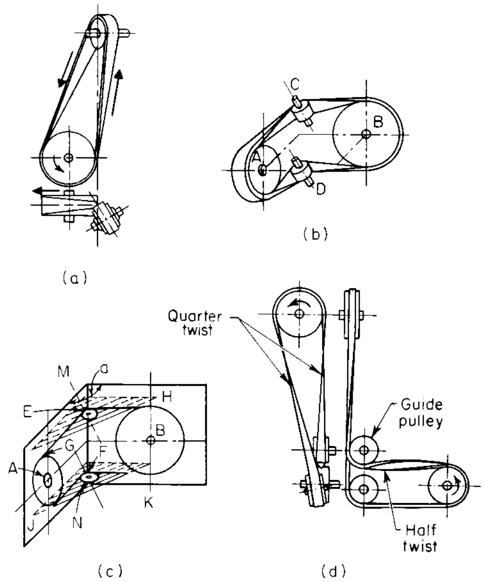


Fig. 8.2.87 Arrangements for flat-belt drives.

A very slight error in a graphical solution drawn to any scale much under full size will introduce an error seriously affecting the equality of belt tensions on the various pairs of pulleys in the set, and where much power is to be transmitted it is advisable to calculate the pulley diameters from the following **formulas** derived from Burmester's graphical method (''Lehrbuch der Mechanik'').

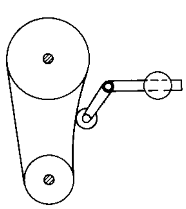


Fig. 8.2.88 Belt tightener.

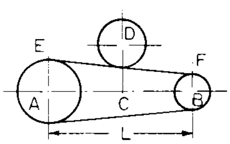


Fig. 8.2.89 Symbols for cone pulley graphical method.

Let D_1 and D_2 be, respectively, the diameters of the smaller and larger pulleys of a pair, $n = D_2/D_1$, and l = distance between shaft centers, all in inches. Also let $m = 1.58114l - D_0$, where D_0 = diam of both pulleys for a speed ratio $n = 1$. Then $(D_1 + m)^2 + (nD_1 + m)^2 = 5l^2$. First settle on values of D_0, l, and n, and then substitute in the equation and solve for D_1. The diameter D_2 of the other pulley of the pair will then be nD_1. The values are correct to the fourth decimal place.

The speeds given by cone pulleys should increase in a **geometric ratio;** i.e., each speed should be multiplied by a constant a in order to obtain the next higher speed. Let n_1 and n_2 be, respectively, the lowest and highest speeds (r/min) desired and k the number of speed changes. Then $a = \sqrt[k-1]{n_2/n_1}$. In practice, a ranges from 1.25 up to 1.75 and even 2. The ideal value for a in machine-tool practice, according to Carl G. Barth, would be 1.189. In the example below, this would mean the use of 18 speeds instead of 8.

EXAMPLE. Let $n_1 = 16$, $n_2 = 400$, and $k = 8$, to be obtained with four pairs of pulleys and a back gear. From formula, $a = \sqrt[7]{25} = 1.584$, whence speeds will be 16, ($16 \times 1.584 =$) 25.34, ($25.34 \times 1.584 =$) 40.14, and similarly 63.57, 100.7, 159.5, 252.6, and 400. The first four speeds are with the back gear in; hence the back-gear ratio must be $100.7 \div 16 = 6.29$.

Transmission of Power by Flat Belts The theory of flat-belt drives takes into account changes in belt tension caused by friction forces between belt and pulley, which, in turn, cause belt elongation or contraction, thus inducing relative movement between belt and pulley. The transmission of power is a complex affair. A lengthy mathematical presentation can be found in Firbank, ''Mechanics of the Flat Belt Drive,'' ASME Paper 72-PTG-21. A simpler, more conventional analysis used for many years yields highly serviceable designs.

The turning force (tangential) on the rim of a pulley driven by a flat belt is equal to $T_1 - T_2$, where T_1 and T_2 are, respectively, the tensions in the driving (tight) side and following (slack) side of the belt. (For the relations of T_1 and T_2 at low peripheral speeds, see Sec. 3.) Log $(T_1/T_2) = 0.0076fa$ when the effect of centrifugal force is neglected and $T_1/T_2 = 10^{0.0076fa}$. Figure 8.2.90 gives values of this function. When the speeds are high, however, the relations of T_1 to T_2 are modified by centrifugal stresses in the belt, in which case $\log(T_1/T_2) = 0.0076f(1 - x)a$, where f = coefficient of friction between the belt and pulley surface, a = angle of wrap, and $x = 12wv^2/(gt)$ in which w =

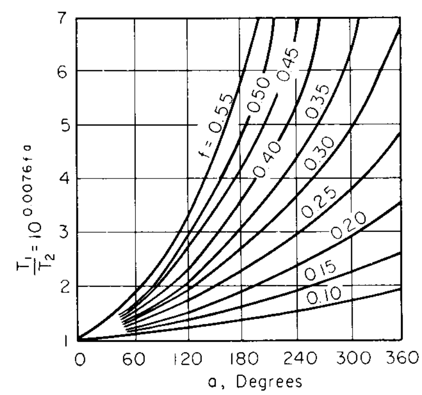


Fig. 8.2.90 Values of $10^{0.0076fa}$.

weight of 1 in^3 of belt material, lb; v = belt speed, ft/s; g = 32.2 ft/s^2; and t = allowable working tension, lb/in^2. Values of x for leather belting (with $w = 0.035$ and $t = 300$) are as follows:

u	30	40	50	60	70	
x	0.039	0.070	0.118	0.157	0.214	
uv	80	90	100	110	120	130
x	0.279	0.352	0.435	0.526	0.626	0.735

Researches by Barth (*Trans. ASME*, 1909) seem to show that f is a function of the belt velocity, varying according to the formula $f = 0.54 - 140/(500 + V)$ for leather belts on iron pulleys, where V = belt velocity for ft/min. For practical design, however, the following values of f may be used: for leather belts on cast-iron pulleys, $f = 0.30$; on wooden pulleys, $f = 0.45$; on paper pulleys, $f = 0.55$. The treatment of belts with belt dressing, pulleys with cork inserts, and dampness are all factors which greatly modify these values, tending to make them higher.

The **arc of contact** on the smaller of two pulleys connected by an open belt, in degrees, is approximately equal to $180 - 60(D - d)/l$, where D and d are the larger and smaller pulley diameters and l the distance between their shaft centers, all in inches. This formula gives an error not exceeding 0.5 percent.

Selecting a Belt Selecting an appropriate belt involves calculating horsepower per inch of belt width as follows:

$$\text{hp/in} = (\text{demanded hp} \times S)/(K \times W)$$

where demanded hp = horsepower required by the job at hand; S = service factor; K = arc factor; W = proposed belt width (determined from pulley width). One enters a belt manufacturer's catalog with *hp/in, belt speed,* and *small pulley diameter*, then selects that belt which has a matching maximum hp/in rating. See Table 8.2.47*a, b, c,* and *d* for typical values of *S, K,* hp/in ratings, and minimum pulley diameters.

V-Belt Drives

V-belt drives are widely used in power transmission, despite the fact that they may range in efficiency from about 70 to 96 percent. Such drives consist essentially of endless belts of trapezoidal cross section which ride in V-shaped pulley grooves (see Fig. 8.2.93*a*). The belts are formed of cord and fabric, impregnated with rubber, the cord material being cotton, rayon, synthetic, or steel. V-belt drives are quiet, able to absorb shock and operate at low bearing pressures. A V belt should ride with the top surface approximately flush with the top of the pulley groove; clearance should be present between the belt base and the base of the groove so that the belt rides on the groove walls. The friction between belt and groove walls is greatly enhanced beyond normal values because sheave groove angles are made somewhat less than belt-section angles, causing the belt to wedge itself into the groove. See Table 8.2.56*a* for standard groove dimensions of sheaves.

The cross section and lengths of V belts have been standardized by ANSI in both inch and SI (metric) units, while ANSI and SAE have standardized the special category of automobile belts, again in both units. Standard designations are shown in Table 8.2.48, which also includes minimum sheave diameters. V belts are specified by combining a standard designation (from Table 8.2.48) and a belt length; inside length for the inch system, and pitch (effective) length for metric system.

Table 8.2.48 V-Belt Standard Designations—A Selection

	Inch standard		Metric standard	
Type	Section	Minimum sheave diameter, in	Section	Minimum sheave diameter, mm
Heavy-duty	A	3.0	13 C	80
	B	5.4	16 C	140
	C	9.0	22 C	214
	D	13.0	32 C	355
	E	21.04		
Automotive	0.25	2.25	6 A	57
	0.315	2.25	8 A	57
	0.380	2.40	10 A	61
	0.440	2.75	11 A	70
	0.500	3.00	13 A	76
	11⁄16	3.00	15 A	76
	¾	3.00	17 A	76
	⅞	3.50	20 A	89
	1.0	4.00	23 A	102
Heavy-duty narrow	3 V	2.65		
	5 V	7.1		
	8 V	12.3		
Notched narrow	3 VX	2.2		
	5 VX	4.4		
Light-duty	2 L	0.8		
	3 L	1.5		
	4 L	2.5		
	5 L	3.5		
Synchronous belts	MXL			
	XL			
	L			
	H			
	XH			
	XXH			

NOTE: The use of smaller sheaves than minimum will tend to shorten belt life.
SOURCE: Compiled from ANSI/RMA IP-20, 21, 22, 23, 24, 25, 26; ANSI/SAE J636C.

Sheaves are specified by their pitch diameters, which are used for velocity ratio calculations in which case inside belt lengths must be converted to pitch lengths for computational purposes. Pitch lengths are calculated by adding a conversion factor to inside length (i.e., $L_p = L_s + \Delta$). See Table 8.2.49 for conversion factors. Table 8.2.50 lists standard inside inch lengths L_s and Table 8.2.51 lists standard metric pitch (effective) lengths L_p.

Table 8.2.49 Length Conversion Factors Δ

Belt section	Size interval	Conversion factor	Belt section	Size interval	Conversion factor
A	26–128	1.3	D	120–210	3.3
B	35–210	1.8	D	≥240	0.8
B	≥240	0.3	E	180–210	4.5
C	51–210	2.9	E	≥240	1.0
C	≥240	0.9			

SOURCE: Adapted from ANSI/RMA IP-20-1977 (R88) by permission.

Table 8.2.50 Standard Lengths L_s, in, and Length Correction Factors K_2: Conventional Heavy-Duty V Belts

	Cross section				
L_s	A	B	C	D	E
26	0.78				
31	0.82				
35	0.85	0.80			
38	0.87	0.82			
42	0.89	0.84			
46	0.91	0.86			
51	0.93	0.88	0.80		
55	0.95	0.89			
60	0.97	0.91	0.83		
68	1.00	0.94	0.85		
75	1.02	0.96	0.87		
80	1.04				
81		0.98	0.89		
85	1.05	0.99	0.90		
90	1.07	1.00	0.91		
96	1.08		0.92		
97		1.02			
105	1.10	1.03	0.94		
112	1.12	1.05	0.95		
120	1.13	1.06	0.96	0.88	
128	1.15	1.08	0.98	0.89	
144		1.10	1.00	0.91	
158		1.12	1.02	0.93	
173		1.14	1.04	0.94	
180		1.15	1.05	0.95	0.92
195		1.17	1.06	0.96	0.93
210		1.18	1.07	0.98	0.95
240		1.22	1.10	1.00	0.97
270		1.24	1.13	1.02	0.99
300		1.27	1.15	1.04	1.01
330			1.17	1.06	1.03
360			1.18	1.07	1.04
390			1.20	1.09	1.06
420			1.21	1.10	1.07
480				1.13	1.09
540				1.15	1.11
600				1.17	1.13
660				1.18	1.15

SOURCE: ANSI/RMA IP-20-1977 (R88), reproduced by permission.

Table 8.2.51 Standard Pitch Lengths L_p (Metric Units) and Length Correction Factors K_2

13C		16C		22C		32C	
L_p	K_2	L_p	K_2	L_p	K_2	L_p	K_2
710	0.83	960	0.81	1,400	0.83	3,190	0.89
750	0.84	1,040	0.83	1,500	0.85	3,390	0.90
800	0.86	1,090	0.84	1,630	0.86	3,800	0.92
850	0.88	1,120	0.85	1,830	0.89	4,160	0.94
900	0.89	1,190	0.86	1,900	0.90	4,250	0.94
950	0.90	1,250	0.87	2,000	0.91	4,540	0.95
1,000	0.92	1,320	0.88	2,160	0.92	4,720	0.96
1,075	0.93	1,400	0.90	2,260	0.93	5,100	0.98
1,120	0.94	1,500	0.91	2,390	0.94	5,480	0.99
1,150	0.95	1,600	0.92	2,540	0.96	5,800	1.00
1,230	0.97	1,700	0.94	2,650	0.96	6,180	1.01
1,300	0.98	1,800	0.95	2,800	0.98	6,560	1.02
1,400	1.00	1,900	0.96	3,030	0.99	6,940	1.03
1,500	1.02	1,980	0.97	3,150	1.00	7,330	1.04
1,585	1.03	2,110	0.99	3,350	1.01	8,090	1.06
1,710	1.05	2,240	1.00	3,550	1.02	8,470	1.07
1,790	1.06	2,360	1.01	3,760	1.04	8,850	1.08
1,865	1.07	2,500	1.02	4,120	1.06	9,240	1.09
1,965	1.08	2,620	1.03	4,220	1.06	10,000	1.10
2,120	1.10	2,820	1.05	4,500	1.07	10,760	1.11
2,220	1.11	2,920	1.06	4,680	1.08	11,530	1.13
2,350	1.13	3,130	1.07	5,060	1.10	12,290	1.14
2,500	1.14	3,330	1.09	5,440	1.11		
2,600	1.15	3,530	1.10	5,770	1.13		
2,730	1.17	3,740	1.11	6,150	1.14		
2,910	1.18	4,090	1.13	6,540	1.15		
3,110	1.20	4,200	1.14	6,920	1.16		
3,310	1.21	4,480	1.15	7,300	1.17		
		4,650	1.16	7,680	1.18		
		5,040	1.18	8,060	1.19		
		5,300	1.19	8,440	1.20		
		5,760	1.21	8,820	1.21		
		6,140	1.23	9,200	1.22		
		6,520	1.24				
		6,910	1.25				
		7,290	1.26				
		7,670	1.27				

SOURCE: ANSI/RMA IP-20-1988 revised, reproduced by permission.

For given large and small sheave diameters and center-to-center distance, the needed V-belt length can be computed from

$$L_p = 2C + 1.57(D + d) + \frac{(D - d)^2}{4C}$$

$$C = \frac{K + \sqrt{K^2 - 32(D - d)^2}}{16} \qquad K = 4L_p - 6.28\,(D + d)$$

where C = center-to-center distance; D = pitch diameter of large sheave; d = pitch diameter of small sheave; L_p = pitch (effective) length.

Arc of contact on the smaller sheave (degrees) is approximately

$$\theta = 180 - \frac{60(D - d)}{C}$$

Transmission of Power by V Belts Unfortunately there is no theory or mathematical analysis that is able to explain all experimental results reliably. Empirical formulations based on experimental results, however, do provide very serviceable design procedures, and together with data published in V-belt manufacturers' catalogs provide the engineer with the necessary V-belt selection tools.

For satisfactory performance under most conditions, ANSI provides the following empirical single V-belt power-rating formulation (inch and metric units) for 180° arc of contact and average belt length:

$$H_r = \left(C_1 - \frac{C_2}{d} - C_3(r - d)^2 - C_4 \log rd\right)rd + C_2 r\left(1 - \frac{1}{K_A}\right)$$

where H_r = rated horsepower for inch units (rated power kW for metric units); C_1, C_2, C_3, C_4 = constants from Table 8.2.53; r = r/min of high-speed shaft times 10^{-3}; K_A = speed ratio factor from Table 8.2.52; d = pitch diameter of small sheave, in (mm).

Selecting a Belt Selecting an appropriate belt involves calculating horsepower per belt as follows:

$$NH_r = (\text{demanded hp} \times K_s)/(K_1 K_2)$$

where H_r = hp/belt rating, either from ANSI formulation above, or from manufacturer's catalog (see Table 8.2.55); demanded hp = horsepower required by the job at hand; K_s = service factor accounting for driver and driven machine characteristics regarding such things as shock, torque level, and torque uniformity (see Table 8.2.54); K_1 = angle of contact correction factor (see Fig. 8.2.91*a*); K_2 = length correction factor (see Tables 8.2.50 and 8.2.51); N = number of belts.

See Fig. 8.2.91*b* for selection of V-belt cross section.

V band belts, effectively joined V belts, serve the function of multiple single V belts (see Fig. 8.2.93*b*).

Long center distances are not recommended for V belts because excess slack-side vibration shortens belt life. In general $D \leqq C \leqq 3(D + d)$. If longer center distances are needed, then link-type V belts can be used effectively.

Since belt-drive capacity is normally limited by slippage of the smaller sheave, V-belt drives can sometimes be used with a flat, larger pulley rather than with a grooved sheave, with little loss in capacity. For instance the flat surface of the flywheel in a large punch press can serve such a purpose. The practical range of application is when speed ratio is over 3 : 1, and center distance is equal to or slightly less than the diameter of the large pulley.

Table 8.2.52 Approximate Speed-Ratio Factor K_A for Use in Power-Rating Formulation

D/d range	K_A	D/d range	K_A
1.00–1.01	1.0000	1.15–1.20	1.0586
1.02–1.04	1.0112	1.21–1.27	1.0711
1.05–1.07	1.0226	1.28–1.39	1.0840
1.08–1.10	1.0344	1.40–1.64	1.0972
1.11–1.14	1.0463	Over 1.64	1.1106

SOURCE: Adapted from ANSI/RMA IP-20-1977 (R88), by permission.

Table 8.2.53 Constants C_1, C_2, C_3, C_4 for Use in Power-Rating Formulation

Belt section	C_1	C_2	C_3	C_4
		Inch		
A	0.8542	1.342	2.436×10^{-4}	0.1703
B	1.506	3.520	4.193×10^{-4}	0.2931
C	2.786	9.788	7.460×10^{-4}	0.5214
D	5.922	34.72	1.522×10^{-3}	1.064
E	8.642	66.32	2.192×10^{-3}	1.532
		Metric		
13C	0.03316	1.088	1.161×10^{-8}	5.238×10^{-3}
16C	0.05185	2.273	1.759×10^{-8}	7.934×10^{-3}
22C	0.1002	7.040	3.326×10^{-8}	1.500×10^{-2}
32C	0.2205	26.62	7.037×10^{-8}	3.174×10^{-2}

SOURCE: Compiled from ANSI/RMA IP-20-1977 (R88), by permission.

Table 8.2.54 Approximate Service Factor K_s for V-Belt Drives

	Load			
Power source torque	Uniform	Light shock	Medium shock	Heavy shock
Average or normal	1.0–1.2	1.1–1.3	1.2–1.4	1.3–1.5
Nonuniform or heavy	1.1–1.3	1.2–1.4	1.4–1.6	1.5–1.8

SOURCE: Adapted from ANSI/RMA IP-20-1977 (R88), by permission.

Table 8.2.55 Horsepower Ratings of V Belts

Belt section	Speed of faster shaft, r/min	Rated horsepower per belt for small sheave pitch diameter, in							Additional horsepower per belt for speed ratio				
		2.60	3.00	3.40	3.80	4.20	4.60	5.00	1.02–1.04	1.08–1.10	1.15–1.20	1.28–1.39	1.65–over
A	200	0.20	0.27	0.33	0.40	0.46	0.52	0.59	0.00	0.01	0.01	0.02	0.03
	800	0.59	0.82	1.04	1.27	1.49	1.70	1.92	0.01	0.04	0.06	0.08	0.11
	1,400	0.87	1.25	1.61	1.97	2.32	2.67	3.01	0.02	0.06	0.10	0.15	0.19
	2,000	1.09	1.59	2.08	2.56	3.02	3.47	3.91	0.03	0.09	0.15	0.21	0.27
	2,600	1.25	1.87	2.47	3.04	3.59	4.12	4.61	0.04	0.12	0.19	0.27	0.35
	3,200	1.37	2.08	2.76	3.41	4.01	4.57	5.09	0.05	0.14	0.24	0.33	0.43
	3,800	1.43	2.23	2.97	3.65	4.27	4.83	5.32	0.06	0.17	0.28	0.40	0.51
	4,400	1.44	2.29	3.07	3.76	4.36	4.86*	5.26*	0.07	0.20	0.33	0.46	0.59
	5,000	1.39	2.28	3.05	3.71	4.24*	4.48*	4.64*	0.07	0.22	0.37	0.52	0.65
	5,600	1.29	2.17	2.92	3.50*				0.08	0.25	0.42	0.58	0.75
	6,200	1.11	1.98	2.65*					0.09	0.28	0.46	0.64	0.83
	6,800	0.87	1.68*	2.24*					0.10	0.30	0.51	0.71	0.91
	7,400	0.56	1.28*						0.11	0.33	0.55	0.77	0.99
	7,800	0.31*							0.12	0.35	0.58	0.81	1.04
		4.60	5.20	5.80	6.40	7.00	7.60	8.00					
B	200	0.69	0.86	1.02	1.18	1.34	1.50	1.61	0.01	0.02	0.04	0.05	0.07
	600	1.68	2.12	2.56	2.99	3.41	3.83	4.11	0.02	0.07	0.12	0.16	0.21
	1,000	2.47	3.16	3.84	4.50	5.14	5.78	6.20	0.04	0.12	0.19	0.27	0.35
	1,400	3.13	4.03	4.91	5.76	6.59	7.39	7.91	0.05	0.16	0.27	0.38	0.49
	1,800	3.67	4.75	5.79	6.79	7.74	8.64	9.21	0.07	0.21	0.35	0.49	0.63
	2,200	4.08	5.31	6.47	7.55	8.56	9.48	10.05	0.09	0.26	0.43	0.60	0.77
	2,600	4.36	5.69	6.91	8.01	9.01	9.87	10.36	0.10	0.30	0.51	0.71	0.91
	3,000	4.51	5.89	7.11	8.17	9.04	9.73*		0.12	0.35	0.58	0.82	1.05
	3,400	4.51	5.88	7.03	7.95*				0.13	0.40	0.66	0.93	1.19
	3,800	4.34	5.64	6.65*					0.15	0.44	0.74	1.04	1.33
	4,200	4.01	5.17*						0.16	0.49	0.82	1.15	1.47
	4,600	3.48							0.18	0.54	0.90	1.25	1.61
	5,000	2.76*							0.19	0.59	0.97	1.36	1.75
		7.00	8.00	9.00	10.00	11.00	12.00	13.00					
C	100	1.03	1.29	1.55	1.81	2.06	2.31	2.56	0.01	0.03	0.05	0.08	0.10
	400	3.22	4.13	5.04	5.93	6.80	7.67	8.53	0.04	0.13	0.22	0.30	0.39
	800	5.46	7.11	8.73	10.31	11.84	13.34	14.79	0.09	0.26	0.43	0.61	0.78
	1,000	6.37	8.35	10.26	12.11	13.89	15.60	17.24	0.11	0.33	0.54	0.76	0.97
	1,400	7.83	10.32	12.68	14.89	16.94	18.83	20.55	0.15	0.46	0.76	1.06	1.36
	1,800	8.76	11.58	14.13	16.40	18.37	20.01	21.31*	0.20	0.59	0.98	1.37	1.75
	2,000	9.01	11.90	14.44	16.61	18.37	19.70*		0.22	0.65	1.08	1.52	1.95
	2,400	9.04	11.87	14.14					0.26	0.78	1.30	1.82	2.34
	2,800	8.38	10.85*						0.30	0.91	1.52	2.12	2.73
	3,000	7.76							0.33	0.98	1.63	2.28	2.92
	3,200	6.44*							0.36	1.07	1.79	2.50	3.22

* For footnote see end of table on next page.

In general the V-belting of skew shafts is discouraged because of the decrease in life of the belts, but where design demands such arrangements, special deep-groove sheaves are used. In such cases center distances should comply with the following:

For 90° turn $C_{min} = 5.5(D + W)$
For 45° turn $C_{min} = 4.0(D + W)$
For 30° turn $C_{min} = 3.0(D + W)$

where W = width of group of individual belts. Selected values of W are shown in Table 8.2.56*d*.

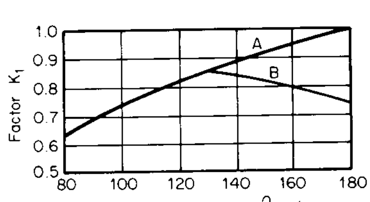


Fig. 8.2.91*a* Angle-of-contact correction factor, where A = grooved sheave to grooved pulley distance (V to V) and B = grooved sheave to flat-face pulley distance (V to flat).

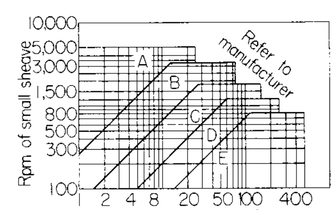


Fig. 8.2.91*b* V-belt section for required horsepower ratings. Letters A, B, C, D, E refer to belt cross section. (See Table 8.2.54 for service factor.)

Figure 8.2.92 shows a 90° turn arrangement, from which it can be seen that the horizontal shaft should lie some distance Z higher than the center of the vertical-shaft sheave. Table 8.2.56*c* lists the values of Z for various center distances in a 90° turn arrangement.

Cogged V belts have cogs molded integrally on the underside of the belt (Fig. 8.2.94*a*). Sheaves can be up to 25 percent smaller in diameter with cogged belts because of the greater flexibility inherent in the cogged construction. An extension of the cogged belt mating with a sheave or pulley notched at the same pitch as the cogs leads to a drive particularly useful for timing purposes.

Table 8.2.55 Horsepower Ratings of V Belts (*Continued*)

Belt section	Speed of faster shaft, r/min	Rated horsepower per belt for small sheave pitch diameter, in							Additional horsepower per belt for speed ratio				
		12.00	14.00	16.00	18.00	20.00	22.00	24.00	1.02–1.04	1.08–1.10	1.15–1.20	1.28–1.39	1.65–over
D	50	1.96	2.52	3.08	3.64	4.18	4.73	5.27	0.02	0.06	0.10	0.13	0.17
	200	6.28	8.27	10.24	12.17	14.08	15.97	17.83	0.08	0.23	0.38	0.54	0.69
	400	10.89	14.55	18.12	21.61	25.02	28.35	31.58	0.15	0.46	0.77	1.08	1.38
	600	14.67	19.75	24.64	29.33	33.82	38.10	42.15	0.23	0.69	1.15	1.61	2.07
	800	17.70	23.91	29.75	35.21	40.24	44.83	48.94	0.31	0.92	1.54	2.15	2.77
	1,000	19.93	26.94	33.30	38.96	43.86	47.93	51.12	0.38	1.15	1.92	2.69	3.46
	1,200	21.32	28.71	35.05	40.24	44.18*			0.46	1.39	2.31	3.23	4.15
	1,400	21.76	29.05	34.76*					0.54	1.62	2.69	3.77	4.84
	1,600	21.16	27.81*						0.62	1.85	3.08	4.30	5.53
	1,800	19.41							0.69	2.08	3.46	4.84	6.22
	1,950	17.28*							0.75	2.25	3.75	5.25	6.74
		18.00	21.00	24.00	27.00	30.00	33.00	36.00					
E	50	4.52	5.72	6.91	8.08	9.23	10.38	11.52	0.04	0.11	0.18	0.26	0.33
	100	8.21	10.46	12.68	14.87	17.04	19.19	21.31	0.07	0.22	0.37	0.51	0.66
	200	14.68	18.86	22.97	27.00	30.96	34.86	38.68	0.15	0.44	0.73	1.03	1.32
	300	20.37	26.29	32.05	37.67	43.13	48.43	53.58	0.22	0.66	1.10	1.54	1.98
	400	25.42	32.87	40.05	46.95	53.55	59.84	65.82	0.29	0.88	1.47	2.06	2.64
	500	29.86	38.62	46.92	54.74	62.05	68.81	75.00	0.37	1.10	1.84	2.57	3.30
	600	33.68	43.47	52.55	60.87	68.36	74.97	80.63	0.44	1.32	2.20	3.08	3.96
	700	36.84	47.36	56.83	65.15	72.22	77.93*		0.51	1.54	2.57	3.60	4.62
	800	39.29	50.20	59.61	67.36	73.30*			0.59	1.76	2.94	4.11	5.28
	900	40.97	51.89	60.73					0.66	1.98	3.30	4.63	5.94
	1,000	41.84	52.32	60.04*					0.73	2.21	3.67	5.14	6.60
	1,100	41.81	51.40*						0.81	2.43	4.04	5.65	7.26
	1,200	40.83							0.88	2.65	4.41	6.17	7.93
	1,300	38.84*							0.95	2.87	4.77	6.68	8.59

* Rim speed above 6,000 ft/min. Special sheaves may be necessary.
SOURCE: Compiled from ANSI/RMA IP-20-1988 revised, by permission.

Ribbed V belts are really flat belts molded integrally with longitudinal ribbing on the underside (Fig. 8.2.94*b*). Traction is provided principally by friction between the ribs and sheave grooves rather than by wedging action between the two, as in conventional V-belt operation. The flat upper portion transmits the tensile belt loads. Ribbed belts serve well when substituted for multiple V-belt drives and for all practical purposes eliminate the necessity for belt-matching in multiple V-belt drives.

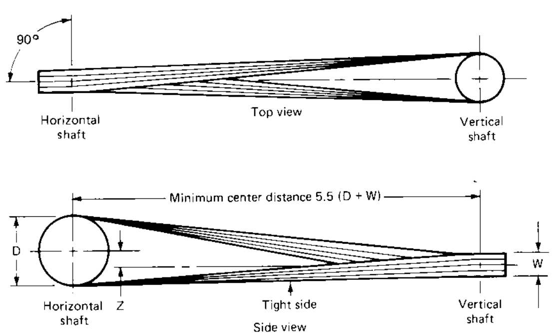


Fig. 8.2.92 Quarter-turn drive for V belts.

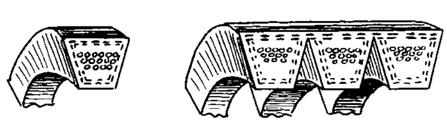


Fig. 8.2.93 V- and V-band belt cross section.

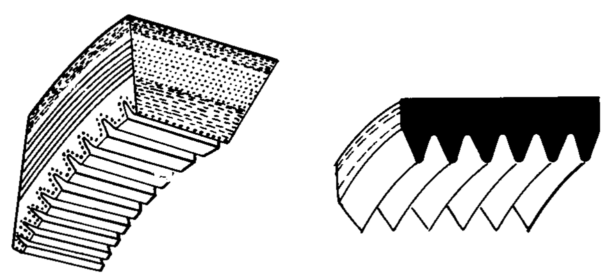


Fig. 8.2.94 Special V belts. (*a*) Cogged V belt; (*b*) ribbed V belt.

Adjustable Motor Bases

To maintain proper belt tensions on short center distances, an adjustable motor base is often used. Figure 8.2.95 shows an embodiment of such a base made by the Automatic Motor Base Co., in which adjustment for proper belt tension is made by turning a screw which opens or closes the center distance between pulleys, as required. The carriage portion of the base is spring loaded so that after the initial adjustment for belt tension

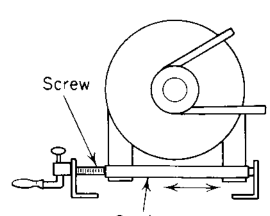
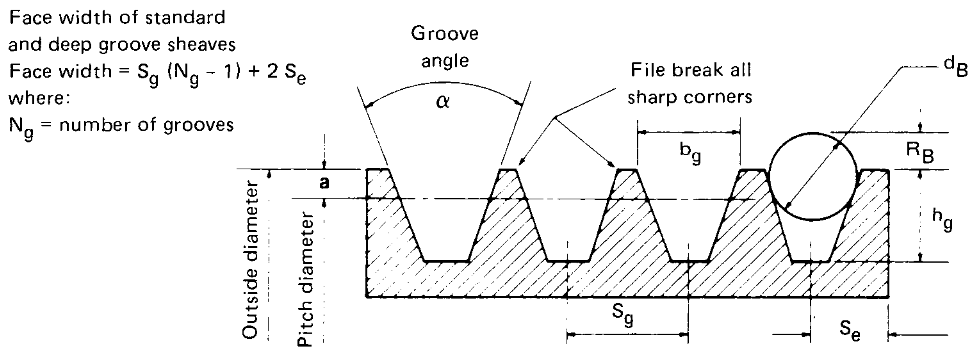


Fig. 8.2.95 Adjustable motor base.

Table 8.2.56*a* Standard Sheave Groove Dimensions, in



	Standard groove dimensions								Drive design factors		
Cross section	Outside diameter range	Groove angle $\alpha \pm 0.33°$	b_g	h_g min	R_B min	$d_B \pm 0.0005$	$S_g \pm 0.025$	S_e	Pitch diameter range	Minimum recommended pitch diameter	$2a$
A	Up through 5.65 Over 5.65	34 38	0.494 0.504 ± 0.005	0.460	0.148 0.149	0.4375 (7/16)	0.625	0.375 + 0.090 − 0.062	Up through 5.40 Over 5.40	3.0	0.250
B	Up through 7.35 Over 7.35	34 38	0.637 0.650 ± 0.006	0.550	0.189 0.190	0.5625 (9/16)	0.750	0.500 + 0.120 − 0.065	Up through 7.00 Over 7.00	5.4	0.350
A/B	Up through 7.35 Over 7.35	34 38	0.612 0.625 ± 0.006	0.612	0.230 0.226	0.5625 (9/16)	0.750	0.500 + 0.120 + 0.065		A = 3.0 B = 5.4	A = 0.620* B = 0.280
C	Up through 8.39 Over 8.39 to and incl. 12.40 Over 12.40	34 36 38	0.879 0.877 ± 0.007 0.895	0.750	0.274 0.276 0.277	0.7812 (25/32)	1.000	0.688 + 0.160 − 0.70	Up through 7.99 Over 7.99 to and incl. 12.00 Over 12.00	9.0	0.400
D	Up through 13.59 Over 13.59 to and incl. 17.60 Over 17.60	34 36 38	1.256 1271 ± 0.008 1.283	1.020	0.410 0.410 0.411	1.1250 (1 1/8)	1.438	+ 0.220 − 0.80	Up through 12.99 Over 12.99 to and incl. 17.00 Over 17.00	13.0	0.600
E	Up through 24.80 Over 24.80	36 38	1.527 1.542 ± 0.010	1.270	0.476 0.477	1.3438 (1 11/32)	1.750	1.125 + 0.280 − 0.090	Up through 24.00 Over 24.00	21.0	0.800

* The *a* values shown for the A/B combination sheaves are the geometrically derived values. These values may be different than those shown in manufacturers' catalogs.
SOURCE: ''Dayco Engineering Guide for V-Belt Drives,'' Dayco Corp., Dayton, OH, 1981, reprinted by permission.

Table 8.2.56*b* Classical Deep Groove Sheave Dimensions, in

Cross section	Deep groove dimensions						
	Outside diameter range	Groove angle $\alpha \pm 0.33°$	b_g	h_g min	$2a$	$S_g \pm 0.025$	S_e
A	Up through 5.96	34	0.539 $\pm$ 0.005	0.615	0.560	0.750	0.438 $+0.090$ -0.062
	Over 5.96	38	0.611				
B	Up through 7.71	34	0.747 $\pm$ 0.006	0.730	0.710	0.875	0.562 $+0.120$ -0.065
	Over 7.71	38	0.774				
C	Up through 9.00	34	1.066 $+$ 0.007				0.812 $+0.160$ -0.070
	Over 9.00 to and incl. 13.01	36	1.085	1.055	1.010	1.250	
	Over 13.01	38	1.105				
D	Up through 14.42	34	1.513				1.062 $+0.220$ -0.080
	Over 14.42 to and incl. 18.43	36	1.541 $\pm$ 0.008	1.435	1.430	1.750	
	Over 18.43	38	1.569				
E	Up through 25.69	36	1.816 $+$ 0.010	1.715	1.690	2.062	1.312 $+0.280$ -0.090
	Over 25.69	38	1.849				

SOURCE: ''Dayco Engineering Guide for V-Belt Drives,'' Dayco Corp., Dayton, OH, 1981, reprinted by permission.

Table 8.2.56*c* *Z* Dimensions for Quarter-Turn Drives, in

Center distance	3V, 5V, 8V *Z* dimension	A B, C, D *Z* dimension
20		0.2
30		0.2
40		0.4
50		0.4
60	0.2	0.5
80	0.3	0.5
100	0.4	1.0
120	0.6	1.5
140	0.9	2.0
160	1.2	2.5
180	1.5	3.5
200	1.8	4.0
220	2.2	5.0
240	2.6	6.0

SOURCE: ''Dayco Engineering Guide for V-Belt Drives,'' Dayco Corp., Dayton, OH, 1981, reprinted by permission.

Table 8.2.56*d* Width *W* of Set of Belts Using Deep-Groove Sheaves, in

No. of belts	V-belt cross section						
	3V	5V	8V	A	B	C	D
1	0.4	0.6	1.0	0.5	0.7	0.9	1.3
2	0.9	1.4	2.3	1.3	1.6	2.2	3.1
3	1.4	2.2	3.6	2.0	2.5	3.4	4.8
4	1.9	3.0	4.0	2.8	3.3	4.7	6.6
5	2.4	3.8	6.2	3.5	4.2	5.9	8.3
6	2.9	4.7	7.6	4.3	5.1	7.2	10.1
7	3.4	5.5	8.9	5.0	6.0	8.4	11.8
8	3.9	6.3	10.2	5.8	6.8	9.7	13.6
9	4.4	7.1	11.5	6.5	7.7	10.9	15.3
10	4.9	7.9	12.8	7.3	8.6	12.2	17.1

SOURCE: ''Dayco Engineering Guide for V-Belt Drives,'' Dayco Corp., Dayton, OH, 1981, reprinted by permission.

has been made by the screw, the spring will compensate for a normal amount of stretch in the belts. When there is more stretch than can be accommodated by the spring, the screw is turned to provide the necessary belt tensions. The carriage can be moved while the unit is in operation, and the motor base is provided for vertical as well as horizontal mounting.

CHAIN DRIVES

Roller-Chain Drives

The advantages of finished steel roller chains are high efficiency (around 98 to 99 percent), no slippage, no initial tension required, and chains may travel in either direction. The basic construction of roller chains is shown in Fig. 8.2.96 and Table 8.2.57.

The shorter the pitch, the higher the permissible operating speed of roller chains. Horsepower capacity in excess of that provided by a single chain may be had by the use of multiple chains, which are essentially parallel single chains assembled on pins common to all strands. Because of its lightness in relation to tensile strength, the effect of centrifugal pull does not need to be considered; even at the unusual speed of 6,000 ft/min, this pull is only 3 percent of the ultimate tensile strength.

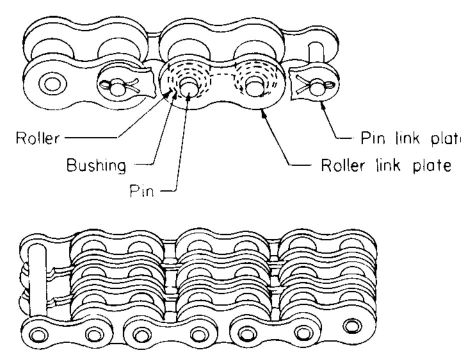


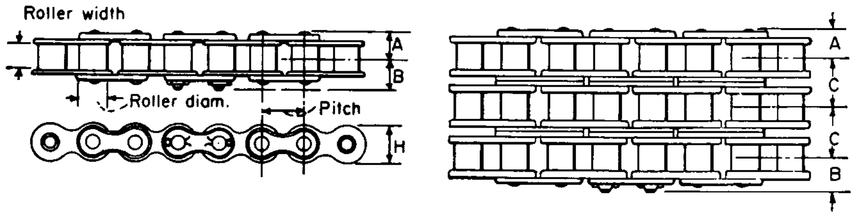
Fig. 8.2.96 Roller chain construction.

Sprocket wheels with fewer than 16 teeth may be used for relatively slow speeds, but 18 to 24 teeth are desirable for high-speed service. Sprockets with fewer than 25 teeth, running at speeds above 500 or 600 r/min, should be heat-treated to give a tough wear-resistant surface testing between 35 and 45 on the Rockwell C hardness scale.

If the speed ratio requires the larger sprocket to have as many as 128 teeth, or more than eight times the number on the smaller sprocket, it is usually better, with few exceptions, to make the desired reduction in two or more steps. The ANSI tooth form ASME B29.1 M-1993 allows roller chain to adjust itself to a larger pitch circle as the pitch of the chain elongates owing to natural wear in the pin-bushing joints. The greater the number of teeth, the sooner the chain will ride out too near the ends of the teeth.

Idler sprockets may be used on either side of the standard roller chain,

Table 8.2.57 Roller-Chain Data and Dimensions, in*



ANSI chain no.	ISO chain no.	Roller				Roller link plate		Dimension			Tensile strength per strand, lb	Recommended max speed r/min		
		Pitch	Width	Diam	Pin diam	Thickness	Height *H*	*A*	*B*	*C*		12 teeth	18 teeth	24 teeth
25	04C-1	¼	⅛	0.130	0.091	0.030	0.230	0.150	0.190	0.252	925	5,000	7,000	7,000
35	06C-1	⅜	3/16	0.200	0.141	0.050	0.344	0.224	0.290	0.399	2,100	2,380	3,780	4,200
41	085	½	¼	0.306	0.141	0.050	0.383	0.256	0.315	—	2,000	1,750	2,725	2,850
40	08A-1	½	5/16	0.312	0.156	0.060	0.452	0.313	0.358	0.566	3,700	1,800	2,830	3,000
50	10A-1	⅝	⅜	0.400	0.200	0.080	0.594	0.384	0.462	0.713	6,100	1,300	2,030	2,200
60	12A-1	¾	½	0.469	0.234	0.094	0.679	0.493	0.567	0.897	8,500	1,025	1,615	1,700
80	16A-1	1	⅝	0.625	0.312	0.125	0.903	0.643	0.762	1.153	14,500	650	1,015	1,100
100	20A-1	1¼	¾	0.750	0.375	0.156	1.128	0.780	0.910	1.408	24,000	450	730	850
120	24A-1	1½	1	0.875	0.437	0.187	1.354	0.977	1.123	1.789	34,000	350	565	650
140	28A-1	1¾	1	1.000	0.500	0.218	1.647	1.054	1.219	1.924	46,000	260	415	500
160	32A-1	2	1¼	1.125	0.562	0.250	1.900	1.250	1.433	2.305	58,000	225	360	420
180		2¼	1 13/32	1.406	0.687	0.281	2.140	1.421	1.770	2.592	76,000	180	290	330
200	40A-1	2½	1½	1.562	0.781	0.312	2.275	1.533	1.850	2.817	95,000	170	260	300
240	48A-1	3	1⅞	1.875	0.937	0.375	2.850	1.722	2.200	3.458	135,000	120	190	210

* For conversion to metric units (mm) multiply table values by 25.4.

Table 8.2.58 Selected Values of Horsepower Ratings of Roller Chains

ANSI no. and pitch, in	Number of teeth in small socket	Small sprocket, r/min										
		50	500	1,200	1,800	2,500	3,000	4,000	5,000	6,000	8,000	10,000
25	11	0.03	0.23	0.50	0.73	0.98	1.15	1.38	0.99	0.75	0.49	0.35
¼	15	0.04	0.32	0.70	1.01	1.36	1.61	2.08	1.57	1.20	0.78	0.56
	20	0.06	0.44	0.96	1.38	1.86	2.19	2.84	2.42	1.84	1.201	0.86
	25	0.07	0.56	1.22	1.76	2.37	2.79	3.61	3.38	2.57	1.67	1.20
	30	0.08	0.68	1.49	2.15	2.88	3.40	4.40	4.45	3.38	2.20	1.57
	40	0.12	.092	2.03	2.93	3.93	4.64	6.00	6.85	5.21	3.38	2.42
35	11	0.10	0.77	1.70	2.45	3.30	2.94	1.91	1.37	1.04	0.67	0.48
⅜	15	0.14	1.08	2.38	3.43	4.61	4.68	3.04	2.17	1.65	1.07	0.77
	20	0.19	1.48	3.25	4.68	6.29	7.20	4.68	3.35	2.55	1.65	1.18
	25	0.24	1.88	4.13	5.95	8.00	9.43	6.54	4.68	3.56	2.31	1.65
	30	0.29	2.29	5.03	7.25	9.74	11.5	8.59	6.15	4.68	3.04	2.17
	40	0.39	3.12	6.87	9.89	13.3	15.7	13.2	9.47	7.20	4.68	—
41	11	0.13	1.01	1.71	0.93	(0.58)	0.43	0.28	0.20	0.15	0.10	
½	15	0.18	1.41	2.73	1.49	(0.76)	0.69	0.45	0.32	0.24	0.16	
	20	0.24	1.92	4.20	2.29	(1.41)	1.06	0.69	0.49	0.38		
	25	0.31	2.45	5.38	3.20	(1.97)	1.49	0.96	0.69	0.53		
	30	0.38	2.98	6.55	4.20	(2.58)	1.95	1.27	0.91	0.69		
	40	0.51	4.07	8.94	6.47	(3.97)	3.01	1.95	1.40			
40	11	0.23	1.83	4.03	4.66	(3.56)	2.17	1.41	1.01	0.77	0.50	
½	15	0.32	2.56	5.64	7.43	(4.56)	3.45	2.24	1.60	1.22	0.79	
	20	0.44	3.50	7.69	11.1	(7.03)	5.31	3.45	2.47	1.88		
	25	0.56	4.45	9.78	14.1	(9.83)	7.43	4.82	3.45	2.63		
	30	0.68	5.42	11.9	17.2	(12.9)	9.76	6.34	4.54	3.45		
	40	0.93	7.39	16.3	23.4	(19.9)	15.0	9.76	6.99			
50	11	0.45	3.57	7.85	5.58	(3.43)	2.59	1.68	1.41	1.20	0.92	
⅝	15	0.63	4.99	11.0	8.88	(5.46)	4.13	2.68	2.25	1.92		
	20	0.86	6.80	15.0	13.7	(8.40)	6.35	4.13	3.46	2.95		
	25	1.09	8.66	19.0	19.1	(11.7)	8.88	5.77	4.83			
	30	1.33	10.5	23.2	25.1	(15.4)	11.7	7.58				
	40	1.81	14.4	31.6	38.7	(23.7)	18.0					

Table 8.2.58 Selected Values of Horsepower Ratings of Roller Chains (*Continued*)

ANSI no. and pitch, in	Number of teeth in small sprocket	Small sprocket, r/min										
		10	50	100	200	500	700	1,000	1,400	2,000	2,700	4,000
60 ¾	11	0.18	0.77	1.44	2.69	6.13	8.30	11.4	9.41	5.51	(3.75)	1.95
	15	0.25	1.08	2.01	3.76	8.57	11.6	16.0	15.0	8.77	(6.18)	3.10
	20	0.35	1.47	2.75	5.13	11.7	15.8	21.8	23.1	13.5	(9.20)	
	25	0.44	1.87	3.50	6.52	14.9	20.1	27.8	32.2	18.9	(12.9)	
	30	0.54	2.28	4.26	7.94	18.1	24.5	33.8	42.4	24.8	(16.9)	
	40	0.73	3.11	5.81	10.8	24.7	33.5	46.1	62.5	38.2		
80 1	11	0.42	1.80	3.36	6.28	14.3	19.4	19.6	11.8	6.93	4.42	
	15	0.59	2.52	4.70	8.77	20.0	27.1	31.2	18.9	11.0	7.04	
	20	0.81	3.44	6.41	12.0	27.3	37.0	48.1	29.0	17.0		
	25	1.03	4.37	8.16	15.2	34.7	47.0	64.8	40.6	23.8		
	30	1.25	5.33	9.94	18.5	42.3	57.3	78.9	53.3	31.2		
	40	1.71	7.27	13.6	25.3	57.7	78.1	108	82.1	48.1		
100 1¼	11	0.81	3.45	6.44	12.0	27.4	37.1	23.4	14.2	8.29		
	15	1.13	4.83	9.01	16.8	38.3	51.9	37.3	22.5	13.2		
	20	1.55	6.58	12.3	22.9	52.3	70.8	57.5	34.7	20.3		
	25	1.97	8.38	15.5	29.2	66.6	90.1	80.3	48.5	28.4		
	30	2.40	10.2	19.0	35.5	81.0	110	106	63.7	10.0		
	40	3.27	13.9	26.0	48.5	111	150	163	98.1			
120 1½	11	1.37	5.83	10.9	20.3	46.3	46.3	27.1	16.4	9.59		
	15	1.91	8.15	15.2	28.4	64.7	73.8	43.2	26.1			
	20	2.61	11.1	20.7	38.7	88.3	114	66.5	40.1			
	25	3.32	14.1	26.4	49.3	112	152	92.9	56.1			
	30	4.05	17.2	32.1	60.0	137	185	122	73.8			
	40	5.52	23.5	43.9	81.8	187	253	188	59.5			
140 1¾	11	2.12	9.02	16.8	31.4	71.6	52.4	30.7	18.5			
	15	2.96	12.6	23.5	43.9	100	83.4	48.9	29.5			
	20	4.04	17.2	32.1	59.9	137	128	75.2	45.4			
	25	5.14	21.9	40.8	76.2	174	180	105	63.5			
	30	6.26	26.7	49.7	92.8	212	236	138				
	40	8.54	36.4	67.9	127	289	363	213				
160 2	11	3.07	13.1	24.4	45.6	96.6	58.3	34.1				
	15	4.30	18.3	34.1	63.7	145	92.8	54.4				
	20	5.86	25.0	46.4	86.9	198	143	83.7				
	25	7.40	31.8	59.3	111	252	200	117				
	30	9.08	38.7	72.2	125	307	263	154				
	40	12.4	52.8	98.5	184	419	404					
180 2¼	11	4.24	18.1	33.7	62.9	106	64.1	37.5				
	15	5.93	25.3	47.1	88.0	169	102	59.7				
	20	8.10	34.5	64.3	120	260	157	92.0				
	25	10.3	43.9	81.8	153	348	220					
	30	12.5	53.4	99.6	186	424	289					
	40	17.1	72.9	136	254	579	398					
200 2½	11	5.64	24.0	44.8	83.5	115						
	15	7.88	33.5	62.6	117	184						
	20	10.7	45.8	85.4	159	283						
	25	13.7	58.2	109	203	396						
240 3	11	9.08	38.6	72.1	135							
	15	12.7	54.0	101	188							
	20	17.3	73.7	138	257							
	25	22.0	93.8	175	327							

NOTE: The sections separated by heavy lines denote the method of lubrication as follows: type A (left section), manual; type B (middle section), bath or disk; type C (right section), oil stream.
SOURCE: Supplementary section of ANSI B29.1-1975 (R93), adapted by permission.

to take up slack, to guide the chain around obstructions, to change the direction of rotation of a driven shaft, or to provide more wrap on another sprocket. Idlers should not run faster than the speeds recommended as maximum for other sprockets with the same number of teeth. It is desirable that idlers have at least two teeth in mesh with the chain, and it is advisable, though not necessary, to have an idler contact the idle span of chain.

Horsepower ratings are based upon the number of teeth and the rotative speed of the smaller sprocket, either driver or follower. The pin-bushing bearing area, as it affects allowable working load, is the important factor for medium and higher speeds. For extremely slow speeds, the chain selection may be based upon the ultimate tensile strength of the chain. For chain speeds of 25 ft/min and less, the chain pull should be not more than ⅕ of the ultimate tensile strength; for 50 ft/min, ⅙; for

100 ft/min, 1⁄7; for 150 ft/min, 1⁄8; for 200 ft/min, 1⁄9; and for 250 ft/min, 1⁄10 of the ultimate tensile strength.

Ratings for **multiple-strand chains** are proportional to the number of strands. The recommended numbers of strands for multiple chains are 2, 3, 4, 6, 8, 10, 12, 16, 20, and 24, with the maximum overall width in any case limited to 24 in.

The **horsepower ratings** in Table 8.2.58 are modified by the **service factors** in Table 8.2.59. Thus for a drive having a nominal rating of 3 hp, subject to heavy shock, abnormal conditions, 24-h/day operation, the chain rating obtained from Table 8.2.58 should be at least $3 \times 1.7 = 5.1$ hp.

Table 8.2.59 Service Factors for Roller Chains

	Load		
Power source	Smooth	Moderate shock	Heavy shock
Internal combustion engine with hydraulic drive	1.0	1.2	1.4
Electric motor or turbine	1.0	1.3	1.5
Internal combustion engine with mechanical drive	1.2	1.4	1.7

SOURCE: ANSI B29.1-1975, adapted by permission.

Chain-Length Calculations Referring to Fig. 8.2.97, L = length of chain, in; P = pitch of chain, in; R and r = pitch radii of large and small sprockets, respectively, in; D = center distance, in; A = tangent length, in; a = angle between tangent and centerline; N and n = number

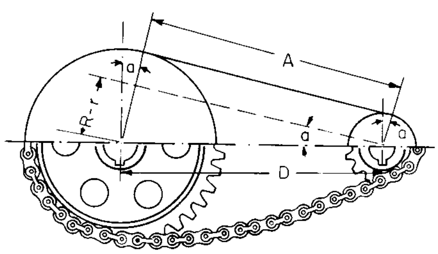


Fig. 8.2.97 Symbols for chain length calculations.

of teeth on larger and smaller sprockets, respectively; $180 + 2a$ and $180 - 2a$ are angles of contact on larger and smaller sprockets, respectively, deg.

$$a = \sin^{-1}[(R - r)/D] \qquad A = D \cos a$$

$$L = NP(180 + 2a)/360 + nP(180 - 2a)/360 + 2D \cos a$$

If L_p = length of chain in pitches and D_p = center distance in pitches,

$$L_p = (N + n)/2 + a(N - n)/180 + 2D_p \cos a$$

Avoiding the use of trigonometric tables,

$$L_p = 2C + (N + n)/2 + K(N - n)^2/C$$

where C is the center distance in pitches and K is a variable depending upon the value of $(N - n)/C$. Values of K are as follows:

$(N - n)/C$	0.1	1.0	2.0	3.0
K	0.02533	0.02538	0.02555	0.02584
$(N - n)/C$	4.0	5.0	6.0	
K	0.02631	0.02704	0.02828	

Formulas for chain length on multisprocket drives are cumbersome except when all sprockets are the same size and on the same side of the chain. For this condition, the chain length in pitches is equal to the sum of the consecutive center distances in pitches plus the number of teeth on one sprocket.

Actual chain lengths should be in even numbers of pitches. When necessary, an odd number of pitches may be secured by the use of an offset link, but such links should be avoided if possible. An offset link is one pitch; half roller link at one end and half pin link at the other end. If center distances are to be nonadjustable, they should be selected to give an initial snug fit for an even number of pitches of chain. For the average application, a center distance equal to 40 ± 10 pitches of chain represents good practice.

There should be at least 120° of wrap in the arc of contact on a power sprocket. For ratios of 3 : 1 or less, the wrap will be 120° or more for any center distance or number of teeth. To secure a wrap of 120° or more, for ratios greater than 3 : 1, the center distance must be not less than the difference between the pitch diameters of the two sprockets.

Sprocket Diameters N = number of teeth; P = pitch of chain, in; D = diameter of roller, in. The pitch of a standard roller chain is measured from the center of a pin to the center of an adjacent pin.

$$\text{Pitch diam} = P/\sin \frac{180}{N}$$

$$\text{Bottom diam} = \text{pitch diam} - D$$

$$\text{Outside diam} = P\left(0.6 + \cot \frac{180}{N}\right)$$

$$\text{Caliper diam} = \left(\text{pitch diam} \times \cos \frac{90}{N}\right) - D$$

The exact bottom diameter cannot be measured for an odd number of teeth, but it can be checked by measuring the distance (caliper diameter) between bottoms of the two tooth spaces nearest opposite to each other. Bottom and caliper diameters must not be oversize—all tolerances must be negative. ANSI negative tolerance $= 0.003 + 0.001P\sqrt{N}$ in.

Design of Sprocket Teeth for Roller Chains The **section profile** for the teeth of roller chain sprockets, recommended by ANSI, has the proportions shown in Fig. 8.2.98. Let P = chain pitch; W = chain width (length of roller); n = number of strands of multiple chain; M = overall width of tooth profile section; H = nominal thickness of the link plate, all in inches. Referring to Fig. 8.2.98, $T = 0.93W - 0.006$, for

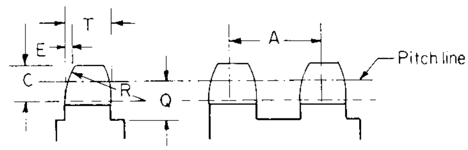


Fig. 8.2.98 Sprocket tooth sections.

single-strand chain; $= 0.90W - 0.006$, for double- and triple-strand chains; $= 0.88W - 0.006$, for quadruple- or quintuple-strand chains; and $= 0.86W - 0.006$, for sextuple-strand chain and over. $C = 0.5P$. $E = \frac{1}{8}P$. $R(\min) = 1.063P$. $Q = 0.5P$. $A = W + 4.15H + 0.003$. $M = A(n - 1) + T$.

Inverted-tooth (silent) chain drives have a typical tooth form shown in Fig. 8.2.99. Such chains should be operated in an oil-retaining casing with provisions for lubrication. The use of offset links and chains with an uneven number of pitches should be avoided.

Horsepower ratings per inch of silent chain width, given in ANSI B29.2-1957 (R1971), for various chain pitches and speeds, are shown in

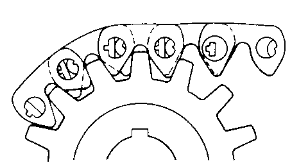


Fig. 8.2.99 Inverted tooth (silent-chain) drive.

Table 8.2.60 Horsepower Rating per Inch of Chain Width, Silent-Chain Drive (Small Pitch)

Pitch, in	No. of teeth in small sprocket	Small sprocket, r/min											
		500	600	700	800	900	1,200	1,800	2,000	3,500	5,000	7,000	9,000
3/16	21	0.41	0.48	0.55	0.62	0.68	0.87	1.22	1.33	2.03	2.58	3.12	3.35
	25	0.49	0.58	0.66	0.74	0.82	1.05	1.47	1.60	2.45	3.13	3.80	4.10
	29	0.57	0.67	0.76	0.86	0.95	1.21	1.70	1.85	2.83	3.61	4.40	4.72
	33	0.64	0.75	0.86	0.97	1.07	1.37	1.90	2.08	3.17	4.02	4.85	—
	37	0.71	0.84	0.96	1.08	1.19	1.52	2.11	2.30	3.48	4.39	5.24	—
	45	0.86	1.02	1.15	1.30	1.43	1.83	2.53	2.75	4.15	5.21	—	—
Type*		I						II			III		

Pitch, in	No. of teeth in small sprocket	Small sprocket, r/min												
		100	500	1,000	1,200	1,500	1,800	2,000	2,500	3,000	3,500	4,000	5,000	6,000
3/8	21	0.58	2.8	5.1	6.0	7.3	8.3	9.0	10	11	12	12	12	10
	25	0.69	3.3	6.1	7.3	8.8	10	11	13	14	15	5	15	14
	29	0.80	3.8	7.3	8.5	10	12	13	15	16	18	19	19	18
	33	0.90	4.4	8.3	9.8	12	14	15	18	19	21	21	21	20
	37	1.0	4.9	9.1	11	14	15	16	20	21	24	24	24	—
	45	1.3	6.0	11	13	16	19	20	24	26	28	29	—	—
Type*		I			II				III					

Pitch, in	No. of teeth in small sprocket	Small sprocket, r/min										
		100	500	700	1,000	1,200	1,800	2,000	2,500	3,000	3,500	4,000
1/2	21	1.0	5.0	6.3	8.8	10	14	14	15	16	16	—
	25	1.2	5.0	7.5	10	13	16	18	20	21	21	20
	29	1.4	6.3	8.8	13	14	19	21	24	25	25	25
	33	1.6	7.5	10	14	16	23	24	28	29	30	29
	37	1.9	8.8	11	16	19	25	26	30	33	33	—
	45	2.5	10	14	19	23	30	30	36	39	—	—
Type*		I		II				III				

Pitch, in	No of teeth in small sprocket	Small sprocket, r/min									
		100	500	700	1,000	1,200	1,800	2,000	2,500	3,000	3,500
5/8	21	1.6	7.5	10	13	15	19	20	20	20	—
	25	1.9	8.8	11	16	19	24	25	26	26	24
	29	2.1	10	14	19	21	28	30	31	31	29
	33	2.5	11	16	21	25	33	34	36	36	34
	37	2.8	13	18	24	28	36	39	43	41	—
	45	3.4	16	21	29	34	44	46	—	—	—
Type*		I		II		III					

Pitch, in	No. of teeth in small sprocket	Small sprocket, r/min								
		100	500	700	1,000	1,200	1,500	1,800	2,000	2,500
3/4	21	2.3	10	14	18	20	23	24	25	24
	25	2.8	13	16	21	25	29	31	31	30
	29	3.1	15	20	26	30	34	36	38	38
	33	3.6	16	23	30	34	39	43	44	44
	37	4.0	19	25	34	39	44	48	49	49
	45	4.9	23	30	40	46	53	56	58	—
Type*		I		II		III				

* Type I: manual, brush, or oil cup. Type II: bath or disk. Type III: circulating pump.
NOTE: For best results, smaller sprocket should have at least 21 teeth.
SOURCE: Adapted from ANSI B29.2M-1982.

Tables 8.2.60 and 8.2.61. These ratings are based on ideal drive conditions with relatively little shock or load variation, an average life of 20,000 h being assumed. In utilizing the horsepower ratings of the tables, the nominal horsepower of the drive should be multiplied by a service factor depending on the application. Maximum, or worst-case scenario, service factors are listed in Table 8.2.62.

For details on lubrication, sprocket dimensions, etc., see ANSI B29.2M-1957(R71). In utilizing Tables 8.2.60 and 8.2.61 (for a complete set of values, see ANSI B29.2M-1982) the required chain width is obtained by dividing the design horsepower by the horsepower ratings given. For calculating silent-chain lengths, the procedure for roller-chain drives may be used.

Table 8.2.61 Horsepower Rating per Inch of Chain Width, Silent-Chain Drive (Large Pitch)

Pitch, in	No. of teeth in small sprocket	Small sprocket, r/min										
		100	200	300	400	500	700	1,000	1,200	1,500	1,800	2,000
1	21	3.8	7.5	11	15	18	23	29	31	33	33	—
	25	5.0	8.8	14	18	21	28	35	39	41	41	41
	29	5.0	11	16	20	25	33	11	46	50	51	50
	33	6.3	13	18	24	29	38	49	54	59	59	58
	37	6.8	14	20	26	33	43	54	60	65	66	—
	45	8.8	16	25	31	39	51	65	71	76	—	—
Type*		I			II				III			

Pitch, in	No. of teeth in small sprocket	Small sprocket, r/min										
		100	200	300	400	500	600	700	800	1,000	1,200	1,500
$1\frac{1}{4}$	21	6.3	11	18	23	26	30	33	36	40	41	—
	25	7.5	14	20	26	31	36	40	44	50	53	53
	29	8.6	16	24	31	38	43	48	53	59	63	64
	33	9.9	19	28	35	43	49	55	60	69	73	74
	37	11	21	30	40	48	55	63	68	76	81	—
	45	13	26	38	49	59	68	75	81	91	—	—
Type*		I			II			III				

Pitch, in	No. of teeth in small sprocket	Small sprocket, r/min										
		100	200	300	400	500	600	700	800	900	1,000	1,200
$1\frac{1}{2}$	21	8.8	16	24	30	36	40	44	46	49	49	—
	25	10	20	29	38	44	50	55	59	61	65	64
	29	13	24	34	44	51	59	65	70	74	75	76
	33	14	28	39	50	59	68	75	80	85	88	89
	37	16	30	44	59	66	76	84	90	96	99	—
	45	19	38	54	68	81	93	101	108	113	—	—
Type*		I		II			III					

Pitch, in	No. of teeth in small sprocket	Small sprocket, r/min								
		100	200	300	400	500	600	700	800	900
2	21	16	29	40	50	53	63	65	—	—
	25	18	35	49	61	70	78	83	85	85
	29	21	41	58	73	84	93	99	103	103
	33	25	46	66	83	96	106	114	118	118
	37	28	53	75	72	110	124	128	131	—
	45	34	64	90	113	131	144	151	—	—
Type*		I	II			III				

* Type I: manual, brush, or oil cup. Type II: bath or disk. Type III: circulating pump.
NOTE: For best results, small sprocket should have at least 21 teeth.
SOURCE: Adapted from ANSI B29.2-1982.

Table 8.2.62 Service Factors* for Silent-Chain Drives

Application	Fluid-coupled engine or electric motor		Engine with straight mechanical drive		Torque converter drives	
	10 h	24 h	10 h	24 h	10 h	24 h
Agitators	1.1	1.4	1.3	1.6	1.5	1.8
Brick and clay machinery	1.4	1.7	1.6	1.9	1.8	2.1
Centrifuges	1.4	1.7	1.6	1.9	1.8	2.1
Compressors	1.6	1.9	1.8	2.1	2.0	2.3
Conveyors	1.6	1.9	1.8	2.1	2.0	2.3
Cranes and hoists	1.4	1.7	1.6	1.9	1.8	2.1
Crushing machinery	1.6	1.9	1.8	2.1	2.0	2.3
Dredges	1.6	1.9	1.8	2.1	2.0	2.3
Elevators	1.4	1.7	1.6	1.9	1.8	2.1
Fans and blowers	1.5	1.8	1.7	2.0	1.9	2.2
Mills—flour, feed, cereal	1.4	1.7	1.6	1.9	1.8	2.1
Generator and excitors	1.2	1.5	1.4	1.7	1.6	1.9
Laundry machinery	1.2	1.5	1.4	1.7	1.6	1.9
Line shafts	1.6	1.9	1.8	2.1	2.0	2.3
Machine tools	1.4	1.7	—	—	—	—
Mills—ball, hardinge, roller, etc.	1.6	1.9	1.8	2.1	—	—
Mixer—concrete, liquid	1.6	1.9	1.8	2.1	2.0	2.3
Oil field machinery	1.6	1.9	1.8	2.1	2.0	2.3
Oil refinery equipment	1.5	1.8	1.7	2.0	1.9	2.2
Paper machinery	1.5	1.8	1.7	2.0	1.9	2.2
Printing machinery	1.5	2.0	1.4	1.7	1.6	1.9
Pumps	1.6	1.9	1.8	2.1	2.0	2.3
Rubber plant machinery	1.5	1.8	1.7	2.0	1.9	2.2
Rubber mill equipment	1.6	1.9	1.8	2.1	2.0	2.3
Screens	1.5	1.8	1.7	2.0	1.9	2.2
Steel plants	1.3	1.6	1.5	1.8	1.7	2.0
Textile machinery	1.1	1.4	—	—	—	—

* The values shown are for the maximum worst-case scenario for each application. The table was assembled from ANSI B29.2M-1982, by permission. Because the listings above are maximum values, overdesign may result when they are used. Consult the ANSI tables for specific design values.

ROTARY AND RECIPROCATING ELEMENTS

Slider Crank Mechanism

Kinematics The slider crank mechanism is widely used in automobile engines, punch presses, feeding mechanisms, etc. For such mechanisms, displacements, velocities, and accelerations of the parts are important design parameters. The basic mechanism is shown in Fig. 8.2.100. The slider displacement x is given by

$$x = r\cos\theta + l\sqrt{1 - [(r/l)\sin\theta]^2}$$

where r = crank length, l = connecting-rod length, θ = crank angle measured from top dead center position. Slider velocity is given by

$$\dot{x} = V = -r\omega\left(\sin\theta + \frac{r\sin 2\theta}{2l\cos\beta}\right)$$

where ω = instantaneous angular velocity of the crank at position θ and

$$\cos\beta = \sqrt{1 - (r/l)^2\sin^2\theta}$$

Slider acceleration is given by

$$a = \ddot{x} = -r\alpha\left(\sin\theta + \frac{r}{l}\,\frac{\sin 2\theta}{2\cos\beta}\right) - r\omega^2\left(\cos\theta + \frac{r}{l}\,\frac{\cos 2\theta}{\cos\beta} + \frac{r}{l^3}\,\frac{\sin^2 2\theta}{4\cos^3\beta}\right)$$

where α = instantaneous angular acceleration of the crank at position θ.

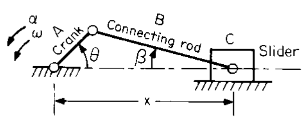


Fig. 8.2.100 Basic slider-crank mechanism.

Forces Neglecting gravity effects, the forces in a mechanism arise from those produced by input and output forces or torques (herewith called static components). Such forces may be produced by driving motors, shaft loads, expanding cylinder gases, etc. The net forces on the various links cause accelerations of the mechanism's masses, and can be thought of as dynamic components. The static components must be borne by the various links, thus giving rise to internal stresses in those parts. The supporting bearings and slide surfaces also feel the effects of these components, as do the support frames. Stresses are also induced by the dynamic components in the links, and such components cause shaking forces and shaking moments in the support frame.

By building onto the basic mechanism appropriately designed countermasses, the support frame and its bearings can be relieved of a significant portion of the dynamic component effects, sometimes called inertia effects. The augmented mechanism is then considered to be ''balanced.'' The static components cannot be relieved by any means, so that the support frame and its bearings must be designed to carry safely the static component forces and not be overstressed. Figure 8.2.101 illustrates a common, simple form of approximate balancing. Sizing of the countermass is somewhat complicated because the total

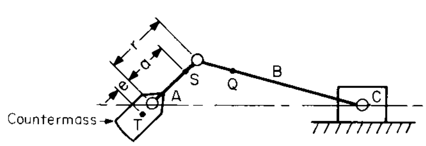


Fig. 8.2.101 ''Balanced'' slider-crank mechanism where T = center of mass, S = center of mass of crank A, and Q = center of mass of connecting rod B.

inertia effect is the vector sum of the separate link inertias, which change in magnitude and direction at each position of the crank. The countermass D is sized to contain sufficient mass to completely balance the crank plus an additional mass (effective mass) to ''balance'' the other links (connecting rod and slider). In simple form, the satisfying condition is approximately

$$M_D e\omega^2 = M_A a\omega^2 + M_{\text{eff}} e\omega^2$$

where a = distance from crankshaft to center of mass of crank (note that most often a is approximately equal to the crank radius r); e = distance from crankshaft to center of mass of countermass; M_{eff} = additional mass of countermass to ''balance'' connecting rod and slider.

From one-half to two-thirds the slider mass (e.g., $\frac{1}{2}M_c \le M_{\text{eff}} \le \frac{2}{3}M_c$) is usually added to the countermass for a single-cylinder engine. For critical field work, single and multiple slider crank mechanisms are dynamically balanced by experimental means.

Forces and Torques Figure 8.2.102*a* shows an exploded view of the slider crank mechanism and the various forces and torques on the links (neglecting gravity and weight effects). Inertial effects are shown as broken-line vectors and are manipulated in the same manner as the actual or real forces. The inertial effects are also known as **D'Alembert forces.** The meanings are as follows: $F_1 = -M_D e\omega^2$, parallel to crank A; $F_2 = -M_a e\alpha$, perpendicular to crank A; $F_3 = (z/l)F_9$; F_4 = as found in Fig. 8.2.102*b*; $F_5 = F_2 - F_7$; $F_6 = F_1 - F_8$; $F_7 = -M_A r\omega^2$, parallel to crank A; $F_8 = -M_A r\alpha$, perpendicular to crank A; $F_9 = -M_B \times$ absolute acceleration of Q, where acceleration of point Q can be found

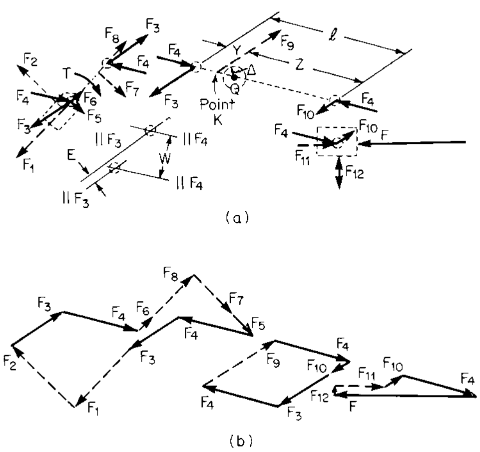


Fig. 8.2.102 (*a*) Forces and torques; (*b*) force polygons.

graphically by constructing an acceleration polygon of the mechanism (see for example Shigley, ''Kinematic Analysis of Mechanisms,'' McGraw-Hill); $F_{10} = (y/l)F_9$; $F_{11} = -M_c \times$ absolute acceleration of slider $\ddot{x}$; F_{12} = normal wall force (neglecting friction); F = external force on slider's face, where the vector sum $F + F_4 + F_{10} + F_{11} + F_{12} = 0$; T = external crankshaft torque, where the algebraic sum $T + F_3E + F_4W + F_2e + F_7a = 0$ (note that signs must reflect direction of torque); $T_t = F_3E + F_wW$ = transmitted torque; K = the effective location of force F_9. The distance Δ of force F_9 from the center of mass of connecting rod B (see Fig. 8.2.102*a*) is given by $\Delta = J_{B(\text{cm})} \times$ angular acceleration of link $B/M_B \times$ absolute acceleration of point Q. Figure 8.2.102*b* shows the force polygons of the separate links of the mechanism.

Flywheel

One can surmise that both F and T may be functions of crank angle θ. Even if one or the other were deliberately kept constant, the remaining one would still be a function of θ. If a steady-speed crank is desired (ω = constant and α = zero), then external crankshaft torque T must be constantly adjusted to equal transmitted torque T_t. In such a situation a motor at the crankshaft would suffer fatigue effects. In a combustion engine the crankshaft would deliver a fluctuating torque to its load.

Inserting a flywheel at the crankshaft allows the peak and valley excursions of ω to be considerably reduced because of the flywheel's ability to absorb energy over periods when $T > T_t$ and to deliver back into the system such excess energy when $T < T_t$. Figure 8.2.103*a* illustrates the above concepts, also showing that over one cycle of a repeated event the excess (+) energies and the deficient (−) energies are equal. The greatest crank speed change tends to occur across a single large positive loop, as illustrated in Fig. 8.2.103*a*.

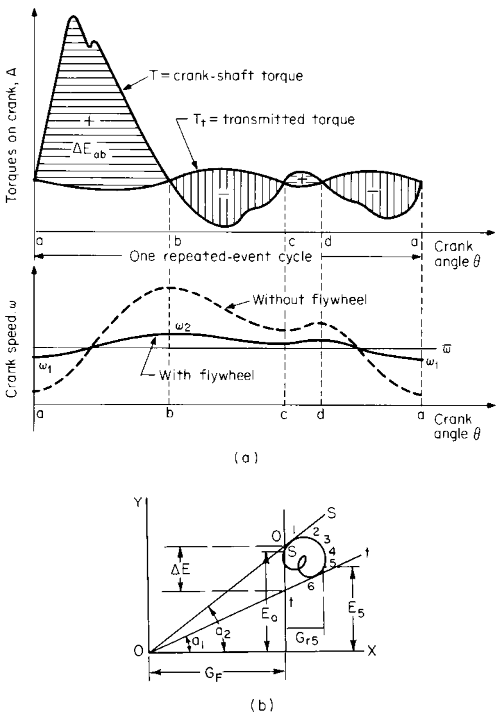


Fig. 8.2.103 Sizing the flywheel. (*a*) Variation of torque and crank speed vs. crank angle, showing ΔE_{ab}; (*b*) graphics for Wittenbauer's analysis.

Sizing the Flywheel For the single largest energy change we can write

$$\Delta E_{ab} \int_a^b (T - T_t)\, d\theta = \frac{J_0}{2}(\omega_2^2 - \omega_1^2)$$
$$= \frac{J_0}{2}(\omega_2 - \omega_1)(\omega_2 + \omega_1)$$

where J_0 = flywheel moment of inertia plus effective mechanism moment of inertia.

Define $\bar{\omega} \doteq (\omega_2 + \omega_1)/2$ and C_s = coefficient of speed fluctuation = $(\omega_2 - \omega_1)/\bar{\omega}$. Hence $\Delta E_{ab} = J_0 C_s \bar{\omega}^2$. Acceptable values of C_s are:

Pumps	0.03–0.05
Machine tools	0.025–0.03
Looms	0.025
Paper mills	0.025
Spinning mills	0.015
Crusher	0.02
Electric generators, ac	0.003
Electric generators, dc	0.002

Evaluating ΔE_{ab} involves finding the integral

$$\int_a^b (T - T_1)\, d\theta$$

which can be done graphically or by a numerical technique such as Simpson's rule.

Wittenbauer's Analysis for Flywheel Performance This method does not involve more computation work than the one described above, but it is more accurate where the reciprocating parts are comparatively heavy. Wittenbauer's method avoids the inaccuracy resulting from the evaluation of the inertia forces on the reciprocating parts on the basis of the uniform nominal speed of rotation for the engine.

Let the crankpin velocity be represented by v_r and the velocity of any moving masses (m_1, m_2, m_3, etc.) at any instant of phase be represented, respectively, by v_1, v_2, v_3, etc. The kinetic energy of the entire engine system of moving masses may then be expressed as

$$E = ½(m_1 v_1^2 + m_2 v_2^2 + m_3 v_3^2 + \cdots) = ½ M_r v_r^2$$

or, the single reduced mass M_r at the crankpin which possesses the equivalent kinetic energy is

$$M_r = [m_1(v_1/v_r)^2 + m_2(v_2/v_r)^2 + m_3(v_3/v_r)^2 + \cdots]$$

In an engine mechanism, sufficiently accurate values of M_r can be obtained if the weight of the connecting rod is divided between the crankpin and the wrist pin so as to retain the center of gravity of the rod in its true position; usually 0.55 to 0.65 of the weight of the connecting rod should be placed on the crankpin, and 0.45 to 0.35 of the weight on the wrist pin. M_r is a variable in engine mechanisms on account of the reciprocating parts and should be found for a number of crank positions. It should include all moving masses except the flywheel.

The total energy E used in accelerating reciprocating parts from the beginning of the forward stroke up to any crank position can be obtained by finding from the indicator cards the total work done in the cylinder (on both sides of the piston) up to that time and subtracting from it the work done in overcoming the resisting torque, which may usually be assumed constant. The mean energy of the moving masses is $E_0 = ½ M_r v_r^2$.

In Fig. 8.2.103*b*, the reduced weights of the moving masses $G_F + G_{r5}$ are plotted on the X axis corresponding to different crank positions. $G_F = gM_F$ is the reduced flywheel weight and $G_{r5} = gM_{r5}$ is the sum of the other reduced weights. Against each of these abscissas is plotted the energy E available for acceleration measured from the beginning of the forward stroke. The curve $O123456$ is the locus of these plotted points.

The diagram possesses the following property: Any straight line drawn from the origin O to any point in the curve is a measure of the velocity of the moving masses; tangents bounding the diagram measure the limits of velocity between which the crankpin will operate. The maximum linear velocity of the crankpin in feet per second is $v_2 = \sqrt{2g}\tan a_2$, and the minimum velocity is $v_1 = \sqrt{2g}\tan a_1$. Any desired change in v_1 and v_2 may be accomplished by changing the value of G_F, which means a change in the flywheel weight or a change in the flywheel weight reduced to the crankpin. As G_F is very large compared with G_r and the point 0 cannot be located on the diagram unless a very large drawing is made, the tangents are best formed by direct calculation:

$$\tan\alpha_2 = \frac{v_r^2}{2g}(1 + k) \qquad \tan\alpha_1 = \frac{v_r^2}{2g}(1 - k)$$

where k is the coefficient of velocity fluctuation. The two tangents *ss* and *tt* to the curve $O123456$, thus drawn, cut a distance ΔE and on the ordinate E_0. The reduced flywheel weight is then found to be

$$G_F(\Delta E)g/(v_r^2 k)$$

SPRINGS

It is assumed in the following formulas that the springs are in no case stressed beyond the elastic limit (i.e., that they are perfectly elastic) and that they are subject to Hooke's law.

Notation

P = safe load, lb
f = deflection for a given load P, in
l = length of spring, in
V = volume of spring, in^3
S_s = safe stress (due to bending), lb/in^2
S_v = safe shearing stress, lb/in^2
U = resilience, in · lb

For sheet metal and wire gages, ferrous and nonferrous, see Table 8.2.76 and metal suppliers' catalogs.

The **work** in inch-pounds performed in **deflecting a spring** from 0 to f (spring duty) is $U = Pf/2 = s_s^2V/(CE)$. This is based upon the assumption that the deflection is proportional to the load, and C is a constant dependent upon the shape of the springs.

The **time of vibration** T (in seconds) **of a spring** (weight not considered) is equal to that of a simple circular pendulum whose length l_0 equals the deflection f (in feet that is produced in the spring by the load P. $T = \pi\sqrt{l_0/g}$, where g = acceleration of gravity, ft/s^2.

Springs Subjected to Bending

1. **Rectangular plate spring** (Fig. 8.2.104).

$$P = bh^2S_s/(6l) \qquad I = bh^3/12 \qquad U = Pf/2 = VS_s^2/(18E)$$
$$f = Pl^3/(3EI) = 4Pl^3/(bh^3E) = 2l^2S_s/(3hE)$$

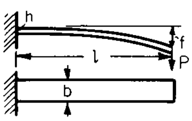


Fig. 8.2.104 Rectangular plate spring.

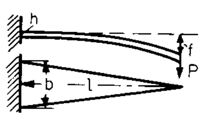


Fig. 8.2.105 Triangular plate spring.

2. **Triangular plate spring** (Fig. 8.2.105). The elastic curve is a circular arc.

$$P = bh^2S_s/(6l) \qquad I = bh^3/12 \qquad U = Pf/2 = S_s^2V/(6E)$$
$$f = Pl^3/(2EI) = 6Pl^3/(bh^3E) = l^2S_s/(hE)$$

3. **Rectangular plate spring with end tapered** in the form of a cubic parabola (Fig. 8.2.106). The elastic curve is a circular arc; P, I, and f same as for triangular plate spring (Fig. 8.2.105); $U = Pf/2 = S_s^2V/(9E)$.

The strength and deflection of **single-leaf flat springs** of various forms are given (Bruce, *Am. Mach.*, July 19, 1900) by the formulas $h = al^2/f$ and $b = cPl/h^2$. The volume of the spring is given by $V = vlbh$. The values of constants a and c and the resilience in inch-pounds per cubic inch are given in Table 8.2.63, in terms of the safe stress S_s. Values of v are given also.

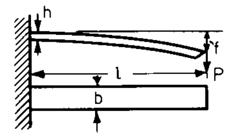


Fig. 8.2.106 Rectangular plate spring: tapered end.

4. **Compound (leaf or laminated) springs.** If several springs of rectangular section are combined, the resulting compound spring should (1) form a beam of uniform strength that (2) does not open between the joints while bending (i.e., elastic curve must be a circular arc). Only the type immediately following meets both requirements, the others meeting only the second requirement.

5. **Laminated triangular plate spring** (Fig. 8.2.107). If the triangular plate spring shown at I is cut into an even number ($= 2n$) of strips of equal width (in this case eight strips of width $b/2$), and these strips are combined, a laminated spring will be formed whose carrying capacity will equal that of the original uncut spring; or $P = nbh^2S_s/(6l)$; $n = 6Pl/(bh^2S_s)$.

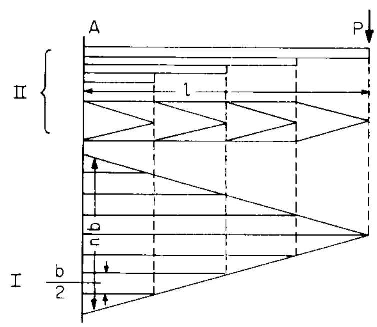
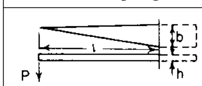
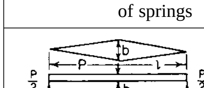
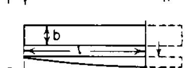
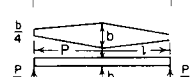
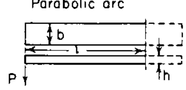
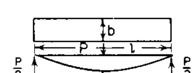
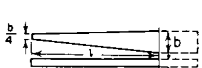
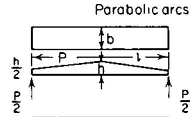
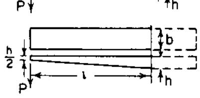
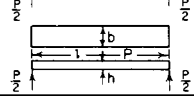


Fig. 8.2.107 Laminated triangular plate spring.

Table 8.2.63 Strength and Deflection of Single-Leaf Flat Springs

Load applied at end of spring; $c = 6/S_s$				Load applied at center of spring; $c = 6/4S_s$			
Plans and elevations of springs	a	U	v	Plans and elevations of springs	a	U	v
	$\frac{S_s}{E}$	$\frac{S_s^2}{6E}$	$\frac{1}{2}$		$\frac{S_s}{4E}$	$\frac{S_s^2}{6E}$	$\frac{1}{2}$
 Parabolic arc	$\frac{4S_s}{3E}$	$\frac{S_s^2}{6E}$	$\frac{2}{3}$		$\frac{0.87S_s}{4E}$	$\frac{0.70S_s^2}{6E}$	$\frac{5}{8}$
	$\frac{2S_s}{3E}$	$\frac{0.33S_s^2}{6E}$	1	 Parabolic arcs	$\frac{S_s}{3E}$	$\frac{S_s^2}{6E}$	$\frac{2}{3}$
	$\frac{0.87S_s}{E}$	$\frac{0.70S_s^2}{6E}$	$\frac{5}{8}$		$\frac{1.09S_s}{4E}$	$\frac{0.725S_s^2}{6E}$	$\frac{3}{4}$
	$\frac{1.09S_s}{E}$	$\frac{0.725S_s^2}{6E}$	$\frac{3}{4}$		$\frac{S_s}{6E}$	$\frac{0.33S_s^2}{6E}$	1

6. **Laminated rectangular plate spring with lead ends tapered** in the form of a cubical parabola (Fig. 8.2.108); see case 3.

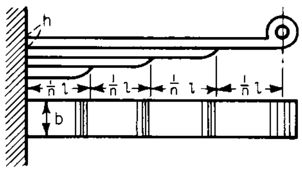


Fig. 8.2.108 Laminated rectangular plate spring with leaf end tapered.

7. **Laminated trapezoidal plate spring with leaf ends tapered** (Fig. 8.2.109). The ends of the leaves are trapezoidal and are tapered according to the formula

$$z = \frac{h}{\sqrt[3]{1 + (b_1/b)(a/x - 1)}}$$

8. **Semielliptic springs** (for locomotives, trucks, etc.). Referring to Fig. 8.2.110, the load $2P$ (lb) acting on the spring center band produces a tensional stress $P/\cos a$ in each of the inclined shackle links. This is resolved into the vertical force P and the horizontal force $P \tan a$, which together produce a bending moment $M = P(l + p \tan a)$. The equations

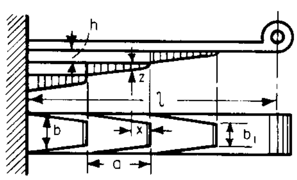


Fig. 8.2.109 Laminated trapezoidal plate spring with leaf ends tapered.

given in (1), (2), and (3) apply to curved as well as straight springs. The bearing force $= 2P = (2nbh^2/6)[S_s/(l + p \tan a)]$, and the deflection $= [6l^2/(nbh^3)]P(l + p \tan a)/E = l^2S_2/(hE)$.

In addition to the bending moment, the leaves are subjected to the tension force $P \tan a$ and the transverse force P, which produce in the upper leaf an additional stress $S = P \tan a/(bh)$, as well as a transverse shearing stress.

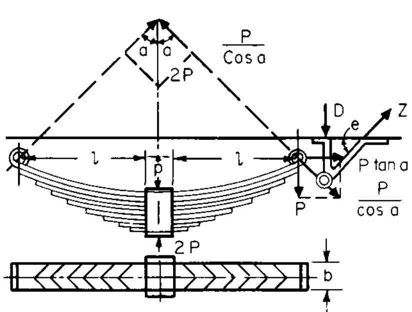


Fig. 8.2.110 Semielliptic springs.

In determining the number of leaves n in a given spring, allowance should be made for an excess load on the spring caused by the vibration. This is usually done by decreasing the allowable stress about 15 percent.

The foregoing does not take account of initial stresses caused by the band. For more detailed information, see Wahl, ''Mechanical Springs,'' Penton.

9. **Elliptic springs.** Safe load $P = nbh^2S_s/(6l)$, where $l = ½$ distance between bolt eyes (less ½ length of center band, where used); deflection $f = 4l^2S_sK/(hE)$, where

$$K = \frac{1}{(1 - r)^3}\left[\frac{1 - r^2}{2} - 2r(1 - r) - r^2 \ln r\right]$$

r being the number of full-length leaves ÷ total number (n) of leaves in the spring. All dimensions in inches. For semielliptic springs, the deflection is only half as great. Safe load $= nbh^2S_s/(3l)$. (Peddle, *Am. Mach.*, Apr. 17, 1913.)

Coiled Springs In these, the load is applied as a couple Pr which turns the spring while winding or holds it in place when wound up. If the spindle is not to be subjected to bending moment, P must be replaced by two equal and opposite forces ($P/2$) acting at the circumference of a circle of radius r. The formulas are the same in both cases. The springs are assumed to be fixed at one end and free at the other. The bending moment acting on the section of least resistance is always Pr. The length of the straightened spring $= l$. See Benjamin and French, Experiments on Helical Springs, *Trans. ASME,* **23**, p. 298.

For **heavy closely coiled helical springs** the usual formulas are inaccurate and result in stresses greatly in excess of those assumed. See Wahl, Stresses in Heavy Closely-Coiled Helical Springs, *Trans. ASME,* 1929. In springs 10 to 12 and 15 to 18, the quantity k is unity for lighter springs and has the stated values (supplied by Wahl) for heavy closely coiled springs.

10. **Spiral coiled springs of rectangular cross section** (Fig. 8.2.111).

$$P = bh^2S_s/(6rk) \qquad I = bh^3/12 \qquad U = Pf/2 = S_s^2V/(6Ek^2)$$
$$f = ra = Plr^2/(EI) = 12Plr^2/(Ebh^3) = 2rlS_s/(hEk)$$

For heavy closely coiled springs, $k = (3c - 1)/(3c - 3)$, where $c = 2R/h$ and R is the minimum radius of curvature at the center of the spiral.

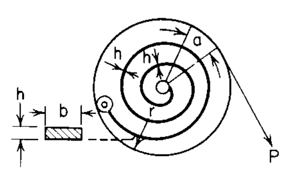


Fig. 8.2.111 Spiral coiled spring: rectangular cross section.

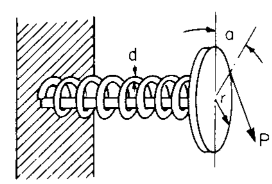


Fig. 8.2.112 Cylindrical helical spring: circular cross section.

11. **Cylindrical helical spring of circular cross section** (Fig. 8.2.112).

$$P = \pi d^3S_s/(32rk) \qquad I = \pi d^4/64 \qquad U = Pf/2 = S_s^2V/(8Ek^2)$$
$$f = ra = Plr^2/(EI) = 64Plr^2/(\pi Ed^4) = 2rlS_s/(dEk)$$

For heavy closely coiled springs, $k = (4c - 1)/(4c - 4)$, where $c = 2r/d$.

12. **Cylindrical helical spring of rectangular cross section** (Fig. 8.2.113).

$$P = bh^2S_s/(6rk) \qquad I = bh^3/12 \qquad U = Pf/2 = S_s^2V/(8Ek^2)$$
$$f = ra = Plr^2/(EI) = 12Plr^2/(Ebh^3) = 2rlS_s/(hEk)$$

For heavy closely coiled springs, $k = (3c - 1)/(3c - 3)$, where $c = 2r/h$.

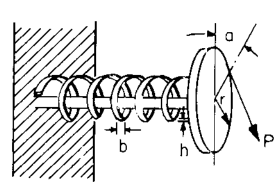


Fig. 8.2.113 Cylindrical helical spring: rectangular cross section.

Springs Subjected to Torsion

The statements made concerning coiled springs subjected to bending apply also to springs 13 and 14.

13. **Straight bar spring of circular cross section** (Fig. 8.2.114).

$$P = \pi d^3 S_v/(16r) = 0.1963 d^3 S_v/r \qquad U = Pf/2 = S_v^2 V/(4G)$$
$$f = ra = 32r^2 lP/(\pi d^4 G) = 2rlS_v/(dG)$$

14. **Straight bar spring of rectangular cross section** (Fig. 8.2.115).

$$P = 2b^2 hS_v/(9r) \qquad K = b/h$$
$$U = Pf/2 = 4S_v^2 V(K^2 + 1)/(45G) \qquad \text{max when } K = 1$$
$$f = ra = 3.6r^2 lP(b^2 + h^2)/(b^3h^3G) = 0.8rlS_v(b^2 + h^2)/(bh^2G)$$

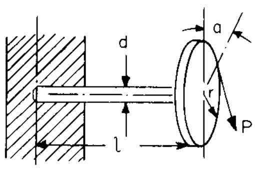


Fig. 8.2.114 Straight bar spring: circular cross section.

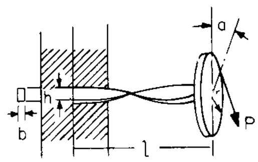


Fig. 8.2.115 Straight bar spring: rectangular cross section.

Springs Loaded Axially in Either Tension or Compression

NOTE. For springs 15 to 18, r = mean radius of coil; n = number of coils.

15. **Cylindrical helical spring of circular cross section** (Fig. 8.2.116).

$$P = \pi d^3 S_v/(16rk) = 0.1963 d^3 S_v/(rk)$$
$$U = Pf/2 = S_v^2 V/(4Gk^2)$$
$$f = 64nr^3 P/(d^4 G) = 4\pi nr^2 S_v/(dGk)$$

For heavy closely coiled springs, $k = (4c - 1)/(4c - 4) + 0.615/c$, where $c = 2r/d$.

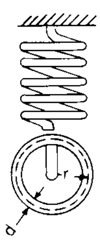


Fig. 8.2.116 Cylindrical helical spring: circular cross section.

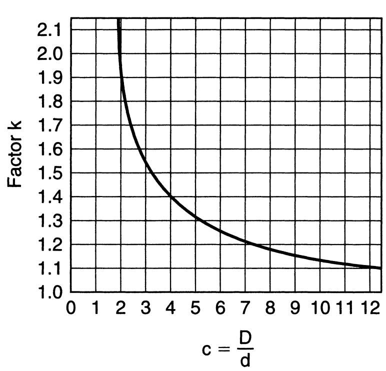


Fig. 8.2.117 Wahl correction factor.

16. **Cylindrical helical spring of rectangular cross section** (Fig. 8.2.118).

$$P = 2b^2 hS_v/(9rk) \qquad K = b/h$$
$$U = Pf/2 = 4S_v^2 V(K^2 + 1)/(45Gk^2) \qquad \text{max when } K = 1$$
$$f = 7.2\pi nr^3 P(b^2 + h^2)/(b^3h^3G) = 1.6\pi nr^2 S_v(b^2 + h^2)/(bh^2Gk)$$

For heavy closely coiled springs, $k = (4c - 1)/(4c - 4) + 0.615/c$, where $c = 2r/b$.

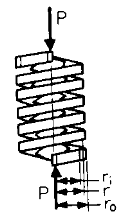


Fig. 8.2.118 Cylindrical helical spring: rectangular cross section.

17. **Conical helical spring of circular cross section** (Fig. 8.2.119*a*).

l = length of developed spring
d = diameter of wire
r = maximum mean radius of coil

$$P = \pi d^3 S_v/(16rk) = 0.1963 d^3 S_v/(rk)$$
$$U = Pf/2 = S_v^2 V/(8Gk^2)$$
$$f = 16r^2 lP/(\pi d^4 G) = 16nr^3 P/(d^4 G)$$
$$= rlS_v/(dGk) = \pi nr^2 S_v/(dGk)$$

For heavy closely coiled springs, $k = (4c - 1)/(4c - 4) + 0.615/c$, where $c = 2r/d$.

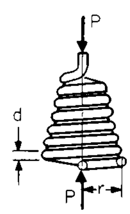


Fig. 8.2.119*a* Conical helical spring: circular cross section.

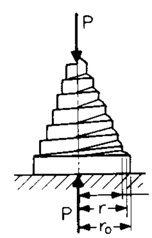


Fig. 8.2.119*b* Conical helical spring: rectangular cross section.

18. **Conical helical spring of rectangular cross section** (Fig. 8.2.119*b*).

b = small dimension of section
d = large dimension of section
r = maximum mean radius of coil

$$K = b/h\ (\le 1) \qquad P = 2b^2 hS_v/(9rk)$$
$$U = Pf/2 = 2S_v^2 V(K^2 + 1)/(45Gk^2) \qquad \text{max when } K = 1$$
$$f = 1.8r^2 lP(b^2 + h^2)/(b^3h^3G) = 1.8\pi nr^3 P(b^2 + h^2)/(b^3h^3G)$$
$$= 0.4rlS_v(b^2 + h^2)/(bh^2Gk) = 0.4\pi nr^2 S_v(b^2 + h^2)/(bh^2Gk)$$

For heavy closed coiled springs, $k = (4c - 1)/(4c - 4) + 0.615c$, where $c = 2r/r_o - r_i$.

19. **Truncated conical springs** (17 and 18). The formulas under 17 and 18 apply for truncated springs. In calculating deflection f, however, it is necessary to substitute $r_1^2 + r_2^2$ for r^2, and $\pi n(r_1 + r_2)$ for πnr, r_1 and r_2 being, respectively, the greatest and least mean radii of the coils.

NOTE. The preceding formulas for various forms of coiled springs are sufficiently accurate when the cross-section dimensions are small in comparison with the radius of the coil, and for small pitch. Springs 15 to 19 are for either tension or compression but formulas for springs 17 and 18 are good for compression only until the largest coil flattens out; then r becomes a variable, depending on the load.

Design of Helical Springs

When sizing a new spring, one must consider the spring's available working space and the loads and deflections the spring must experience. Refinements dictated by temperature, corrosion, reliability, cost, etc. may also enter design considerations. The two basic formulas of load and deflection (see item 15, Fig. 8.2.116) contain eight variables (f, P, d, S, r, k, n, G) which prevent one from being able to use a one-step solution. For instance, if f and P are known and S and G are chosen, there still remain d, r, k, and n to be found.

A variety of solution approaches are available, including: (1) slide-rule-like devices available from spring manufacturers, (2) nomographic methods (Chironis, ''Spring Design and Application,'' McGraw-Hill; Tsai, Speedy Design of Helical Compression Springs by Nomography Method, *J. of Eng. for Industry,* Feb. 1975), (3) table methods (Carlson, ''Spring Designer's Handbook,'' Marcel Dekker), (4) formula method (ibid.), and (5) computer programs and subroutines.

Design by Tables

Safe working loads and deflections of cylindrical helical springs of round steel wire in tension or compression are given in Table 8.2.64. The table is based on the formulas given for spring 15. d = diameter of steel wire, in; D = pitch diameter (center to center of wire), in; P = safe working load for given unit stress, lb; f = deflection of 1 coil for safe working load, in.

The table is based on the values of unit stress indicated, and G = 12,500,000. For any other value of unit stress, divide the tabular value by the unit stress used in the table and multiply by the unit stress to be used in the design. For any other value of G, multiply the value of f in the table by 12,500,000 and divide by the value of G chosen. For **square steel wire,** multiply values of P by 1.06, and values of f by 0.75. For **round brass wire,** take S_s = 10,000 to 20,000, and multiply values of f by 2 (Howe).

EXAMPLES OF USE OF TABLE 8.2.64. 1. Required the safe load (P) for a spring of ⅜-in round steel with a pitch diameter (D) of 3½ in. In the line headed D, under 3½, is given the value of P, or 678 lb. This is for a unit stress of 115,000 lb/in². The load P for any other unit stress may be found by dividing the 678 by 115,000 and multiplying by the unit stress to be used in the design. To determine the number of coils this spring would need to compress (say) 6 in under a load of (say) 678 lb, take the value of f under 678, or 0.938, which is the deflection of one coil under the given load. Therefore, 6/0.938 = 6.4, say 7, equals the number of coils required. The spring will therefore be 2⅝ in long when closed (7 × ⅜), counting the working coils only, and must be 8⅝ in long when unloaded. Whether there is an extra coil at one end which does not deflect will depend upon the details of the particular design. The deflection in the above example is for a unit stress of 115,000 lb/in². The rule is, divide the deflection by 115,000 and multiply by the unit stress to be used in the design.

2. A 7⁄16-in steel spring of 3½-in OD has its coils in close contact. How much can it be extended without exceeding the limit of safety? The maximum safe load for this spring is found to be 1,074 lb, and the deflection of one coil under this load is 0.810 in. This is for a unit stress of 115,000 lb/in². Therefore, 0.810 is the greatest admissible opening between any two coils. In this way, it is possible to ascertain whether or not a spring is overloaded, without knowledge of the load carried.

Design by Formula

A design formula can be constructed by equating calculated stress s_v (from load formula, Fig. 8.2.117) to an allowable working stress in torsion:

$$s_v = \sigma_{\max} = \frac{16rP}{\pi d^3}\left(\frac{4c-1}{4c-4} + \frac{0.615}{c}\right) = \frac{16rPk}{\pi d^3} = \frac{S_v}{K_{sf}}$$

where P = axial load on spring, lb; $r = D/2$ = mean radius of coil, in; D = mean diameter of coil, in (outside diameter minus wire diameter); d = wire diameter, in; $\sigma_{\max}$ = torsional stress, lb/in²; K_{sf} = safety factor and $c = D/d$. Note that the expression in parentheses $[(4c-1)(4c-4) + 0.615/c]$ is the **Wahl correction factor** k, which accounts for the added stresses in the coils due to curvature and direct shear. See Fig. 8.2.117. Values of S_v, yield point in shear from standard tests, are strongly dependent on d, hence the availability of S_v in the literature is limited. However, an empirical relationship between S_{uT} and d is available (see Shigley, ''Mechanical Engineering Design,'' McGraw-Hill, 4th ed., p. 452). Using also the approximate relations $S_y = 0.75S_{ut}$ and $S_v = 0.577S_y$ results in the following relationship:

$$S_v = \frac{0.43A}{d^m}$$

where A and m are constants (see Table 8.2.65).

Substituting S_v and rearranging yield the following useful formula:

$$d^{3-m} = \frac{K_{sf}16rPk}{\pi 0.43A}$$

EXAMPLE. A slow-speed follower is kept in contact with its cam by means of a helical compression spring, in which the minimum spring force desired is 20 lb to assure firm contact, while maximum spring force is not to exceed 60 lb to prevent excessive surface wear on the cam. The follower rod is ¼ inch in diameter, and the rod enclosure where the spring is located is ⅞ inch in diameter. Maximum displacement is 1.5 in.

Choose $r = 0.5 \times 0.75 = 0.375$. From Table 8.2.65 choose $m = 0.167$ and A = 169,000. Choose $K_{sf} = 2$. Assume $k = 1$ to start.

$$d^{3-0.167} = \frac{(2)(0.375)(60)(16)}{(\pi)(0.43)(169{,}000)} = 0.00315$$

$$d = (0.00315)^{0.35298} = 0.131 \text{ in}$$

$$\text{Spring constant} = \frac{\Delta P}{\Delta f} = \frac{40}{1.5} = 26.667$$

and from the deflection formula, the number of active turns

$$n = \frac{(0.131)^4(11{,}500{,}000)}{(26.667)(64)(0.375)^3} = 37.6$$

For squared and ground ends add two dead coils, so that

$$\begin{aligned} n_{\text{total}} &= 40 \\ H &= \text{solid height} = (40)(0.131) = 5.24 \text{ in} \\ f_0 &= \text{displacement from zero to maximum load} \\ &= 60/26.667 = 2.249 \\ L_0 &= \text{approximate free length} = 5.24 + 2.249 = 7.49 \text{ in} \end{aligned}$$

NOTE. Some clearance should be added between coils so that at maximum load the spring is not closed to its solid height. Also, the nearest commercial stock size should be selected, and recalculations made on this stock size for S_v, remembering to include k at this juncture. If recalculated S_v is satisfactory as compared to published values, the design is retained, otherwise enough iterations are performed to arrive at a satisfactory result. Figure 8.2.120 shows a plot of S_{uT} versus d. To convert S_{uT} to S_v, multiply S_{uT} by 0.43.

$$c = 2(0.375/0.131) = 5.73$$

$$k = \frac{4(5.73)-1}{4(5.73)-4} + \frac{0.615}{5.73} = 1.257$$

$$S_v = \frac{(K_{sf})(16)(0.375)(1.257)(60)}{\pi(0.131)^3} = (K_{sf})(64{,}072)$$

Now S_{uT} = 235,000 lb/in² (from Fig. 8.2.120) and S_v (tabulated) = (0.43)(235,000) = 101,696 lb/in² so that K_{sf} = 101,696/64,072 = 1.59, a satisfactory value.

NOTE. The original choice of a generous $K_{sf} = 2$ was made to hedge against the large statistical variations implied in the empirical formula $S_v = 0.43A/d^m$.

The basis for design of springs in parallel or in series is shown in Fig. 8.2.121.

Belleville Springs Often called dished, or conical spring, washers, Belleville springs occupy a very small space. They are stressed in a very complex manner, and provide unusual spring rate curves (Fig. 8.2.122*a*). These springs are nonlinear, but for some proportions, they behave with approximately linear characteristics in a limited range. Likewise, for some proportions they can be used through a spectrum of spring rates, from positive to flat and then through a negative region. The snap-through action, shown at point A in Fig. 8.2.122*b*, can be useful in particular applications requiring reversal of spring rates. These

Table 8.2.64 Safe Working Loads *P* and Deflections *f* of Cylindrical Helical Steel Springs of Circular Cross Section

(For closely coiled springs, divide given load and deflection values by the curvature factor *k*.)

Allowable unit stress, lb/in²	Diam, in		Pitch diameter *D*, in																				
		D	5/32	3/16	1/4	5/16	3/8	7/16	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/4	2 1/2
150,000	0.035	*P*	16.2	13.4	10.0	8.10	6.66	5.75	4.96	4.05	3.39												
		f	.026	.037	.067	.105	.149	.200	.276	.420	.608												
	0.041	*P*	26.2	21.6	16.2	13.0	10.8	9.27	8.10	6.52	5.35	4.57											
		f	.023	.032	.057	.089	.128	.175	.229	.362	.512	.697											
	0.047	*P*	39.1	32.6	24.5	19.6	16.4	13.9	12.3	9.80	8.10	6.92	6.14										
		f	.019	.028	.049	.078	.112	.153	.200	.311	.449	.610	.800										
	0.054	*P*	59.4	49.6	37.2	29.7	24.6	21.2	18.5	14.7	12.4	10.5	9.25	8.23									
		f	.016	.024	.043	.067	.098	.133	.174	.273	.390	.532	.695	.878									
	0.062	*P*		74.9	56.1	44.9	37.3	32.0	28.0	22.4	18.6	16.1	13.9	12.5	11.2								
		f		.021	.037	.058	.084	.115	.151	.235	.340	.460	.605	.760	.947								
	0.063	*P*		78.2	58.7	46.9	39.2	33.9	29.4	23.5	19.6	16.8	14.7	13.2	11.9	10.7							
		f		.020	.037	.057	.083	.113	.148	.233	.335	.445	.591	.748	.930	1.12							
	0.072	*P*		117.	80.7	70.0	58.7	50.2	43.6	35.2	29.0	25.0	21.9	19.5	17.5	16.0							
		f		.018	.032	.050	.077	.100	.130	.203	.294	.405	.521	.652	.802	.986							
	0.080	*P*			121	96.6	80.5	69.1	60.4	48.2	48.2	34.6	30.1	26.7	24.2	22.1	20.2						
		f			.029	.045	.065	.090	.117	.183	.262	.359	.470	.593	.735	.886	1.105						
140,000	0.092	*P*			171	136	113	97.6	85.5	68.9	57.3	48.8	42.6	37.8	34.5	31.3	28.6						
		f			.023	.037	.053	.072	.098	.148	.214	.291	.388	.481	.596	.720	.854						
	0.093	*P*			178	142	118	99.5	89.0	71.2	59.1	50.9	44.3	39.6	35.7	32.3	29.6	27.3					
		f			.023	.036	.052	.071	.093	.146	.211	.286	.376	.473	.585	.707	.841	.986					
	0.105	*P*				204	170	147	127	102	85.4	73.0	63.4	56.6	51.1	46.3	42.6	38.9					
		f				.032	.047	.064	.083	.122	.188	.256	.336	.425	.512	.632	.755	.880					
	0.120	*P*				303	253	217	190	152	126	108	95.2	84.2	76.2	69.2	63.5	58.5	54.3				
		f				.028	.041	.055	.073	.114	.174	.223	.296	.368	.449	.551	.657	.768	.893				
	0.125	*P*					286	245	214	171	143	121	107	95.5	85.2	78.0	71.5	65.8	60.8	57.2			
		f					.039	.053	.069	.109	.169	.213	.278	.353	.437	.528	.626	.731	.855	.981			
	0.135	*P*					359	309	270	217	171	154	135	120	108	98.7	90.2	82.7	72.2	71.8	67.5		
		f					.036	.049	.064	.106	.145	.198	.260	.327	.399	.486	.581	.680	.791	.908	1.04		
	0.148	*P*						408	356	285	237	207	178	158	142	130	118	109	102	95.0	89.0		
		f						.045	.059	.092	.132	.180	.236	.293	.370	.448	.530	.620	.723	.828	.945		

Table 8.2.64 Safe Working Loads *P* and Deflections *f* of Cylindrical Helical Steel Springs of Circular Cross Section (*Continued*)

Allowable unit stress, lb/in²	Diam, in		Pitch diameter *D*, in																					
		D	7⁄16	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2	2¼	2½	2¾	3	3½	4	4½	5
140,000	0.156	*P*	480	418	330	270	234	208	185	167	152	139	128	119	111	104	92.7							
		f	0.42	.056	.087	.125	.171	.223	.282	.350	.422	.509	.588	.685	.785	.896	1.12							
	0.162	*P*		468	376	311	276	234	207	187	170	156	143	134	125	117	103							
		f		.054	.085	.121	.165	.216	.273	.338	.409	.488	.566	.663	.757	.863	1.09							
	0.177	*P*		608	487	406	347	305	270	243	223	205	187	174	163	152	135	122						
		f		.049	.077	.115	.151	.198	.251	.311	.375	.447	.522	.606	.695	.793	1.00	1.24						
125,000	0.187	*P*		642	522	426	367	320	284	256	233	213	197	183	170	160	142	128						
		f		.041	.065	.093	.127	.166	.210	.260	.314	.373	.487	.510	.584	.665	.832	1.04						
	0.192	*P*		696	556	465	396	348	309	278	254	233	214	199	186	174	154	139	126					
		f		.040	.063	.091	.124	.160	.205	.252	.308	.366	.428	.499	.571	.652	.825	1.02	1.23					
	0.207	*P*			694	579	495	432	385	346	315	288	266	247	232	216	192	173	158	144				
		f			.059	.085	.115	.151	.191	.236	.286	.342	.396	.462	.570	.607	.757	.943	1.11	1.36				
	0.218	*P*			812	678	580	509	452	408	360	339	310	291	270	255	225	204	185	169				
		f			.055	.080	.109	.142	.180	.223	.269	.321	.374	.437	.488	.570	.710	.891	1.08	1.28				
	0.225	*P*			895	746	640	560	498	447	407	372	345	320	299	280	248	224	203	187				
		f			.054	.078	.106	.138	.175	.213	.262	.312	.372	.425	.486	.565	.691	.866	1.05	1.24				
	0.244	*P*			1120	950	811	711	632	570	517	475	438	406	381	356	316	284	259	237				
		f			.049	.071	.098	.138	.161	.200	.240	.287	.336	.391	.449	.537	.646	.800	.965	1.14				
	0.250	*P*				1027	880	760	685	617	560	513	476	440	410	385	342	308	281	266	222			
		f				.070	.095	.131	.157	.191	.236	.281	.328	.385	.439	.524	.624	.780	.946	1.12	1.53			

	0.263	*P*	1195	1125	895	795	717	652	598	551	501	478	448	400	359	326	298	256			
		f	.066	.089	.118	.149	.183	.224	.266	.312	.363	.416	.475	.592	.740	.896	1.06	1.44			
	0.281	*P*	1450	1240	1087	969	863	794	724	665	620	580	543	482	437	395	362	310			
		f	.062	.085	.111	.140	.172	.209	.250	.292	.340	.390	.443	.562	.692	.840	1.02	1.36			
	0.283	*P*		1264	1110	985	886	805	740	682	634	592	564	492	439	402	370	317			
		f		.084	.111	.139	.169	.207	.246	.289	.338	.386	.440	.559	.690	.883	.990	1.35			
115,000	0.312	*P*		1575	1376	1220	1100	1000	915	845	775	733	687	610	550	500	460	392	343		
		f		.070	.092	.116	.144	.174	.207	.242	.283	.322	.368	.467	.577	.697	.829	1.12	1.47		
	0.331	*P*			1636	1455	1316	1187	1090	1000	932	870	818	725	653	594	545	468	410		
		f			.088	.109	.135	.163	.194	.227	.264	.343	.346	.437	.541	.654	.770	1.05	1.30		
	0.341	*P*			1820	1620	1452	1325	1214	1120	1040	970	910	808	728	661	608	520	454		
		f			.082	.105	.127	.156	.186	.218	.256	.293	.330	.413	.522	.625	.745	1.02	1.32		
	0.362	*P*			2140	1910	1714	1560	1430	1318	1220	1147	1070	950	858	778	714	612	535		
		f			.079	.100	.123	.149	.177	.207	.243	.273	.317	.400	.495	.598	.713	.965	.126		
	0.375	*P*				2110	1940	1780	1580	1458	1354	1265	1185	1058	950	860	790	678	592	528	
		f				.079	.117	.144	.172	.201	.234	.268	.308	.382	.478	.579	.688	.938	1.22	1.54	
	0.393	*P*				2430	2180	1984	1820	1680	1560	1458	1365	1212	1092	990	910	780	682	670	
		f				.092	.114	.137	.164	.195	.223	.256	.292	.369	.457	.550	.657	.890	1.16	1.47	
	0.406	*P*					2400	2170	2000	1840	1710	1600	1500	1330	1200	1090	1000	855	750	666	
		f					.108	.134	.159	.168	.217	.248	.284	.353	.444	.525	.640	.867	1.13	1.43	
	0.430	*P*					2875	2610	2400	2210	2050	1918	1798	1598	1440	1308	1200	1028	900	800	
		f					.104	.126	.150	.175	.204	.234	.267	.338	.418	.503	.600	.815	1.06	1.35	
	0.437	*P*					3000	2730	2500	2310	2140	2000	1800	1665	1500	1365	1250	1074	940	835	750
		f					.100	.124	.148	.173	.201	.231	.264	.327	.412	.490	.593	.810	1.05	1.33	1.64

Table 8.2.64 Safe Working Loads *P* and Deflections *f* of Cylindrical Helical Steel Springs of Circular Cross Section (*Continued*)

Allowable unit stress, lb/in²	Diam, in	Pitch diameter *D*, in																	
		D	1¼	1⅜	1½	1⅝	1¾	1⅞	2	2¼	2½	2¾	3	3½	4	4½	5	5½	6
110,000	0.460	*P*		3065	2800	2580	2400	2230	2100	1865	1680	1530	1400	1200	1058	952	840		
		f		.112	.134	.157	.183	.209	.239	.303	.374	.447	.536	.729	.956	1.21	1.49		
	0.468	*P*		3265	2940	2725	2530	2375	2210	1970	1770	1610	1472	1265	1110	935	885		
		f		.111	.132	.154	.182	.206	.235	.295	.368	.444	.530	.720	.943	1.19	1.47		
	0.490	*P*		3675	3270	3115	2890	2710	2535	2245	2025	1840	1690	1445	1268	1125	1015	920	
		f		.106	.126	.148	.172	.196	.225	.284	.351	.424	.506	.688	.900	1.13	1.40	1.70	
	0.500	*P*			3610	3320	3090	2890	2710	2410	2160	1970	1810	1550	1352	1205	1082	985	
		f			.123	.144	.168	.192	.220	.274	.347	.415	.495	.672	.880	1.11	1.37	1.65	
	0.562	*P*				4700	4390	4090	3830	3420	3080	2790	2565	2190	1913	1710	1535	1395	1280
		f				.128	.149	.175	.195	.248	.306	.372	.440	.596	.782	.990	1.22	1.47	1.75
	0.625	*P*					6100	5600	5260	4660	4210	3825	3505	3000	2630	2340	2110	1913	1750
		f					.134	.154	.176	.218	.275	.328	.397	.538	.705	.875	1.05	1.33	1.58
100,000	0.687	*P*							6325	5660	5090	4630	4250	3625	3195	2825	2560	2330	2125
		f							.145	.183	.228	.274	.3278	.443	.580	.733	.908	1.00	1.30
	0.750	*P*								7400	6640	6030	5540	4745	4150	3690	3325	3025	2770
		f								.178	.218	.252	.299	.402	.532	.671	.832	1.00	1.19
	0.812	*P*									8420	7660	7000	6000	5260	4675	4200	3825	3500
		f									.192	.232	.276	.376	.490	.620	.766	.880	1.10
	0.875	*P*									10830	9550	8700	7500	6560	5740	5250	4770	4730
		f									.179	.218	.257	.348	.456	.577	.712	.860	1.02
90,000	0.937	*P*										10600	9700	8400	7160	6470	5810	5290	4850
		f										.179	.217	.290	.383	.480	.591	.715	.855
	1.000	*P*											11780	10100	8800	7850	7050	6330	5870
		f											.206	.276	.360	.454	.561	.680	.803
	1,125	*P*												14400	12600	11230	10100	9200	8400
		f												.244	.320	.405	.496	.600	.718
	1.250	*P*												24700	18200	15300	13250	12540	11500
		f												.260	.287	.364	.442	.545	.648
80,000	1.375	*P*													20400	18100	16150	14850	13600
		f													.280	.294	.364	.440	.522

Table 8.2.65 Constants for Use in $S_v = 0.43A/d^m$

Material	Size range, in	Size range, mm	Exponent m	Constant A kips	Constant A MPa
Music wire*	0.004–0.250	0.10–6.55	0.146	196	2,170
Oil-tempered wire†	0.020–0.500	0.50–12	0.186	149	1,880
Hard-drawn wire‡	0.028–0.500	0.70–12	0.192	136	1,750
Chrome vanadium§	0.032–0.437	0.80–12	0.167	169	2,000
Chrome silicon¶	0.063–0.375	1.6–10	0.112	202	2,000

* Surface is smooth, free from defects, and has a bright, lustrous finish.
† Has a slight heat-treating scale which must be removed before plating.
‡ Surface is smooth and bright, with no visible marks.
§ Aircraft-quality tempered wire; can also be obtained annealed.
¶ Tempered to Rockwell C49 but may also be obtained untempered.
SOURCE: Adapted from ''Mechanical Engineering Design,'' Shigley, McGraw-Hill, 1983 by permission.

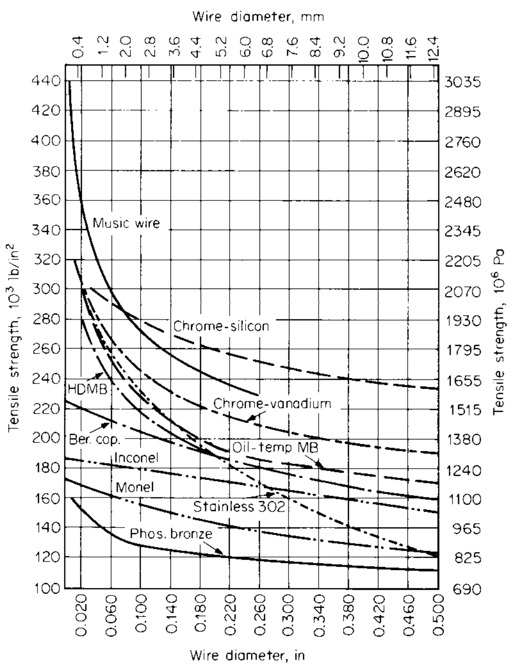
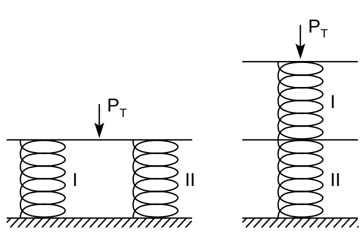


Fig. 8.2.120 Minimum tensile strength for the most popular spring materials, spring-quality wire. *(Reproduced from Carlson, ''Spring Designer's Handbook,'' Marcel Dekker, by permission.)*



Parallel

$P_T = P_I + P_{II}$
$\Delta_T = \Delta_I = \Delta_{II}$
$K_T = K_I + K_{II}$

Series

$P_T = P_I = P_{II}$
$\Delta_T = \Delta_I + \Delta_{II}$
$K_T = \dfrac{K_I K_{II}}{K_I + K_{II}}$

where P = load, lb
Δ = deflection, in.
K = spring rate, lb/in.

Fig. 8.2.121 Springs in parallel and in series.

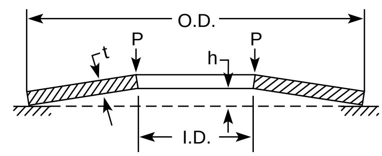


Fig. 8.2.122*a* Sectional view of Belleville spring.

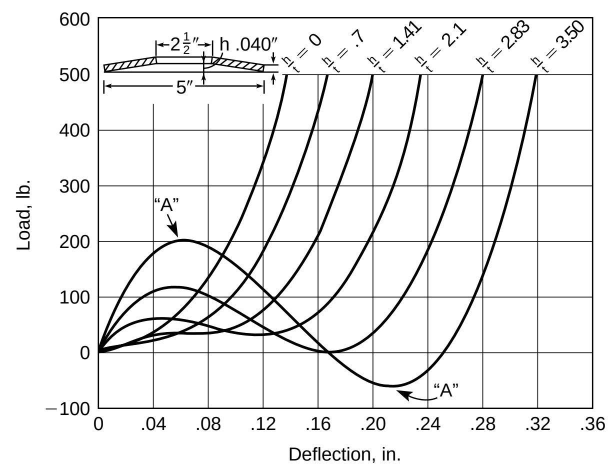


Fig. 8.2.122*b* Load deflection curves for a family of Belleville springs. *(Associated Spring Corp.)*

springs are used for very high and special spring rates. They are extremely sensitive to slight variations in their geometry. A wide range is available commercially.

WIRE ROPE

When power source and load are located at extreme distances from one another, or loads are very large, the use of wire rope is suggested. Design and use decisions pertaining to wire ropes rest with the user, but manufacturers generally will help users toward appropriate choices. The following material, based on the Committee of Wire Rope Producers, ''Wire Rope User's Manual,'' 2d ed., 1981, may be used as an initial guide in selecting a rope.

Wire rope is composed of (1) wires to form a strand, (2) strands wound helically around a core, and (3) a core. Classification of wire ropes is made by giving the number of strands, number of minor strands in a major strand (if any), and nominal number of wires per strand. For example 6 × 7 rope means 6 strands with a nominal 7 wires per strand (in this case no minor strands, hence no middle number). A nominal value simply represents a range. A nominal value of 7 can mean anywhere from 3 to 14, of which no more than 9 are outside wires. A full

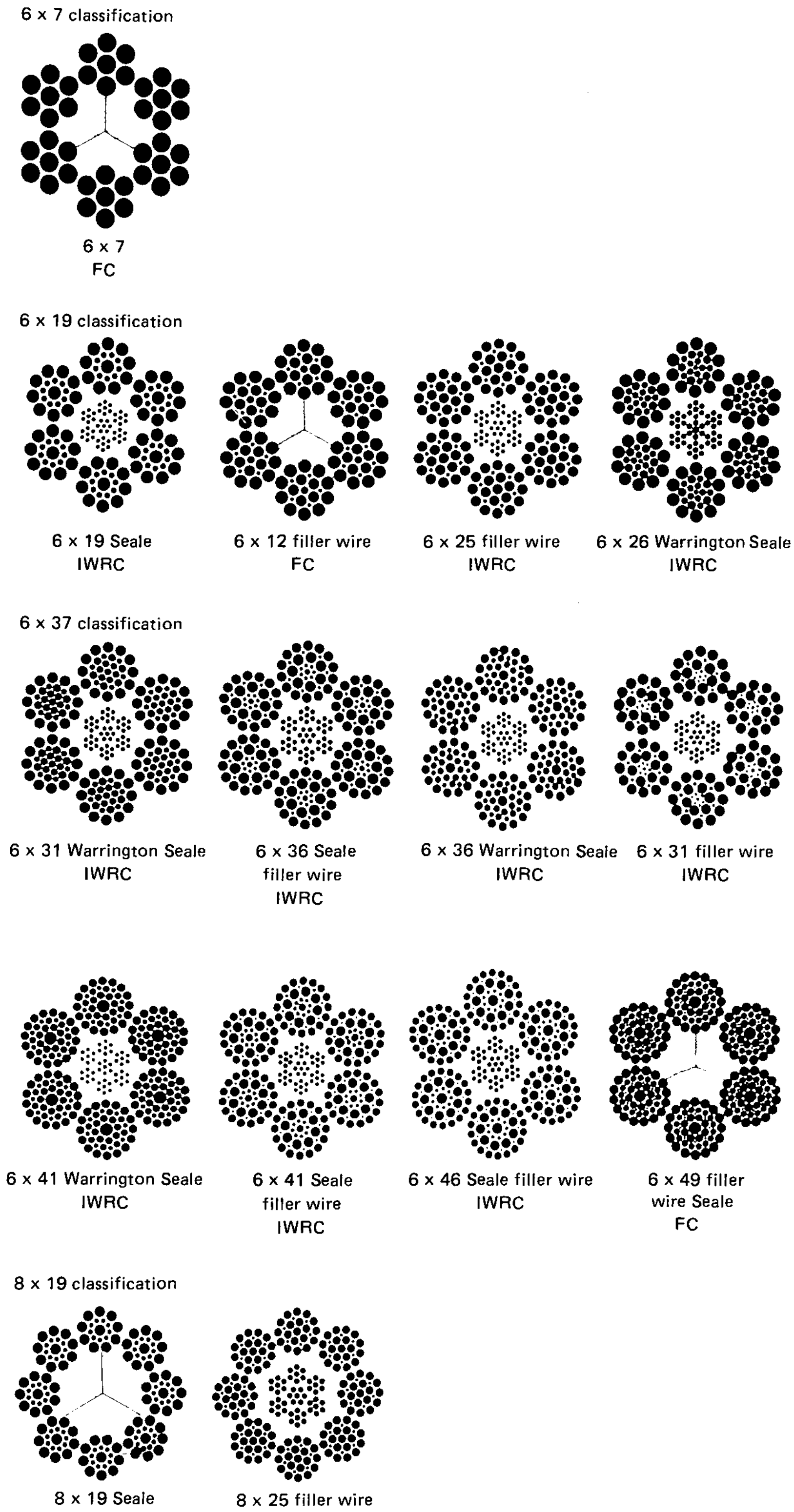


Fig. 8.2.123 Cross sections of some commonly used wire rope construction. *(Reproduced from ''Wire Rope User's Manual,'' AISI, by permission.)*

rope description will also include length, size (diameter), whether wire is preformed or not prior to winding, direction of lay (right or left, indicating the direction in which strands are laid around the core), grade of rope (which reflects wire strength), and core. The most widely used classifications are: 6 × 7, 6 × 19, 6 × 37, 6 × 61, 6 × 91, 6 × 127, 8 × 19, 18 × 7, 19 × 7. Some special constructions are: 3 × 7 (guardrail rope); 3 × 19 (slusher), 6 × 12 (running rope); 6 × 24 and 6 × 30 (hawsers); 6 × 42 and 6 × 6 × 7 (tiller rope); 6 × 3 × 19 (spring lay); 5 × 19 and 6 × 19 (marlin clad); 6 × 25B, 6 × 27H, and 6 × 30G (flattened strand). The diameter of a rope is the circle which just contains the rope. The right-regular lay (in which the wire is twisted in one direction to form the strands and the strands are twisted in the opposite direction to form the rope) is most common. Regular-lay ropes do not kink or untwist and handle easily. Lang-lay ropes (in which wires and strands are twisted in the same direction) are more resistant to abrasive wear and fatigue failure.

Cross sections of some commonly used wire rope are shown in Fig. 8.2.123. Figure 8.2.124 shows rotation-resistant ropes, and Fig. 8.2.125 shows some special-purpose constructions.

The core provides support for the strands under normal bending and loading. Core materials include fibers (hard vegetable or synthetic) or steel (either a strand or an independent wire rope). Most common core designations are: fiber core (FC), independent wire-rope core (IWRC), and wire-strand core (WSC). Lubricated fiber cores can provide lubrication to the wire, but add no real strength and cannot be used in high temperature environments. Wire-strand or wire-rope cores add from 7 to 10 percent to strength, but under nonstationary usage tend to wear from interface friction with the outside strands. Great flexibility can be achieved when wire rope is used as strands. Such construction is very pliable and friction resistant. Some manufacturers will provide plastic coatings (nylon, Teflon, vinyl, etc.) upon request. Such coatings help provide resistance to abrasion, corrosion, and loss of lubricant. *Crushing* refers to rope damage caused by excessive pressures against drum or sheave, improper groove size, and multiple layers on drum or sheave. Consult wire rope manufacturers in doubtful situations.

Wire-rope materials and their strengths are reflected as grades. These are: traction steel (TS), mild plow steel (MPS), plow steel (PS), improved plow steel (IPS), and extra improved plow (EIP). The plow steel strength curve forms the basis for calculating the strength of all steel rope wires. American manufacturers use color coding on their ropes to identify particular grades.

The grades most commonly available and tabulated are IPS and EIP. Two specialized categories, where selection requires extraordinary attention, are elevator and rotation-resistant ropes.

Elevator rope can be obtained in four principal grades: iron, traction steel, high-strength steel, and extra-high-strength steel.

Bronze rope has limited use; iron rope is used mostly for older existing equipment.

Selection of Wire Rope

Appraisal of the following is the key to choosing the rope best suited to the job: resistance to breaking, resistance to bending fatigue, resistance to vibrational fatigue, resistance to abrasion, resistance to crushing, and reserve strength. Along with these must be an appropriate choice of safety factor, which in turn requires careful consideration of all loads, acceleration-deceleration, shocks, rope speed, rope attachments, sheave

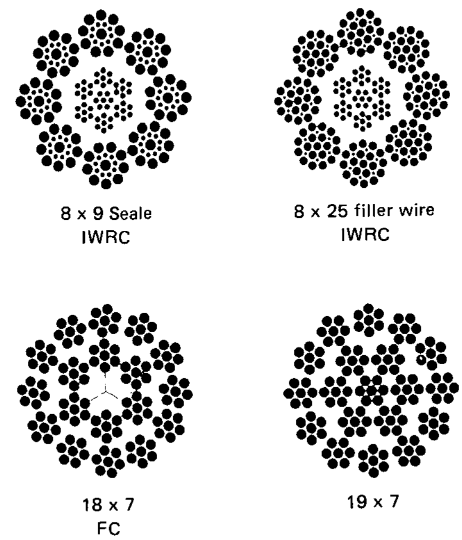


Fig. 8.2.124 Cross section of some rotation-resistant wire ropes. *(Reproduced from ''Wire Rope User's Manual,'' AISI, by permission.)*

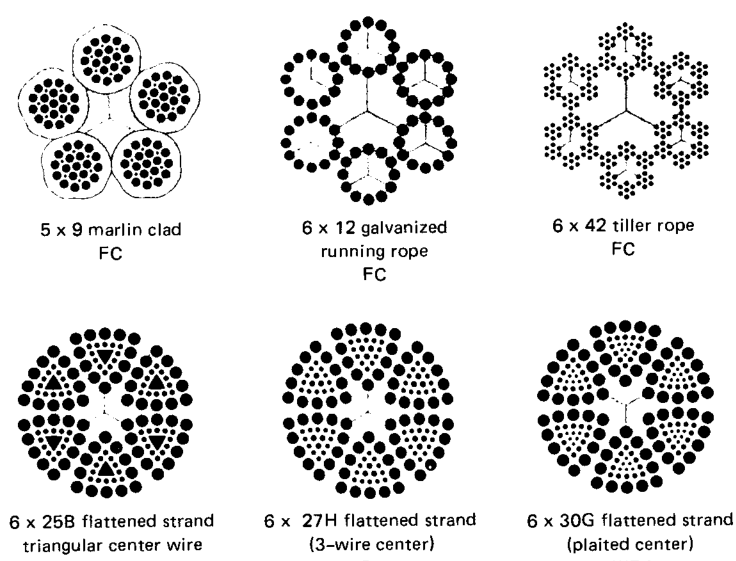


Fig. 8.2.125 Some special constructions. *(Reproduced from ''Wire Rope User's Manual,'' AISI, by permission.)*

Table 8.2.66 Selected Values of Nominal Strengths of Wire Rope

			Fiber core				IWRC					
	Nominal diameter		Approximate mass		Nominal strength IPS		Approximate mass		Nominal strength			
									IPS		EIP	
Classification	in	mm	lb/ft	kg/m	tons	t	lb/ft	kg/m	tons	t	tons	t
6 × 7 Bright	¼	6.4	0.09	0.14	2.64	2.4	0.10	0.15	2.84	2.58		
(uncoated)	⅜	9.5	0.21	0.31	5.86	5.32	0.23	0.34	6.30	5.72		
	½	13	0.38	0.57	10.3	9.35	0.42	0.63	11.1	10.1		
	⅝	16	0.59	0.88	15.9	14.4	0.65	0.97	17.1	15.5		
	⅞	22	1.15	1.71	30.7	27.9	1.27	1.89	33.0	29.9		
	1⅛	29	1.90	2.83	49.8	45.2	2.09	3.11	53.5	48.5		
	1⅜	35	2.82	4.23	73.1	66.3	3.12	4.64	78.6	71.3		
6 × 19 Bright	¼	6.4	0.11	0.16	2.74	2.49	0.12	0.17	2.94	2.67	3.40	3.08
(uncoated)	⅜	9.5	0.24	0.35	6.10	5.53	0.26	0.39	6.56	5.95	7.55	6.85
	½	13	0.42	0.63	10.7	9.71	0.46	0.68	11.5	10.4	13.3	12.1
	⅝	16	0.66	0.98	16.7	15.1	0.72	1.07	17.7	16.2	20.6	18.7
	⅞	22	1.29	1.92	32.2	29.2	1.42	2.11	34.6	31.4	39.8	36.1
	1⅛	29	2.13	3.17	52.6	47.7	2.34	3.48	56.5	51.3	65.0	59.0
	1⅜	35	3.18	4.73	77.7	70.5	3.5	5.21	83.5	75.7	96.0	87.1
	1⅝	42	4.44	6.61	107	97.1	4.88	7.26	115	104	132	120
	1⅞	48	5.91	8.8	141	128	6.5	9.67	152	138	174	158
	2⅛	54	7.59	11.3	179	162	8.35	12.4	192	174	221	200
	2⅜	60	9.48	14.1	222	201	10.4	15.5	239	217	274	249
	2⅝	67	11.6	17.3	268	243	12.8	19.0	288	261	331	300
6 × 37 Bright	¼	6.4	0.11	0.16	2.74	2.49	0.12	0.17	2.94	2.67	3.4	3.08
(uncoated)	⅜	9.5	0.24	0.35	6.10	5.53	0.26	0.39	6.56	5.95	7.55	6.85
	½	13	0.42	0.63	10.7	9.71	0.46	0.68	11.5	10.4	13.3	12.1
	⅝	16	0.66	0.98	16.7	15.1	0.72	1.07	17.9	16.2	20.6	18.7
	⅞	22	1.29	1.92	32.2	29.2	1.42	2.11	34.6	31.4	39.5	36.1
	1⅛	29	2.13	3.17	52.6	47.7	2.34	3.48	56.5	51.3	65.0	59.0
	1⅜	35	3.18	4.73	77.7	70.5	3.50	5.21	83.5	75.7	96.0	87.1
	1⅝	42	4.44	6.61	107	97.1	4.88	7.26	115	104	132	120
	1⅞	48	5.91	8.8	141	128	6.5	9.67	152	138	174	158
	2⅛	54	7.59	11.3	179	162	8.35	12.4	192	174	221	200
	2⅜	60	9.48	14.1	222	201	10.4	15.5	239	217	274	249
	2⅞	67	11.6	17.3	268	243	12.8	19.0	288	261	331	300
	3⅛	74	13.9	20.7	317	287	15.3	22.8	341	309	392	356
		80	16.4	24.4	371	336	18.0	26.8	399	362	458	415
6 × 61 Bright	1⅛	29	2.13	3.17	50.1	45.4	2.34	3.48	53.9	48.9	61.9	56.2
(uncoated)	1⅝	42	4.44	6.61	103	93.4	4.88	7.62	111	101	127	115
	2	52	6.77	10.1	154	140	7.39	11.0	165	150	190	172
	2⅝	67	11.6	17.3	260	236	12.8	18.3	279	253	321	291
	3	77	15.1	22.5	335	304	16.6	24.7	360	327	414	376
	4	103	26.9	40.0	577	523	29.6	44.1	620	562	713	647
	5	128	42.0	62.5	872	791	46.2	68.8	937	850	1,078	978
6 × 91 Bright	2	51	6.77	10.1	146	132	7.39	11.0	157	142	181	164
(uncoated)	3	77	15.1	22.5	318	288	16.6	24.7	342	310	393	357
	4						29.6	44.1	589	534	677	614
	5						46.2	68.7	891	808	1,024	929
	6						65.0	96.7	1,240	1,125	1,426	1,294
6 × 25B	½	13	0.45	0.67	11.8	10.8	0.47	0.70	12.6	11.4	14	12.7
6 × 27H	9/16	14.5	0.57	0.85	14.9	13.5	0.60	0.89	16.0	14.5	17.6	16
6 × 30G	¾	19	1.01	1.50	26.2	23.8	1.06	1.58	28.1	25.5	31	28.1
Flattened strand bright	1	26	1.80	2.68	46.0	41.7	1.89	2.83	49.4	44.8	54.4	49.4
(uncoated)	1¼	32	2.81	4.18	71.0	64.4	2.95	4.39	76.3	60.2	84	76.2
	1½	38	4.05	6.03	101	91.6	4.25	6.32	108	98	119	108
	1¾	45	5.51	8.20	136	123	5.78	8.60	146	132	161	146
	2	52	7.20	10.70	176	160	7.56	11.3	189	171	207	188
8 × 19 Bright	¼	6.4	0.10	0.15	2.35	2.13	0.47	0.70	10.1	9.16	11.6	10.5
(uncoated)	⅜	9.5	0.22	0.33	5.24	4.75	0.73	1.09	15.7	14.2	18.1	16.4
	½	13	0.39	0.58	9.23	8.37	1.44	2.14	30.5	27.7	35.0	31.8
	⅝	16	0.61	0.91	14.3	13.0	2.39	3.56	49.8	45.2	57.3	51.7
	1	26	1.57	2.34	36.0	32.7	4.24	6.31	87.3	79.2	100.0	90.7
	1½	38	3.53	5.25	79.4	72.0						
18 × 7	½	13	0.43	0.64	9.85	8.94	0.45	0.67	9.85	8.94	10.8	9.8
Rotation resistant, bright	¾	19	0.97	1.44	21.8	19.8	1.02	1.52	21.8	19.8	24.0	21.8
(uncoated)	1	26	1.73	2.57	38.3	34.7	1.82	2.71	38.3	34.7	42.2	38.3
	1¼	32	2.70	4.02	59.2	53.7	2.84	4.23	59.2	53.7	65.1	59.1
	1½	38	3.89	5.79	84.4	76.6	4.08	6.07	84.4	76.6	92.8	84.2

SOURCE: ''Wire Rope User's Manual,'' AISI, adapted by permission.

arrangements as well as their number and size, corrosive and/or abrasive environment, length of rope, etc. An approximate selection formula can be written as:

$$\mathrm{DSL} = \frac{(\mathrm{NS})K_b}{K_{sf}}$$

where DSL (demanded static load) = known or dead load **plus** additional loads caused by sudden starts or stops, shocks, bearing friction, etc., tons; NS (nominal strength) = published test strengths, tons (see Table 8.2.66); K_b = a factor to account for the reduction in nominal strength due to bending when a rope passes over a curved surface such as a stationary sheave or pin (see Fig. 8.2.126); K_{sf} = safety factor. (For average operation use $K_{sf} = 5$. If there is danger to human life or other critical situations, use $8 \le K_{sf} \le 12$. For instance, for elevators moving at 50 ft/min, $K_{sf} = 8$, while for those moving at 1,500 ft/min, $K_{sf} = 12$.)

Having made a tentative selection of a rope based on the demanded static load, one considers next the wear life of the rope. A loaded rope

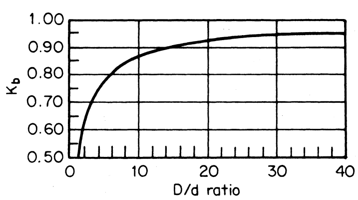


Fig. 8.2.126 Values of K_{bend} vs. D/d ratios (D = sheave diameter, d = rope diameter), based on standard test data for 6 × 9 and 6 × 17 class ropes. *(Compiled from ''Wire Rope User's Manual,'' AISI, by permission.)*

bent over a sheave stretches elastically and so rubs against the sheave, causing wear of both members. Drum or sheave size is of paramount importance at this point.

Sizing of Drums or Sheaves

Diameters of drums or sheaves in wire rope applications are controlled by two main considerations: (1) the radial pressure between rope and groove and (2) degree of curvature imposed on the rope by the drum or sheave size.

Radial pressures can be calculated from $p = 2T/(Dd)$, where p = unit radial pressure, lb/in^2; T = rope load, lb; D = tread diameter of drum or sheave, in; d = nominal diameter of rope, in. Table 8.2.67 lists suggested allowable radial bearing pressures of ropes on various sheave materials.

All wire ropes operating over drums or sheaves are subjected to cyclical stresses, causing shortened rope life because of fatigue. Fatigue resistance or relative service life is a function of the ratio D/d. Adverse effects also arise out of relative motion between strands during passage around the drum or sheave. Additional adverse effects can be traced to poor match between rope and groove size, and to lack of rope lubrication. Table 8.2.68 lists suggested and minimum sheave and drum ratios for various rope construction. Table 8.2.69 lists relative bending life factors; Figure 8.2.127 shows a plot of relative rope service life versus D/d. Table 8.2.70 lists minimum drum (sheave) groove dimensions. Periodic groove inspection is recommended, and worn or corrugated grooves should be remachined or the drum replaced, depending on severity of damage.

Seizing and Cutting Wire Rope Before a wire rope is cut, seizings (bindings) must be applied on either side of the cut to prevent rope distortion and flattening or loosened strands. Normally, for preformed ropes, one seizing on each side of the cut is sufficient, but for ropes that

Table 8.2.68 Sheave and Drum Ratios

Construction*	Suggested	Minimum
6 × 7	72	42
19 × 7 or 18 × 7 Rotation-resistant	51	34
6 × 19 S	51	34
6 × 25 B flattened strand	45	30
6 × 27 H flattened strand	45	30
6 × 30 G flattened strand	45	30
6 × 21 FW	45	30
6 × 26 WS	45	30
6 × 25 FW	39	26
6 × 31 WS	39	26
6 × 37 SFW	39	26
6 × 36 WS	35	23
6 × 43 FWS	35	23
6 × 41 WS	32	21
6 × 41 SFW	32	21
6 × 49 SWS	32	21
6 × 46 SFW	28	18
6 × 46 WS	28	18
8 × 19 S	41	27
8 × 25 FW	32	21
6 × 42 Tiller	21	14

* WS—Warrington Seale; FWS—Filler Wire Seale; SFW—Seale Filler Wire; SWS—Seale Warrington Seale; S—Seale; FW—Filler Wire.

† D = tread diameter of sheave; d = nominal diameter of rope. To find any tread diameter from this table, the diameter for the rope construction to be used is multiplied by its nominal diameter d. For example, the minimum sheave tread diameter for a ½-in 6 × 21 FW rope would be ½ in (nominal diameter) × 30 (minimum ratio), or 15 in.

NOTE: These values are for reasonable service. Other values are permitted by various standards such as ANSI, API, PCSA, HMI, CMAA, etc. Similar values affect rope life.

SOURCE: ''Wire Rope User's Manual,'' AISI, reproduced by permission.

Table 8.2.67 Suggested Allowable Radial Bearing Pressures of Ropes on Various Sheave Materials

Material	Regular lay rope, lb/in^2				Lang lay rope, lb/in^2			Flattened strand lang lay, lb/in^2	Remarks
	6 × 7	6 × 19	6 × 37	8 × 19	6 × 7	6 × 19	6 × 37		
Wood	150	250	300	350	165	275	330	400	On end grain of beech, hickory, gum.
Cast iron	300	480	585	680	350	550	660	800	Based on minimum Brinell hardness of 125.
Carbon-steel casting	550	900	1,075	1,260	600	1,000	1,180	1,450	30–40 carbon. Based on minimum Brinell hardness of 160.
Chilled cast iron	650	1,100	1,325	1,550	715	1,210	1,450	1,780	Not advised unless surface is uniform in hardness.
Manganese steel	1,470	2,400	3,000	3,500	1,650	2,750	3,300	4,000	Grooves must be ground and sheaves balanced for high-speed service.

SOURCE: ''Wire Rope User's Manual,'' AISI, reproduced by permission.

Table 8.2.69 Relative Bending Life Factors

Rope construction	Factor	Rope construction	Factor
6 × 7	0.61	6 × 36 WS	1.16
19 × 7 or 18 × 7	0.67	6 × 43 FWS	1.16
Rotation-resistant	0.81	6 × 41 WS	1.30
6 × 19 S	0.90	6 × 41 SFW	1.30
6 × 25 B flattened strand	0.90	6 × 49 SWS	1.30
6 × 27 H flattened strand	0.90	6 × 43 FW (2 op)	1.41
6 × 30 G flattened strand	0.89	6 × 46 SFW	1.41
6 × 21 FW	0.89	6 × 46 WS	1.41
6 × 26 WS	1.00	8 × 19 S	1.00
6 × 25 FW	1.00	8 × 25 FW	1.25
6 × 31 WS	1.00	6 × 42 Tiller	2.00
6 × 37 SFW			

SOURCE: ''Wire Rope User's Manual,'' AISI, reproduced by permission.

Table 8.2.70 Minimum Sheave- and Drum-Groove Dimensions*

Nominal rope diameter		Groove radius			
		New		Worn	
in	nm	in	mm	in	mm
¼	6.4	0.135	3.43	.129	3.28
5⁄16	8.0	0.167	4.24	.160	4.06
⅜	9.5	0.201	5.11	.190	4.83
7⁄16	11	0.234	5.94	.220	5.59
½	13	0.271	6.88	.256	6.50
9⁄16	14.5	0.303	7.70	.288	7.32
⅝	16	0.334	8.48	.320	8.13
¾	19	0.401	10.19	.380	9.65
⅞	22	0.468	11.89	.440	11.18
1	26	0.543	13.79	.513	13.03
1⅛	29	0.605	15.37	.577	14.66
1¼	32	0.669	16.99	.639	16.23
1⅜	35	0.736	18.69	.699	17.75
1½	38	0.803	20.40	.759	19.28
1⅝	42	0.876	22.25	.833	21.16
1¾	45	0.939	23.85	.897	22.78
1⅞	48	1.003	25.48	.959	24.36
2	52	1.085	27.56	1.025	26.04
2⅛	54	1.137	28.88	1.079	27.41
2¼	58	1.210	30.73	1.153	29.29
2⅜	60	1.271	32.28	1.199	30.45
2½	64	1.338	33.99	1.279	32.49
2⅝	67	1.404	35.66	1.339	34.01
2¾	71	1.481	37.62	1.409	35.79
2⅞	74	1.544	39.22	1.473	37.41
3	77	1.607	40.82	1.538	39.07
3⅛	80	1.664	42.27	1.598	40.59
3¼	83	1.731	43.97	1.658	42.11
3⅜	87	1.807	45.90	1.730	43.94
3½	90	1.869	47.47	1.794	45.57
3¾	96	1.997	50.72	1.918	48.72
4	103	2.139	54.33	2.050	52.07
4¼	109	2.264	57.51	2.178	55.32
4½	115	2.396	60.86	2.298	58.37
4¾	122	2.534	64.36	2.434	61.82
5	128	2.663	67.64	2.557	64.95
5¼	135	2.804	71.22	2.691	68.35
5½	141	2.929	74.40	2.817	71.55
5¾	148	3.074	78.08	2.947	74.85
6	154	3.198	81.23	3.075	78.11

* Values given are applicable to grooves in sheaves and drums; they are not generally suitable for pitch design since this may involve other factors. Further, the dimensions do not apply to traction-type elevators; in this circumstance, drum- and sheave-groove tolerances should conform to the elevator manufacturer's specifications. Modern drum design embraces extensive considerations beyond the scope of this publication. It should also be noted that drum grooves are now produced with a number of oversize dimensions and pitches applicable to certain service requirements.

SOURCE: ''Wire Rope User's Manual,'' AISI, reproduced by permission.

are not preformed a minimum of two seizings on each side is recommended, and these should be spaced six rope diameters apart (see Fig. 8.2.128). Seizings should be made of soft or annealed wire or strand, and the width of the seizing should never be less than the diameter of the rope being seized. Table 8.2.71 lists suggested seizing wire diameters.

Wire Rope Fittings or Terminations End terminations allow forces to be transferred from rope to machine, or load to rope, etc. Figure 8.2.129 illustrates the most commonly used end fittings or terminations. Not all terminations will develop full strength. In fact, if all of the rope elements are not held securely, the individual strands will sustain unequal loads causing unequal wear among them, thus shortening the effective rope service life. Socketing allows an end fitting which reduces the chances of unequal strand loading.

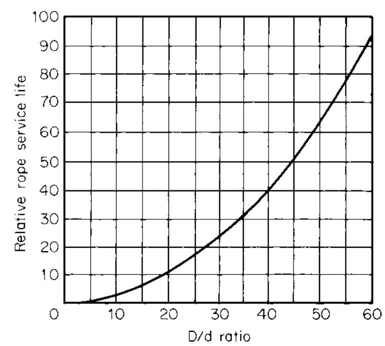


Fig. 8.2.127 Service life curves for various D/d ratios. Note that this curve takes into account only bending and tensile stresses. *(Reproduced from ''Wire Rope User's Manual,'' AISI, by permission.)*

Wire rope manufacturers have developed a recommended procedure for socketing. A tight wire serving band is placed where the socket base will be, and the wires are unlaid, straightened, and ''broomed'' out. Fiber core is cut close to the serving band and removed, wires are cleaned with a solvent such as SC-methyl chloroform, and brushed to remove dirt and grease. If additional cleaning is done with muriatic acid this must be followed by a neutralizing rinse (if possible, ultrasonic cleaning is preferred). The wires are dipped in flux, the socket is positioned, zinc (spelter) is poured and allowed to set, the serving band is removed, and the rope lubricated.

A somewhat similar procedure is used in thermoset resin socketing.

Socketed terminations generally are able to develop 100 percent of nominal strength.

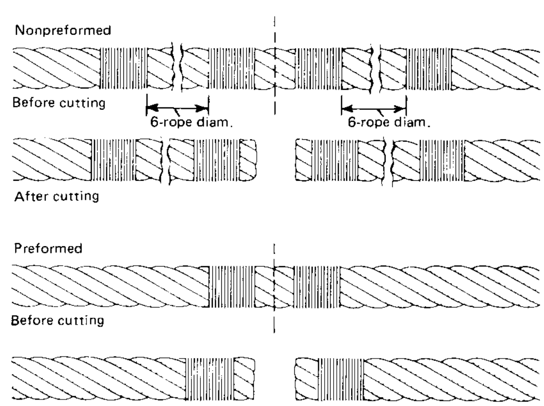


Fig. 8.2.128 Seizings. *(Reproduced from ''Wire Rope User's Manual,'' AISI, by permission.)*

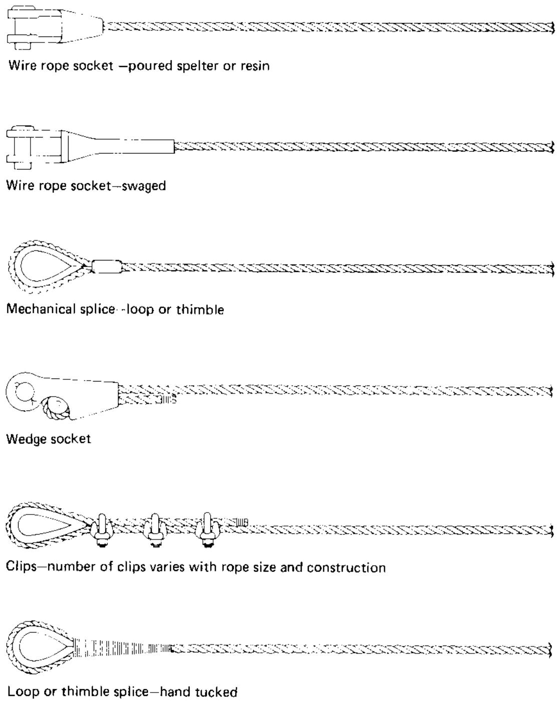


Fig. 8.2.129 End fittings, or terminations, showing the six most commonly used. *(Reproduced from ''Wire Rope User's Manual,'' AISI, by permission.)*

Table 8.2.71 Seizing*

Rope diameter		Suggested seizing wire diameter†	
in	mm	in	mm
1/8–5/16	3.5–8.0	0.032	0.813
3/8–9/16	9.4–14.5	0.048	1.21
5/8–15/16	16.0–24.0	0.063	1.60
1–1 5/16	26.0–33.0	0.080	2.03
1 3/8–1 11/16	35.0–43.0	0.104	2.64
1 3/4 and larger	45.0 and larger	0.124	3.15

* Length of the seizing should not be less than the rope diameter.
† The diameter of seizing wire for elevator ropes is usually somewhat smaller than that shown in this table. Consult the wire rope manufacturer for specific size recommendations. Soft annealed seizing strand may also be used.
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FIBER LINES

The breaking strength of various fiber lines is given in Table 8.2.72.

Knots, Hitches, and Bends

No two parts of a knot which would move in the same direction if the rope were to slip should lie alongside of and touching each other. The knots shown in Fig. 8.2.130 are known by the following names:

A, bight of a rope; *B*, simple or overhand knot; *C*, figure 8 knot; *D*, double knot; *E*, boat knot; *F*, bowline, first step; *G*, bowline, second step; *H*, bowline, completed; *I*, square or reef knot; *J*, sheet bend or weaver's knot; *K*, sheet bend with a toggle; *L*, carrick bend;

Table 8.2.72 Breaking Strength of Fiber Lines, Lb

Size, in Diam	Size, in Cir.	Manila	Composite	Sisal	Sisal mixed	Sisal hemp	Agave or jute	Nylon	Dacron	Poly-ethylene	Poly-propylene (mono-filament)	Esterlon (polyester)
3/16	5/8	450	—	360	340	310	270	1,000	850	700	800	720
1/4	3/4	600	—	480	450	420	360	1,500	1,380	1,200	1,200	1,150
5/16	1	1,000	—	800	750	700	600	2,500	2,150	1,750	2,100	1,750
3/8	1 1/8	1,350	—	1,080	1,010	950	810	3,500	3,000	2,500	3,100	2,450
7/16	1 1/4	1,750	—	1,400	1,310	1,230	1,050	4,800	4,500	3,400	3,700	3,400
1/2	1 1/2	2,650	—	2,120	1,990	1,850	1,590	6,200	5,500	4,100	4,200	4,400
9/16	1 3/4	3,450	—	2,760	2,590	2,410	2,070	8,300	7,300	4,600	5,100	5,700
5/8	2	4,400	—	3,520	3,300	3,080	2,640	10,500	9,500	5,200	5,800	7,300
3/4	2 1/4	5,400	—	4,320	4,050	3,780	3,240	14,000	12,500	7,400	8,200	9,500
13/16	2 1/2	6,500	—	5,200	4,880	4,550	3,900	17,000	15,000	8,900	9,800	11,500
7/8	2 3/4	7,700	—	—	—	—	—	20,000	17,500	10,400	11,500	13,500
1	3	9,000	—	7,200	6,750	6,300	5,400	24,000	20,000	12,600	14,000	16,500
1 1/16	3 1/4	10,500	—	8,400	7,870	7,350	6,300	28,000	22,500	14,500	16,100	19,000
1 1/8	3 1/2	12,000	—	9,600	9,000	8,400	7,200	32,000	25,000	16,500	18,300	21,500
1 1/4	3 3/4	13,500	—	10,800	10,120	9,450	8,100	36,500	28,500	18,600	21,000	24,300
1 5/16	4	15,000	—	12,000	11,250	10,500	9,000	42,000	32,000	21,200	24,000	28,000
1 1/2	4 1/2	18,500	16,600	14,800	13,900	12,950	11,100	51,000	41,000	26,700	30,000	34,500
1 5/8	5	22,500	20,300	18,000	16,900	15,800	13,500	62,000	50,000	32,700	36,500	41,500
1 3/4	5 1/2	26,500	23,800	21,200	19,900	18,500	15,900	77,500	61,000	39,500	44,000	51,000
2	6	31,000	27,900	24,800	23,200	21,700	18,600	90,000	72,000	47,700	53,000	61,000
2 1/8	6 1/2	36,000	—	—	—	—	—	105,000	81,000	55,800	62,000	70,200
2 1/4	7	41,000	36,900	32,800	30,800	28,700	—	125,000	96,000	63,000	70,000	81,000
2 1/2	7 1/2	46,500	—	—	—	—	—	138,000	110,000	72,500	80,500	92,000
2 5/8	8	52,000	46,800	41,600	39,000	36,400	—	154,000	125,000	81,000	90,000	103,000
2 7/8	8 1/2	58,000	—	—	—	—	—	173,000	140,000	92,000	100,000	116,000
3	9	64,000	57,500	51,200	48,000	44,800	—	195,000	155,000	103,000	116,000	130,000
3 1/4	10	77,000	69,300	61,600	57,800	53,900	—	238,000	190,000	123,000	137,000	160,000
3 1/2	11	91,000	—	—	—	—	—	288,000	230,000	146,000	162,000	195,000
4	12	105,000	94,500	84,000	78,800	73,500	—	342,000	275,000	171,000	190,000	230,000

Breaking strength is the maximum load the line will hold at the time of breaking. The working load of a line is one-fourth to one-fifth of the breaking strength.
SOURCE: Adapted, by permission of the U.S. Naval Institute, Annapolis, MD, and Wall Rope Works, Inc., New York, NY.

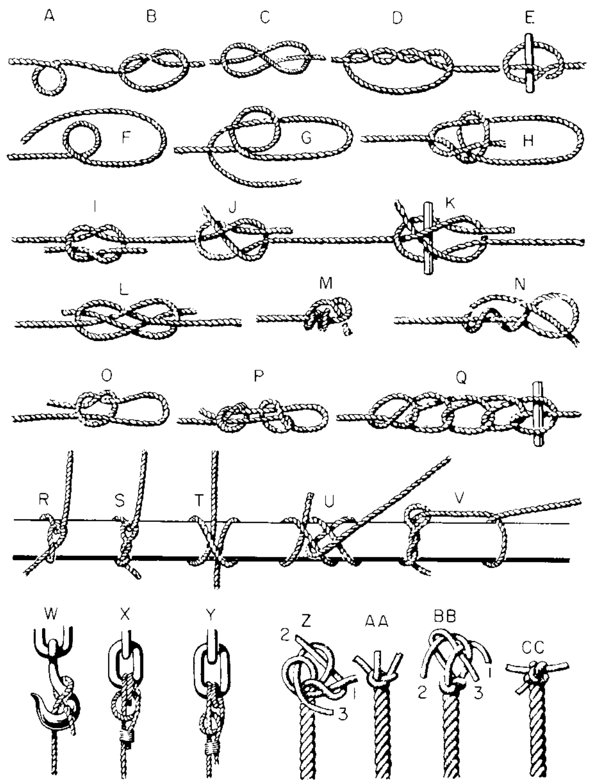


Fig. 8.2.130 Knots, hitches, and bends.

M, ''stevedore'' knot completed; *N*, ''stevedore'' knot commenced; *O*, slip knot; *P*, Flemish loop; *Q*, chain knot with toggle; *R*, half hitch; *S*, timber hitch; *T*, clove hitch; *U*, rolling hitch; *V*, timber hitch and half hitch; *W*, blackwall hitch; *X*, fisherman's bend; *Y*, round turn and half hitch; *Z*, wall knot commenced; *AA*, wall knot completed; *BB*, wall-knot crown commenced; *CC*, wall-knot crown completed.

The bowline *H*, one of the most useful knots, will not slip, and after being strained is easily untied. Knots *H*, *K*, and *M* are easily untied after being under strain. The knot *M* is useful when the rope passes through an eye and is held by the knot, as it will not slip, and is easily untied after being strained. The wall knot is made as follows: Form a bight with strand 1 and pass the strand 2 around the end of it, and the strand 3 around the end of 2, and then through the bight of 1, as shown at *Z* in the figure. Haul the ends taut when the appearance is as shown in *AA*. The end of the strand 1 is now laid over the center of the knot, strand 2 laid over 1, and 3 over 2, when the end of 3 is passed through the bight of 1, as shown at *BB*. Haul all the strands taut, as shown at *CC*. The ''stevedore'' knot (*M*, *N*) is used to hold the end of a rope from passing through a hole. When the rope is strained, the knot draws up tight, but it can be easily untied when the strain is removed. If a knot or hitch of any kind is tied in a rope, its failure under stress is sure to occur at that place. The shorter the bend in the standing rope, the weaker is the knot. The approximate strength of knots compared with the full strength of (dry) rope (= 100), based on Miller's experiments (*Mach.*, 1900, p. 198), is as follows: eye splice over iron thimble, 90; short splice in rope, 80; *S* and *Y*, 65; *H*, *O*, and *T*, 60; *I* and *J*, 50; *B* and *P*, 45.

NAILS AND SPIKES

Nails are either **wire nails** of circular cross section and constant diameter or **cut nails** of rectangular cross section with taper from head to point. The larger sizes are called **spikes.** The length of the nail is expressed in the **''penny''** system, the equivalents in inches being given in Tables 8.2.73 to 8.2.75. The letter d is the accepted symbol for penny. A keg of nails weighs 100 lb. **Heavy hinge nails** or **track nails** with countersunk heads have chisel points unless diamond points are specified. **Plasterboard nails** are smooth with circumferential grooves and have diamond points. **Spikes** are made either with flat heads and diamond points or with oval heads and chisel points.

Table 8.2.73 Wire Nails for Special Purposes
(Steel wire gage)

	Barrel nails		Barbed roofing nails		Barbed dowel nails	
Length, in	Gage	No. per lb	Gage	No. per lb	Gage	No. per lb
⅝	15½	1,570	—	—	8	394
¾	15½	1,315	13	729	8	306
⅞	14½	854	12	478	8	250
1	14½	750	12	416	8	212
1⅛	14½	607	12	368	8	183
1¼	14	539	11	250	8	16
1⅜	13	386	11	228	8	145
1½	13	355	10	167	8	131
1¾	—	—	10	143		
2	—	—	9	104		

	Clout nails		Slating nails		Fine nails		
Length, in	Gage	No. per lb	Gage	No. per lb	Length, in	Gage	No. per lb
¾	15	999					
⅞	14	733					
1	14	648	12	425	1	16½	1,280
1⅛	14	580	10½	229			
1¼	13	398	—	—	1	17	1,492
1⅜	13	365					
1½	13	336	10½	190	1⅛	15	757
1¾	—	—	10	144	1⅛	16	984
2	—	—	9	104			

Table 8.2.74 Wire Nails and Spikes
(Steel wire gage)

		Casing nails		Finishing nails		Clinch nails		Shingle nails	
Size of nail	Length, in	Gage	No. per lb	Gage	No. per lb	Gage	No. per lb	Gage	No. per lb
2d	1	15½	940	16½	1.473	14	723	13	434
3d	1¼	14½	588	15½	880	13	432	12	271
4d	1½	14	453	15	634	12	273	12	233
5d	1¾	14	389	15	535	12	234	12	203
6d	2	12½	223	13	288	11	158		
7d	2¼	12½	200	13	254	11	140		
8d	2½	11½	136	12½	196	10	101		
9d	2¾	11½	124	12½	178	10	91.4		
10d	3	10½	90	11½	124	9	70		
12d	3¼	10½	83	11½	113	9	64.1		
16d	3½	10	69	11	93	8	50		
20d	4	9	51	10	65	7	36.4		
30d	4½	9	45						
40d	5	8	37						

		Boat nails				Hinge nails				Flooring nails	
		Heavy		Light		Heavy		Light			
Size of nail	Length, in	Diam, in	No. per lb	Diam, in	No. per lb	Diam, in	No. per lb	Diam, in	No. per lb	Gage	No. per lb
4d	1½	¼	47	3/16	82	¼	53	3/16	90		
6d	2	¼	36	3/16	62	¼	39	3/16	66	11	168
8d	2½	¼	29	3/16	50	¼	31	3/16	53	10	105
10d	3	⅜	11	¼	24	⅜	12	¼	25	9	72
12d	3¼	⅜	10.4	¼	22	⅜	11	¼	23	8	56
16d	3½	⅜	9.6	¼	20	⅜	10	¼	22	7	44
20d	4	⅜	8	¼	18	⅜	8	¼	19	6	32

		Common wire nails and brads		Barbed car nails				Spikes			
				Heavy		Light					
Size of nail	Length, in	Gage	No. per lb	Gage	No. per lb	Gage	No. per lb	Size	Length, in	Gage	Approx no. per lb
2d	1	15	847	—	—	—	—	10d	3	6	43
3d	1¼	14	548	—	—	—	—	12d	3¼	6	39
4d	1½	12½	294	10	179	12	284	16d	3½	5	31
5d	1¾	12½	254	9	124	10	152	20d	4	4	23
6d	2	11½	167	9	108	10	132	30d	4½	3	18
7d	2¼	11½	150	8	80	9	95	40d	5	2	14
8d	2½	10¼	101	8	72	9	88	50d	5½	1	11
9d	2¾	10¼	92	7	55	8	65	50d	6	1	10
10d	3	9	66	7	50	8	59	—	7	5/16 in.	7
12d	3¼	9	61	6	39	7	46	—	8	⅜	4.1
16d	3½	8	47	6	36	7	43	—	9	⅜	3.7
20d	4	6	30	5	27	6	32	—	10	⅜	3.3
30d	4½	5	23	5	24	6	28	—	12	⅜	2.7
40d	5	4	18	4	18	5	22				
50d	5½	3	14	3	14	4	17				
60d	6	2	11	3	13	4	15				

Table 8.2.75 Cut Steel Nails and Spikes
(Sizes, lengths, and approximate number per lb)

Size	Length, in	Common	Clinch	Finishing	Casing and box	Fencing	Spikes	Barrel	Slating	Tobacco	Brads	Shingle
2d	1	740	400	1,100	—	—	—	450	340			
3d	1¼	460	260	880	—	—	—	280	280			
4d	1½	280	180	530	420	—	—	190	220			
5d	1¾	210	125	350	300	100	—	—	180	130		
6d	2	160	100	300	210	80	—	—	—	97	120	
7d	2¼	120	80	210	180	60	—	—	—	85	94	
8d	2½	88	68	168	130	52	—	—	—	68	74	90
9d	2¾	73	52	130	107	38	—	—	—	58	62	72
10d	3	60	48	104	88	26	—	—	—	48	50	60
12d	3¼	46	40	96	70	20	—	—	—	—	40	
16d	3½	33	34	86	52	18	17	—	—	—	27	
20d	4	23	24	76	38	16	14					
25d	4¼	20	—	—	—	—	—					
30d	4½	16½	—	—	30	—	11					
40d	5	12	—	—	26	—	9					
50d	5½	10	—	—	20	—	7½					
60d	6	8	—	—	16	—	6					
—	6½	—	—	—	—	—	5½					
—	7	—	—	—	—	—	5					

WIRE AND SHEET-METAL GAGES

In the metal industries, the word *gage* has been used in various systems, or scales, for expressing the thickness or weight per unit area of thin plates, sheet, and strip, or the diameters of rods and wire. Specific diameters, thicknesses, or weights per square foot have been or are denoted in gage systems by certain numerals followed by the word *gage*, for example, no. 12 gage, or simply 12 gage. Gage numbers for flat rolled products have been used only in connection with thin materials (Table 8.2.76). Heavier and thicker, flat rolled materials are usually designated by thickness in English or metric units.

There is considerable danger of confusion in the use of gage number in both foreign and domestic trade, which can be avoided by specifying thickness or diameter in inches or millimeters.

DRILL SIZES

See Table 8.2.77.

Table 8.2.76 Comparison of Standard Gages*
Thickness of diameter, in

Gage no.	BWG; Stubs Iron Wire	AWG; B&S	U.S. Steel Wire; Am. Steel & Wire; Washburn & Moen; Steel Wire	Galv. sheet steel	Manufacturers' standard
0000000	—	—	0.4900	—	—
000000	—	0.580000	0.4615	—	—
00000	—	0.516500	0.4305	—	—
0000	0.454	0.460000	0.3938	—	—
000	0.425	0.409642	0.3625	—	—
00	0.380	0.364796	0.3310	—	—
0	0.340	0.324861	0.3065	—	—
1	0.300	0.289297	0.2830	—	—
2	0.284	0.257627	0.2625	—	—
3	0.259	0.229423	0.2437	—	0.2391
4	0.238	0.204307	0.2253	—	0.2242
5	0.220	0.181940	0.2070	—	0.2092
6	0.203	0.162023	0.1920	—	0.1943
7	0.180	0.144285	0.1770	—	0.1793
8	0.165	0.128490	0.1620	0.1681	0.1644
9	0.148	0.114423	0.1483	0.1532	0.1495
10	0.134	0.101897	0.1350	0.1382	0.1345
11	0.120	0.090742	0.1205	0.1233	0.1196
12	0.109	0.080808	0.1055	0.1084	0.1046
13	0.095	0.071962	0.0915	0.0934	0.0897
14	0.083	0.064084	0.0800	0.0785	0.0747
15	0.072	0.057068	0.0720	0.0710	0.0673
16	0.065	0.050821	0.0625	0.0635	0.0598
17	0.058	0.045257	0.0540	0.0575	0.0538
18	0.049	0.040303	0.0475	0.0516	0.0478
19	0.042	0.035890	0.0410	0.0456	0.0418
20	0.035	0.031961	0.0348	0.0396	0.0359
21	0.032	0.028462	0.03175	0.0366	0.0329
22	0.028	0.025346	0.0286	0.0336	0.0299
23	0.025	0.022572	0.0258	0.0306	0.0269
24	0.022	0.020101	0.0230	0.0276	0.0239
25	0.020	0.017900	0.0204	0.0247	0.0209
26	0.018	0.015941	0.0181	0.0217	0.0179
27	0.016	0.014195	0.0173	0.0202	0.0164
28	0.014	0.012641	0.0162	0.0187	0.0149
29	0.013	0.011257	0.0150	0.0172	0.0135
30	0.012	0.010025	0.0140	0.0157	0.0120
31	0.010	0.008928	0.0132	0.0142	0.0105
32	0.009	0.007950	0.0128	0.0134	0.0097
33	0.008	0.007080	0.0118	—	0.0090
34	0.007	0.006305	0.0104	—	0.0082
35	0.005	0.005615	0.0095	—	0.0075
36	0.004	0.005000	0.0090	—	0.0067
37	—	0.004453	0.0085	—	0.0064
38	—	0.003965	0.0080	—	0.0060
39	—	0.003531	0.0075	—	—
40	—	0.003144	0.0070	—	—

* Principal uses—BWG: strips, bands, hoops, and wire; AWG or B&S: nonferrous sheets, rod, and wire; U.S. Steel Wire: steel wire except music wire; manufacturers' standard: uncoated steel sheets.

Table 8.2.77 Diameters of Small Drills

Number, letter, metric, and fractional drills in order of size (rounded to 4 decimal places)

No.	Ltr	mm	in	Diam, in
		0.10		0.0039
		0.15		0.0059
		0.20		0.0079
		0.25		0.0098
		0.30		0.0118
80				0.0135
		0.35		0.0137
79				0.0145
			1/64	0.0156
		0.40		0.0157
78				0.0160
		0.45		0.0177
77				0.0180
		0.50		0.0197
76				0.0200
75				0.0210
		0.55		0.0216
74				0.0225
		0.60		0.0236
73				0.0240
72				0.0250
		0.65		0.0255
71				0.0260
		0.70		0.0275
70				0.0280
69				0.0292
		0.75		0.0295
68				0.0310
			1/32	0.0312
		0.80		0.0314
67				0.0320
66				0.0330
		0.85		0.0334
65				0.0350
		0.90		0.0354
64				0.0360
63				0.0370
		0.95		0.0374
62				0.0380
61				0.0390
		1.00		0.0393
60				0.0400
59				0.0410
		1.05		0.0413
58				0.0420
57				0.0430
		1.10		0.0433
		1.15		0.0452
56				0.0465
			3/64	0.0468
		1.20		0.0472
		1.25		0.0492
		1.30		0.0512
55				0.0520
		1.35		0.0531
54				0.0550
		1.40		0.0551
		1.45		0.0570
		1.50		0.0590
53				0.0595
		1.55		0.0610
			1/16	0.0625
		1.60		0.0630
52				0.0635
		1.65		0.0649
		1.70		0.0669
51				0.0670
		1.75		0.0688
50				0.0700
		1.80		0.0709
		1.85		0.0728
49				0.0730
		1.90		0.0748
48				0.0760
		1.95		0.0767
			5/64	0.0781
47				0.0785
		2.00		0.0787
		2.05		0.0807
46				0.0810
45				0.0820
		2.10		0.0827
		2.15		0.0846
44				0.0860
		2.20		0.0866
		2.25		0.0885
43				0.0890
		2.30		0.0906
		2.35		0.0925
42				0.0935
			3/32	0.0937
		2.40		0.0945
41				0.0960
		2.45		0.0964
40				0.0980
		2.50		0.0984
39				0.0995
38				0.1015
		2.60		0.1024
37				0.1040
		2.70		0.1063
36				0.1065
		2.75		0.1082
			7/64	0.1093
35				0.1100
		2.80		0.1102
34				0.1110
33				0.1130
		2.90		0.1142
32				0.1160
		3.00		0.1181
31				0.1200
		3.10		0.1220
			1/8	0.1250
		3.20		0.1260
		3.25		0.1279
30				0.1285
		3.30		0.1299
		3.40		0.1339
29				0.1360
		3.50		0.1378
28				0.1405
			9/64	0.1406
		3.60		0.1417
27				0.1440
		3.70		0.1457
26				0.1470
		3.75		0.1476
25				0.1495
		3.80		0.1496
24				0.1520
		3.90		0.1535
23				0.1540
			5/32	0.1562
22				0.1570
		4.00		0.1575
21				0.1590
20				0.1610
		4.10		0.1614
		4.2		0.1654
19				0.1660
		4.25		0.1673
		4.30		0.1693
18				0.1695
			11/64	0.1718
17				0.1730
		4.40		0.1732
16				0.1770
		4.50		0.1772
15				0.1800
		4.60		0.1811
14				0.1820
13				0.1850
		4.70		0.1850
		4.75		0.1850
			3/16	0.1875
		4.80		0.1890
12				0.1890
11				0.1910
		4.90		0.1920
10				0.1935
9				0.1960
		5.00		0.1968
8				0.1990
		5.10		0.2008
7				0.2010
			13/64	0.2031
6				0.2040
		5.20		0.2047
5				0.2055
		5.25		0.2066
		5.30		0.2087
4				0.2090
		5.40		0.2126
3				0.2130
		5.50		0.2165
			7/32	0.2187
		5.60		0.2205
2				0.2210
		5.70		0.2244
		5.75		0.2263
1				0.2280
		5.80		0.2283
		5.90		0.2323
	A			0.2340
			15/64	0.2340
		6.00		0.2362
	B			0.2380
		6.10		0.2402
	C			0.2420
		6.20		0.2441
	D			0.2460
		6.25		0.2460
		6.30		0.2480
	E		1/4	0.2500
		6.40		0.2519
		6.50		0.2559
	F			0.2570
		6.60		0.2598
	G			0.2610
		6.70		0.2637
			17/64	0.2656
		6.75		0.2657
	H			0.2660
		6.80		0.2677
		6.90		0.2717
	I			0.2720
		7.00		0.2756
	J			0.2770
		7.10		0.2795
	K			0.2810
			9/32	0.2812
		7.20		0.2835
		7.25		0.2854
		7.30		0.2874
	L			0.2900
		7.40		0.2913
	M			0.2950
		7.50		0.2953

Table 8.2.77 Diameters of Small Drills (*Continued*)

No.	Ltr	mm	in	Diam, in	No.	Ltr	mm	in	Diam, in	No.	Ltr	mm	in	Diam, in
			19/64	0.2968			9.70		0.3819			17.00		0.6693
		7.60		0.2992			9.75		0.3838				43/64	0.6718
	N			0.3020			9.80		0.3858				11/16	0.6875
		7.70		0.3031		W			0.3860			17.50		0.6890
		7.75		0.3051			9.90		0.3898				45/64	0.7031
		7.80		0.3071				25/64	0.3906			18.00		0.7087
		7.90		0.3110			10.00		0.3937				23/32	0.7187
			5/16	0.3125		X			0.3970			18.50		0.7283
		8.00		0.3150		Y			0.4040				47/64	0.7374
	O			0.3160				13/32	0.4062			19.00		0.7480
		8.10		0.3189		Z			0.4130				3/4	0.7500
		8.20		0.3228			10.50		0.4134				49/64	0.7656
	P			0.3230				27/64	0.4218			19.50		0.7677
		8.25		0.3248			11.00		0.4331				25/32	0.7812
		8.30		0.3268				7/16	0.4375			20.00		0.7874
			21/64	0.3281			11.50		0.4528				51/64	0.7968
		8.40		0.3307				29/64	0.4531			20.50		0.8070
	Q			0.3320				15/32	0.4687				13/16	0.8125
		8.50		0.3346			12.00		0.4724			21.00		0.8267
		8.60		0.3386				31/64	0.4843				53/64	0.8281
	R			0.3390			12.50		0.4921				27/32	0.8437
		8.70		0.3425				1/2	0.5000			21.50		0.8464
			11/32	0.3437			13.00		0.5118				55/64	0.8593
		8.75		0.3444				33/64	0.5156			22.00		0.8661
		8.80		0.3464				17/32	0.5312				7/8	0.8750
	S			0.3480			13.50		0.5315			22.50		0.8858
		8.90		0.3504				35/64	0.5468				57/64	0.8906
		9.00		0.3543			14.00		0.5512			23.00		0.9055
	T			0.3580				9/16	0.5625				29/32	0.9062
		9.10		0.3583			14.50		0.5708				59/64	0.9218
			23/64	0.3593				37/64	0.5781			23.50		0.9251
		9.20		0.3622			15.00		0.5905				15/16	0.9375
		9.25		0.3641				19/32	0.5937			24.00		0.9448
		9.30		0.3661				39/64	0.6093				61/64	0.9531
	U			0.3680			15.50		0.6102			24.50		0.9646
		9.40		0.3701				5/8	0.6250				31/32	0.9687
		9.50		0.3740			16.00		0.6299			25.00		0.9842
			3/8	0.3750				41/64	0.6406				63/64	0.9843
	V			0.3770			16.50		0.6496				1.0	1.0000
		9.60		0.3780				21/32	0.6562			25.50		1.0039

SOURCE: Adapted from Colvin and Stanley, ''American Machinists' Handbook,'' 8th ed., McGraw-Hill, New York, 1945.

8.3 GEARING

by George W. Michalec

REFERENCES: Buckingham, ''Manual of Gear Design,'' Industrial Press. Cunningham, Noncircular Gears, *Mach. Des.*, Feb. 19, 1957. Cunningham and Cunningham, Rediscovering the Noncircular Gear, *Mach. Des.*, Nov. 1, 1973. Dudley, ''Gear Handbook,'' McGraw-Hill. Dudley, ''Handbook of Practical Gear Design,'' McGraw-Hill. Michalec, ''Precision Gearing: Theory and Practice,'' Wiley. Shigely, ''Engineering Design,'' McGraw-Hill. AGMA Standards. ''Gleason Bevel and Hypoid Gear Design,'' Gleason Works, Rochester. ''Handbook of Gears: Inch and Metric'' and ''Elements of Metric Gear Technology,'' Designatronics, New Hyde Park, NY. Adams, ''Plastics Gearing: Selection and Application,'' Marcel Dekker.

Notation

a = addendum
b = dedendum
B = backlash, linear measure along pitch circle
c = clearance
C = center distance
d = pitch diam of pinion
d_b = base circle diam of pinion
d_o = outside diam of pinion
d_r = root diam of pinion
D = pitch diameter of gear
D_P = pitch diam of pinion
D_G = pitch diam of gear
D_o = outside diam of gear
D_b = base circle diam of gear
D_t = throat diam of wormgear
F = face width
h_k = working depth
h_t = whole depth
$\text{inv}\ \phi$ = involute function ($\tan \phi - \phi$)

l = lead (advance of worm or helical gear in 1 rev)
$l_p(l_G)$ = lead of pinion (gear) in helical gears
L = lead of worm in one revolution
m = module
m_G = gear ratio ($m_G = N_G/N_P$)
m_p = contact ratio (of profiles)
M = measurement of over pins
$n_P(n_G)$ = speed of pinion (gear), r/min
$N_P(N_G)$ = number of teeth in pinion (gear)
n_w = number of threads in worm
p = circular pitch
p_b = base pitch
p_n = normal circular pitch of helical gear
P_d = diametral pitch
P_{dn} = normal diametral pitch
R = pitch radius
R_c = radial distance from center of gear to center of measuring pin
$R_P(R_G)$ = pitch radius of pinion (gear)
R_T = testing radius when rolled on a variable-center-distance inspection fixture
s = stress
t = tooth thickness
t_n = normal circular tooth thickness
$T_P(T_G)$ = formative number of teeth in pinion (gear) (in bevel gears)
v = pitch line velocity
X = correction factor for profile shift
α = addendum angle of bevel gear
γ = pitch angle of bevel pinion
γ_R = face angle at root of bevel pinion tooth
γ_o = face angle at tip of bevel pinion tooth
Γ = pitch angle of bevel gear
Γ_R = face angle at root of bevel gear tooth
Γg_o = face angle at tip of bevel gear tooth
δ = dedendum angle of bevel gear
$\overline{\Delta C}$ = relatively small change in center distance C
ϕ = pressure angle
ϕ_n = normal pressure angle
ψ = helix or spiral angle
$\psi_P(\psi_G)$ = helix angle of teeth in pinion (gear)
Σ = shaft angle of meshed bevel pair

BASIC GEAR DATA

Gear Types Gears are grouped in accordance with tooth forms, shaft arrangement, pitch, and quality. Tooth forms and shaft arrangements are:

Tooth form	*Shaft arrangement*
Spur	Parallel
Helical	Parallel or skew
Worm	Skew
Bevel	Intersecting
Hypoid	Skew

Pitch definitions (see Fig. 8.3.1). **Diametral pitch** P_d is the ratio of number of teeth in the gear to the diameter of the pitch circle D measured in inches, $P_d = N/D$. **Circular pitch** p is the linear measure in inches along the pitch circle between corresponding points of adjacent teeth. From these definitions, $P_d p = \pi$. The **base pitch** p_b is the distance along the line of action between successive involute tooth surfaces. The base and circular pitches are related as $p_b = p \cos \phi$, where ϕ = the pressure angle.

Pitch circle is the imaginary circle that rolls without slippage with a pitch circle of a mating gear. The pitch (circle) diameter equals $D = N/P_d = Np/\pi$. The basic relation between P_d and p is $P_d p = \pi$.

Tooth size is related to pitch. In terms of diametral pitch P_d, the relationship is inverse; i.e., large P_d implies a small tooth, and small P_d implies a large tooth. Conversely, there is a direct relationship between tooth size and circular pitch p. A small tooth has a small p, but a large tooth has a large p. (See Fig. 8.3.1*b*.) In terms of P_d, coarse teeth comprise P_d less than 20; fine teeth comprise P_d of 20 and higher. (See Fig. 8.3.1*b*.) **Quality** of gear teeth is classified as commercial, precision, and ultraprecision.

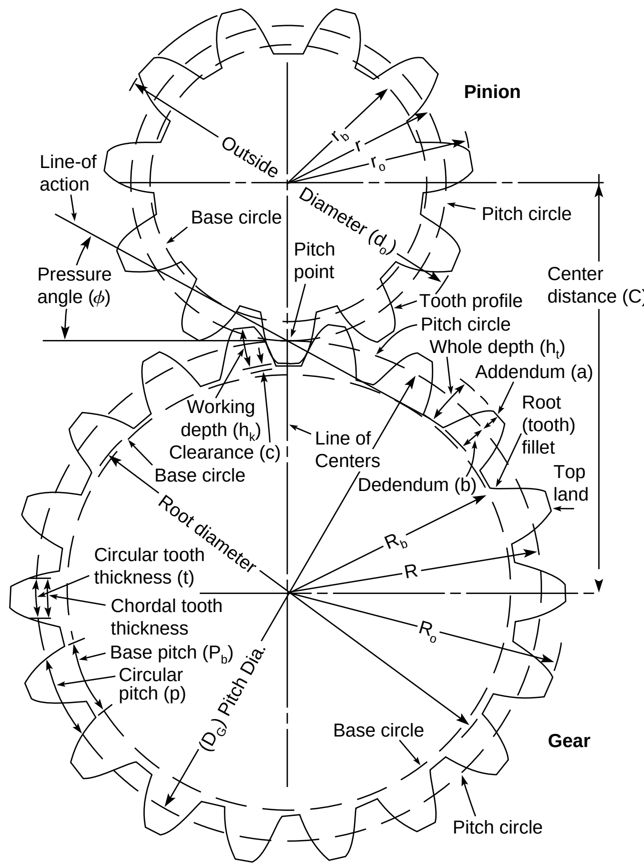


Fig. 8.3.1*a* Basic gear geometry and nomenclature.

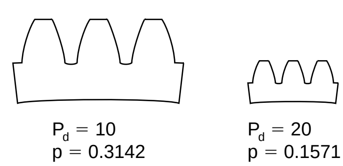


Fig. 8.3.1*b* Comparison of pitch and tooth size.

Pressure angle ϕ for all gear types is the acute angle between the common normal to the profiles at the contact point and the common pitch plane. For spur gears it is simply the acute angle formed by the common tangent between base circles of mating gears and a normal to the line of centers. For standard gears, pressure angles of 14½°, 20°, and 25° have been adopted by ANSI and the gear industry (see Fig. 8.3.1*a*). The 20° pressure angle is most widely used because of its versatility. The higher pressure angle 25° provides higher strength for highly loaded gears. Although 14½° appears in standards, and in past decades was extensively used, it is used much less than 20°. The 14½° standard is still used for replacement gears in old design equipment, in applications where backlash is critical, and where advantage can be taken of lower backlash with change in center distance.

The **base circle** (or **base cylinder**) is the circle from which the involute tooth profiles are generated. The relationship between the base-circle and pitch-circle diameter is $D_b = D \cos \phi$.

Tooth proportions are established by the addendum, dedendum, working depth, clearance, tooth circular thickness, and pressure angle (see Fig. 8.3.1). In addition, gear face width F establishes thickness of the gear measured parallel to the gear axis.

For involute teeth, proportions have been standardized by ANSI and AGMA into a limited number of systems using a basic rack for specification (see Fig. 8.3.2 and Table 8.3.1). Dimensions for the basic rack are normalized for diametral pitch = 1. Dimensions for a specific pitch are

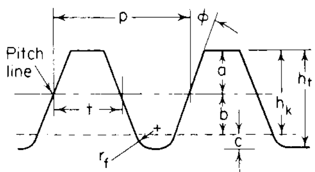


Fig. 8.3.2 Basic rack for involute gear systems. a = addendum; b = dedendum; c = clearance; h_k = working depth; h_t = whole depth; p = circular pitch; r_f = fillet radius; t = tooth thickness; ϕ = pressure angle.

obtained by dividing by the pitch. Standards for basic involute spur, helical and face gear designs, and noninvolute bevel and wormgear designs are listed in Table 8.3.2.

Gear ratio (or **mesh ratio**) m_G is the ratio of number of teeth in a meshed pair, expressed as a number greater than 1; $m_G = N_G/N_P$, where the pinion is the member having the lesser number of teeth. For spur and parallel-shaft helical gears, the base circle ratio must be identical to the gear ratio. The speed ratio of gears is inversely proportionate to their numbers of teeth. Only for standard spur and parallel-shaft helical gears is the pitch diameter ratio equal to the gear ratio and inversely proportionate to the speed ratio.

Metric Gears—Tooth Proportions and Standards

Metric gearing not only is based upon different units of length measure but also involves its own unique design standard. This means that metric gears and American-standard-inch diametral-pitch gears are not interchangeable.

In the metric system the *module m* is analogous to pitch and is defined as

$$m = \frac{D}{N} = \text{mm of pitch diameter per tooth}$$

Table 8.3.2 Gear System Standards

Gear type	ANSI/AGMA no.	Title
Spur and helical	201.02	Tooth Proportions for Coarse-Pitch Involute Spur Gears
Spur and helical	1003-G93	Tooth Proportions for Fine-Pitch Spur and Helical Gearing
Spur and helical	370.01	Design Manual for Fine-Pitch Gearing
Bevel gears	2005-B88	Design Manual for Bevel Gears (Straight, Zerol, Spiral, and Hypoid)
Worm gearing	6022-C93	Design of General Industrial Coarse-Pitch Cylindrical Worm Gearing
Worm gearing	6030-C87	Design of Industrial Double-Enveloping Worm Gearing
Face gears	203.03	Fine-Pitch on Center-Face Gears for 20-Degree Involute Spur Pinions

Note that, for the module to have proper units, the pitch diameter must be in millimeters.

The **metric module** was developed in a number of versions that differ in minor ways. The German DIN standard is widely used in Europe and other parts of the world. The Japanese have their own version defined in JS standards. Deviations among these and other national standards are minor, differing only as to dedendum size and root radii. The differences have been resolved by the new unified module standard promoted by the International Standards Organization (ISO). This unified version (Fig. 8.3.3) conforms to the new SI in all respects. All major industrial

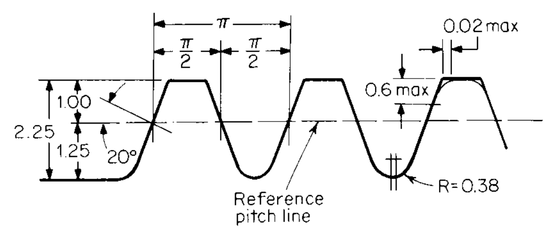


Fig. 8.3.3 The ISO basic rack for metric module gears.

countries on the metric system have shifted to this ISO standard, which also is the basis for American metric gearing. Table 8.3.3 lists pertinent current ISO metric standards.

Table 8.3.1 Tooth Proportions of Basic Rack for Standard Involute Gear Systems

Tooth parameter (of basic rack)	Symbol, Figs. 8.3.1*a* and 8.3.2	Tooth proportions for various standard systems					
		1	2	3	4	5	6
		Full-depth involute, 14½°	Full-depth involute, 20°	Stub involute, 20°	Coarse-pitch involute spur gears, 20°	Coarse-pitch involute spur gears, 25°	Fine-pitch involute, 20°
1. System sponsors		ANSI and AGMA	ANSI	ANSI and AGMA	AGMA	AGMA	ANSI and AGMA
2. Pressure angle	ϕ	14½°	20°	20°	20°	25°	20°
3. Addendum	a	$1/P_d$	$1/P_d$	$0.8/P_d$	$1.000/P_d$	$1.000/P_d$	$1.000/P_d$
4. Min dedendum	b	$1.157/P_d$	$1.157/P_d$	$1/P_d$	$1.250/P_d$	$1.250/P_d$	$1.200/P_d + 0.002$
5. Min whole depth	h_t	$2.157/P_d$	$2.157/P_d$	$1.8/P_d$	$2.250/P_d$	$2.250/P_d$	$2.2002/P_d$ 0.002 in
6. Working depth	h_k	$2/P_d$	$2/P_d$	$1.6/P_d$	$2.000/P_d$	$2.000/P_d$	$2.000/P_d$
7. Min clearance	h_c	$0.157/P_d$	$0.157/P_d$	$0.200/P_d$	$0.250/P_d$	$0.250/P_d$	$0.200/P_d + 0.002$ in
8. Basic circular tooth thickness on pitch line	t	$1.5708/P_d$	$1.5708/P_d$	$1.5708/P_d$	$\pi/(2P_d)$	$\pi/(2P_d)$	$1.5708/P_d$
9. Fillet radius in basic rack	r_f	1⅓ × clearance	1½ × clearance	Not standardized	$0.300/P_d$	$0.300/P_d$	Not standardized
10. Diametral pitch range		Not specified	Not specified	Not specified	19.99 and coarser	19.99 and coarser	20 and finer
11. Governing standard:							
ANSI		B6.1	B6.1	B6.1			
AGMA		201.02		201.02	201.02	201.02	1,003–G93

Table 8.3.3 ISO Metric Gearing Standards

ISO 53 : 1974	Cylindrical gears for general and heavy engineering—Basic rack
ISO 54 : 1977	Cylindrical gears for general and heavy engineering—Modules and diametral pitches
ISO 677 : 1976	Straight bevel gears for general and heavy engineering—Basic rack
ISO 678 : 1976	Straight bevel gears for general and heavy engineering—Modules and diametral pitches
ISO 701 : 1976	International gear notation—symbols for geometric data
ISO 1122-1 : 1983	Glossary of gear terms—Part 1: Geometric definitions
ISO 1328 : 1975	Parallel involute gears—ISO system of accuracy
ISO 1340 : 1976	Cylindrical gears—Information to be given to the manufacturer by the purchaser in order to obtain the gear required
ISO 1341 : 1976	Straight bevel gears—Information to be given to the manufacturer by the purchaser in order to obtain the gear required
ISO 2203 : 1973	Technical drawings—Conventional representation of gears

Tooth proportions for standard spur and helical gears are given in terms of the basic rack. Dimensions, in millimeters, are normalized for module $m = 1$. Corresponding values for other modules are obtained by multiplying each dimension by the value of the specific module m. Major tooth parameters are described by this standard:

Tooth form: Straight-sided and full-depth, forming the basis of a family of full-depth interchangeable gears.

Pressure angle: 20°, conforming to worldwide acceptance.

Addendum: Equal to module m, which corresponds to the American practice of $1/P_d$ = addendum.

Dedendum: Equal to $1.25m$, which corresponds to the American practice of $1.25/P_d$ = dedendum.

Root radius: Slightly greater than American standards specifications.

Tip radius: A maximum is specified, whereas American standards do not specify. In practice, U.S. manufacturers can specify a tip radius as near zero as possible.

Note that the basic racks for metric and American inch gears are essentially identical, but metric and American standard gears are not interchangeable.

The preferred standard gears of the metric system are not interchangeable with the preferred diametral-pitch sizes. Table 8.3.4 lists commonly used pitches and modules of both systems (preferred values are boldface).

Metric gear use in the United States, although expanding, is still a small percentage of total gearing. Continuing industry conversions and imported equipment replacement gearing are building an increasing demand for metric gearing. The reference list cites a domestic source of stock metric gears of relatively small size, in medium and fine pitches. Large diameter coarse pitch metric gears are made to order by many gear fabricators.

Table 8.3.4 Metric and American Gear Equivalents

Diametral pitch P_d	Module m	Circular pitch, in	Circular pitch, mm	Circular tooth thickness, in	Circular tooth thickness, mm	Addendum, in	Addendum, mm
1/2	50.8000	6.2832	159.593	3.1416	79.7965	2.0000	50.8000
0.5080	**50**	6.1842	157.080	3.0921	78.5398	1.9685	50
0.5644	45	5.5658	141.372	2.7850	70.6858	1.7730	45
0.6048	**42**	5.1948	131.947	2.5964	65.9734	1.6529	42
0.6513	39	4.8237	122.522	2.4129	61.2610	1.5361	39
0.7056	**36**	4.4527	113.097	2.2249	56.5487	1.4164	36
3/4	33.8667	4.1888	106.396	2.0943	53.1977	1.3333	33.8667
0.7697	33	4.0816	103.673	2.0400	51.8363	1.2987	33
0.8467	**30**	3.7105	94.248	1.8545	47.1239	1.1806	30
0.9407	27	3.3395	84.823	1.6693	42.4115	1.0627	27
1	25.4000	3.1416	79.800	1.5708	39.8984	1.0000	25.4001
1.0583	**24**	2.9685	75.398	1.4847	37.6991	0.9452	24
1.1546	22	2.7210	69.115	1.3600	34.5575	0.8658	22
1.2700	**20**	2.4737	62.832	1.2368	31.4159	0.7874	20
1.4111	**18**	2.2263	56.548	1.1132	28.2743	0.7087	18
1.5	16.9333	2.0944	53.198	1.0472	26.5988	0.6667	16.933
1.5875	**16**	1.9790	50.267	0.9894	25.1327	0.6299	16
1.8143	**14**	1.7316	43.983	0.8658	21.9911	0.5512	14
2	12.7000	1.5708	39.898	0.7854	19.949	0.5000	12.7000
2.1167	**12**	1.4842	37.699	0.7420	18.8496	0.4724	12
2.5	10.1600	1.2566	31.918	0.6283	15.9593	0.4000	10.1600
2.5400	**10**	1.2368	31.415	0.6184	15.7080	0.3937	10
2.8222	9	1.1132	28.275	0.5565	14.1372	0.3543	9
3	8.4667	1.0472	26.599	0.5235	13.2995	0.3333	8.4667
3.1416	8.0851	1.0000	25.400	0.5000	12.7000	0.3183	0.0851
3.1750	**8**	0.9895	25.133	0.4948	12.5664	0.3150	8.00
3.5	7.2571	0.8976	22.799	0.4488	11.3994	0.2857	7.2571
3.6286	**7**	0.8658	21.991	0.4329	10.9956	0.2756	7.000
3.9078	**6.5**	0.8039	20.420	0.4020	10.2101	0.2559	6.5
4	6.3500	0.7854	19.949	0.3927	9.9746	0.2500	6.3500
4.2333	**6**	0.7421	18.850	0.3710	9.4248	0.2362	6.0000
4.6182	**5.5**	0.6803	17.279	0.3401	8.6394	0.2165	5.5
5	5.0801	0.6283	15.959	0.3142	7.9794	0.2000	5.080
5.0802	**5**	0.6184	15.707	0.3092	7.8537	0.1968	5.000
5.3474	**4.75**	0.5875	14.923	0.2938	7.4612	0.1870	4.750
5.6444	**4.5**	0.5566	14.138	0.2783	7.0688	0.1772	4.500
6	4.2333	0.5236	13.299	0.2618	6.6497	0.1667	4.233
6.3500	**4**	0.4947	12.565	0.2473	6.2827	0.1575	4.000
6.7733	**3.75**	0.4638	11.781	0.2319	5.8903	0.1476	3.750
7	3.6286	0.4488	11.399	0.2244	5.6998	0.1429	3.629

Table 8.3.4 Metric and American Gear Equivalents (*Continued*)

Diametral pitch P_d	Module m	Circular pitch, in	Circular pitch, mm	Circular tooth thickness, in	Circular tooth thickness, mm	Addendum, in	Addendum, mm
7.2571	**3.5**	0.4329	10.996	0.2164	5.4979	0.1378	3.500
7.8154	**3.25**	0.4020	10.211	0.2010	5.1054	0.1279	3.250
8	3.1750	0.3927	9.974	0.1964	4.9886	0.1250	3.175
8.4667	**3**	0.3711	9.426	0.1855	4.7130	0.1181	3.000
9	2.8222	0.3491	8.867	0.1745	4.4323	0.1111	2.822
9.2364	**2.75**	0.3401	8.639	0.1700	4.3193	0.1082	2.750
10	2.5400	0.3142	7.981	0.1571	3.9903	0.1000	2.540
10.1600	**2.50**	0.3092	7.854	0.1546	3.9268	0.0984	2.500
11	2.3091	0.2856	7.254	0.1428	3.6271	0.0909	2.309
11.2889	**2.25**	0.2783	7.069	0.1391	3.5344	0.0886	2.250
12	2.1167	0.2618	6.646	0.1309	3.3325	0.0833	2.117
12.7000	**2**	0.2474	6.284	0.1236	3.1420	0.0787	2.000
13	1.9538	0.2417	6.139	0.1208	3.0696	0.0769	1.954
14	1.8143	0.2244	5.700	0.1122	2.8500	0.0714	1.814
14.5143	**1.75**	0.2164	5.497	0.1082	2.7489	0.0689	1.750
15	1.6933	0.2094	5.319	0.1047	2.6599	0.0667	1.693
16	1.5875	0.1964	4.986	0.0982	2.4936	0.0625	1.587
16.9333	**1.5**	0.1855	4.712	0.0927	2.3562	0.0591	1.500
18	1.4111	0.1745	4.432	0.0873	2.2166	0.0556	1.411
20	1.2700	0.1571	3.990	0.0785	1.9949	0.0500	1.270
20.3200	**1.25**	0.1546	3.927	0.0773	1.9635	0.0492	1.250
22	1.1545	0.1428	3.627	0.0714	1.8136	0.0455	1.155
24	1.0583	0.1309	3.325	0.0655	1.6624	0.0417	1.058
25.4000	**1**	0.1237	3.142	0.0618	1.5708	0.0394	1.000
28	0.90701	0.1122	2.850	0.0561	1.4249	0.0357	0.9071
28.2222	**0.9**	0.1113	2.827	0.0556	1.4137	0.0354	0.9000
30	0.84667	0.1047	2.659	0.0524	1.3329	0.0333	0.8467
31.7500	**0.8**	0.0989	2.513	0.04945	1.2566	0.0315	0.8000
32	0.79375	0.0982	2.494	0.04909	1.2468	0.0313	0.7937
33.8667	**0.75**	0.0928	2.357	0.04638	1.1781	0.0295	0.7500
36	0.70556	0.0873	2.217	0.04363	1.1083	0.0278	0.7056
36.2857	**0.7**	0.0865	2.199	0.04325	1.0996	0.0276	0.7000
40	0.63500	0.0785	1.994	0.03927	0.9975	0.0250	0.6350
42.3333	**0.6**	0.0742	1.885	0.03710	0.9423	0.0236	0.6000
44	0.57727	0.0714	1.814	0.03570	0.9068	0.0227	0.5773
48	0.52917	0.0655	1.661	0.03272	0.8311	0.0208	0.5292
50	0.50800	0.0628	1.595	0.03141	0.7976	0.0200	0.5080
50.8000	**0.5**	0.06184	1.5707	0.03092	0.7854	0.0197	0.5000
63.5000	**0.4**	0.04947	1.2565	0.02473	0.6283	0.0157	0.4000
64	0.39688	0.04909	1.2469	0.02454	0.6234	0.0156	0.3969
67,7333	0.375	0.04638	1.1781	0.02319	0.5890	0.0148	0.3750
72	0.35278	0.04363	1.1082	0.02182	0.5541	0.0139	0.3528
72.5714	0.35	0.04329	1.0996	0.02164	0.5498	0.0138	0.3500
78.1538	0.325	0.04020	1.0211	0.02010	0.5105	0.0128	0.3250
80	0.31750	0.03927	0.9975	0.01964	0.4987	0.0125	0.3175
84.6667	**0.3**	0.03711	0.9426	0.01856	0.4713	0.0118	0.3000
92.3636	0.275	0.03401	0.8639	0.01700	0.4319	0.0108	0.2750
96	0.26458	0.03272	0.8311	0.01636	0.4156	0.0104	0.2646
101.6000	0.25	0.03092	0.7854	0.01546	0.3927	0.00984	0.2500
120	0.21167	0.02618	0.6650	0.01309	0.3325	0.00833	0.2117
125	0.20320	0.02513	0.6383	0.01256	0.3192	0.00800	0.2032
127.0000	**0.2**	0.02474	0.6284	0.01237	0.3142	0.00787	0.2000
150	0.16933	0.02094	0.5319	0.01047	0.2659	0.00667	0.1693
169.3333	0.15	0.01855	0.4712	0.00928	0.2356	0.00591	0.1500
180	0.14111	0.01745	0.4432	0.00873	0.2216	0.00555	0.1411
200	0.12700	0.01571	0.3990	0.00786	0.1995	0.00500	0.1270
203.2000	0.125	0.01546	0.3927	0.00773	0.1963	0.00492	0.1250

FUNDAMENTAL RELATIONSHIPS OF SPUR AND HELICAL GEARS

Center distance is the distance between axes of mating gears and is determined from $C = (n_G + N_P)/(2P_d)$, or $C = (D_G + D_P)/2$. Deviation from ideal center distance of involute gears is not detrimental to proper (conjugate) gear action which is one of the prime superiority features of the involute tooth form.

Contact Ratio Referring to the top part of Fig. 8.3.4 and assuming no tip relief, the pinion engages in the gear at *a*, where the outside circle of the gear tooth intersects the line of action *ac*. For the usual spur gear and pinion combinations there will be two pairs of teeth theoretically in contact at engagement (a gear tooth and its mating pinion tooth considered as a pair). This will continue until the pair ahead (bottom part of Fig. 8.3.4) disengages at *c*, where the outside circle of the pinion intersects the line of action *ac*, the movement along the line of action being *ab*. After disengagement the pair behind will be the only pair in contact until another pair engages, the movement along the line of action for single-pair contact being *bd*. Two pairs are theoretically in contact during the remaining intervals, $ab + dc$.

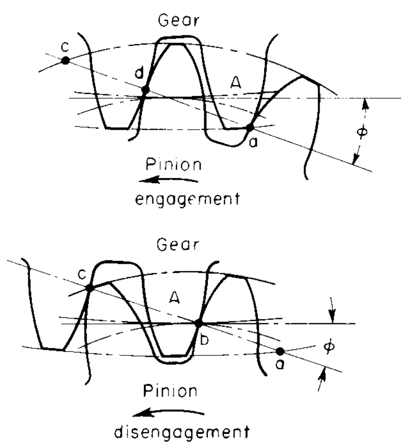


Fig. 8.3.4 Contact conditions at engagement and disengagement.

Contact ratio expresses the average number of pairs of teeth theoretically in contact and is obtained numerically by dividing the length of the line of action by the normal pitch. For full-depth teeth, without undercutting, the contact ratio is $m_p = (\sqrt{D_0^2 - D_b^2} + \sqrt{d_0^2 - d_b^2} - 2C \sin \phi)/(2p \cos \phi)$. The result will be a mixed number with the integer portion the number of pairs of teeth always in contact and carrying load, and the decimal portion the amount of time an additional pair of teeth are engaged and share load. As an example, for m_p between 1 and 2:

Load is carried by one pair, $(2 - m_p)/m_p$ of the time.

Load is carried by two pairs, $2(m_p - 1)/m_p$ of the time.

In Figs. 8.3.5 to 8.3.7, contact ratios are given for standard generated gears, the lower part of Figs. 8.3.5 and 8.3.6 representing the effect of undercutting.

These charts are applicable to both standard dimetral pitch gears made in accordance with American standards and also standard metric gears that have an addendum of one module.

Tooth Thickness For standard gears, the tooth thickness t of mating gears is equal, where $t = p/2 = \pi/(2P_d)$ measured linearly along the arc of the pitch circle. The tooth thickness t_1 at any radial point of the tooth (at diameter D_1) can be calculated from the known thickness t at the pitch radius $D/2$ by the relationship $t_1 = t(D_1/D) - D_1$ (inv ϕ_1 − inv ϕ), where inv $\phi = \tan \phi - \phi$ = involute function. Units for ϕ must be radians. Tables of values for inv ϕ from 0 to 45° can be found in the references (Buckingham and Dudley).

Over-plus measurements (spur gears) are another means of deriving tooth thickness. If cylindrical pins are inserted in tooth spaces diametrically opposite one another (or nearest space for an odd number of teeth) (Fig. 8.3.8), the tooth thickness can be derived from the measurement M as follows:

$$t = D(\pi/N + \text{inv } \phi_1 - \text{inv } \phi - d_w/D_b)$$
$$\cos \phi_1 = (D \cos \phi)/2R_c$$
$$R_c = (M - d_w)/2 \qquad \text{for even number of teeth}$$
$$R_c = (M - d_w)/[2 \cos (90/N)] \qquad \text{for odd number of teeth}$$

where d_w = pin diameter, R_c = distance from gear center to center of pin, and M = measurement over pins.

For the reverse situation, the over-pins measurement M can be found for a given tooth thickness t at diameter D and pressure angle ϕ by the following: inv $\phi_1 = t/D +$ inv $\phi + d_w/(D \cos \phi) - \pi/N$, $M = D \cos \phi/\cos \phi_1 + d_w$ (for even number of teeth), $M = (D \cos \phi/\cos \phi_1) \cos (90°/N) + d_w$ (for odd number of teeth).

Table values of over-pins measures (see Dudley and Van Keuren) facilitate measurements for all standard gears including those with slight departures from standard. (For correlation with tooth thickness and testing radius, see Michalec, *Product Eng.*, May 1957, and ''Precision Gearing: Theory and Practice,'' Wiley.)

Testing radius R_T is another means of determining tooth thickness and refers to the effective pitch radius of the gear when rolled intimately with a master gear of known size calibration. (See Michalec, *Product Eng.*, Nov. 1956, and ''Precision Gearing: Theory and Practice,'' op. cit.) For standard design gears the testing radius equals the pitch radius. The testing radius may be corrected for small departures Δt from ideal tooth thickness by the relationship, $R_T = R + \Delta t/2 \tan \phi$, where $\Delta t = t_1 - t$ and is positive and negative respectively for thicker and thinner tooth thicknesses than standard value t.

Backlash B is the amount by which the width of a tooth space exceeds the thickness of the engaging tooth measured on the pitch circle. Backlash does not adversely affect proper gear function except for lost mo-

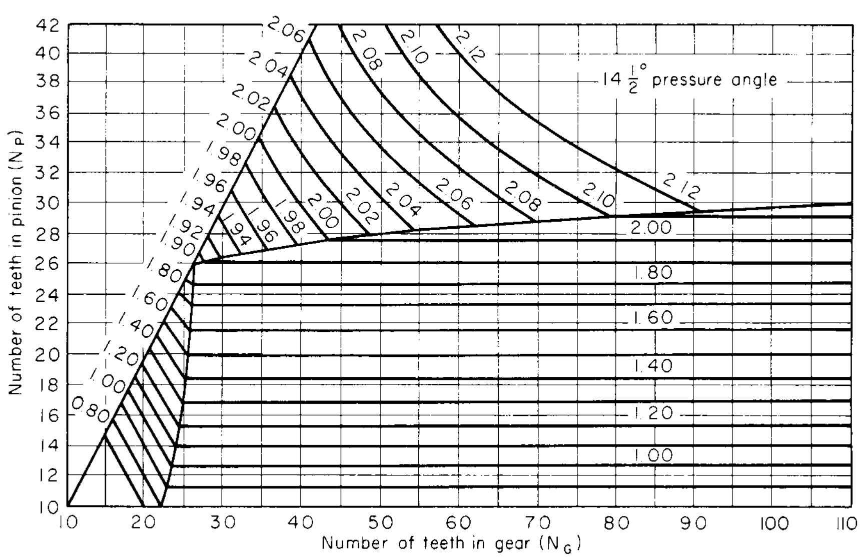


Fig. 8.3.5 Contact ratio, spur gear pairs—full depth, standard generated teeth, 14½° pressure angle.

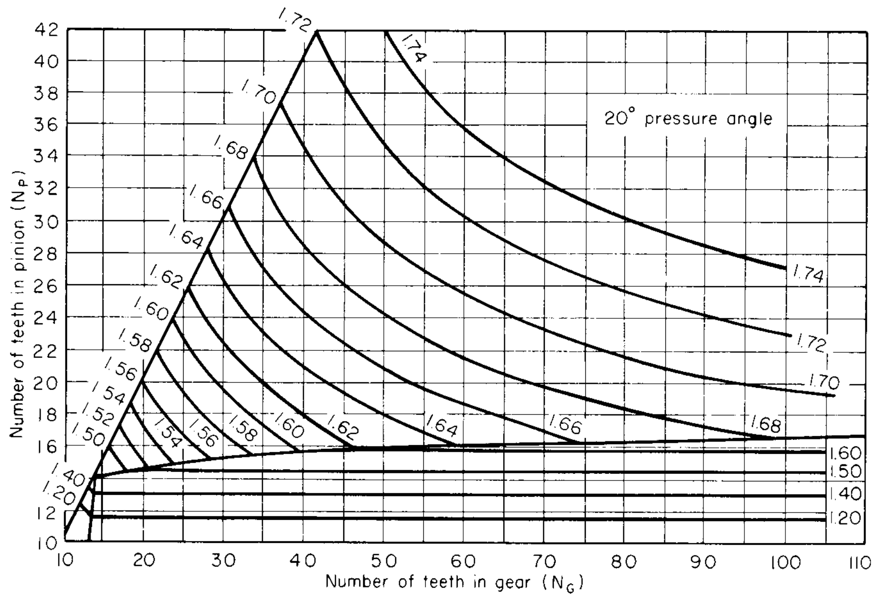


Fig. 8.3.6 Contact ratio, spur gear pairs—full-depth standard generated teeth, 20° pressure angle.

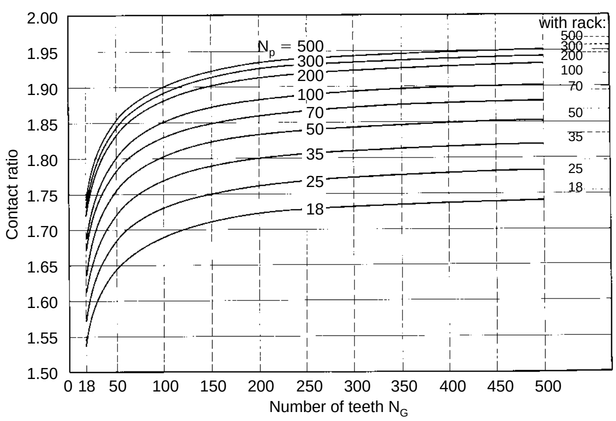


Fig. 8.3.7 Contact ratio for large numbers of teeth—spur gear pairs, full-depth standard teeth, 20° pressure angle. *(Data by R. Feeney and T. Wall.)*

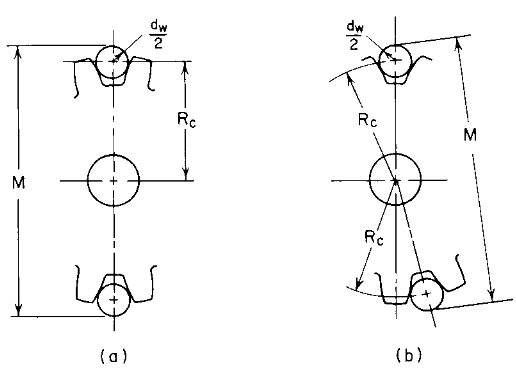


Fig. 8.3.8 Geometry of over-pins measurements (*a*) for an even number of teeth and (*b*) for an odd number of teeth.

tion upon reversal of gear rotation. Backlash inevitably occurs because of necessary fabrication tolerances on tooth thickness and center distance plus need for clearance to accommodate lubricant and thermal expansion. Proper backlash can be introduced by a specified amount of tooth thinning or slight increase in center distance. The relationship between small change in center distance $\overline{\Delta C}$ and backlash is $B = 2\,\overline{\Delta C}\tan\phi$ (see Michalec, ''Precision Gearing: Theory and Practice'').

Total composite error (tolerance) is a measure of gear quality in terms of the net sum of irregularity of its testing radius R_T due to pitch-circle runout and tooth-to-tooth variations (see Michalec, op. cit.).

Tooth-to-tooth composite error (tolerance) is the variation of testing radius R_T between adjacent teeth caused by tooth spacing, thickness, and profile deviations (see Michalec, op. cit.).

Profile shifted gears have tooth thicknesses that are significantly different from nominal standard value; excluded are deviations caused by normal allowances and tolerances. They are also known as **modified gears, long and short addendum gears,** and **enlarged gears.** They are produced by cutting the teeth with standard cutters at enlarged or reduced outside diameters. The result is a relative shift of the two families of involutes forming the tooth profiles, simultaneously with a shift of the tooth radially outward or inward (see Fig. 8.3.9). Calculation of operating conditions and tooth parameters are

$$C_1 = \frac{(C\cos\phi)}{\cos\phi_1}$$

$$\text{inv}\,\phi_1 = \text{inv}\,\phi + \frac{N_P(t'_G + t'_P) - \pi D_P}{D_P(N_P + N_G)}$$

$$t'_G = t + 2X_G\tan\phi$$
$$t'_P = t + 2X_P\tan\phi$$
$$D'_G = (N_G/P_d) + 2X_G$$
$$D'_P = (N_P/P_d) + 2X_P$$
$$D'_o = D'_G + (2/P_d)$$
$$d'_o = D'_P + (2/P_d)$$

where ϕ = standard pressure angle, ϕ_1 = operating pressure angle, C = standard center distance = $(N_G + N_P)/2P_d$, C_1 = operating center distance, X_G = profile shift correction of gear, and X_P = profile shift correction of pinion. The quantity X is positive for enlarged gears and negative for thinned gears.

Table 8.3.5 Metric Spur Gear Design Formulas

To obtain:	From known	Use this formula*
Pitch diameter D	Module; diametral pitch	$D = mN$
Circular pitch p_c	Module; diametral pitch	$p_c = m\pi = \frac{D}{N}\pi = \frac{\pi}{P}$
Module m	Diametral pitch	$m = \frac{25.4}{P}$
No. of teeth N	Module and pitch diameter	$N = \frac{D}{m}$
Addendum a	Module	$a = m$
Dedendum b	Module	$b = 1.25m$
Outside diameter D_o	Module and pitch diameter or number of teeth	$D_o = D + 2m = m(N + 2)$
Root diameter D_r	Pitch diameter and module	$D_r = D - 2.5m$
Base circle diameter D_b	Pitch diameter and pressure angle ϕ	$D_b = D \cos \phi$
Base pitch p_b	Module and pressure angle	$p_b = m\pi \cos \phi$
Tooth thickness at standard pitch diameter T_{std}	Module	$T_{\text{std}} = \frac{\pi}{2} m$
Center distance C	Module and number of teeth	$C = \frac{m(N_1 + N_2)}{2}$
Contact ratio m_p	Outside radii, base-circle radii, center distance, pressure angle	$m_p = \frac{\sqrt{{}_1R_o^2 - {}_1R_b^2} + \sqrt{{}_2R_o^2 - {}_2R_b^2} - C \sin \phi}{m\pi \cos \phi}$
Backlash (linear) B (along pitch circle)	Change in center distance	$B = 2(\Delta C) \tan \phi$
Backlash (linear) B (along pitch circle)	Change in tooth thickness, T	$B = \Delta T$
Backlash (linear) (along line of action) B_{LA}	Linear backlash (along pitch circle)	$B_{LA} = B \cos \phi$
Backlash (angular) B_a	Linear backlash (along pitch circle)	$B_a = 6{,}880 \frac{B}{D}$ (arc minutes)
Min. number teeth for no undercutting, N_c	Pressure angle	$N_c = \frac{2}{\sin^2 \phi}$

* All linear dimensions in millimeters.

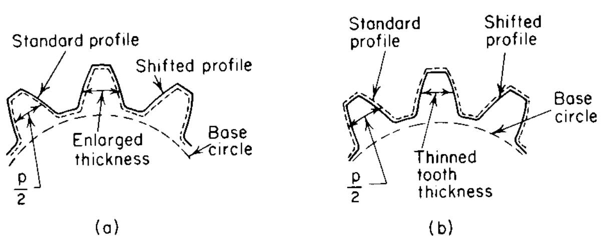


Fig. 8.3.9 Geometry of profile-shifted teeth. (*a*) Enlarged case; (*b*) thinned tooth thickness case.

Metric Module Gear Design Equations Basic design equations for spur gearing utilizing the metric module are listed in Table 8.3.5. (See Designatronics, ''Elements of Metric Gear Technology.'')

HELICAL GEARS

Helical gears divide into two general applications: for driving parallel shafts and for driving skew shafts (mostly at right angles), the latter often referred to as *crossed-axis* helical gears. The helical tooth form may be imagined as consisting of an infinite number of staggered laminar spur gears, resulting in the curved cylindrical helix.

Pitch of helical gears is definable in two planes. The diametral and circular pitches measured in the plane of rotation (transverse) are defined as for spur gears. However, pitches measured normal to the tooth are related by the cosine of the helix angle; thus normal diametral pitch $= P_{dn} = P_d/\cos \psi$, normal circular pitch $= p_n = p \cos \psi$, and $P_{dn}p_n = \pi$. **Axial pitch** is the distance between corresponding sides of adjacent teeth measured parallel to the gear axis and is calculated as $p_a = p \cot \psi$.

Pressure angle of helical gears is definable in the normal and transverse planes by $\tan \phi_n = \tan \phi \cos \psi$. The transverse pressure angle, which is effectively the real pressure angle, is always greater than the normal pressure angle.

Tooth thickness t of helical gears can be measured in the plane of rotation, as with spur gears, or normal to the tooth surface t_n. The relationship of the two thicknesses is $t_n = t \cos \psi$.

Over-Pins Measurement of Helical Gears Tooth thicknesses t at diameter d can be found from a known over-pins measurement M at known pressure angle ϕ, corresponding to diameter D as follows:

$$R_c = (M - d_w)/2 \qquad \text{for even number of teeth}$$
$$R_c = (M - d_w)/[2 \cos (180/2N)] \qquad \text{for odd number of teeth}$$
$$\cos \phi_1 = (D \cos \phi)/2R_c$$
$$\tan \phi_n = \tan \phi \cos \psi$$
$$\cos \psi_b = \sin \phi_n / \sin \phi$$
$$t = D[\pi/N + \text{inv } \phi_1 - \text{inv } \phi - d_w/(D \cos \phi \cos \psi_b)]$$

Parallel-shaft helical gears must conform to the same conditions and requirements as spur gears with parameters (pressure angle and pitch) consistently defined in the transverse plane. Since standard spur gear cutting tools are usually used, normal plane values are standard, resulting in nonstandard transverse pitches and nonstandard pitch diameters and center distances. For parallel shafts, helical gears must have identical helix angles, but must be of opposite hand (left and right helix directions). The commonly used helix angles range from 15 to 35°. To make most advantage of the helical form, the advance of a tooth should be greater than the circular pitch; recommended ratio is 1.5 to 2 with 1.1 minimum. This overlap provides two or more teeth in continual contact with resulting greater smoothness and quietness than spur gears. Because of the helix, the normal component of the tangential pressure on the teeth produces end thrust of the shafts. To remove this objection, gears are made with helixes of opposite hand on each half of the face and are then known as herringbone gears (see Fig. 8.3.10).

Crossed-axis helical gears, also called *spiral* or *screw gears* (Fig. 8.3.11), are a simple type of involute gear used for connecting nonparallel, nonintersecting shafts. Contact is point and there is considerably more sliding than with parallel-axis helicals, which limits the load capacity. The individual gear of this mesh is identical in form and specification to a parallel-shaft helical gear. Crossed-axis helicals can connect

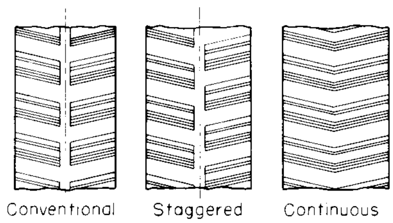


Fig. 8.3.10 Herringbone gears.

any shaft angle Σ, although 90° is prevalent. Usually, the helix angles will be of the same hand, although for some extreme cases it is possible to have opposite hands, particularly if the shaft angle is small.

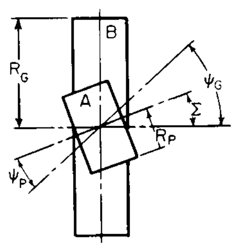


Fig. 8.3.11 Crossed-axis helical gears.

Helical Gear Calculations For parallel shafts the center distance is a function of the helix angle as well as the number of teeth, that is, $C = (N_G + N_P)/(2P_{dn}\cos\psi)$. This offers a powerful method of gearing shafts at any specified center distance to a specified velocity ratio. For crossed-axis helicals the problem of connecting a pair of shafts for any velocity ratio admits of a number of solutions, since both the pitch radii and the helix angles contribute to establishing the velocity ratio. The formulas given in Tables 8.3.6 and 8.3.7 are of assistance in calculations. The notation used in these tables is as follows:

$N_P(N_G)$ = number of teeth in pinion (gear)
$D_P(D_G)$ = pitch diam of pinion (gear)
$p_P(p_G)$ = circular pitch of pinion (gear)
p = circular pitch in plane of rotation for both gears
P_d = diametral pitch in plane of rotation for both gears
p_n = normal circular pitch for both gears
P_{dn} = normal diametral pitch for both gears
ψ_G = tooth helix angle of gear
ψ_P = tooth helix angle of pinion
$l_P(l_G)$ = lead of pinion (gear)
= lead of tooth helix
$n_P(n_G)$ = r/min of pinion (gear)
Σ = angle between shafts in plan
C = center distance

Table 8.3.6 Helical Gears on Parallel Shafts

To find:	Formula
Center distance C	$\dfrac{N_G + N_P}{2P_{dn}\cos\psi}$
Pitch diameter D	$\dfrac{N}{P_d} = \dfrac{N}{P_{dn}\cos\psi}$
Normal diametral pitch P_{dn}	$\dfrac{P_d}{\cos\psi}$
Normal circular pitch p_n	$p\cos\psi$
Pressure angle ϕ	$\tan^{-1}\dfrac{\tan\phi_n}{\cos\psi}$
Contact ratio m_p	$\dfrac{\sqrt{{}_GD_o^2 - {}_GD_b^2} + \sqrt{{}_PD_o^2 - {}_PD_b^2} + 2C\sin\phi}{2p\cos\phi} + \dfrac{F\sin\psi}{p_n}$
Velocity ratio m_G	$\dfrac{N_G}{N_P} = \dfrac{D_G}{D_P}$

Table 8.3.7 Crossed Helical Gears on Skew Shafts

To find:	Formula
Center distance C	$\dfrac{P_n}{2\pi}\left(\dfrac{N_G}{\cos\psi_G} + \dfrac{N_P}{\cos\psi_P}\right)$
Pitch diameter D_G, D_P	$D_G = \dfrac{N_G}{P_{dG}} = \dfrac{N_G}{P_{dn}\cos\psi_G} = \dfrac{N_G p_n}{\pi\cos\psi_G}$
	$D_P = \dfrac{N_P}{P_{dP}} = \dfrac{N_P}{P_{dn}\cos\psi_P} = \dfrac{n_P p_n}{\pi\cos\psi_P}$
Gear ratio m_G	$\dfrac{N_G}{N_P} = \dfrac{D_G\cos\psi_G}{D_P\cos\psi_P}$
Shaft angle Σ	$\psi_G + \psi_P$

NONSPUR GEAR TYPES*

Bevel gears are used to connect two intersecting shafts in any given speed ratio. The tooth shapes may be designed in any of the shapes shown in Fig. 8.3.12. A special type of gear known as a **hypoid** was developed by Gleason Works for the automotive industry (see *Jour. SAE,* **18,** no. 6). Although similar in appearance to a spiral bevel, it is not a true bevel gear. The basic pitch rolling surfaces are hyperbolas of revolution. Because a ''spherical involute'' tooth form has a curved crown tooth (the basic tool for generating all bevel gears), Gleason used a straight-sided crown tooth which resulted in bevel gears differing slightly from involute form. Because of the figure 8 shape of the complete theoretical tooth contact path, the tooth form has been called ''octoid.'' Straight-sided bevel gears made by reciprocating cutters are of this type. Later, when curved teeth became widely used (spiral and Zerol), practical limitations of such cutters resulted in introduction of the ''spherical'' tooth form which is now the basis of all curved tooth bevel gears. (For details see Gleason's publication, ''Guide to Bevel Gears.'') Gleason Works also developed the generated tooth form

* In the following text relating to bevel gearing, all tables and figures have been extracted from Gleason Works publications, with permission.

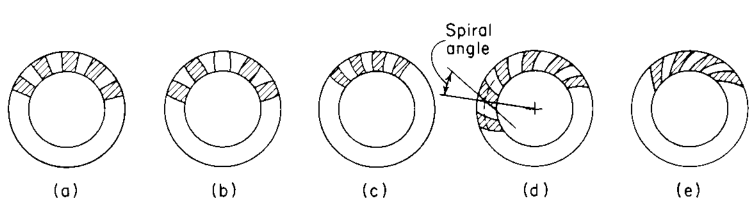


Fig. 8.3.12 Bevel gear types. (*a*) Old-type straight teeth; (*b*) modern Coniflex straight teeth (exaggerated crowning); (*c*) Zerol teeth; (*d*) spiral teeth; (*e*) hypoid teeth.

Revecycle and the nongenerated tooth forms Formate and Helixform, used principally for mass production of hypoid gears for the automotive industry.

Referring to Fig. 8.3.13, we see that the pitch surfaces of bevel gears are frustums of cones whose vertices are at the intersection of the axes; the essential elements and definitions follow.

Addendum angle α: The angle between elements of the face cone and pitch cone.

Back angle: The angle between an element of the back cone and a plane of rotation. It is equal to the pitch angle.

Back cone: The angle of a cone whose elements are tangent to a sphere containing a trace of the pitch circle.

Back-cone distance: The distance along an element of the back cone from the apex to the pitch circle.

Cone distance A_o: The distance from the end of the tooth (heel) to the pitch apex.

Crown: The sharp corner forming the outside diameter.

Crown-to-back: The distance from the outside diameter edge (crown) to the rear of the gear.

Dedendum angle δ: The angle between elements of the root cone and pitch cone.

Face angle γ_o: The angle between an element of the face cone and its axis.

Face width F: The length of teeth along the cone distance.

Front angle: The angle between an element of the front cone and a plane of rotation.

Generating mounting surface, GMS: The diameter and/or plane of rotation surface or shaft center which is used for locating the gear blank during fabrication of the gear teeth.

Heel: The portion of a bevel gear tooth near the outer end.

Mounting distance, MD: For assembled bevel gears, the distance from the crossing point of the axes to the registering surface, measured along the gear axis. Ideally, it should be identical to the pitch apex to back.

Mounting surface, MS: The diameter and/or plane of rotation surface which is used for locating the gear in the application assembly.

Octoid: The mathematical form of the bevel tooth profile. Closely resembles a spherical involute but is fundamentally different.

Pitch angle Γ: The angle formed between an element of the pitch cone and the bevel gear axis. It is the half angle of the pitch.

Pitch apex to back: The distance along the axis from apex of pitch cone to a locating registering surface on back.

Registering surface, RS: The surface in the plane of rotation which locates the gear blank axially in the generating machine and the gear in application. These are usually identical surfaces, but not necessarily so.

Root angle γ_R: The angle formed between a tooth root element and the axis of the bevel gear.

Shaft angle Σ: The angle between mating bevel-gear axes; also, the sum of the two pitch angles.

Spiral angle ψ: The angle between the tooth trace and an element of the pitch cone, corresponding to helix angle in helical gears. The spiral angle is understood to be at the mean cone distance.

Toe: The portion of a bevel tooth near the inner end.

Bevel gears are described by the parameter dimensions at the large end (heel) of the teeth. Pitch, pitch diameter, and tooth dimensions, such as addendum are measurements at this point. At the large end of the gear, the tooth profiles will approximate those generated on a spur gear pitch circle of radius equal to the back cone distance. The formative number of teeth is equal to that contained by a complete spur gear. For pinion and gear, respectively, this is $T_P = N_P/\cos\gamma$; $T_G = N_G/\cos\Gamma$, where T_P and T_G = formative number teeth and N_P and N_G = actual number teeth.

Although bevel gears can connect intersecting shafts at any angle, most applications are for right angles. When such bevels are in a 1 : 1 ratio, they are called **mitre gears.** Bevels connecting shafts other than 90° are called **angular bevel gears.** The speeds of the shafts of bevel gears are determined by $n_P/n_G = \sin\Gamma/\sin\gamma$, where $n_P(n_G)$ = r/min of pinion (gear), and $\gamma(\Gamma)$ = pitch angle of pinion (gear).

All standard bevel gear designs in the United States are in accordance with the **Gleason bevel gear system.** This employs a basic pressure angle of 20° with long and short addendums for ratios other than 1 : 1 to avoid undercut pinions and to increase strength.

20° Straight Bevel Gears for 90° Shaft Angle Since straight bevel gears are the easiest to produce and offer maximum precision, they are frequently a first choice. Modern straight-bevel-gears generators produce a tooth with localized tooth bearing designated by the Gleason registered tradename Coniflex. These gears, produced with a circular cutter, have a slightly crowned tooth form (see Fig. 8.3.12*b*). Because of the superiority of Coniflex bevel gears over the earlier reciprocating cutter produced straight bevels and because of their faster production, they are the standards for all bevel gears. The design parameters of Fig. 8.3.13 are calculated by the formulas of Table 8.3.8. Backlash data are given in Table 8.3.9.

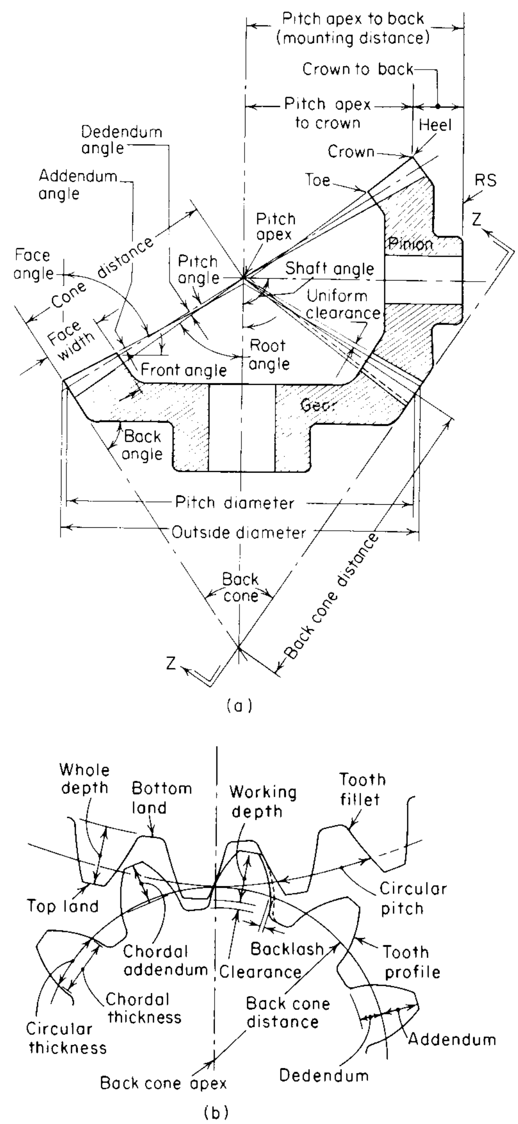


Fig. 8.3.13 Geometry of bevel gear nomenclature. (*a*) Section through axes; (*b*) view along axis *Z*/*Z*.

Table 8.3.8 Straight Bevel Gear Dimensions*
(All linear dimensions in inches)

1. Number of pinion teeth†	n	5. Working depth	$h_k = \dfrac{2.000}{P_d}$
2. Number of gear teeth†	N	6. Whole depth	$h_i = \dfrac{2.188}{P_d} + 0.002$
3. Diametral pitch	P_d	7. Pressure angle	ϕ
4. Face width	F	8. Shaft angle	Σ

	Pinion	Gear
9. Pitch diameter	$d = \dfrac{n}{P_d}$	$D = \dfrac{N}{P_d}$
10. Pitch angle	$\gamma = \tan^{-1} \dfrac{n}{N}$	$\Gamma = 90° - \gamma$
11. Outer cone distance	$A_O = \dfrac{D}{2 \sin \Gamma}$	
12. Circular pitch	$p = \dfrac{3.1416}{P_d}$	
13. Addendum	$a_{OP} = h_k - a_{OG}$	$a_{OG} = \dfrac{0.540}{P_d} + \dfrac{0.460}{P_d(N/n)^2}$
14. Dedendum‡	$b_{OP} = \dfrac{2.188}{P_d} - a_{OP}$	$b_{OG} = \dfrac{2.188}{P_d} - a_{OG}$
15. Clearance	$c = h_t - h_k$	
16. Dedendum angle	$\delta_P = \tan^{-1} \dfrac{b_{OP}}{A_O}$	$\delta_G = \tan^{-1} \dfrac{b_{OG}}{A_O}$
17. Face angle of blank	$\gamma_O = \gamma + \delta_G$	$\Gamma_O = \Gamma + \delta_P$
18. Root angle	$\gamma_R = \gamma - \delta_P$	$\Gamma_R = \Gamma - \delta_G$
19. Outside diameter	$d_O = d + 2a_{OP} \cos \gamma$	$D_O = D + 2a_{OG} \cos \Gamma$
20. Pitch apex to crown	$x_O = \dfrac{D}{2} - a_{OP} \sin \gamma$	$X_O = \dfrac{d}{2} - a_{OG} \sin \Gamma$
21. Circular thickness	$t = p - T$	$T = \dfrac{p}{2} - (a_{OP} - a_{OG}) \tan \phi - \dfrac{K}{P_d}$ (see Fig. 8.3.14)
22. Backlash	B See Table 8.3.9	
23. Chordal thickness	$t_C = t - \dfrac{t^2}{6d^2} - \dfrac{B}{2}$	$T_C = T - \dfrac{T^2}{6D^2} - \dfrac{B}{2}$
24. Chordal addendum	$a_{CP} = a_{OP} + \dfrac{t^2 \cos \gamma}{4d}$	$a_{CG} = a_{OG} + \dfrac{T^2 \cos \Gamma}{4D}$
25. Tooth angle	$\dfrac{3{,}438}{A_O}\left(\dfrac{t}{2} + b_{OP} \tan \phi\right)$ minutes	$\dfrac{3{,}438}{A_O}\left(\dfrac{T}{2} + b_{OG} \tan \phi\right)$ minutes
26. Limit-point width (L.F.)	$W_{LOP} = (T - 2b_{OP} \tan \phi) - 0.0015$	$W_{LOG} = (t - 2b_{OG} \tan \phi) - 0.0015$
27. Limit-point width (S.E.)	$W_{LiP} = \dfrac{A_O - F}{A_O}(T - 2b_{OP} \tan \phi) - 0.0015$	$W_{LiG} = \dfrac{A_O - F}{A_O}(t - 2b_{OG} \tan \phi) - 0.0015$
28. Tool-point width	$W = W_{LiP}$ − stock allowance	$W = W_{LiG}$ − stock allowance

* Abstracted from ''Gleason Straight Bevel Gear Design,'' Tables 8.3.8 and 8.3.9 and Fig. 8.3.4. Gleason Works, Inc.
† Numbers of teeth; ratios with 16 or more teeth in pinion: 15/17 and higher; 14/20 and higher; 13/31 and higher. These can be cut with 20° pressure angle without undercut.
‡ The actual dedendum will be 0.002 in greater than calculated.

Table 8.3.9 Recommended Normal Backlash for Bevel Gear Meshes*

P_d	Backlash range	P_d	Backlash range
1.00–1.25	0.020–0.030	3.50–4.00	0.007–0.009
1.25–1.50	0.018–0.026	4–5	0.006–0.008
1.50–1.75	0.016–0.022	5–6	0.005–0.007
1.75–2.00	0.014–0.018	6–8	0.004–0.006
2.00–2.50	0.012–0.016	8–10	0.003–0.005
2.50–3.00	0.010–0.013	10–12	0.002–0.004
3.00–3.50	0.008–0.011	Finer than 12	0.001–0.003

* The table gives the recommended normal backlash for gears assembled ready to run. Because of manufacturing tolerances and changes resulting from heat treatment, it is frequently necessary to reduce the theoretical tooth thickness by slightly more than the tabulated backlash in order to obtain the correct backlash in assembly. In case of choice, use the smaller backlash tolerances.

Angular straight bevel gears connect shaft angles other than 90° (larger or smaller), and the formulas of Table 8.3.8 are not entirely applicable, as shown in the following:

Item 8, shaft angle, is the specified non-90° shaft angle.

Item 10, pitch angles. Shaft angle Σ less than 90°, $\tan\gamma = \sin\Sigma/(N/n + \cos\Sigma)$; shaft angle Σ greater than 90°, $\tan\gamma = \sin(180 - \Sigma)/[N/n - \cos(180° - \Sigma)]$.

For all shaft angles, $\sin\gamma/\sin\Gamma = n/N$; $\Gamma = \Sigma - \gamma$.

Item 13, addendum, requires calculation of the equivalent 90° bevel gear ratio m_{90}, $m_{90} = [N\cos\gamma/(n\cos\Gamma)]^{1/2}$. The value m_{90} is used as the ratio N/n when applying the formula for addendum. The quantity under the radical is always the absolute value and is therefore always positive.

Item 20, pitch apex to crown, $x_o = A_0\cos\gamma - a_{op}\sin\gamma$, $X_o = A_0\cos\Gamma - a_{oG}\sin\Gamma$.

Item 21, circular thickness, except for high ratios, K may be zero.

Spiral Bevel Gears for 90° Shaft Angle The spiral curved teeth produce additional overlapping tooth action which results in smoother gear action, lower noise, and higher load capacity. The spiral angle has been standardized by Gleason at 35°. Design parameters are calculated by formulas of Table 8.3.10.

Angular Spiral Bevel Gears Several items deviate from the formulas of Table 8.3.10 in the same manner as angular straight bevel gears. Therefore, the same formulas apply for the deviating items with only the following exception:

Item 21, circular thickness, the value of K in Fig. 8.3.15 must be determined from the equivalent 90° bevel ratio (m_{90}) and the equivalent 90° bevel pinion. The latter is computed as $n_{90} = n\sin\Gamma_{90}/\cos\gamma$, where $\tan\Gamma_{90} = m_{90}$.

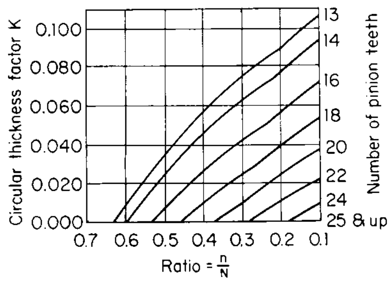


Fig. 8.3.14 Circular thickness factor for straight bevel gears.

Table 8.3.10 Spiral Bevel Gear Dimensions
(All linear dimensions in inches)

1. Number of pinion teeth	n	5. Working depth	$h_k = \dfrac{1.700}{P_d}$
2. Number of gear teeth	N	6. Whole depth	$h_i = \dfrac{1.888}{P_d}$
3. Diametral pitch	P_d	7. Pressure angle	ϕ
4. Face width	F	8. Shaft angle	Σ

	Pinion	Gear
9. Pitch diameter	$d = \dfrac{n}{P_d}$	$D = \dfrac{N}{P_d}$
10. Pitch angle	$\gamma = \tan^{-1}\dfrac{n}{N}$	$1 = 90° - \gamma$
11. Outer cone distance	$A_O = \dfrac{D}{2\sin\Gamma}$	
12. Circular pitch	$p = \dfrac{3.1416}{P_d}$	
13. Addendum	$A_{OP} = h_k - a_{OG}$	$a_{OG} = \dfrac{0.460}{P_d} + \dfrac{0.390}{P_d(N/n)^2}$
14. Dedendum	$b_{OP} = h_t - a_{OP}$	$b_{OG} = h_t - a_{OG}$
15. Clearance	$c = h_t - h_k$	
16. Dedendum angle	$\delta_p = \tan^{-1}\dfrac{b_{OP}}{A_O}$	$\delta_G = \tan^{-1}\dfrac{b_{OG}}{A_O}$
17. Face angle of blank	$\gamma_O = \gamma + \delta_G$	$\Gamma_O = \Gamma + \delta_P$
18. Root angle	$\gamma_R = \gamma - \delta_P$	$\Gamma_R = \Gamma - \delta_G$
19. Outside diameter	$d_O = d + 2a_{OP}\cos\gamma$	$D_O = D + 2a_{OG}\cos\Gamma$
20. Pitch apex to crown	$x_O = \dfrac{D}{2} - a_{OP}\sin\gamma$	$X_O = \dfrac{d}{2} - a_{OG}\sin\Gamma$
21. Circular thickness	$t = p - T$	$T = \dfrac{p}{2}(a_{OP} - a_{OG})\dfrac{\tan\phi}{\cos\phi} - \dfrac{K}{P_d}$ (see Fig. 8.3.15)
22. Backlash	See Table 8.3.9	
23. Hand of spiral	Left or right	Right or left
24. Spiral angle	35°	
25. Driving member	Pinion or gear	
26. Direction of rotation	Clockwise or counterclockwise	

SOURCE: Gleason, ''Spiral Bevel Gear System.''

The zerol bevel gear is a special case of a spiral bevel gear and is limited to special applications. Design and fabrication details can be obtained from Gleason Works.

Hypoid gears are special and are essentially limited to automotive applications.

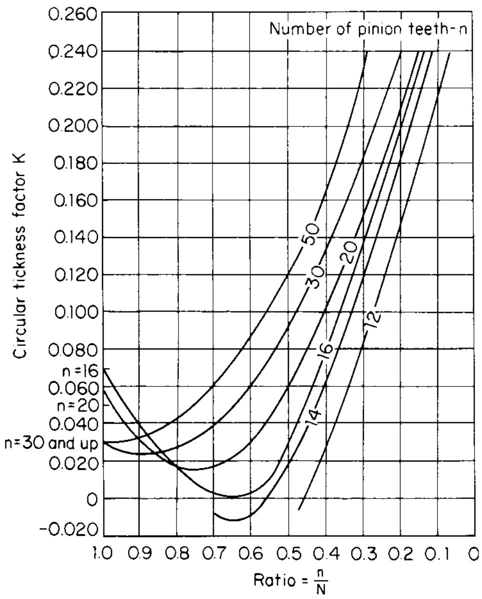


Fig. 8.3.15 Circular thickness factors for spiral bevel gears with 20° pressure angle and 35° spiral angle. Left-hand pinion driving clockwise or right-hand pinion driving counterclockwise.

WORMGEARS AND WORMS

Worm gearing is used for obtaining large speed reductions between nonintersecting shafts making an angle of 90° with each other. If a wormgear such as shown in Fig. 8.3.16 engages a straight worm, as shown in Fig. 8.3.17, the combination is known as **single enveloping worm gearing.** If a wormgear of the kind shown in Fig. 8.3.16 engages a worm as shown in Fig. 8.3.18, the combination is known as **double enveloping worm gearing.**

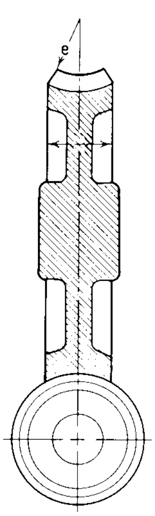


Fig. 8.3.16 Single enveloping worm gearing.

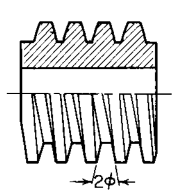


Fig. 8.3.17 Straight worm.

With worm gearing, the **velocity ratio** is the ratio between the number of teeth on the wormgear and the number of threads on the worm. Thus, a 30-tooth wormgear meshing with a single threaded worm will have a velocity ratio of 1:30; that is, the worm must make 30 rv in order to revolve the wormgear once. For a double threaded worm, there will be 15 rv of the worm to one of the wormgear, etc. High-velocity ratios are thus obtained with relatively small wormgears.

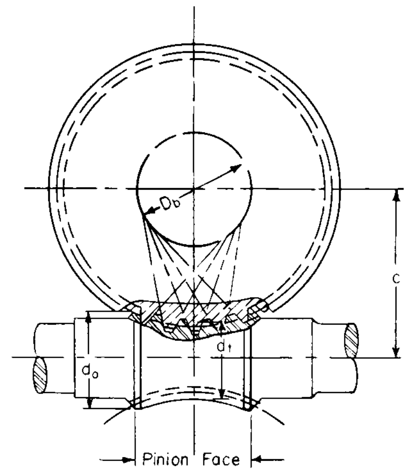


Fig. 8.3.18 Double enveloping worm gearing.

Tooth proportions of the worm in the central section (Fig. 8.3.17) follow standard rack designs, such as 14½, 20, and 25°. The mating wormgear is cut conjugate for a unique worm size and center distance. The geometry and related design equations for a straight-sided cylindrical worm are best seen from a development of the pitch plane (Fig. 8.3.19).

$$D_w = \text{pitch diameter of worm} = \frac{n_w p_n}{\pi \sin \lambda}$$

$$p_n = p \cos \lambda = \frac{\pi D_w}{n_w} \sin \lambda$$

$$L = \text{lead of worm} = n_w p$$

$$D_g = \text{pitch diameter of wormgear}$$

$$= \frac{N_g}{P_d} = \frac{pN_g}{\pi} = \frac{P_n N_g}{\pi \cos \lambda}$$

$$C = \text{center distance}$$

$$= \frac{D_w + D_g}{2} = \frac{p_n}{2\pi}\left(\frac{N_g}{\cos \lambda} + \frac{n_w}{\sin \lambda}\right)$$

where n_w = number of threads in worm; N_g = number of teeth in wormgear; Z = velocity ratio = N_g/n_w.

The pitch diameter of the wormgear is established by the number of teeth, which in turn comes from the desired gear ratio. The pitch diameter of the worm is somewhat arbitrary. The lead must match the wormgear's circular pitch, which can be satisfied by an infinite number of worm diameters; but for a fixed lead value, each worm diameter has a unique lead angle. AGMA offers a design formula that provides near optimized geometry:

$$D_w = \frac{C^{0.875}}{2.2}$$

where C = center distance. Wormgear face width is also somewhat arbitrary. Generally it will be ⅗ to ⅔ of the worm's outside diameter.

Worm mesh nonreversibility, a unique feature of some designs, occurs because of the large amount of sliding in this type of gearing. For a given coefficient of friction there is a critical value of lead angle below which the mesh is nonreversible. This is generally 10° and lower but is related to the materials and lubricant. Most single thread worm meshes are in this category. This locking feature can be a disadvantage or in some designs can be put to advantage.

Double enveloping worm gearing is special in both design and fabrication. Application is primarily where a high load capacity in small space is desired. Currently, there is only one source of manufacture in the United States: Cone Drive Division of Ex-Cello Corp. For design details and load ratings consult publications of Cone Drive and AGMA Standards.

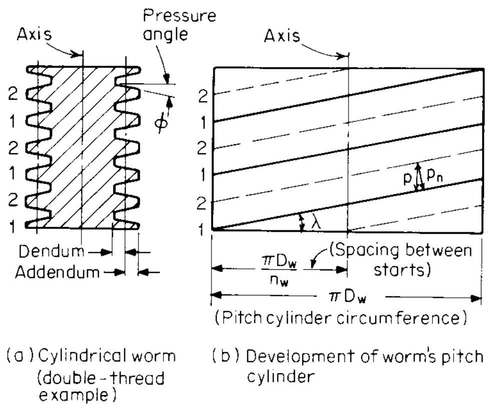


Fig. 8.3.19 Cylindrical worm geometry and design parameters.

Other Gear Types

Gears for special purposes include the following (details are to be found in the references):

Spiroid (Illinois Tool Works) gears, used to connect skew shafts, resemble a hypoid-type bevel gear but in performance are more like worm meshes. They offer very high ratios and a large contact ratio resulting in high strength. The **Helicon** (Illinois Tool Works) gear is a variation in which the pinion is not tapered, and ratios under 10 : 1 are feasible.

Beveloid (Vinco Corp.) gears are tapered involute gears which can couple intersecting shafts, skew shafts, and parallel shafts.

Face gears have teeth cut on the rotating face plane of the gear and mate with standard involute spur gears. They can connect intersecting or nonparallel, nonintersecting shafts.

Noncircular gears or **function gears** are used for special motions or as elements of analog computers. They can be made with elliptical, logarithmic, spiral, and other functions. See Cunningham references; also, Cunningham Industries, Inc., Stamford, CT.

DESIGN STANDARDS

In addition to the ANSI and AGMA standards on basic tooth proportions, the AGMA sponsors a large number of national standards dealing with gear design, specification, and inspection. (Consult AGMA, 1500 King St., Arlington, VA 22314, for details.) Helpful general references are AGMA, ''Gear Handbook,'' 390.03 and ANSI/AGMA, ''Gear Classification and Inspection Handbook,'' 2000-A88, which establish a system of quality classes for all gear sizes and pitches, ranging from crude coarse commercial gears to the highest orders of fine and coarse ultra-precision gears.

There are 13 quality classes, numbered from 3 through 15 in ascending quality. Tolerances are given for key functional parameters: runout, pitch, profile, lead, total composite error, tooth-to-tooth composite error, and tooth thickness. Also, tooth thickness tolerances and recommended mesh backlash are included. These are related to diametral pitch and pitch diameter in recognition of fabrication achievability. Data are available for spur, helical, herringbone, bevel, and worm gearing; and spur and helical racks. Special sections cover gear applications and suggested quality number; gear materials and treatments; and standard procedure for identifying quality, material, and other pertinent parameters. These data are too extensive for inclusion in this handbook, and the reader is referred to the cited AGMA references.

STRENGTH AND DURABILITY

Gear teeth fail in two classical manners: tooth breakage and surface fatigue pitting. Instrument gears and other small, lightly loaded gears are designed primarily for tooth-bending beam strength since minimizing size is the priority. Power gears, usually larger, are designed for both strengths, with surface durability often more critical. Expressions for calculating the beam and surface stresses started with the Lewis-Buckingham formulas and now extend to the latest AGMA formulas.

The Lewis formula for analysis of beam strength, now relegated to historical reference, serves to illustrate the fundamentals that current formulas utilize. In the Lewis formula, a tooth layout shows the load assumed to be at the tip (Fig. 8.3.20). From this Lewis demonstrated that the beam strength $W_b = FSY/P_d$, where F = face width; S = allowable stress; Y = Lewis form factor; P_d = diametral pitch. The form factor Y is derived from the layout as $Y = 2P_d/3$. The value of Y varies with tooth design (form and pressure angle) and number of teeth. In the case of a helical gear tooth, there is a thrust force W_{th} in the axial direction that arises and must be considered as a component of bearing load. See Fig. 8.3.21*b*. Buckingham modified the Lewis formula to include dynamic effects on beam strength and developed equations for evaluating surface stresses. Further modifications were made by other investigators, and have resulted in the most recent AGMA rating formulas which are the basis of most gear designs in the United States.

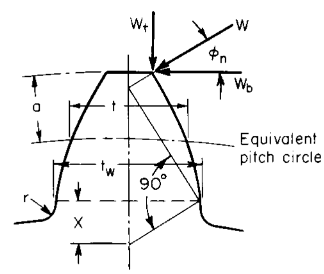


Fig. 8.3.20 Layout for beam strength (Lewis formula).

AGMA Strength and Durability Rating Formulas

For many decades the AGMA Gear Rating Committee has developed and provided tooth beam strength and surface durability (pitting resistance) formulas suitable for modern gear design. Over the years, the formulas have gone through a continual evolution of revision and improvement. The intent is to provide a common basis for rating various gear types for differing applications and thus have a uniformity of practice within the gear industry. This has been accomplished via a series of standards, many of which have been adopted by ANSI.

The latest standards for rating bending beam strength and pitting resistance are ANSI/AGMA 2001-C95, ''Fundamental Rating Factors and Calculation Methods for Involute, Spur, and Helical Gear Teeth'' (available in English and metric units) and AGMA 908-B89, ''Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical, and Herringbone Gear Teeth.'' These standards have replaced AGMA 218.01 with improved formulas and details.

The rating formulas in Tables 8.3.11 and 8.3.12 are abstracted from ANSI/AGMA 2001-B88, ''Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth,'' with permission.

Overload factor K_o is intended to account for an occasional load in excess of the nominal design load W_t. It can be established from experience with the particular application. Otherwise use $K_o = 1$.

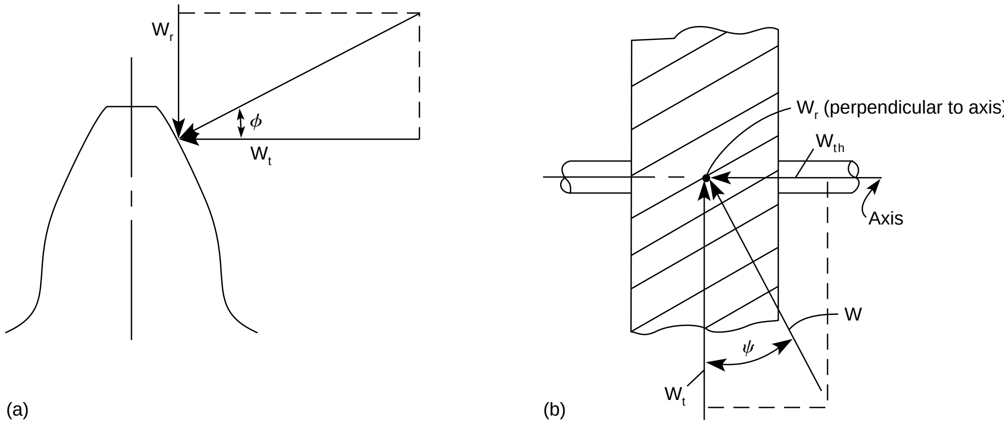


Fig. 8.3.21 Forces on spur and helical teeth. (*a*) Spur gear; (*b*) helical gear.

Table 8.3.11 AGMA Pitting Resistance Formula for Spur and Helical Gears
(See Note 1 below.)

$$s_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{dF} \frac{C_f}{I}}$$

where s_c = contact stress number, lb/in²
C_p = elastic coefficient,* $(\mathrm{lb/in^2})^{0.5}$ (see text and Table 8.3.13)
W_t = transmitted tangential load, lb
K_o = overload factor (see text)
K_v = dynamic factor (see Fig. 8.3.22)
K_s = size factor (see text)
K_m = load distribution factor (see text and Table 8.3.14)
C_f = surface condition factor for pitting resistance (see text)
F = net face width of narrowest member, in
I = geometry factor for pitting resistance (see text and Figs. 8.3.23 and 8.3.24)
d = operating pitch diameter of pinion, in

$$= \frac{2C}{m_G + 1} \qquad \text{for external gears}$$

$$= \frac{2C}{m_G - 1} \qquad \text{for internal gears}$$

where C = operating center distance, in
m_G = gear ratio (never less than 1.0)

Allowable contact stress number s_{ac}

$$s_c \le \frac{s_{ac} Z_N C_H}{S_H K_T K_R}$$

where s_{ac} = allowable contact stress number, lb/in² (see Tables 8.3.15 and 8.3.16; Fig. 8.3.34)
Z_N = stress cycle factor for pitting resistance (see Fig. 8.3.35)
C_H = hardness ratio factor for pitting resistance (see text and Figs. 8.3.36 and 8.3.37)
S_H = safety factor for pitting (see text)
K_T = temperature factor (see text)
K_R = reliability factor (see Table 8.3.19)

* **Elastic coefficient C_p** can be calculated from the following equation when the paired materials in the pinion-gear set are not listed in Table 8.3.13:

$$C_p = \sqrt{\frac{1}{\pi[(1 - \mu_P^2)/E_P + (1 - \mu_G^2)/E_G]}}$$

where $\mu_P (\mu_G)$ = Poisson's ratio for pinion (gear)
$E_P (E_G)$ = modulus of elasticity for pinion (gear), lb/in²

Note 1: If the rating is calculated on the basis of uniform load, the transmitted tangential load is

$$W_t = \frac{33{,}000P}{v_t} = \frac{2T}{d} = \frac{126{,}000P}{n_p d}$$

where P = transmitted power, hp
T = transmitted pinion torque, lb · in
v_t = pitch line velocity at operating pitch diameter, ft/min = $\dfrac{\pi n_p d}{12}$

Table 8.3.12 AGMA Bending Strength Fundamental Formula for Spur and Helical Gears
(See Note 1 in Table 8.3.11.)

$$s_t = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J}$$

where s_t = bending stress number, lb/in²
K_B = rim thickness factor (see Fig. 8.3.38)
J = geometry factor for bending strength (see text and Figs. 8.3.25 to 8.3.31)
P_d = transverse diametral pitch, in^{-1}*;
P_{dn} for helical gears

$$P_d = \frac{\pi}{p_x \tan \psi_s} = P_{dn} \cos \psi_s \qquad \text{for helical gears}$$

where P_{dn} = normal diametral pitch, in^{-1}
p_x = axial pitch, in
ψ_s = helix angle at standard pitch diameter

$$\psi_s = \arcsin \frac{\pi}{p_x P_{dn}}$$

Allowable bending stress numbers s_{at}

$$s_t \le \frac{s_{at} Y_N}{S_F K_T K_R}$$

where s_{at} = allowable bending stress number, lb/in² (see Tables 8.3.17 and 8.3.18 and Figs. 8.3.39 to 8.3.41)
Y_N = stress cycle factor for bending strength (see Fig. 8.3.42)
S_F = safety factor for bending strength (see text)

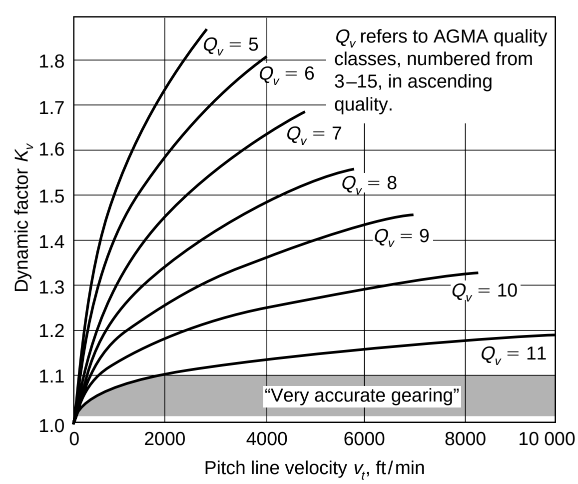


Fig. 8.3.22 Dynamic factor K_v. (*Source: ANSI/AGMA 2001-C95, with permission.*)

Table 8.3.13 Elastic Coefficient C_p

Pinion material	Pinion modulus of elasticity E_P, lb/in² (MPa)	Gear material and modulus of elasticity E_G, lb/in² (MPa): Steel 30×10^6 (2×10^5)	Malleable iron 25×10^6 (1.7×10^5)	Nodular iron 24×10^6 (1.7×10^5)	Cast iron 22×10^6 (1.5×10^5)	Aluminum bronze 17.5×10^6 (1.2×10^5)	Tin bronze 16×10^6 (1.1×10^5)
Steel	30×10^6 (2×10^5)	2,300 (191)	2,180 (181)	2,160 (179)	2,100 (174)	1,950 (162)	1,900 (158)
Malleable iron	25×10^6 (1.7×10^5)	2,180 (181)	2,090 (174)	2,070 (172)	2,020 (168)	1,900 (158)	1,850 (154)
Nodular iron	24×10^6 (1.7×10^5)	2,160 (179)	2,070 (172)	2,050 (170)	2,000 (166)	1,880 (156)	1,830 (152)
Cast iron	22×10^6 (1.5×10^5)	2,100 (174)	2,020 (168)	2,000 (166)	1,960 (163)	1,850 (154)	1,800 (149)
Aluminum bronze	17.5×10^6 (1.2×10^5)	1,950 (162)	1,900 (158)	1,880 (156)	1,850 (154)	1,750 (145)	1,700 (141)
Tin bronze	16×10^6 (1.1×10^5)	1,900 (158)	1,850 (154)	1,830 (152)	1,800 (149)	1,700 (141)	1,650 (137)

Poisson's ratio = 0.30.
SOURCE: ANSI/AGMA 2001-B88; with permission.

Size factor K_s is intended to factor in material nonuniformity due to tooth size, diameter, face width, etc. AGMA has not established factors for general gearing; use $K_s = 1$ unless there is information to warrant using a larger value.

Load distribution factor K_m reflects the nonuniform loading along the lines of contact due to gear errors, installation errors, and deflections. Analytical and empirical methods for evaluating this factor are presented in ANSI/AGMA 2001-C95 but are too extensive to include here. Alternately, if appropriate for the application, K_m can be extrapolated from values given in Table 8.3.14.

Surface condition factor C_f is affected by the manufacturing method (cutting, shaving, grinding, shotpeening, etc.). Standard factors have not been established by AGMA. Use $C_f = 1$ unless experience can establish confidence for a larger value.

Geometry factors *I* and *J* relate to the shape of the tooth at the point of contact, the most heavily loaded point. AGMA 908-B89 (Information Sheet, Geometry Factors for Determining the Pitting Resistance and Bending Strength for Spur, Helical and Herringbone Gear Teeth) presents detailed procedures for calculating these factors. The standard also includes a collection of tabular values for a wide range of gear tooth designs, but they are too voluminous to be reproduced here in their entirety. Earlier compact graphs of *I* and *J* values in AGMA 218.01 are still valid. They are presented here along with several curves from AGMA 610-E88, which are based upon 218.01. See Figs. 8.3.23 to 8.3.31.

Allowable contact stress s_{ac} and **allowable bending stress s_{at}** are obtainable from Tables 8.3.15 to 8.3.18. Contact stress hardness specification applies to the start of active profile at the center of the face width, and for bending stress at the root diameter in the center of the tooth space and face width. The lower stress values are for general design purposes; upper values are for high-quality materials and high-quality control. (See ANSI/AGMA 2001-C95, tables 7 through 10, regarding detailed metallurgical specifications; stress grades 1, 2, and 3; and type A and B hardness patterns.)

For **reversing loads,** allowable bending stress values, s_{at} are to be reduced to 70 percent. If the rim thickness cannot adequately support the load, an additional derating factor K_B is to be applied. See Fig. 8.3.38.

Hardness ratio factor C_H applies when the pinion is substantially harder than the gear, and it results in work hardening of the gear and increasing its capacity. Factor C_H applies to only the gear, not the pinion. See Figs. 8.3.36 and 8.3.37.

Safety factors S_H and S_F are defined by AGMA as factors beyond K_O and K_R; they are used in connection with extraordinary risks, human or economic. The values of these factors are left to the designer's judgment as she or he assesses all design inputs and the consequences of possible failure.

Temperature factor $K_T = 1$ when gears operate with oil temperature not exceeding 250°F.

Reliability factor K_R accounts for statistical distribution of material failures. Typically, material strength ratings (Tables 8.3.13, and 8.3.15 to 8.3.18; Fig. 8.3.34 and 8.3.42) are based on probability of one failure in 100 at 10^7 cycles. Table 8.3.19 lists reliability factors that may be used to modify the allowable stresses and the probability of failure.

Strength and Durability of Bevel, Worm, and Other Gear Types

For bevel gears, consult the referenced Gleason publications; for worm gearing refer to AGMA standards; for other special types, refer to Dudley, ''Gear Handbook.''

Table 8.3.14 Load-Distribution Factor K_m for Spur Gears*

Characteristics of support	Face width, in: 0–2	6	9	16 up
Accurate mountings, small bearing clearances, minimum deflection, precision gears	1.3	1.4	1.5	1.8
Less rigid mountings, less accurate gears, contact across full face	1.6	1.7	1.8	2.2
Accuracy and mounting such that less than full face contact exists	Over 2.2			

* An approximate guide only. See ANSI/AGMA 2001-C95 for derivation of more exact values.
SOURCE: Darle W. Dudley, ''Gear Handbook,'' McGraw-Hill, New York, 1962.

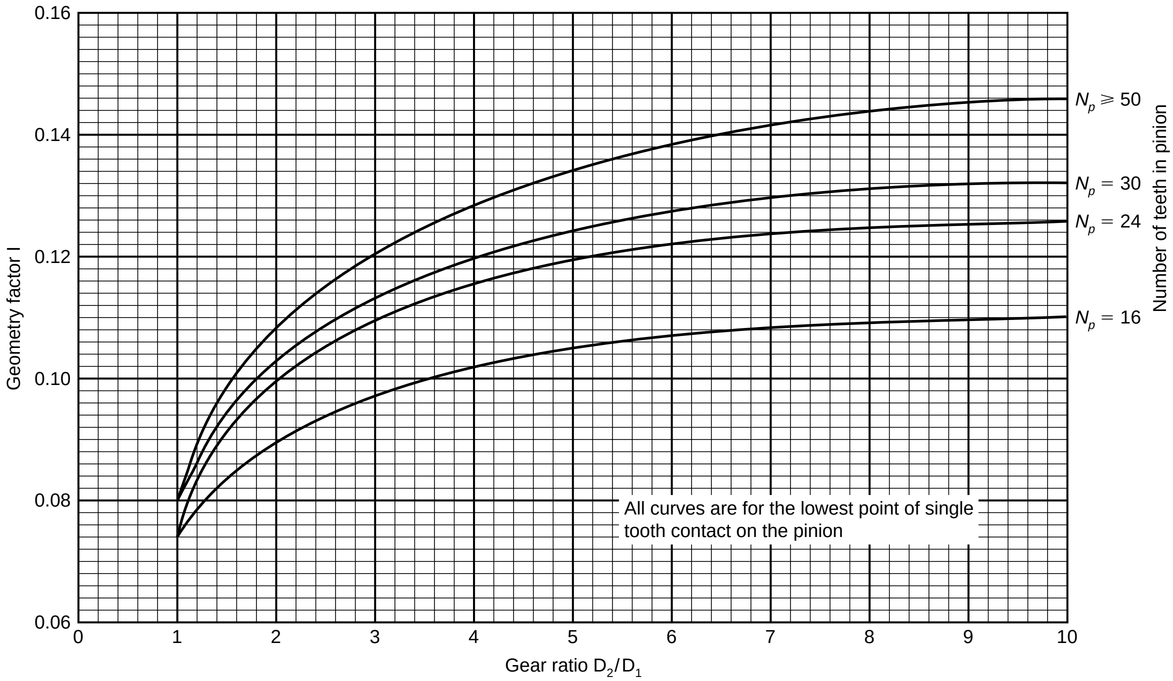


Fig. 8.3.23 Geometry factor *I* for 20° full-depth standard spur gears. (*Source: ANSI/AGMA 2018-01, with permission.*)

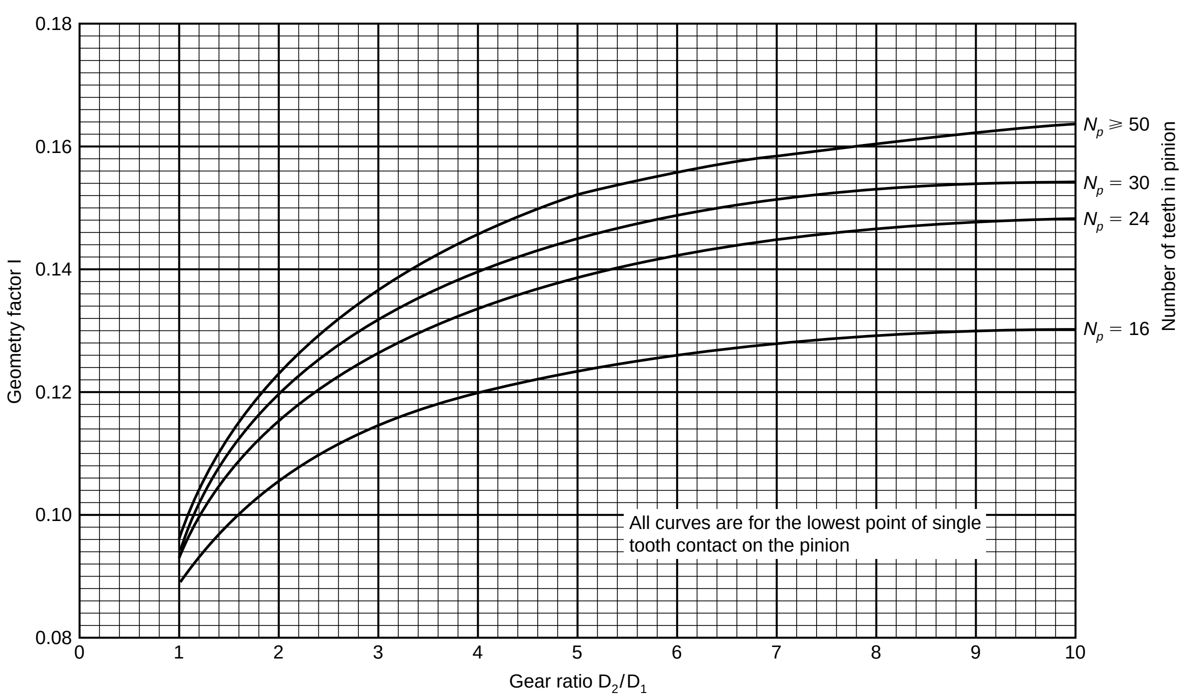


Fig. 8.3.24 Geometry factor *I* for 25° full-depth standard spur gears. (*Source: ANSI/AGMA 2018-01, with permission.*)

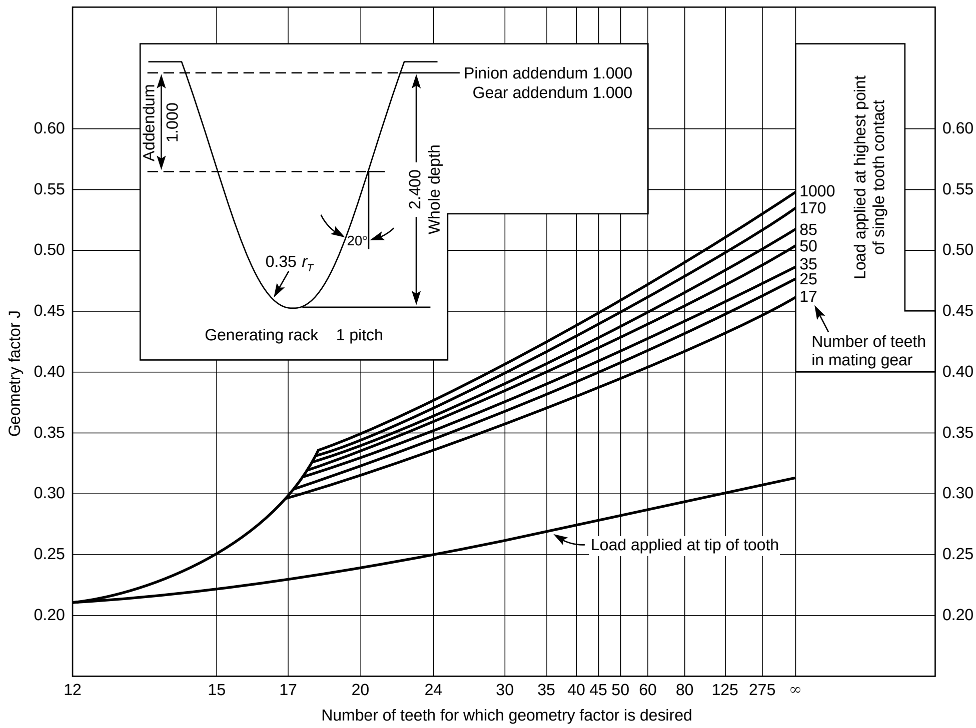


Fig. 8.3.25 Geometry factor J for 20° standard addendum spur gears. (*Source: ANSI/AGMA 2018-01, with permission.*)

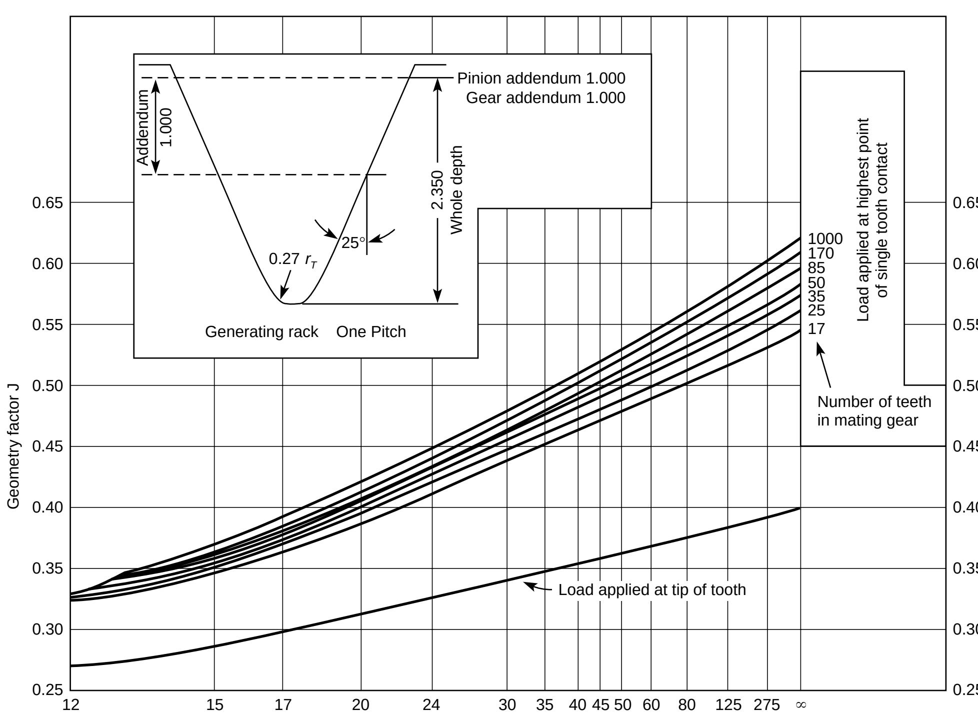


Fig. 8.3.26 Geometry factor J for 25° standard addendum spur gears. (*Source: ANSI/AGMA 2018-01, with permission.*)

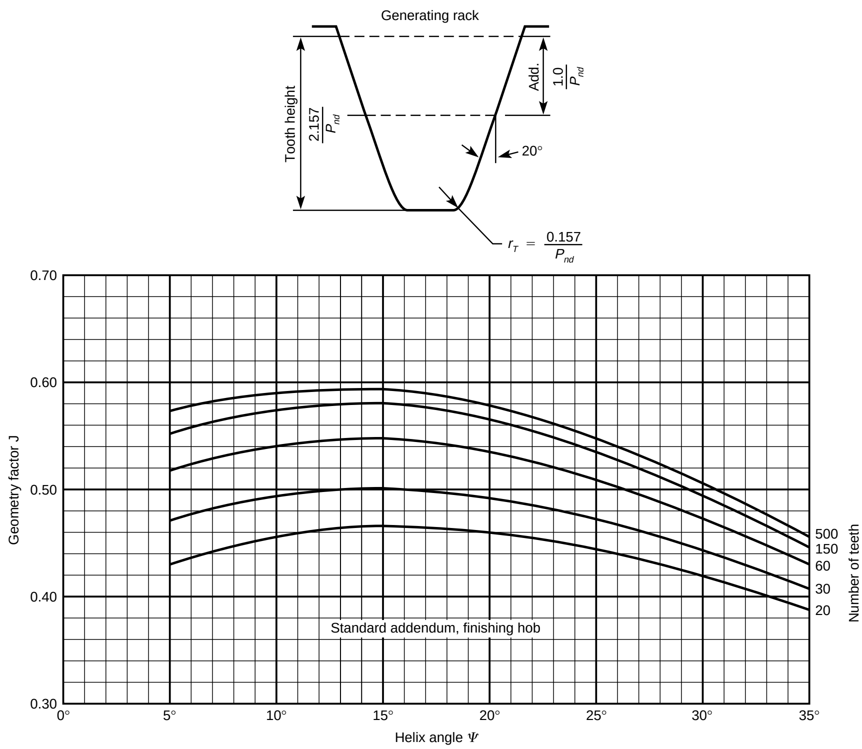


Fig. 8.3.27 Geometry factor J for 20° normal pressure angle helical gears. (Standard addendum, finishing hob.) (*Source: ANSI/AGMA 2018-01, with permission.*)

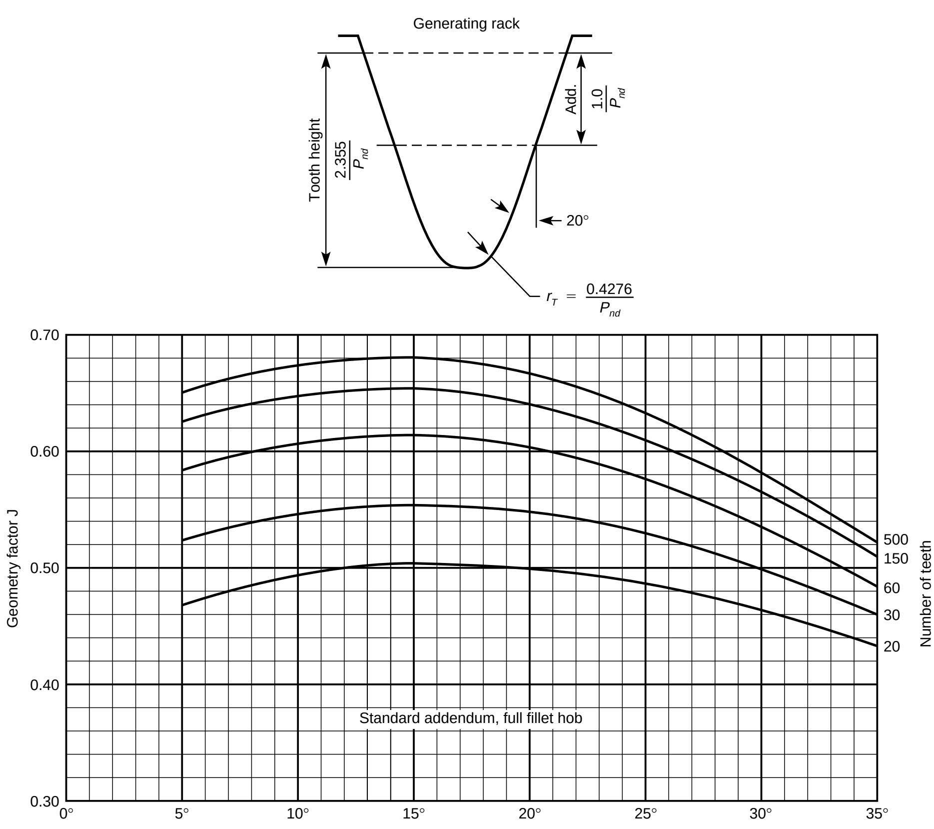


Fig. 8.3.28 Geometry factor J for 20° normal pressure angle helical gears. (Standard addendum, full fillet hob.) (*Source: ANSI/AGMA 2018-01, with permission.*)

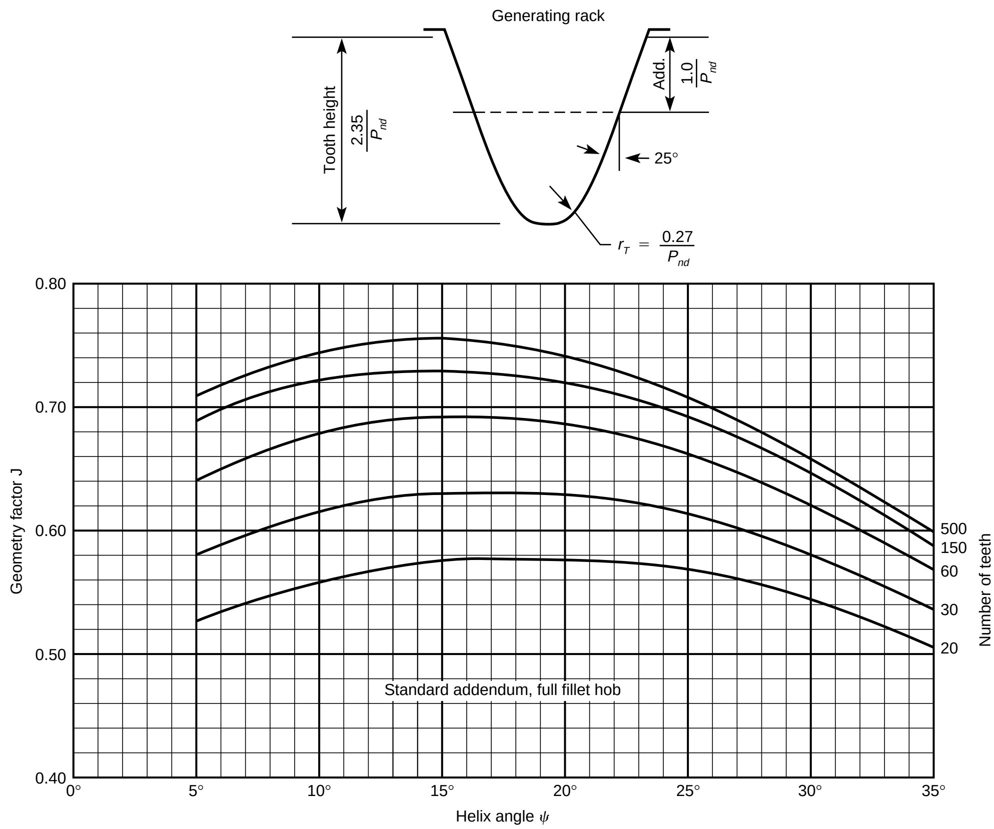


Fig. 8.3.29 Geometry factor J for 25° normal pressure angle helical gears. (Standard addendum, full fillet hob.) (*Source: ANSI/AGMA 2018-01, with permission.*)

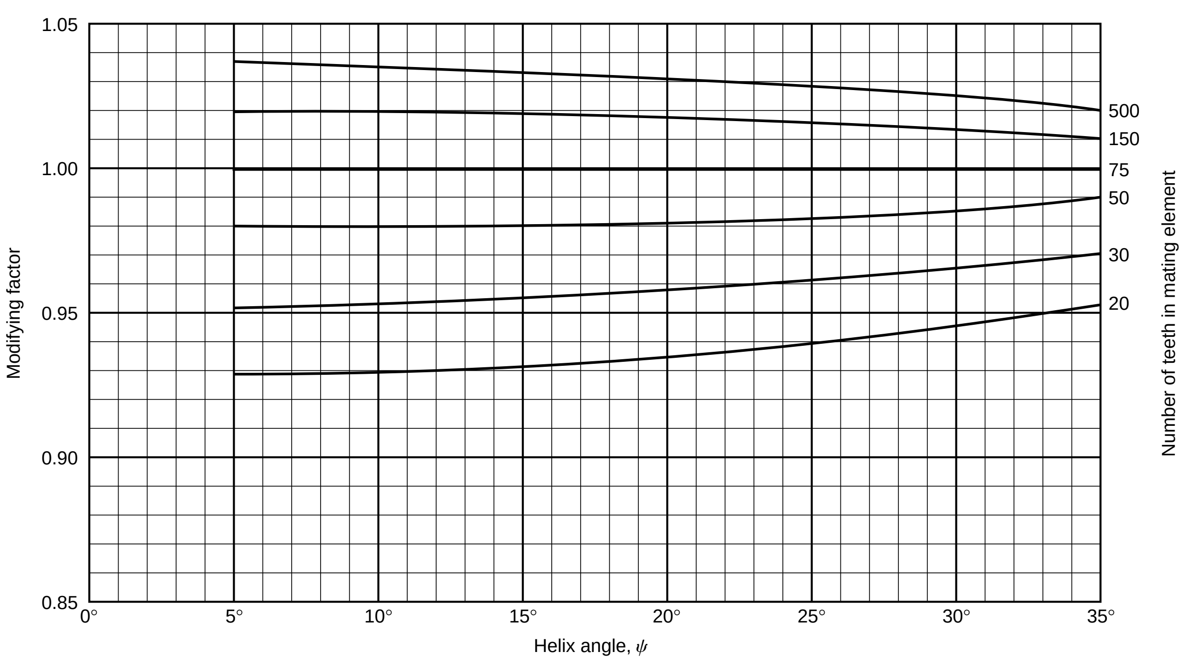


Fig. 8.3.30 Factor J multipliers for 20° normal pressure angle helical gears. The modifying factor can be applied to the J factor when other than 75 teeth are used in the mating element. (*Source: ANSI/AGMA 2018-01, with permission.*)

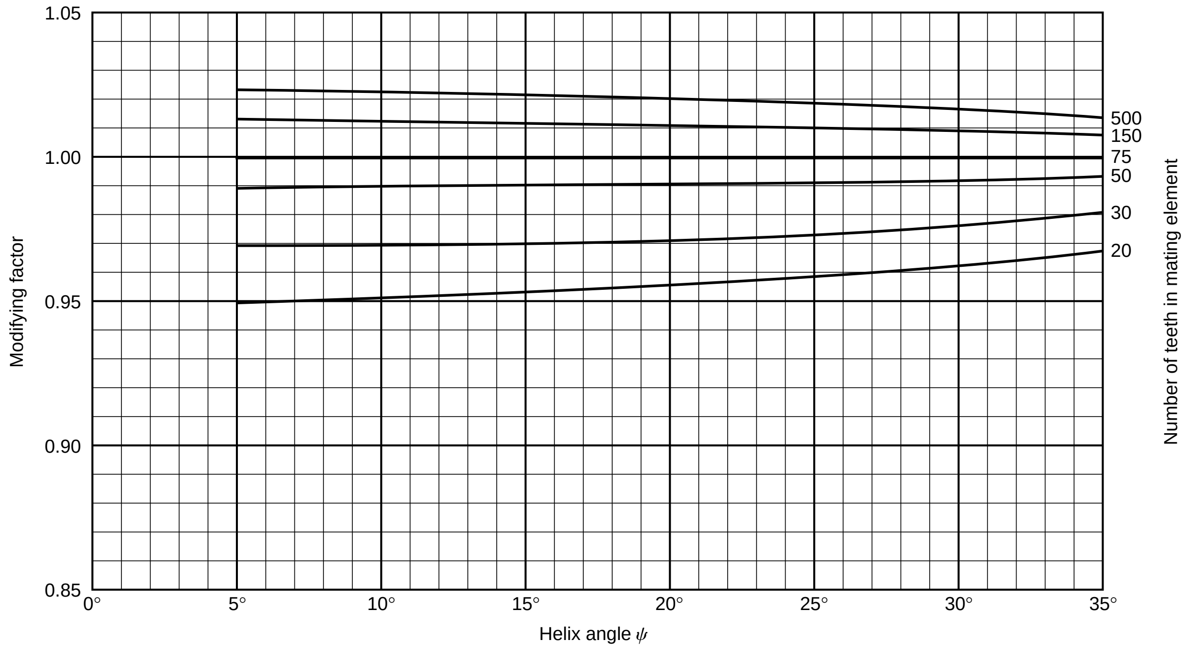


Fig. 8.3.31 Factor *J* multipliers for 25° normal pressure angle helical gears. The modifying factor can be applied to the *J* factor when other than 75 teeth are used in the mating element. (*Source: ANSI/AGMA 6010-E88, with permission.*)

Table 8.3.15 Allowable Contact Stress Number s_{ac} for Steel Gears

Material designation	Heat treatment	Minimum surface hardness*	Allowable contact stress number s_{ac}, lb/in²		
			Grade 1	Grade 2	Grade 3
Steel	Through-hardened*	Fig. 8.3.34	Fig. 8.3.34	Fig. 8.3.34	—
	Flame-* or induction-hardened*	50 HRC 54 HRC	170,000 175,000	190,000 195,000	— —
	Carburized and hardened*	Table 9 Note 1	180,000	225,000	275,000
	Nitrided* (through-hardened steels)	83.5 HR15N 84.5 HR15N	150,000 155,000	163,000 168,000	175,000 180,000
2.5% Chrome (no aluminum)	Nitrided*	87.5 HR15N	155,000	172,000	189,000
Nitralloy 135M	Nitrided*	90.0 HR15N	170,000	183,000	195,000
Nitralloy N	Nitrided*	90.0 HR15N	172,000	188,000	205,000
2.5% Chrome (no aluminum)	Nitrided*	90.0 HR15N	176,000	196,000	216,000

Note 1: Table 9 and Tables 7, 8, and 10 cited in Tables 8.3.16 to 8.3.18 are in ANSI/AGMA 2001-95.
* The allowable-stress numbers indicated may be used with the case depths shown in Figs. 8.3.32 and 8.3.33.
SOURCE: Abstracted from ANSI/AGMA 2001-C95, with permission.

Table 8.3.16 Allowable Contact Stress Number s_{ac} for Iron and Bronze Gears

Material	Material designation*	Heat treatment	Typical minimum surface hardness	Allowable contact stress number s_{ac}, lb/in²
ASTM A48 gray cast iron	Class 20 Class 30 Class 40	As cast As cast As cast	— 174 HB 201 HB	50,000–60,000 65,000–75,000 75,000–85,000
ASTM A536 ductile (nodular) iron	Grade 60-40-18 Grade 80-55-06 Grade 100-70-03 Grade 120-90-02	Annealed Quenched & tempered Quenched & tempered Quenched & tempered	140 HB 179 HB 229 HB 269 HB	77,000–92,000 77,000–92,000 92,000–112,000 103,000–126,000
Bronze	—	Sand-cast	Minimum tensile strength 40,000 lb/in²	30,000
	ASTM B-148 Alloy 954	Heat-treated	Minimum tensile strength 90,000 lb/in²	65,000

* See ANSI/AGMA 2004–B89, ''Gear Materials and Heat Treatment Manual.''
SOURCE: Abstracted from ANSI/AGMA 2001-C95, with permission.

Table 8.3.17 Allowable Bending Stress Number s_{at} for Steel Gears

Material designation	Heat treatment	Minimum surface hardness	Allowable bending stress number s_{at}, lb/in² Grade 1	Grade 2	Grade 3
Steel	Through-hardened	Fig. 8.3.39	Fig. 8.3.39	Fig. 8.3.39	—
	Flame* or induction-hardened* with type A pattern	Table 8§	45,000	55,000	—
	Flame* or induction hardened* with type B pattern	Table 8§ Note 1	22,000	22,000	—
	Carburized and hardened*	Table 9§ Note 2	55,000	65,000 or 70,000†	75,000
	Nitrided*‡ (through-hardened steels)	83.5 HR15N	Fig. 8.3.40	Fig. 8.3.40	—
Nitralloy 135M, nitralloy N, and 2.5% Chrome (no aluminum)	Nitrided‡	87.5 HR15N	Fig. 8.3.41	Fig. 8.3.41	Fig. 8.3.41

Note 1: See Table 8 in ANSI/AGMA 2001-C95.
Note 2: See Table 9 in ANSI/AGMA 2001-C95.
* The allowable-stress numbers indicated may be used with the case depths shown in Figs. 8.3.32 and 8.3.33.
† If bainite and microcracks are limited to grade 3 levels, 70,000 lb/in² may be used.
‡ The overload capacity of nitrided gears is low. Since the shape of the effective $S-N$ curve is flat, the sensitivity to shock should be investigated before one proceeds with the design.
§ The tabular material is too extensive to record here. Refer to ANSI/AGMA 2001-C95, tables 7 to 10.
SOURCE: Abstracted from ANSI/AGMA 2001-C95, with permission.

Table 8.3.18 Allowable Bending Stress Number s_{at} for Iron and Bronze Gears

Material	Material designation*	Heat treatment	Typical minimum surface hardness	Allowable bending stress number s_{at}, lb/in²
ASTM A48 gray cast iron	Class 20	As cast	—	5,000
	Class 30	As cast	174 HB	8,500
	Class 40	As cast	201 HB	13,000
ASTM A536 ductile (nodular) iron	Grade 60-40-18	Annealed	140 HB	22,000–33,000
	Grade 80-55-06	Quenched & tempered	179 HB	22,000–33,000
	Grade 100-70-03	Quenched & tempered	229 HB	27,000–40,000
	Grade 120-90-02	Quenched & tempered	269 HB	31,000–44,000
Bronze		Sand-cast	Minimum tensile strength 40,000 lb/in²	5,700
	ASTM B-148 Alloy 954	Heat-treated	Minimum tensile strength 90,000 lb/in²	23,600

* See ANSI/AGMA 2004-B89, ''Gear Materials and Heat Treatment Manual.''
SOURCE: Abstracted from ANSI/AGMA 2001-C95, with permission.

Table 8.3.19 Reliability Factors K_R

Requirements of application	K_R*
Fewer than one failure in 10,000	1.50
Fewer than one failure in 1,000	1.25
Fewer than one failure in 100	1.00
Fewer than one failure in 10	0.85†
Fewer than one failure in 2	0.70†,‡

* Tooth breakage is sometimes considered a greater hazard than pitting. In such cases a greater value of K_R is selected for bending.
† At this value, plastic flow might occur rather than pitting.
‡ From test data extrapolation.
SOURCE: Abstracted from ANSI/AGMA 2001-C95, with permission.

GEAR MATERIALS

(See Tables 8.3.15 to 8.3.18.)

Metals

Plain carbon steels are most widely used as the most economical; similarly, cast iron is used for large units or intricate body shapes. Heat-treated carbon and alloy steels are used for the more severe load- and wear-resistant applications. Pinions are usually made harder to equalize wear. Strongest and most wear-resistant gears are a combination of heat-treated high-alloy steel cores with case-hardened teeth. (See Dudley, ''Gear Handbook,'' chap. 10.) Bronze is particularly recommended for wormgears and crossed helical gears. Stainless steels are limited to special corrosion-resistant environment applications. Aluminum alloys are used for light-duty instrument gears and airborne lightweight requirements.

Sintered powdered metals technology offers commercial high-quality gearing of high strength at very economical production costs. Die-cast gears for light-duty special applications are suitable for many products.

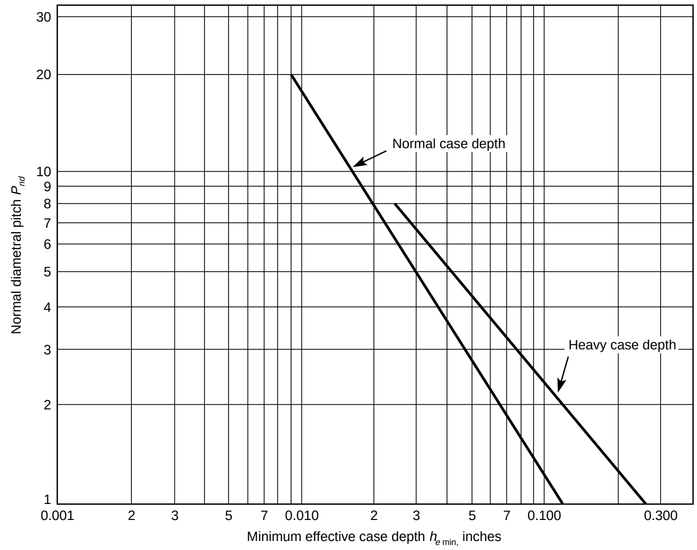


Fig. 8.3.32 Minimum effective case depth for carburized gears $h_{e\,\mathrm{min}}$. Effective case depth is defined as depth of case with minimun hardness of 50 RC. Total case depth to core carbon is approximately 1.5 times the effective case depth. (*Source: ANSI/AGMA 2001-C95, with permission.*)

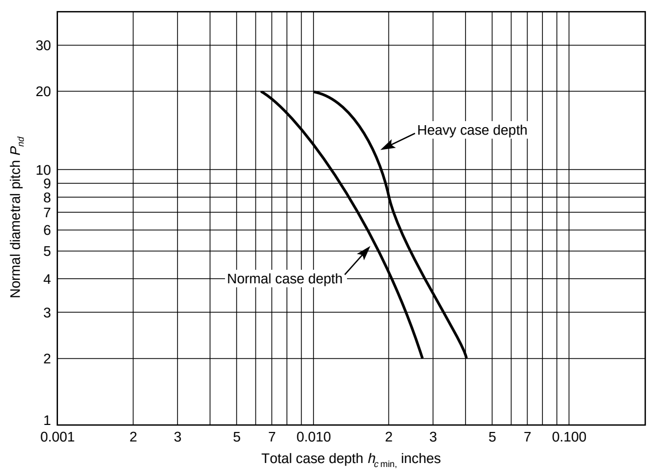


Fig. 8.3.33 Minimum total case depth for nitrided gears $h_{c\,\mathrm{min}}$. (*Source: ANSI/AGMA 2001-C95, with permission.*)

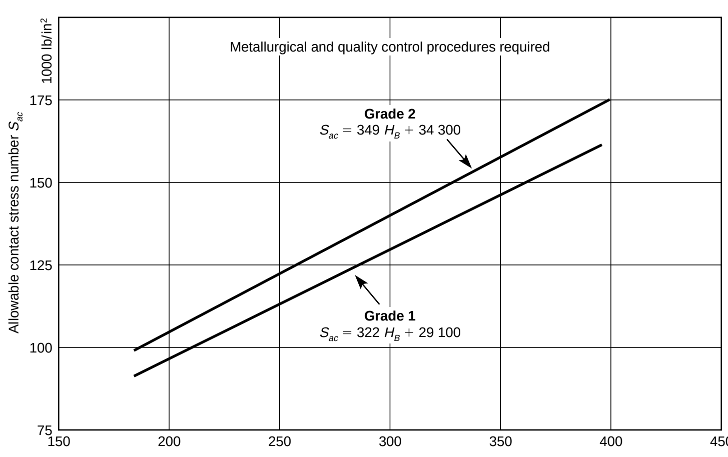


Fig. 8.3.34 Allowable contact stress number for through-hardened steel gears s_{ac}. (*Source: ANSI/AGMA 2001-C95, with permission.*)

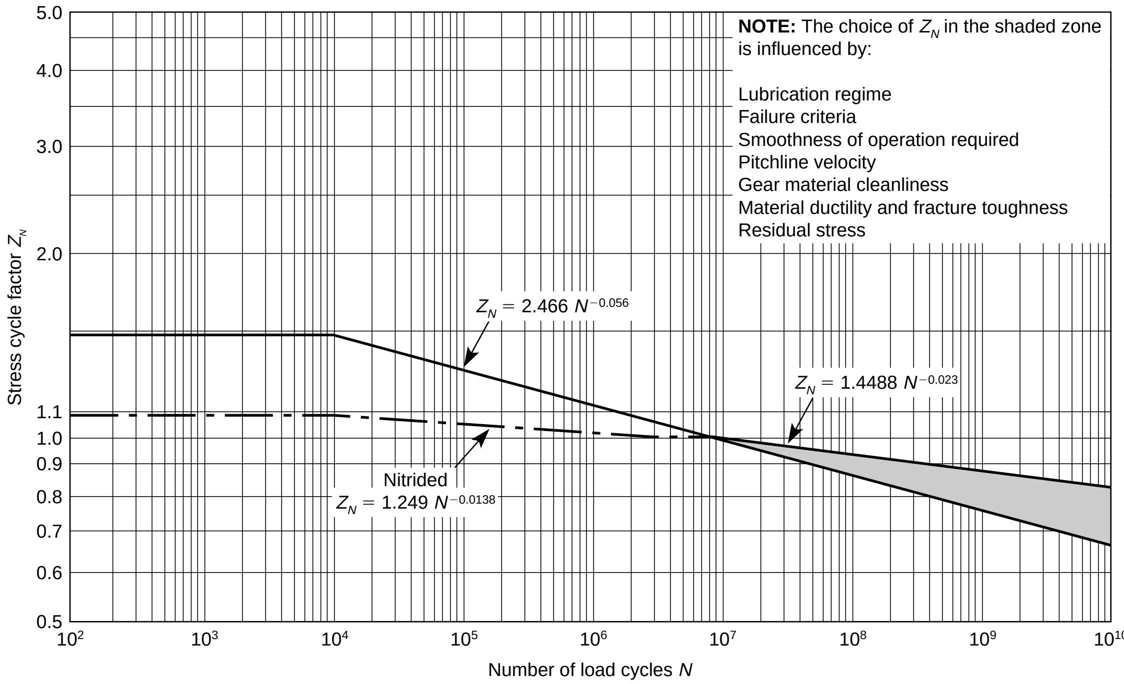


Fig. 8.3.35 Pitting resistance stress cycle factor Z_N. (*Source: ANSI/AGMA 2001-C95, with permission.*)

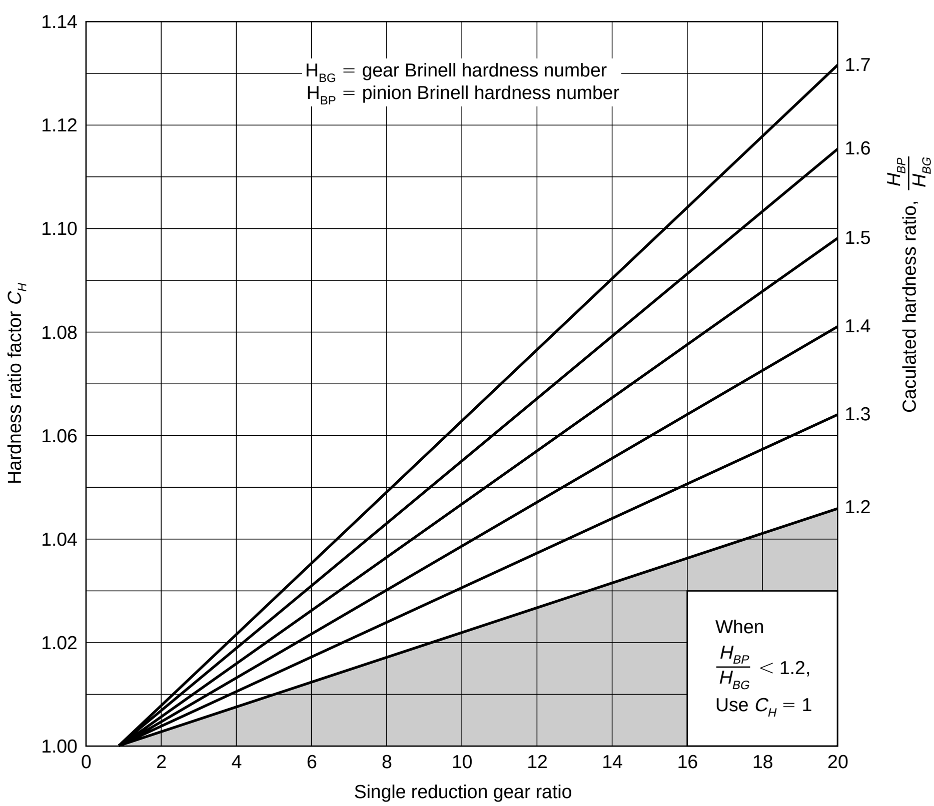


Fig. 8.3.36 Hardness ratio factor C_H (through-hardened). (*Source: ANSI/AGMA 2001-C95, with permission.*)

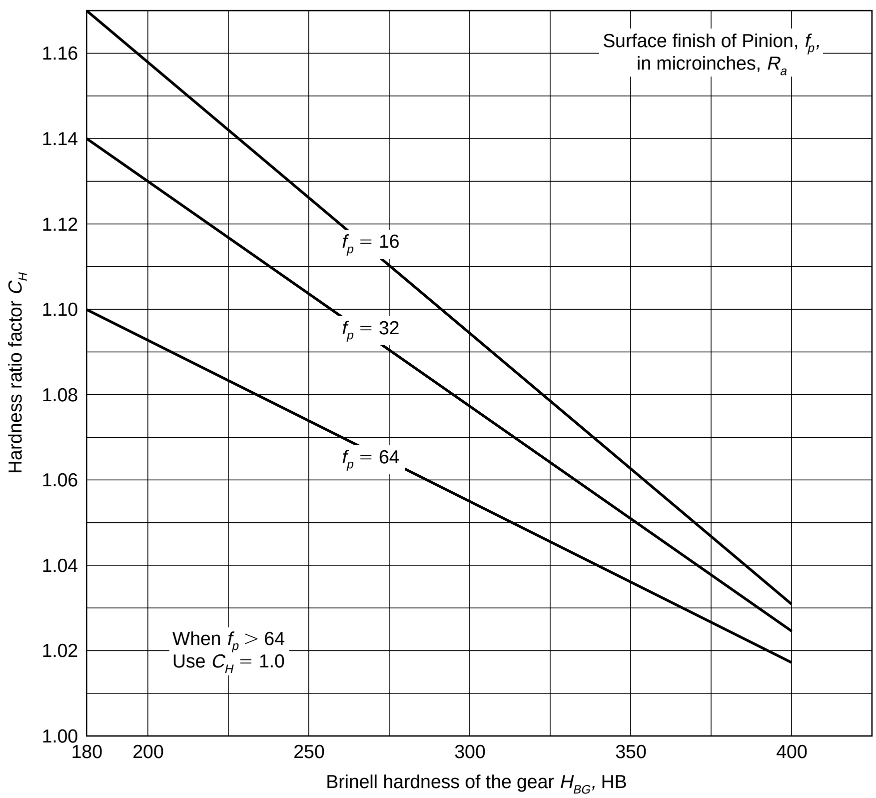


Fig. 8.3.37 Hardness ratio factor C_H (surface-hardened pinions). (*Source: ANSI/AGMA 2001-C95, with permission.*)

For power gear applications, heat treatment is an important part of complete and proper design and specification. Heat treatment descriptions and specification tolerances are given in the reference cited below.

Precision gears of the small device and instrument types often require protective coatings, particularly for aircraft, marine, space, and military applications. There is a wide choice of chemical and electroplate coatings offering a variety of properties and protection.

For pertinent properties and details of the above special materials and protective coatings see Michalec, ''Precision Gearing,'' chap. 9, Wiley.

Plastics

In recent decades, various forms of nonmetallic gears have displaced metal gears in particular applications. Most plastics can be hobbed or shaped by the same methods used for metallic gears. However, high-strength composite plastics suitable for good-quality gear molding have become available, along with the development of economical high-speed injection molding machines and improved methods for producing accurate gear molds.

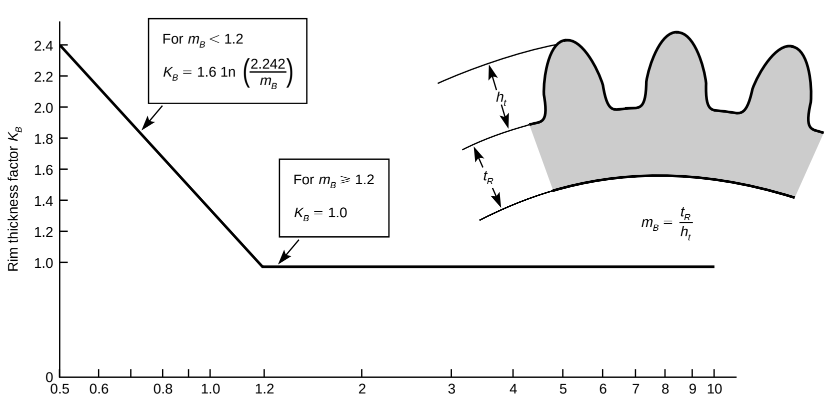


Fig. 8.3.38 Rim thickness factor K_B. (*Source: ANSI/AGMA 2001-C95, with permission.*)

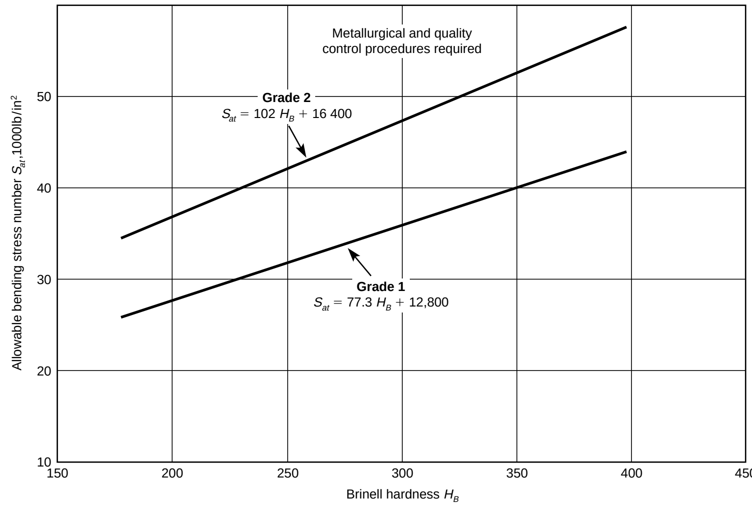


Fig. 8.3.39 Allowable bending stress numbers for through-hardened steel gears s_{at}. (*Source: ANSI/AGMA 2001-C95, with permission.*)

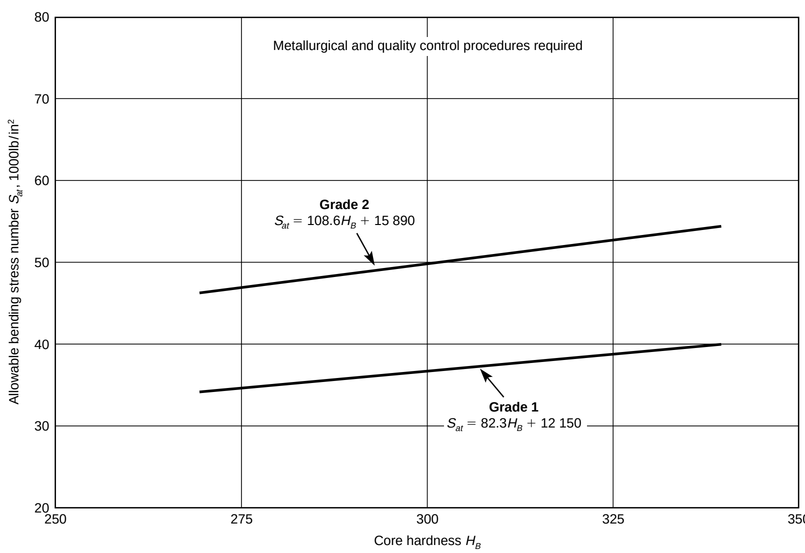


Fig. 8.3.40 Allowable bending stress numbers s_{at} for nitrided through-hardened steel gears (i.e., AISI 4140 and 4340). (*Source: ANSI/AGMA 2001-C95, with permission.*)

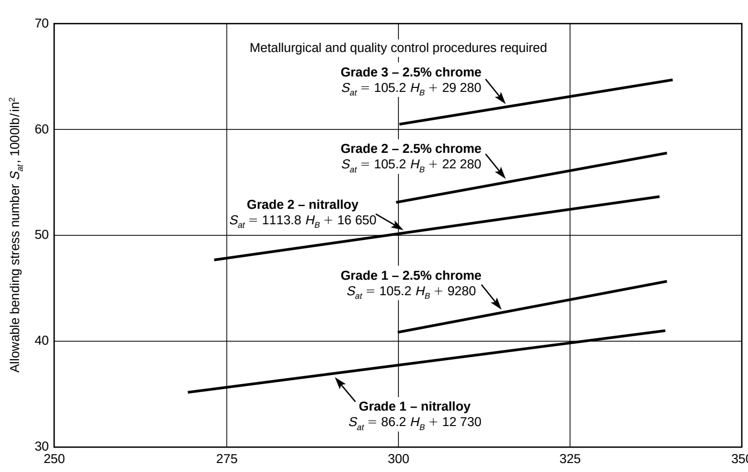


Fig. 8.3.41 Allowable bending stress numbers for nitrided steel gears s_{at}. (*Source: ANSI/AGMA 2001-C95, with permission.*)

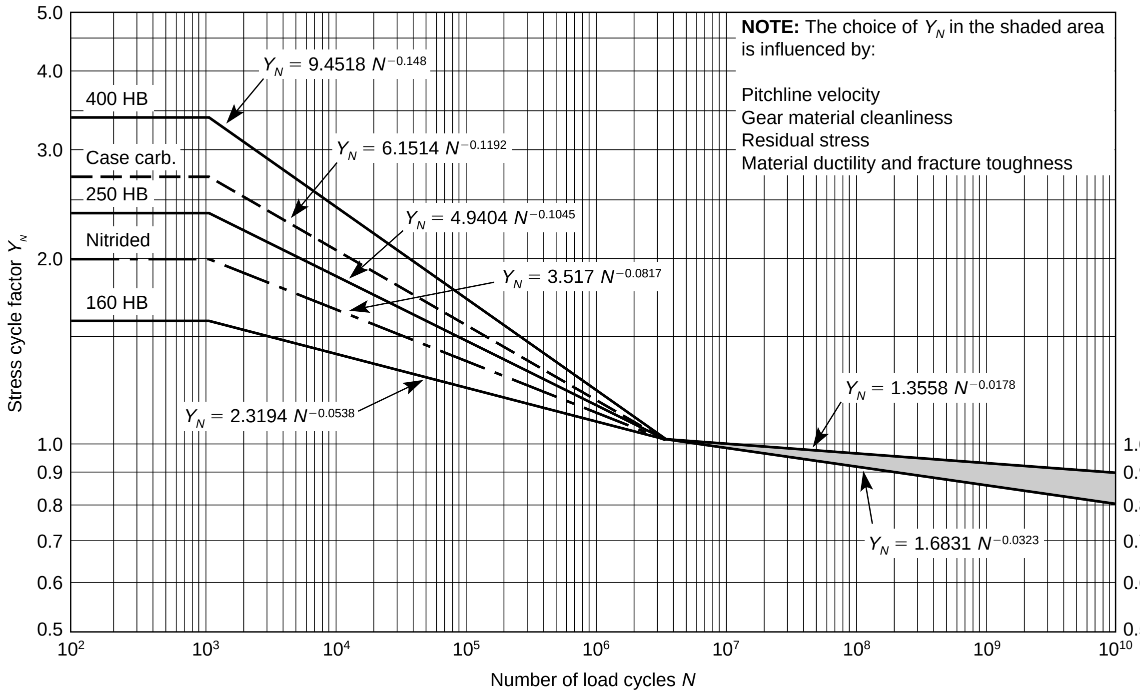


Fig. 8.3.42 Bending strength stress cycle factor Y_N. (*Source: ANSI/AGMA 2001-C95, with permission.*)

The most significant features and advantages of plastic gear materials are:

Cost-effectiveness of injection molding process
Wide choice of characteristics: mechanical strength, density, friction, corrosion resistance, etc.
One-step production; no preliminary or secondary operations
Uniformity of parts
Ability to integrate special shapes, etc., into gear body
Elimination of machining operations
Capability to mold with metallic insert hubs, if required, for more precise bore diameter or body stability
Capability to mold with internal solid lubricants
Ability to operate without lubrication
Quietness of operation
Consistent with trend toward greater use of plastic housings and components

Plastic gears do have limitations relative to metal gears. The most significant are:

Less load-carrying capacity
Cannot be molded to as high accuracy as machined metal gears
Much larger coefficient of expansion compared to metals
Less environmentally stable with regard to temperature and water absorption
Can be negatively affected by some chemicals and lubricants
Initial high cost in mold manufacture to achieve proper tooth geometry accuracies
Narrower range of temperature operation, generally less than 250°F and not lower than 0°F

For further information about plastic gear materials and achievable precision, consult the cited reference: Michalec, ''Precision Gearing: Theory and Practice.''

For a comprehensive presentation of gear molding practices, design, plastic materials, and strength and durability of plastic gears, consult the cited reference: Designatronics, ''Handbook of Gears: Inch and Metric,'' pp. T131–T158.

GEAR LUBRICATION

Proper lubrication is important to prevention of premature wear of tooth surfaces. In the basic action of involute tooth profiles there is a significant sliding component along with rolling action. In worm gearing sliding is the predominant consideration. Thus, a lubricant is essential for all gearing subject to measurable loadings, and even for lightly or negligibly loaded instrument gearing it is needed to reduce friction. Excellent oils and greases are available for high unit load, high speed gearing. See Secs. 6.11 and 8.4; also consult lubricant suppliers for recommendations and latest available high-quality lubricants with special-purpose additives.

General and specific information for lubrication of gearing is found in ANSI/AGMA 9005-D94, ''Industrial Gear Lubrication,'' which covers open and enclosed gearing of all types. AGMA lists a family of lubricants in accordance with viscosities, numbered 1 through 13, with a cross-reference to equivalent ISO grades. See Table 8.3.20. For AGMA's lubrication recommendations for open and closed gearing related to pitch line speed and various types of lubrication systems, refer to Tables 8.3.21 to 8.3.23. For worm gearing and other information, see ANSI/AGMA 9005-D94.

Information in Table 8.3.24 may be used as a quick guide to gear lubricants and their sources for general-purpose instrument and medium-size gearing. Lubricant suppliers should be consulted for specific high-demand applications.

Often gear performance can be enhanced by special additives to the oil. For this purpose, colloidal additives of graphite, molybdenum disulfide (MoS_2), and Teflon are very effective. These additives are particularly helpful to reduce friction and prevent wear; they are also very beneficial to reduce the rate of wear once it has begun, and thus they prolong gear life. The colloidal additive combines chemically with the metal surface material, resulting in a tenacious layer of combined material interposed between the base metals of the meshing teeth. The size of the colloidal additives is on the order of 2 μm, sufficiently fine not to interfere with the proper operation of the lubricant system filters. Table 8.3.25 lists some commercial colloidal additives and their sources of supply.

Plastic gears are often operated without any external lubrication, and

Table 8.3.20 Viscosity Ranges for AGMA Lubricants

Rust- and oxidation-inhibited gear oils, AGMA lubricant no.	Viscosity range[a] mm²/s (cSt) at 40°C	Equivalent ISO grade[a]	Extreme pressure gear lubricants,[b] AGMA lubricant no.	Synthetic gear oils,[c] AGMA lubricant no.
0	28.8–35.2	32		0 S
1	41.4–50.6	46		1 S
2	61.2–74.8	68	2 EP	2 S
3	90–110	100	3 EP	3 S
4	135–165	150	4 EP	4 S
5	198–242	220	5 EP	5 S
6	288–352	320	6 EP	6 S
7, 7 Comp[d]	414–506	460	7 EP	7 S
8, 8 Comp[d]	612–748	680	8 EP	8 S
8A Comp[d]	900–1,100	1,000	8A EP	—
9	1,350–1,650	1,500	9 EP	9 S
10	2,880–3,520	—	10 EP	10 S
11	4,140–5,060	—	11 EP	11 S
12	6,120–7,480	—	12 EP	12 S
13	190–220 cSt at 100°C (212°F)[e]	—	13 EP	13 S

Residual compounds,[f] AGMA lubricant no.	Viscosity ranges[e] cSt at 100°C (212°F)
14R	428.5–857.0
15R	857.0–1,714.0

[a] Per ISO 3448, ''Industrial Liquid Lubricants—ISO Viscosity Classification,'' also ASTM D2422 and British Standards Institution B.S. 4231.
[b] Extreme-pressure lubricants should be used only when recommended by the gear manufacturer.
[c] Synthetic gear oils 9S to 13S are available but not yet in wide use.
[d] Oils marked *Comp* are compounded with 3% to 10% fatty or synthetic fatty oils.
[e] Viscosities of AGMA lubricant no. 13 and above are specified at 100°C (210°F) since measurement of viscosities of these heavy lubricants at 40°C (100°F) would not be practical.
[f] Residual compounds—diluent type, commonly known as solvent cutbacks—are heavy oils containing a volatile, nonflammable diluent for ease of application. The diluent evaporates, leaving a thick film of lubricant on the gear teeth. Viscosities listed are for the base compound without diluent.
CAUTION: These lubricants may require special handling and storage procedures. Diluent can be toxic or irritating to the skin. Do not use these lubricants without proper ventilation. Consult lubricant supplier's instructions.
SOURCE: Abstracted from ANSI/AGMA 9005-D94, with permission.

Table 8.3.21 AGMA Lubricant Number Guidelines for Open Gearing (Continuous Method of Application)[a,b]

		Pressure lubrication		Splash lubrication		Idler immersion
		Pitch line velocity		Pitch line velocity		Pitch line velocity
Ambient temperature[c] °C (°F)	Character of operation	Under 5 m/s (1,000 ft/min)	Over 5 m/s (1,000 ft/min)	Under 5 m/s (1,000 ft/min)	5–10 m/s (1,000–3,000 ft/min)	Up to 1.5 m/s (300 ft/min)
−10 to 15[d] (15–60)	Continuous	5 or 5 EP	4 or 4 EP	5 or 5 EP	4 or 4 EP	8–9 8 EP–9 EP
	Reversing or frequent ''start/stop''	5 or 5 EP	4 or 4 EP	7 or 7 EP	6 or 6 EP	8–9 8 EP–9 EP
10 to 50[d] (50–125)	Continuous	7 or 7 EP	6 or 6 EP	7 or 7 EP	8–9[f] 8 EP–9 EP	11 or 11 EP
	Reversing or frequent ''start/stop''	7 or 7 EP	6 or 6 EP	9–10[e] 9 EP–10 EP		

[a] AGMA lubricant numbers listed above refer to gear lubricants shown in Table 8.3.20. Physical and performance specifications are shown in Tables 1 and 2 of ANSI/ASMA 9005-D94. Although both R & O and EP oils are listed, the EP is preferred. Synthetic oils in the corresponding viscosity grades may be substituted where deemed acceptable by the gear manufacturer.
[b] Does not apply to worm gearing.
[c] Temperature in vicinity of the operating gears.
[d] When ambient temperatures approach the lower end of the given range, lubrication systems must be equipped with suitable heating units for proper circulation of lubricant and prevention of channeling. Check with lubricant and pump suppliers.
[e] When ambient temperature remains between 30°C (90°F) and 50°C (125°F) at all times, use 10 or 10 EP.
[f] When ambient temperature remains between 30°C (90°F) and 50°C (125°F) at all times, use 9 or 9 EP.
SOURCE: Abstracted from ANSI/AGMA 9005-D94, with permission.

Table 8.3.22 AGMA Lubricant Number Guidelines for Open Gearing Intermittent Applications[a,b,c]

Gear pitch line velocity does not exceed 7.5 m/s (1,500 ft/min)

	Intermittent spray systems[e]			Gravity feed or forced-drip method[g]	
Ambient temperature,[d] °C (°F)	R&O or EP lubricant	Synthetic lubricant	Residual compound[f]	R&O or EP lubricant	Synthetic lubricant
− 10–15 (15–60)	11 or 11 EP	11 S	14 R	11 or 11 EP	11 S
5–40 (40–100)	12 or 12 EP	12 S	15 R	12 or 12 EP	12 S
20–50 (70–125)	13 or 13 EP	13 S	15 R	13 or 13 EP	13 S

[a] AGMA viscosity number guidelines listed above refer to gear oils shown in Table 8.3.20.
[b] Does not apply to worm gearing.
[c] Feeder must be capable of handling lubricant selected.
[d] Ambient temperature is temperature in vicinity of gears.
[e] Special compounds and certain greases are sometimes used in mechanical spray systems to lubricate open gearing. Consult gear manufacturer and spray system manufacturer before proceeding.
[f] Diluents must be used to facilitate flow through applicators.
[g] EP oils are preferred, but may not be available in some grades.
SOURCE: Abstracted from ANSI/AGMA 9005-D94, with permission.

Table 8.3.23 AGMA Lubricant Number Guidelines for Enclosed Helical, Herringbone, Straight Bevel, Spiral Bevel, and Spur Gear Drives[a]

Pitch line velocity[b,c] of final reduction stage	AGMA lubricant numbers,[a,d,e] ambient temperature °C (°F)[f,g]			
	− 40 to − 10 (− 40 to + 14)	− 10 to + 10 (14 to 50)	10 to 35 (50 to 95)	35 to 55 (95 to 131)
Less than 5 m/s (1,000 ft/min)[h]	3 S	4	6	8
5–15 m/s (1,000–3,000 ft/min)	3 S	3	5	7
15–25 m/s (3,000–5,000 ft/min)	2 S	2	4	6
Above 25 m/s (5,000 ft/min)[h]	0 S	0	2	3

[a] AGMA lubricant numbers listed above refer to R&O and synthetic gear oil shown in Table 8.3.20. Physical and performance specifications are shown in Tables 1 and 3 of ANSI/AGMA 9005-D94. EP or synthetic gear lubricants in the corresponding viscosity grades may be substituted where deemed acceptable by the gear drive manufacturer.

[b] Special considerations may be necessary at speeds above 40 m/s (8,000 ft/min). Consult gear drive manufacturer for specific recommendations.

[c] Pitch line velocity replaces previous standards' center distance as the gear drive parameter for lubricant selection.

[d] Variations in operating conditions such as surface roughness, temperature rise, loading, speed, etc., may necessitate use of a lubricant of one grade higher or lower. Contact gear drive manufacturer for specific recommendations.

[e] Drives incorporating wet clutches or overrunning clutches as backstopping devices should be referred to the gear manufacturer, as certain types of lubricants may adversely affect clutch performance.

[f] For ambient temperatures outside the ranges shown, consult the gear manufacturer.

[g] Pour point of lubricant selected should be at least 5°C (9°F) lower than the expected minimum ambient starting temperature. If the ambient starting temperature approaches lubricant pour point, oil sump heaters may be required to facilitate starting and ensure proper lubrication (see 5.1.6 in ANSI/AGMA 9005-D94).

[h] At the extreme upper and lower pitch line velocity ranges, special consideration should be given to all drive components, including bearing and seals, to ensure their proper performance.

SOURCE: Abstracted from ANSI/AGMA 9005-D94, with permission.

Table 8.3.24 Typical Gear Lubricants

Lubricant type	Military specification	Useful temp range, °F	Commercial source and specification (a partial listing): Source	Identification	Remarks and applications
Oils					
Petroleum	MIL-L-644B	− 10 to 250	Exxon Corporation Franklin Oil and Gas Co. Royal Lubricants Co. Texaco	#4035 or Unvis P-48 L-499B Royco 380 1692 Low Temp. Oil	Good general-purpose lubricant for all quality gears having a narrow range of operating temperature
Diester	MIL-L-6085A	− 67 to 350	Anderson Oil Co. Eclipse Pioneer Div., Bendix Shell Oil Co. E. F. Houghton and Co.	Windsor Lube I-245X Pioneer P-10 AeroShell Fluid 12 Cosmolubric 270	General-purpose, low-starting torque, and stable over a wide temperature range. Particularly suited for precision instrument gears and small machinery gears
Diester	MIL-L-7808C	− 67 to 400	Sinclair Refining Co. Socony Mobil Oil Co. Bray Oil Co. Exxon Corporation	Aircraft Turbo S Oil Avrex S Turbo 251 Brayco 880 Exxon Turbine Oil 15	Suitable for oil spay or mist system at high temperature. Particularly suitable for high-speed power gears
Silicone		− 75 to 350	Dow-Corning Corp.	DC200	Rated for low-starting torque and lightly loaded instrument gears
Silicone		− 100 to 600	General Electric Co.	Versilube 81644	Best load carrier of silicone oils with widest temperature range. Applicable to power gears requiring wide temperature ranges
Greases					
Diester oil-lithium soap	MIL-G-7421A	− 100 to 200	Royal Lubricants Co. Texaco	Royco 21 Low Temp. No. 1888	For moderately loaded gears requiring starting torques at low temperatures
Diester oil-lithium soap	MIL-G-3278A	− 67 to 250	Exxon Corporation Shell Oil Co. Sinclair Refining Co. Bray Oil Co.	Beacon 325 AeroShell Grease II Sinclair 3278 Grease Braycote 678	General-purpose light grease for precision instrument gears, and generally lightly loaded gears
Petroleum oil-sodium soap	MIL-L-3545	− 20 to 300	Exxon Corporation Standard Oil Co. of Calif.	Andok 260 RPM Aviation Grease #2	A high-temperature lubricant for high speed and high loads
Mineral oil-sodium soap	—	− 25 to 250	Exxon Corporation	Andok C	Stiff grease that channels readily. Suitable for high speeds and highly loaded gears

Table 8.3.25 Solid Oil Additives

Lubricant type	Temperature range, °F	Source	Identification	Remarks
Colloidal graphite	Up to 1,000	Acheson Colloids Co.	SLA 1275	Good load capacity, excellent temperature resistance
Colloidal MoS_2	Up to 750	Acheson Colloids Co.	SLA 1286	Good antiwear
Colloidal Teflon	Up to 575	Acheson Colloids Co.	SLA 1612	Low coefficient of friction

they will provide long service life if the plastic chosen is correct for the application. Plastics manufacturers and their publications can be consulted for guidance. Alternatively, many plastic gear materials can be molded with internal solid lubricants, such as MoS_2, Teflon, and graphite.

GEAR INSPECTION AND QUALITY CONTROL

Gear performance is not only related to the design, but also depends upon obtaining the specified quality. Details of gear inspection and control of subtle problems relating to quality are given in Michalec, ''Precision Gearing,'' Chap. 11.

COMPUTER MODELING AND CALCULATIONS

A feature of the latest AGMA rating standards is that the graphs, including those presented here, are accompanied by equations which allow application of computer-aided design. Gear design equations and strength and durability rating equations have been computer modeled by many gear manufacturers, users, and university researchers. Numerous software programs, including integrated CAD/CAM, are available from these places, and from computer system suppliers and specialty software houses. It is not necessary for gear designers, purchasers, and fabricators to create their own computer programs.

With regard to gear tooth strength and durability ratings, many custom gear house designers and fabricators offer their own computer modeling which incorporates modifications of AGMA formulas based upon experiences from a wide range of applications.

The following organizations offer software programs for design and gear ratings according to methods outlined in AGMA publications: Fairfield Manufacturing Company Gear Software; Geartech Software, Inc.; PC Gears; Universal Technical Systems, Inc. For details and current listings, refer to AGMA's latest ''Catalog of Technical Publications.''

8.4 FLUID FILM BEARINGS

by Vittorio (Rino) Castelli

REFERENCES: ''General Conference on Lubrication and Lubricants,'' ASME. Fuller, ''Theory and Practice of Lubrication for Engineers,'' 2d ed., Wiley. Booser, ''Handbook of Lubrication, Theory and Design,'' vol. 2, CRC Press. Barwell, ''Bearing Systems, Principles and Practice,'' Oxford Univ. Press. Cameron, ''Principles of Lubrication,'' Longmans Greene. ''Proceedings,'' Second International Symposium on Gas Lubrication, ASME. Gross, ''Fluid-Film Lubrication,'' Wiley. Gunter, ''Dynamic Stability of Rotor-Bearing Systems,'' NASA SP-113, Government Printing Office.

Plain bearings, according to their function, may be

Journal bearings, cylindrical, carrying a rotating shaft and a radial load

Thrust bearings, the function of which is to prevent axial motion of a rotating shaft

Guide bearings, to guide a machine element in its translational motion, usually without rotation of the element

In exceptional cases of design, or with a complete **failure of lubrication,** a bearing may run dry. The coefficient of friction is then between 0.25 and 0.40, depending on the materials of the rubbing surfaces. With the **bearing barely greasy,** or when the bearing is well lubricated but the speed of rotation is very slow, boundary lubrication takes place. The coefficient of friction may vary from 0.08 to 0.14. This condition occurs also in any bearing when the shaft is starting from rest if the bearing is not equipped with an oil lift.

Semifluid, or **mixed,** lubrication exists between the journal and bearing when the conditions are not such as to form a load-carrying fluid film and thus separate the surfaces. Semifluid lubrication takes place at comparatively low speed, with intermittent or oscillating motion, heavy load, insufficient oil supply to the bearing (wick or waste-lubrication, drop-feed lubrication). Semifluid lubrication may also exist in thrust bearings with fixed parallel-thrust collars, in guide bearings of machine tools, in bearings with copious lubrication where the shaft is bent or the bearing is misaligned, or where the bearing surface is interrupted by improperly arranged oil grooves. The coefficient of friction in such bearings may range from 0.02 to 0.08 (Fuller, Mixed Friction Conditions in Lubrication, *Lubrication Eng.,* 1954).

Fluid or **complete lubrication,** when the rubbing surfaces are completely separated by a fluid film, provides the lowest friction losses and prevents wear. A certain amount of oil must be fed to the oil film in order to compensate for end leakage and maintain its carrying capacity. Such lubrication can be provided under pressure from a pump or gravity tank, by automatic lubricating devices in self-contained bearings (oil rings or oil disks), or by submersion in an oil bath (thrust bearings for vertical shafts).

Notation

R = radius of bearing, length
r = radius of journal, length
$c = mr = R - r$ = radial clearance, length
W = bearing load, force
μ = viscosity = force × time/length^2
Z = viscosity, centipoise (cP); 1 cP = 1.45×10^{-7} lb · s/in^2 (0.001 N · s/m^2)
β = angle between load and entering edge of oil film
η = coefficient for side leakage of oil
ν = kinematic viscosity = μ/ρ, length^2/time
R_e = Reynolds number = umr/ν
P_a = absolute ambient pressure, force/area
$P = W/(ld)$ = unit pressure, lb/in^2
N = speed of journal, r/min
m = clearance ratio (diametral clearance/diameter)
F = friction force, force
A = operating characteristic of plain cylindrical bearing
P' = alternate operating characteristic of plain cylindrical bearing
h_0 = minimum film thickness, length
ε = eccentricity ratio, or ratio of eccentricity to radial clearance
e = eccentricity = distance between journal and bearing centers, length
f = coefficient of friction
f' = friction factor = $F/(\pi r l \rho u^2)$
l = length of bearing, length
$d = 2r$ = diameter of journal, length

K_f = friction factor of plain cylindrical bearing
t_w = temperature of bearing wall
t_0 = temperature of air
t_1 = temperature of oil film
u = surface speed, length/time
ω = angular velocity, rad/time
ρ = mass density, mass/length^3
Λ = bearing compressibility parameter $= 6\mu\omega r^2/(P_a c^2)$

INCOMPRESSIBLE AND COMPRESSIBLE LUBRICATION

Depending on the fluid employed and the pressure regime, the fluid density may or may not vary appreciably from the ambient value in the load-carrying film. Typically, oils, water, and liquid metals can be considered incompressible, while gases exhibit compressibility effects even at modest loads. The difference comes from the fact that, in incompressible lubricants, fluid flow rates are linearly proportional to pressure differences, whereas for compressible lubricants the mass flow rates are proportional to the difference of some power of the pressure. This is because the pressure affects the fluid density. The bearing behavior is somewhat dissimilar. In incompressible lubrication, gage pressures can be used and the value of the ambient pressure has no effect on the load-carrying capacity, which is linearly related to viscosity and speed. This is not true in compressible lubrication, where the value of ambient pressure has a direct effect on the load-carrying capacity which, in turn, increases with viscosity and speed, but only up to a limit dependent on the bearing geometry. In what follows, incompressible lubrication is treated first and compressible lubrication second.

Incompressible (Plain Cylindrical Journal Bearings)

Fluid lubrication in plain cylindrical bearings depends on the viscosity of the lubricant, the speed of the bearing components, the geometry of the film, and possible external sources of pressurized lubricant. The oil is entrained by the journal into the film by the action of the viscosity which, if the passage is convergent, causes the creation of a pressure field, resulting in a force sufficient to float the journal and carry the load applied to it.

The **minimum film thickness** h_0 determines the closest approach of the journal and bearing surfaces (Fig. 8.4.1). The allowable closest approach depends on the finish of these surfaces and on the rigidity of the journal and bearing structures. In practice, $h_0 = 0.00075$ in (0.019 mm) is common in electric motors and generators of medium speed, with

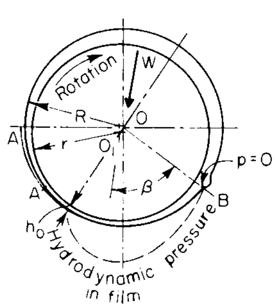


Fig. 8.4.1 Journal bearing with perfect lubrication.

steel shafts in babbitted bearings; $h_0 = 0.003$ in (0.076 mm) to 0.005 in (0.127 mm) for large steel shafts running at high speed in babbitted bearings (turbogenerators, fans), with pressure oil-supply for lubrication; $h_0 = 0.0001$ in (0.0025 mm) to 0.0002 in (0.005 mm) in automotive and aviation engines, with very fine finish of the surfaces.

Figure 8.4.2 gives the relationship between ε and the load-carrying coefficient A for a plain cylindrical journal. The operating characteristic of the bearing is

$$A = (132/\eta)(1{,}000m)^2[P/(ZN)]$$

In Fig. 8.4.1, β is the angle between the direction of the load W and the entering edge of the load-carrying oil film, in degrees. The entering edge is at the place where the hydrodynamic pressure is equal or nearly equal to the atmospheric pressure and may be at the location of the

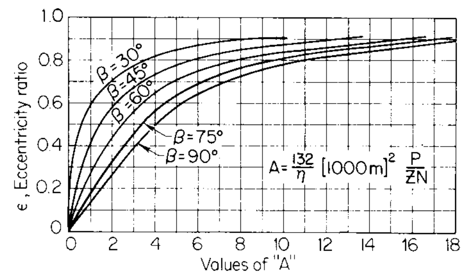


Fig. 8.4.2 Eccentricity ratio for a plain cylindrical journal.

oil-distributing groove B, or at the end of the machined recess pocket as at AA. For **complete bearings,** i.e., when the inner surface of the bearing is not interrupted by grooves, β may be taken as 90°. The reason for this assumption is the fact that, where the film diverges, the bearing pumping action tends to generate negative pressure, which liquids cannot sustain. The film **cavitates;** i.e., it breaks up in regions of fluid intermixed with either air or fluid vapor, while the pressure does not deviate substantially from ambient. For a 120° bearing with a central load, β may be taken as 60°.

The coefficient η corrects for side leakage. There is a loss of load-carrying capacity caused by the drop in the hydrodynamic pressure p in the oil film from the midsection of the bearing toward its ends; $p = 0$ at the ends. The value of η depends on the length-diameter ratio l/d and ε, the eccentricity ratio. Values of η are given in Fig. 8.4.3.

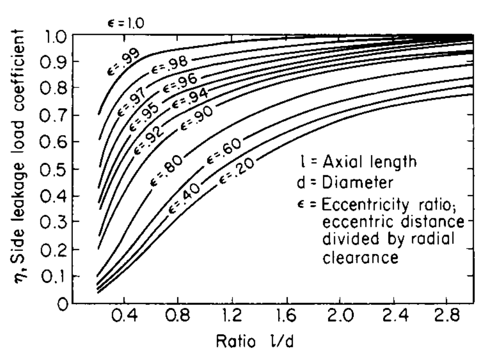


Fig. 8.4.3

EXAMPLE 1. A generator bearing, 6 in diam by 9 in long, carries a vertical downward load of 8,650 lb; $N = 720$ r/min. The diametral clearance of the bearing is 0.012 in; the bearing is split on its horizontal diameter, and the lower half is relieved 40° down on each side, for oil distribution along journal; the bearing arc is therefore 100°; with the load vertical, $\beta = 50°$; bearing temperature 160°F. The absolute viscosity of the oil in the film is 12 centipoises (medium turbine oil). $P = W/ld = 160$ lb/in^2; $\mu = 12 \times 1.45 \times 10^{-7} = 17.4 \times 10^{-7}$ lb · s/in^2. The solution is one of trial and error. By using Fig. 8.4.3 in conjunction with Fig. 8.4.2, only a few trials are necessary to obtain the answer. As a first trial assume $\varepsilon = 0.85$. For an l/d ratio of 1.5 in Fig. 8.4.3, η, the end-leakage factor, will be 0.77. Compute A using this value of η. $m = 0.012/6 = 0.002$.

$$A = \frac{132}{0.77}(2)^2\frac{160}{12 \times 720} = 12.7$$

Enter Fig. 8.4.2 with this value of a and at $\beta = 50°$, and find that $\varepsilon = 0.9$. This value is larger than the initial assumption for ε. As a second trial, $\varepsilon = 0.88$. Then $\eta = 0.8$, $A = 12.2$, and $\varepsilon = 0.89$. This is a sufficiently close check. The minimum film thickness is $h_0 = mr(1 - \varepsilon) = 0.002 \times 3 \times 0.12 = 0.0007$ in (0.01778 mm).

For severe operating conditions the value of A may exceed 18, the limit of Fig. 8.4.2. For complete journal bearings under extreme operating conditions, Fig. 8.4.4 should be used. The ordinate is P', defined as shown. The curves are drawn for various values of l/d instead of values of β as in Fig. 8.4.2. Values of ε may thus be obtained directly (Dennison, Film-Lubrication Theory and Engine-Bearing Design, *Trans. ASME,* **58,** 1936).

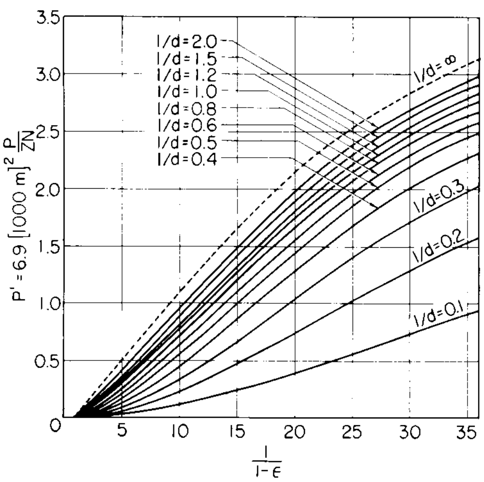


Fig. 8.4.4 Load-carrying parameter in terms of eccentricity.

EXAMPLE 2. A 360° journal bearing 2½ in diam and 3⅞ in long carries a steady load of 3,875 lb. Speed $N = 500$ r/min; diametral clearance, 0.0064 in; average viscosity of the oil in the film, 23.4 centipoises (SAE 20 light motor oil at 105°F). $P = 3{,}875/(2.5 \times 3.875) = 400$ lb/in². Value of $m = 0.0064/2.5 = 0.00256$. Value of $l/d = 1.55$. First, attempt to use Figs. 8.4.2 and 8.4.3 in this solution. Assume eccentricity ratio ε is 0.9. Then, in Fig. 8.4.3, with $l/d = 1.55$, value of η is determined as 0.8. A is calculated as 37. This is completely off scale in Fig. 8.4.2. Consider instead Fig. 8.4.4. Value of P' is computed as

$$P' = 6.9(2.56)^2 \frac{400}{23.4 \times 500} = 1.54$$

In Fig. 8.4.4, enter the curves with $P' = 1.54$, and move left to intersect the curve for $l/d = 1.5$. Drop downward to read a value for $1/(1 - \varepsilon)$ of 16. Then $1/16 = 1 - \varepsilon$, or the eccentricity ratio $\varepsilon = 15/16$, or 0.94. The minimum film thickness, as in Example $1 = h_0 = mr(1 - \varepsilon)$, or

$$h_0 = 0.00256 \times 1.25(1 - 0.94) = 0.0002 \text{ in } (0.0051 \text{ mm})$$

Allowable mean bearing pressures in bearings with fluid film lubrication are given in Table 8.4.1. If the load maintains the same magnitude and direction when the journal is at rest (heavily loaded shafts, heavy gears), the mean bearing pressure should be somewhat less than when bearings are loaded only when running.

For internal-combustion-engine bearing design, Etchells and Underwood (*Mach. Des.,* Sept. 1942) list the following maximum design pressures for bearing alloys, pounds per square inch of projected area: lead-base babbitt (75 to 85 percent lead, 4 to 10 percent tin, 9 to 15 percent antimony) 600 to 800; tin-base babbitt (0.35 to 0.6 percent lead, 86 to 90 percent tin, 4 to 9 percent antimony, 4 to 6 percent copper) 800 to 1,000; cadmium-base alloy (0.4 to 0.75 percent copper, 97 percent cadmium, 1 to 1.5 percent nickel, 0.5 to 1.0 percent silver) 1,200 to 1,500; copper-lead alloy (45 percent lead, 55 percent copper) 2,000 to 3,000; copper-lead (25 percent lead, 3 percent tin, 72 percent copper) 3,000 to 4,000; silver (0.5 to 1.0 percent lead on surface, 99 percent silver) 5,000 up. The above pressures are based on fatigue life of 500 h at 300°F bearing temperature, and a bearing metal thickness 0.01 to 0.015 in for lead-, tin-, and cadmium-base metals and 0.25 in for copper, lead, and silver. At lower temperatures the life will be greatly extended.

Much higher pressures are encountered in rolling element bearings, such as ball and roller bearings, and gears. In these situations, the formation of fluid films capable of preventing contact between surface asperities is aided by the increase of viscosity with pressure, as exhibited by most lubricating oils. The relation is typically exponential, $\mu = \mu_0 e^{\alpha p}$, where α is the so-called pressure coefficient of viscosity.

Length-diameter ratios are usually chosen between $l/d = 1$ and $l/d = 2$, although many engine bearings are designed with $l/d = 0.5$, or even less. In shorter bearings, the carrying capacity of the oil film is greatly impaired by the effect of side leakage. Longer bearings are used to restrain the shaft from vibration, as in line shafts, or to position the shaft accurately, as in machine tools. In power machines, the tendency is toward shorter bearings. Typical values are as follows: turbogenerators, 0.8 to 1.5; gasoline and diesel engines for main and crankpin bearings, 0.4 to 1.0, with most values between 0.5 and 0.8; generators and motors, 1.5 to 2.0; ordinary shafting, heavy, with fixed bearings, 2 to 3; light, with self-aligning bearings, 3 to 4; machine-tool bearings, 2 to 4; railroad journal bearings, 1.2 to 1.8.

For the **clearance between journal and bearing** see Fits in Sec. 8. Medium fits may be used for journals running at speeds under 600 r/min, and free fits for speeds over 600 r/min. Kingsbury suggests for these journals a diametral clearance $= 0.002 + 0.001d$ in. In journals running at high speed, diametral clearance $= 0.002d$ should be used in order to lower the friction losses in the bearing. All units are in inches. The most satisfactory clearance should, of course, be based on a complete bearing analysis which includes both load-carrying capacity and heat generation due to friction. For example, a bearing designed to run at the extremely high speed of 50,000 r/min uses a diametral clearance of 0.0025 in for a journal with 0.8-in diameter, giving a clearance ratio, clearance/diameter, of 0.00316.

Table 8.4.1 Current Practice in Mean Bearing Pressures

Type of bearing	Permissible pressure, lb/in², of projected area	Type of bearing	Permissible pressure, lb/in², of projected area
Diesel engines, main bearings	800–1,500	Automotive gasoline engines, main bearings	500–1,000
Crankpin	1,000–2,000	Crankpin	1,500–2,500
Wrist pin	1,800–2,000	Air compressors, main bearings	120–240
Electric motor bearings	100–200	Crankpin	240–400
Marine diesel engines, main bearings	400–600	Crosshead pin	400–800
Crankpin	1,000–1,400	Aircraft engine crankpin	700–2,000
Marine line-shaft bearings	25–35	Centrifugal pumps	80–100
Steam engines, main bearings	150–500	Generators, low or medium speed	90–140
Crankpin	800–1,500	Roll-neck bearings	1,500–2,500
Crosshead pin	1,000–1,800	Locomotive crankpins	1,500–1,900
Flywheel bearings	200–250	Railway-car axle bearings	300–350
Marine steam engine, main bearings	275–500	Miscellaneous ordinary bearings	80–150
Crankpin	400–600	Light line shaft	15–25
Steam turbines and reduction gears	100–220	Heavy line shaft	100–150

For high-speed internal-combustion-engine bearings using forced-feed lubrication, medium fits are used. Federal-Mogul recommends the following diametral clearances in inches per inch of shaft diameter for insert-type bearings: tin-base and high-lead babbitts, 0.0005; cadmium-silver-copper, 0.0008; copper-lead, 0.001.

The dependence of the **coefficient of friction** for journal bearings on the bearing clearance, lubricant viscosity, rotational speed, and loading pressure, as reported by McKee and others, is shown in Sec. 3. A plot of the coefficient of friction against the parameter ZN/P is a convenient method for showing this relationship. ZN/P is a parameter based on mixed units. Z is the viscosity in centipoise, N is r/min, P is the mean pressure on the bearing due to the load, pounds per square inch of projected area, and m is the clearance ratio. Values of ZN/P greater than about 30 indicate fluid film conditions in the bearings. If the viscosity of the lubricant becomes lower or if there is a reduction in rotational speed or an increase in load, the value of ZN/P will become smaller until the coefficient of friction reaches a minimum value. Any further reduction in ZN/P will produce breakdown of the oil film, marking the transition from fluid film lubrication with complete separation of the moving surfaces to semifluid or mixed lubrication, where there is partial contact. As soon as semifluid conditions are initiated, there will be a sharp increase in the coefficient of friction. The critical value of ZN/P, where this transition takes place, will be lowest for a rigid bearing and shaft with finely finished surfaces.

Figure 8.4.5 shows a generalization of the relationship between the coefficient of friction for a journal bearing and the parameter ZN/P,

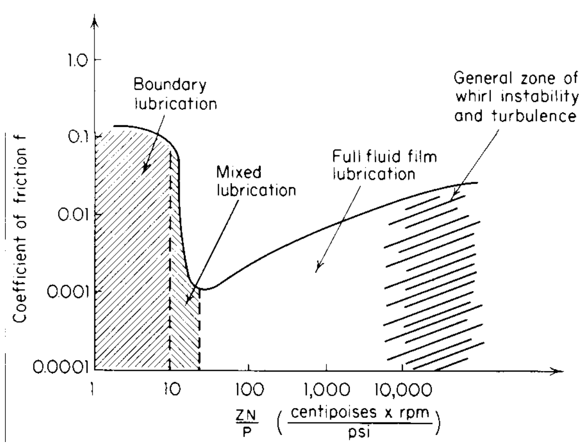


Fig. 8.4.5 Various zones of possible lubrication for a journal bearing.

indicating the various possible lubrication regimes that may be expected. For optimum design, a value of ZN/P somewhere between 30 and 300 would be recommended, but, in any case, the determination of minimum film thickness h_0 should be the deciding parameter. For extremely large values of ZN/P, resulting from high speeds and low loads,

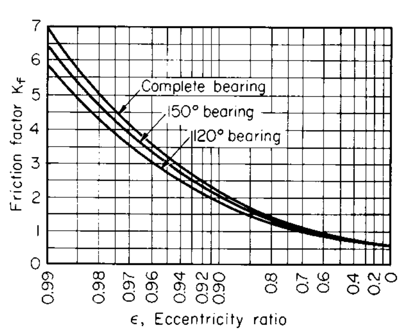


Fig. 8.4.6 Variation of the friction factor of a bearing with eccentricity ratio.

whirl instability may be developed. (See material on gas-lubricated bearings in this section.) With large values of ZN/P and a lubricant having a low kinematic viscosity, turbulent conditions may develop in the bearing clearance.

The friction force in plain journal bearings may be estimated by the use of the expression $F = K_f \mu Nrl/m$, where μ is in lb · s/in² units. The value of K_f depends upon the magnitude of ε and the type of bearing. Figure 8.4.6 shows values of K_f for a complete bearing, a 150° partial bearing, and a 120° partial bearing, assuming that the clearance space is at all times filled with lubricant. Note that F is the friction force at the surface of the bearing. Consequently, the friction torque is obtained by multiplying F by the bearing radius.

EXAMPLE 3. As an illustration of the use of Fig. 8.4.6, determine the friction force in the bearing of Example 2. This is a complete journal bearing 2½-in diam by 3⅞ in. The value of ε was determined as 0.94. From Fig. 8.4.6, $K_f = 2.8$. Then

$$F = \frac{2.8 \times 23.4 \times 1.45 \times 10^{-7} \times 500 \times 1.25 \times 3.875}{0.00256}$$
$$= 8.97 \text{ lb } (4.08 \text{ kg})$$

The coefficient of friction $F/W = 8.97/3875 = 0.00231$. The mechanical loss in the bearing is $FV/33{,}000$ hp, where V is the peripheral velocity of the journal, ft/min.

$$\text{Friction hp} = (8.97 \times 500 \times \pi \times 2.5)/(33{,}000 \times 12)$$
$$= 0.089 \text{ hp } (66.37 \text{ W})$$

Departure from laminarity in the fluid film of a journal bearing will increase the friction loss. Figure 8.4.7 (Smith and Fuller, Journal Bearing Operation at Super-laminar Speeds, *Trans. ASME*, **78**, 1956) shows test results for such bearings, expressed in terms of a Reynolds number for the fluid film, $R_e = umr/\nu$. Laminar conditions hold up to an R_e of about 1,000. Friction may be calculated for laminar flow by using Fig. 8.4.6 or the left branch of the curve in Fig. 8.4.7, where $f' = 2/R_e$, and which applies to low values of the eccentricity ratio ($K_f = 0.66$). The values from Fig. 8.4.7 may be converted to friction torque T by the use of the expression $T = f' \pi \rho u^2 r^2 l$, where ρ is the mass density of the lubricant. In Fig. 8.4.7, a transition region spans values of the Reynolds number from 1,000 to 1,600. Here, two types of flow instability can occur. Usually, the first is due to **Taylor vortices** which are wrapped in

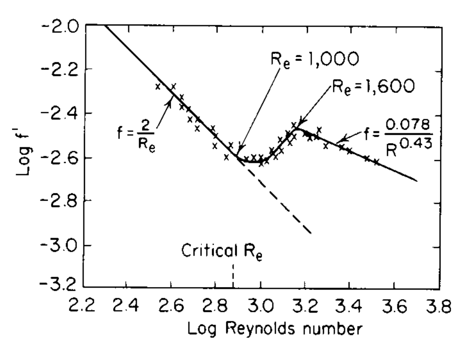


Fig. 8.4.7 Friction f' as a function of the Reynolds number for an unloaded journal bearing with $l/d = 1$. *(Smith and Fuller.)*

regular circumferential structures, each of which occupies the entire clearance. The onset of this phenomenon takes place at a value of the Reynolds number exceeding the threshold $R_e = 41.1(r/c)^{1/2}$. The second instability is due to turbulence, occurring at $R_e > 2{,}000$.

EXAMPLE 4. A journal bearing is 4.5 in diameter by 4.5 in long. Speed 22,000 r/min. $mr = 0.002$ in. Viscosity μ, 1 cP (water) $= 1.45 \times 10^{-7}$ lb · s/in²; mass density $\rho = 62.4/1{,}728 \times 386 = 9.35 \times 10^{-5}$ lb · s²/in⁴; $\nu = \mu/\rho = 1.45 \times 10^{-7}/9.35 \times 10^{-5} = 0.155 \times 10^{-2}$ in²/s; $u = 22{,}000 \times 2\pi \times 2.25/60 = 5{,}180$ in/s; $R_e = 5{,}180 \times 0.002/0.155 \times 10^{-2} = 6{,}680$. This would indicate turbulence in the film. Value of f' is then $0.078/6{,}680^{0.43} = 0.078/44.2 = 1.765 \times 10^{-3}$. Friction torque $T = 1.765 \times 10^{-3} \times \pi \times 9.35 \times 10^{-5} \times 5{,}180^2 \times 2.25^2 \times 4.5$, $T = 317.5$ in · lb. Friction horsepower $= 2\pi TN/12 \times 33{,}000 = 2\pi \times 317.5 \times 22{,}000/12 \times 33{,}000$, FHP $= 111$ (82.77 kW).

In self-contained bearings (electric motor, line shaft, etc.) without external oil or water cooling, the **heat dissipation** is equal to the heat generated by friction in the bearing.

The heat dissipated from the outside bearing wall to the surrounding air is governed by the laws of heat transfer $Q = hS(t_w - t_0)$, where S is the surface area from which the heat is convected, Q is the rate of energy flow; t_w and t_0 are the temperatures of the wall and ambient air, respectively; and h is the heat convection coefficient, which has values from 2.2 Btu/(h · ft² · °F) for still air to 6.5 Btu/(h · ft² · °F) for air moving at 500 ft/min. Calculations of heat loss are extremely important due to the strong temperature dependence of the viscosity of most oils.

The temperature of the oil film will be higher than the temperature of the bearing wall. Typical ranges of values according to Karelitz (*Trans. ASME*, **64**, 1942), Pearce (*Trans. ASME*, **62**, 1940), and Needs (*Trans. ASME*, **68**, 1948) for self-contained bearings with oil bath, oil ring, and waste-packed lubrication are shown in Fig. 8.4.8.

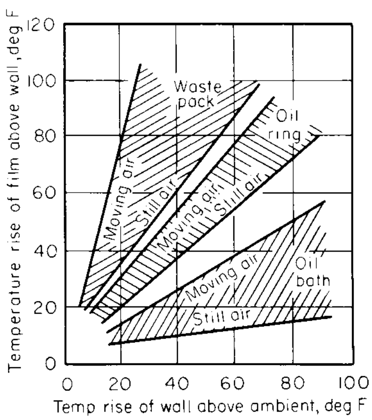


Fig. 8.4.8 Temperature rise of the film.

EXAMPLE 5. The frictional loss for the generator bearing of Example 1, computed by the method outlined in Example 3, is 0.925 hp with $\varepsilon = 0.88$, $K_f = 1.6$, and $F = 27$ lb. Operating in moving air the heat dissipated by the bearing housing will be $L = 6.5S(t_w - t_0)$. Since this is a self-contained bearing, the heat dissipated is also equal to the heat generated by friction in the oil film, or $L = 0.925 \times 2{,}545 = 2{,}355$ Btu/h. With $S = 25 \times 6 \times 9/144 = 9.4$ ft², $t_w = t_0 = 2{,}355/6.5 \times 9.4 = 38.5$°F. This is the temperature rise of the bearing wall above the ambient room temperature. For an 80°F room, the wall temperature of the bearing would be about 118°F. In Fig. 8.4.8 an oil-ring bearing in moving air with a temperature rise of wall over ambient of 38°F should have a film temperature 50°F higher than that of the wall. The film temperature on the basis of Fig. 8.4.8 will then be 80 + 38 + 50, or 168°F. This is close enough to the value of the film temperature of 160°F from Example 1, with which the friction loss in the bearing was computed, to indicate that this bearing can operate without the need for external cooling.

To predict the operating temperature of a self-contained bearing, the cut-and-try method shown above may be used. First, an oil-film temperature is assumed. Viscosity and friction losses are calculated. Then the temperature rise of the wall over ambient is computed so as to dissipate to the atmosphere an amount of heat equal to the friction loss. Lastly from Fig. 8.4.8 the corresponding oil-film temperature is estimated and compared to the value that was originally assumed. A few adjustments of the assumed film temperature will produce satisfactory agreement and indicate the leveling-off temperature of the bearing. Self-contained bearings have been built with diameters of 3, 8, and 24 in (7.62, 20.32, and 60.96 cm) to operate at shaft speeds of 3,600, 1,000, and 200 r/min, respectively. These designs indicate a rough limit for bearings with no external cooling. The highest bearing temperature permissible with normal lubricants is about 210°F (100°C).

The temperature of automotive-type bearings is held within safe limits by using a **pressure-feed oil supply.** Sufficient lubricant is forced through the bearing to act as a coolant and prevent overheating. One widely used practice is to place a circumferential groove at the center of the bearing to which the oil supply is fed. This is effective as far as cooling is concerned but has the disadvantage of interrupting the active length of the bearing and lowering its *l/d* ratio (see Fig. 8.4.9). The axial flow through each side of the bearing is given by

$$Q_1 = \frac{\Delta P m^3 r^4 \pi}{6\mu b}\left(1 + \frac{3}{2}\varepsilon^2\right)$$

where b is the effective axial length of the half bearing and ΔP is the difference between the oil pressure in the circumferential groove and

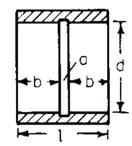


Fig. 8.4.9 Bearing with central circumferential groove.

the pressure at the ends of the bearing. The value of the last term in this equation will vary from 1.0 for a concentric shaft and bearing indicated by $\varepsilon = 0$ to a value of 2.5 for the extreme case of the shaft touching the bearing wall, indicated when $\varepsilon = 1$. Most of the heat caused by friction in the bearing is carried away by the circulating oil. Permissible temperature rises for this type of bearing may range from 15 to 50°F (8 to 28°C). In extreme cases a rise of 100°F (55°C) can be tolerated for high-strength bearing materials. The lower values of temperature rise usually indicate needlessly large oil flow. Such a condition will result in an excessive friction loss in the bearing.

EXAMPLE 6. The bearing of Examples 2 and 3 is lubricated by a circumferential groove with an oil supply pressure of 30 lb/in² and, as before, $\varepsilon = 0.94$, $m = 0.0026$, and $\mu = 23.4 \times 1.45 \times 10^{-7}$ lb · s/in². Length b is about 1.93 in.

$$Q_1 \text{ flow out one side} = \frac{30 \times 0.0026^3 \times 1.25^4 \times \pi}{6 \times 23.4 \times 1.45 \times 10^{-7} \times 1.93}$$

$$\times [1 + 3/2(0.94)^2] = 0.240 \text{ in}^3/\text{s } (3.93 \text{ cm}^3/\text{s})$$

Total flow (two sides) = 0.48 in³/s = 53 lb/h for sp gr = 0.85. The friction loss from Example 3 = 0.089 hp = 226 Btu/h. With a specific heat of 0.5 Btu/(lb · °F) and assuming that all the friction energy is given up to the oil in the form of heat, the temperature rise $\Delta t = 226/0.5 \times 53 = 8.5$°F (4.72°C).

A definite **minimum rate of oil feed** is required to maintain a fluid film in journal bearings. This makes no allowance for the additional flow that may be needed to cool the bearings. However, many industrial bearings run at relatively low speeds with light loads and, as a consequence, additional oil flow to provide cooling is not necessary. But if a fluid film is desired, a definite minimum amount of lubricant is required. If the volume of lubricant fed to the bearing is less than this minimum requirement, there will not be a complete fluid film in the bearing. Friction will rise, wear will become greater, and the satisfactory service life of such a bearing will be reduced. This minimum lubricant supply can be evaluated by using the equation

$$Q_M = K_M urml$$

where Q_M is the flow rate and K_M is approximately 0.006.

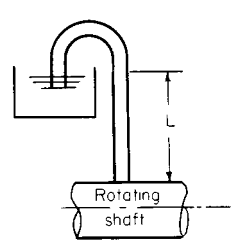


Fig. 8.4.10 Siphon wick.

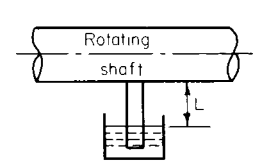


Fig. 8.4.11 Bottom wick.

EXAMPLE 7. The minimum feed rate for a journal bearing 2⅛-in diam by 2⅛ in long will be determined. Diametral clearance is 0.0045 in; speed, 1,230 r/min; load, 40 lb/in² based on projected area. $u = 1{,}230 \times \pi \times 2.125 = 10{,}220$ in/min, $r = 1.062$ in, $m = 0.0045/2.125 = 0.00212$, $l = 2.125$ in. Substituting,

$$Q_M = 0.006 \times 10{,}220 \times 1.062 \times 0.00212 \times 2.125 = 0.28 \text{ in}^3/\text{min}$$

(Fuller and Sternlicht, Preliminary Investigation of Minimum Lubricant Requirements of Journal Bearings, *Trans. ASME,* **78**, 1956.)

Many bearings are supplied with oil at low rates of feed by **felts, wicks,** and **drop-feed oilers.** Wicks can supply substantial rates of feed if they are properly designed. The two basic types of wick feed are siphon wicks, as shown in Fig. 8.4.10, and bottom wicks, as shown in Fig. 8.4.11. Data on oil delivery for these wicks are shown in Figs. 8.4.12 and 8.4.13. The data, from the American Felt Co., are for SAE Fl felts, based on a cross-sectional area of 0.1 in². The flow rate is indicated in drops per minute. One drop equals 0.0026 in³ or 0.043 cm³.

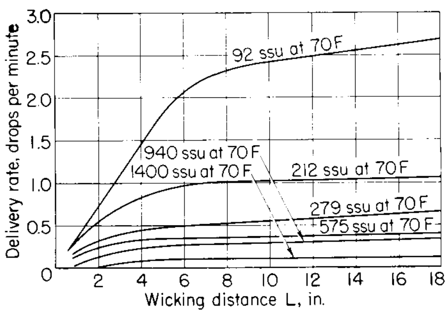


Fig. 8.4.12 Oil delivery with siphon wick (Fig. 8.4.10).

EXAMPLE 8. If it is desired to deliver 12.5 drops/min to a journal bearing, and if the viscosity of the oil is 212 s Saybolt Universal at 70°F, and if L, Fig. 8.4.10, is 5 in, what size of round wick would be required? From Fig. 8.4.12, for the stated conditions the delivery rate would be 0.9 drop/min for an area of 0.1 in². If 12.5 drops/min is needed, this would mean an area of 12.5 divided by 0.9 and multiplied by 0.1, or 1.4 in². For a round wick this would mean a diameter of 1⅜ in (3.49 cm).

If a **bottom wick** is considered with $L = 4$ in, Fig. 8.4.11, then in Fig. 8.4.13 the delivery rate using the same oil would be 1.6 drops/min; and if 12.5 drops/min is required, the area would be 12.5 divided by 1.6 and multiplied by 0.1, or 0.78 in². This would mean a bottom wick of 1 in diam if it is round (2.54 cm).

When journal **bearings** are **started, stopped,** or **reversed,** or whenever conditions are such that the operating value of ZN/P falls below the critical value for that bearing, the oil film will be ruptured and metal-to-metal contact will increase friction and cause wear. This condition can be eliminated by using a **hydrostatic oil lift.** High-pressure oil is introduced to the area between the bottom of the journal and the bearing (Fig. 8.4.14). If the pressure and quantity of flow are great enough, the shaft, whether it is rotating or not, will be raised and supported by an oil film. Neglecting axial flow, which is small, the flow up one side is

$$Q_1 = \frac{Wrm^3}{A\mu} \qquad \text{in}^2/\text{s}$$

and the inlet pressure required, $P_o = \mu Q_1 B/(br^2m^3)$, where b is the axial length of the high-pressure recess. Values of A and B are dimensionless factors which represent geometric effects and are given in the following table as a function of ε:

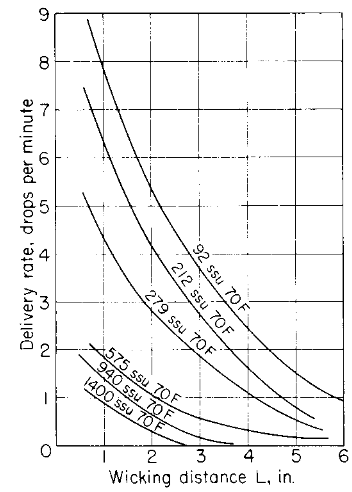


Fig. 8.4.13 Oil delivery with bottom wick (Fig. 8.4.11).

Current practice is to make the total area of the high-pressure recess in a bearing 2½ to 5 percent of the projected area ld of the bearing. It is generally desirable to use a check valve in the supply line to the oil lift so that, when the journal builds up a hydrodynamic oil-film pressure, reverse flow of oil in the supply line will be prevented.

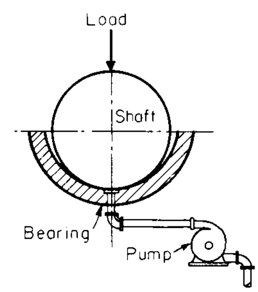


Fig. 8.4.14 Diagram of oil lift.

EXAMPLE 9. A 4,000-in-diam journal rests in a bearing of 4.012-in-diam. SAE 30 oil at 100°F (105 cP) is supplied under pressure to a groove at the lowest point in the bearing. Length of bearing, 6 in, length of groove, 3 in, load on bearing, 3,600 lb. What inlet pressure and oil flow are needed to raise the journal 0.004 in?

$$\begin{aligned} h_0 &= mr(1 - \varepsilon) \\ 0.004 &= 0.006(1 - \varepsilon) \\ \varepsilon &= 0.333 \end{aligned}$$

From the table, $A = 44.5$, $B = 42$.

$$\begin{aligned} Q_1 &= \frac{3{,}600 \times 2}{44.5 \times 105 \times 1.45 \times 10^{-7}} (0.003)^3 \\ &= 0.287 \text{ in}^3/\text{s, one side } (4.70 \text{ cm}^3/\text{s}) \end{aligned}$$

Flow from both sides = (0.287 × 2) × 60⁄231 = 0.149 gal/min (0.564 l/min).

Oil supply pressure is

$$P_o = \frac{105 \times 1.45 \times 10^{-7} \times 0.287 \times 42}{3 \times 4} \times \frac{1}{0.003^3} = 566 \text{ lb/in}^2$$

ε	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.75	0.8	0.85	0.9
A	24.0	28.1	33.8	41.6	53.3	72.0	105	173	237	360	613	1,320
B	18.9	23.2	29.0	38.2	52.7	77.9	128	246	344	634	1,260	3,360

ε	0.91	0.92	0.93	0.94	0.95	0.96	0.97	0.98	0.99
A	1,620	2,070	2,620	3,530	5,040	7,800	13,700	30,600	121,000
B	4,340	5,810	8,040	11,800	18,400	32,100	65,300	179,000	348,000

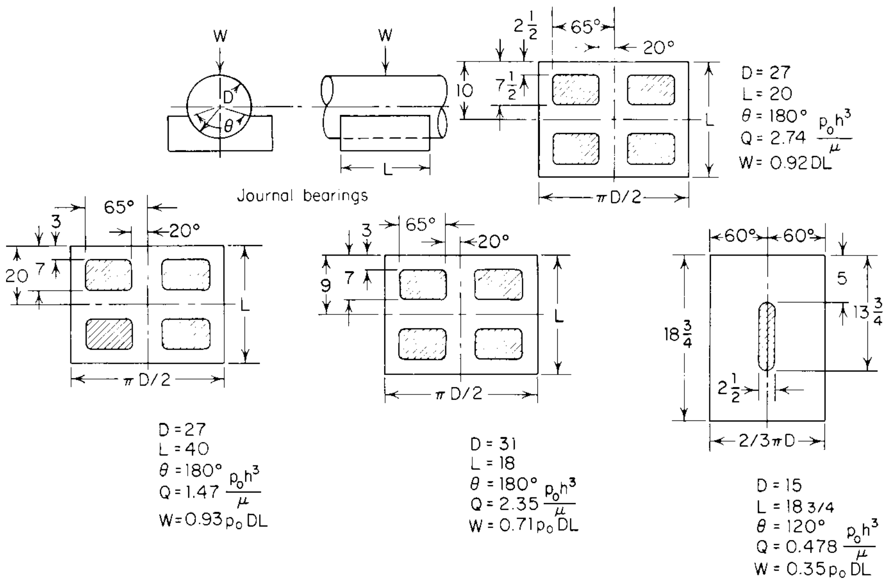


Fig. 8.4.15 Load-carrying capacity and flow for journal bearings *(Loeb).* Lengths in inches.

An adjustable constant-volume pump or a spur-gear pump with a capacity of about 1,000 lb/in^2 (6.894 kN/m^2) should be used to allow for pressure that may be built up in the line before the journal begins to rise.

Other configurations for hydrostatically lubricated journal bearings are shown in Fig. 8.4.15. These were obtained by means of electric analog solutions (Loeb, Determination of Flow, Film Thickness and Load-Carrying Capacity of Hydrostatic Bearings through the Use of the Electric Analog Field Plotter, *Trans. ASLE,* **1**, 1958). The data from Fig. 8.4.15 are exact for a uniform film thickness corresponding to $\varepsilon = 0$ but may be used with discretion for other values of ε.

Multiple recesses are used in externally pressurized bearings in order to provide local **stiffness**. This term indicates that the bearing resists shaft motions in any direction, and it is achieved by properly arranging the feeding network according to a strategy called **compensation**. Three main types are employed: orifice (and its variant, inherent), capillary, and fixed flow rates. In the first two, the idea is to insert a hydraulic resistance in each of the recess feeding lines and to use a single pump to feed all recesses. The flow rate q through orifices varies with the square root of the pressure drop Δp

$$q \propto \sqrt{\Delta p}$$

while for capillary tubes the relation is linear:

$$q = \frac{\pi \, \Delta p \, d^4}{64 \, l_1 \, \mu}$$

The general rule of thumb in designing orifices or capillary restrictors is to generate a pressure drop approximately equal to that taking place through the bearing, i.e., from the recesses to the ambient. The recess geometry and distribution, on the other hand, are designed so that $W = 0.5 p_{\text{recess}} \, DL$. Thus, the pump supply pressure is 4 times the average bearing pressure. The bearing stiffness is usually equal to $K = 0.5 p_{\text{recess}} \, DL/c$.

The third method of compensation consists of forcing the same amount of flow to reach each recess regardless of clearance distribution. This can be achieved either by using separate pumps for each recess or by using a hydraulic device called a *flow divider.* With recess distributions as indicated above, the pump pressure need only be double the average bearing pressure; thus, this method of compensation leads to half the power dissipation of the other two. It is commonly used in large machinery, where power consumption must be limited. The polar axis bearings of the 200-in Hale telescope on Mount Palomar were the first large-scale demonstration of this technique. The azimuth axis thrust bearing of the 270-ft-diameter Goldstone radio telescope is probably the largest example of this type of bearing.

ELEMENTS OF JOURNAL BEARINGS

Typical dimensions of solid and split **bronze bushings** are given in Table 8.4.2.

Bronze bushings made from hard-drawn sheets and rolled into cylindrical shape are made with a wall thickness of only 1⁄32 in for bearings up to ½ in diam and with a wall thickness of 1⁄16 in for bearings from 1 in diam up. The wall thickness of these bearings depends chiefly upon the strength of the material which supports them. Bushings of this type are pressed into place, and the bearing surface is finished by burnishing with a slightly tapered bar to a mirror finish. The allowable bearing pressures may exceed those of cast bronze shown in Table 8.4.1 by 10 to 20 percent.

Babbitt linings in larger bearings are generally employed in thickness of ⅛ in or over and must be provided with sufficient anchorage in the

Table 8.4.2 Wall Thickness of Bronze Bushings, in

	Diam of journal, in						
	¼	¼–½	½–1	1–1½	1½–2½	2½–4	4–5½
Solid bushing, normal	1⁄16	3⁄32	⅛	3⁄16	¼	⅜	½
Split bushing, normal	3⁄32	⅛	5⁄32	7⁄32	5⁄16	15⁄32	⅝
Solid bushing, thin	1⁄16	3⁄32	3⁄32	⅛	3⁄16	¼	⅜
Split bushing, thin	1⁄16	3⁄32	⅛	3⁄16	¼	⅜	½

supporting shell. The anchors take the form of dovetailed grooves or holes drilled in the shell and counterbored from the outside.

Improved conditions are obtained by sweating or bonding the babbitt to the shell by tinning the latter, using potassium chlorate as flux. Tin-base babbitts and other low-strength materials evidence some yielding when subjected to heavy pressures. This tendency may be alleviated by the use of a thinner layer of the bearing material, fused either to a bronze or to a steel shell. This improves the fatigue life of the bearing material. Standard bearing inserts of this type are available in tin-base babbitts, high-lead babbitts, cadmium alloys, and copper-lead mixtures in diameters up to about 6 in (15.24 cm) (Fig. 8.4.16). A few materials can be obtained in sizes up to 8 in (20.32 cm). Some types are available with flanges or with other special features. The bearing lining may vary from about 0.001 in (0.025 mm) to 0.1 in (2.5 mm) in thickness depending upon the size of the bearing.

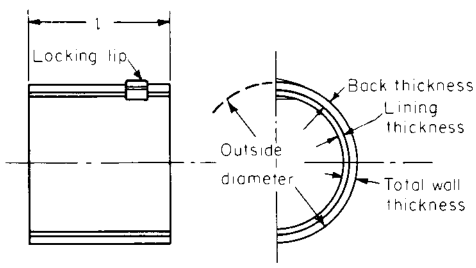


Fig. 8.4.16 Bearing insert.

Figure 8.4.17 shows the principal types of bonded babbitt linings. Figure 8.4.17*a* is for normal operating conditions. Figure 8.4.17*b* is for more severe operating conditions.

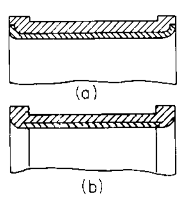


Fig. 8.4.17

General practice for the **thickness of babbitt lining and shells** is as follows: Fig. 8.4.18, $b = \frac{1}{32}d + \frac{1}{8}$ in, $S = 0.18d$ for bronze or steel $= 0.2d$ for cast iron; Fig. 8.4.18*a*, $t = b/2 + \frac{1}{16}$ in, $W = 1.8t$, $W_1 = 2.2t$.

Solid bronze or steel bushings, when pressed into the bearing housing, must be finished after pressing in. Light press fits and securing by

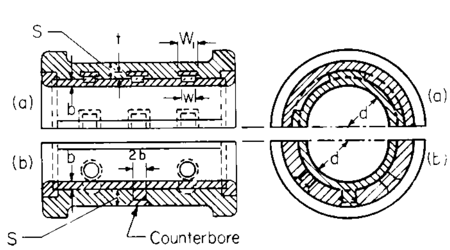


Fig. 8.4.18

setscrews or keys are preferable to heavy press fits and no keying, since heavy pressure, especially in thin-walled bushings, will set up stresses which will release themselves if bearings should run hot in service and will result in closing in on the journal and scoring when cooling.

Uniform Load Distribution Misalignment between journal and bearing should never be so great as to cause metallic contact. The maximum allowable inclination α of the shaft to the bearing is given by $\tan \alpha = md/l$.

Whenever the deflection angle of the bearing installation is greater than α, either the bearing length should be reduced or, if that is not feasible, the bearing should be mounted on a spherical seat to permit self-alignment.

Oil grooves are of two kinds, axial and circumferential; the former distribute the oil lengthwise in the bearing; the latter distribute it around the shaft at the oil hole, and also collect and return oil which would otherwise be forced out at the ends of the bearing. Grooves have often been put into bearings indiscriminatingly, with the result that they scrape off the oil and interrupt the film.

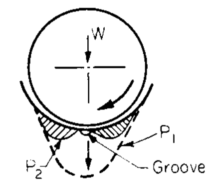


Fig. 8.4.19

In Fig. 8.4.19, *W* is the resultant force or load, pounds, on the bearing or journal. The radial ordinates P_1, to the dotted curve, show the pressures, lb/in², of the journal on the oil film due to the load when there is no axial groove, while the ordinates P_2, to the solid curve, show the pressures with an incorrectly located groove. Since there is no oil pressure near the groove, the permissible load *W* must be reduced or the film will be ruptured.

Groove dimensions (Fig. 8.4.20) are given by the following relations: $a = \frac{1}{3}$ wall thickness; $W_o = 2.5a$; $W_d = 3a$; $c = 0.5W_d$; $f = \frac{1}{16}$ in to $0.5W_d$.

In order to maintain the oil film, **the axial distributing groove should be placed in the unloaded sector** of the bearing. The location of grooves in a variety of cases is shown in Figs. 8.4.21 to 8.4.30.

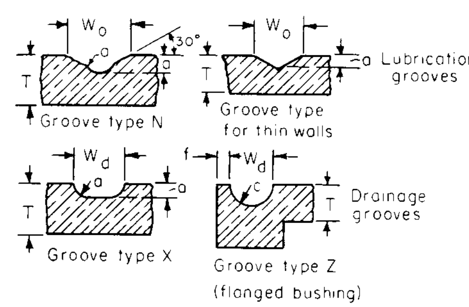


Fig. 8.4.20 Lubrication and drainage grooves.

Horizontal Bearings, Rotational Motion

DIRECTION OF LOAD KNOWN AND CONSTANT

Load downward or inside the lower 60° segment as in the case of ring-oiling bearings (Fig. 8.4.21).

Load at an angle more than 45° to the vertical centerline (Fig. 8.4.22).

In force- or drop-feed oiling, the oil inlet may be anywhere within the no-load sector (Fig. 8.4.23).

Oil can be introduced through the center of the revolving shaft (Fig. 8.4.24).

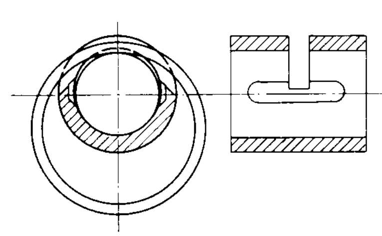


Fig. 8.4.21

Where oil-ring electric-motor bearings will be subjected by the purchaser to belt loads varying from vertical downward to horizontal, a continuous type of oil groove developed by General Electric Co. has proved very successful (Fig. 8.4.25). There are no critical spots with this groove because only a small percentage of the babbitt surface is removed along any axial line.

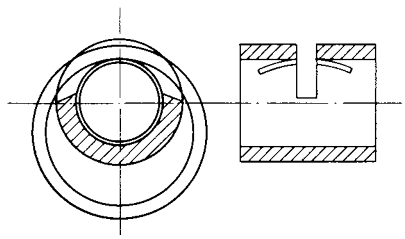


Fig. 8.4.22

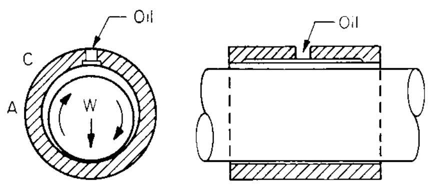


Fig. 8.4.23

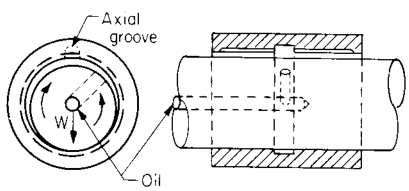
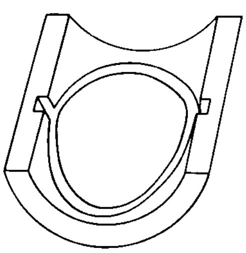


Fig. 8.4.24



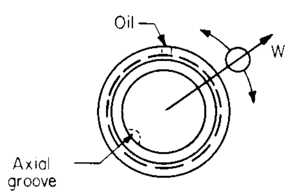
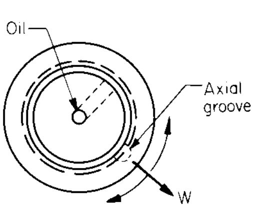


Fig. 8.4.25 **Fig. 8.4.26**

ROTATING LOAD

For rotating shafts, a circumferential groove at the middle of the bearing and an axial groove on the no-load side (Fig. 8.4.26).

For stationary shafts and rotating bearings, a circumferential groove in the bearing and an axial groove on the no-load side. The oil hole is in the shaft at the midlength of the bearing (Fig. 8.4.27).



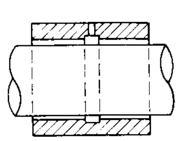


Fig. 8.4.27 **Fig. 8.4.28**

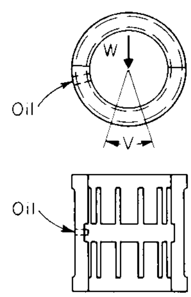
LOAD DIRECTION UNCERTAIN

Oil-ring bearings (Figs. 8.4.21 and 8.4.22) may be used, although they have defects under certain load directions. With forced or drop feed, the oil hole enters a circumferential groove at the middle of the bearing and the axial groove is omitted (Fig. 8.4.28). Arrangements for introducing oil through the rotating shaft can be made.

Bearings with Oscillatory Motion

DIRECTION OF LOAD CONSTANT

No oil film can be built up owing to the small sliding velocity, and boundary lubrication will exist. Axial grooves in the loaded sector distribute the lubricant to all parts of the bearing and avoid dry spots (Fig. 8.4.29).



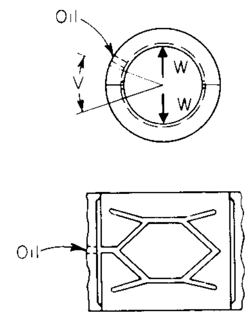


Fig. 8.4.29 **Fig. 8.4.30**

LOAD DIRECTION REVERSED DURING OSCILLATION

Fluid film lubrication is possible, at least during part of the motion, owing to the vacuum caused by shaft moving back and forth. Figure 8.4.30 shows grooving which may be modified to suit local conditions. This arrangement is also advisable for bearings under a load which reverses in direction periodically without any rotation of the bearing. The lubrication may then provide an oil cushion to soften shocks.

Bearing seals are used to prevent oil leakage from the bearing housing and to protect the bearing from outside dust, water, vapors, etc. A drainage groove at the end of the bearing is effective to divert the oil passing through the bearing back into the oil well (Fig. 8.4.31*a*). The drain holes at the bottom of the groove must be ample for passage of the oil flow.

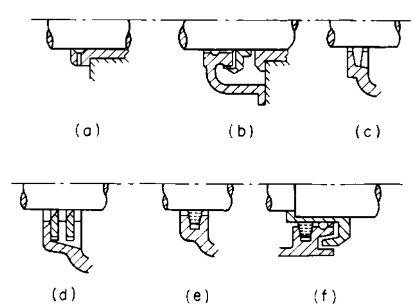


Fig. 8.4.31 Sealing end grooves.

An oil thrower mounted on the shaft is shown in Fig. 8.4.31*b*. The bearing housing may be provided with a single (Fig. 8.4.31*c*) or double collecting groove, or with brass or aluminum strip scrapers (Fig. 8.4.31*d*), to collect the oil creeping along the shaft.

For protection from dust, etc., felt packing rings are often used (Fig. 8.4.31*e*). The felt ring is soaked in oil to prevent charring by friction heat. In severe cases, additional protection by a labyrinth runner is very effective (Fig. 8.4.31*f*).

Standard seals are available for oil and grease retention as shown in

Fig. 8.4.32*a*, *b*, and *c*. The seal material that is pressed against the rotating shaft is typically made of synthetic rubber, which is satisfactory for temperatures as high as about 250°F (121°C). Figure 8.4.32*a* shows the seal material pressed against the shaft by a series of flexible fingers

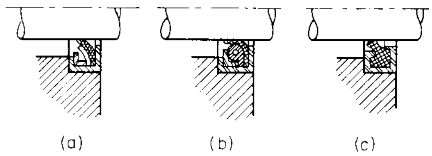


Fig. 8.4.32 Seals for oil and grease retention.

or leaf springs. In Fig. 8.4.32*b* a helical garter spring provides the gripping force. In Fig. 8.4.32*c* the rubber acts as its own spring.

Types of bearings are shown in Figs. 8.4.33 to 8.4.38. They include the principal methods of lubrication and types of construction.

Oiless bearings is the accepted term for self-lubricating bearings containing lubricants in solid or liquid form in their material. Graphite, molybdenum disulfide, and Teflon are used as solid lubricants in one group, and another group consists of porous structures (wood, metal), containing oil, grease, or wax.

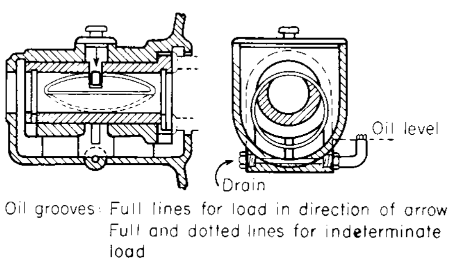


Fig. 8.4.33 Ring-oiled bearing solid bushing.

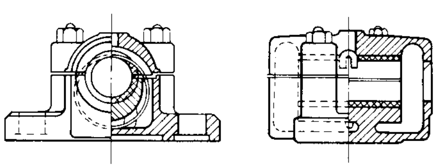


Fig. 8.4.34 Rigid ring-oiling pillow block. *(Link Belt Co.)*

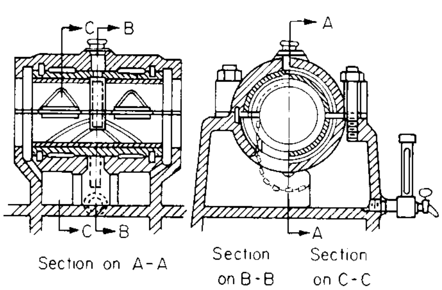


Fig. 8.4.35 Split bearing with one chain. Main crankshaft bearing; vertical oil engine.

Graphite-lubricated bearings (bridge bearings, sheaves, trolley wheels, high-temperature applications) consist generally of cast bearing bronze as a supporting structure containing various overlapping designs of grooves which are filled with graphite. The graphite is mixed with a binder, and the plastic mass is pressed into the cavities to the hardness of a lead pencil; 45 percent of the bearing area may be graphite.

Porous-metal bearings, compressed from metal powders and sintered, contain up to 35 percent of liquid lubricant. See ASTM B202-45T for sintered bronze and iron bearings, and also Army and Navy Specification AN-B-7G. The porous metal generally consists of a 90-10 copper-

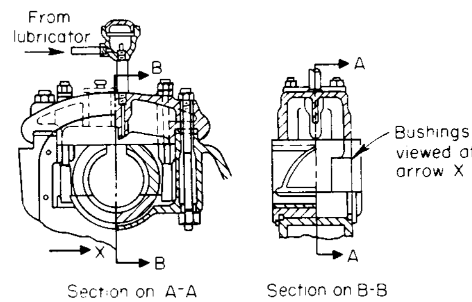


Fig. 8.4.36 Crankshaft main bearing. Horizontal engine with drop-feed lubrication.

tin bronze with 1½ percent graphite. These bearings do not require oil grooves since capillarity distributes the oil and maintains an oil film. If additional lubrication from an oil well should be provided, oil will be absorbed through the porous wall as required. For high temperatures where oil will carburize, a higher percentage of graphite (6 to 15 percent) is used.

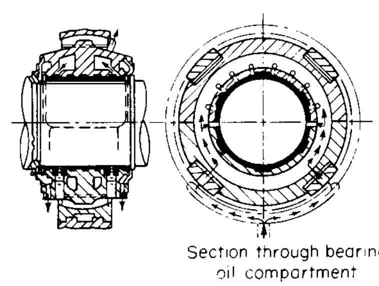


Fig. 8.4.37

Porous-metal bearings are used where plain metal bearings are impractical because of lack of space, cost, or inaccessibility for lubrication, as in automotive generators and motors, hand power tools, vacuum cleaner motors, and the like.

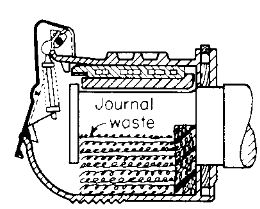


Fig. 8.4.38

THRUST BEARINGS

At low speeds, shaft shoulders or collars bear against flat bearing rings. The lubrication may be semifluid, and the friction is comparatively high.

For hardened-steel collars on bronze rings, with intermittent service, pressures up to 2,000 lb/in^2 (13,790 kN/m^2) are permissible; for continuous low-speed operation, 1,500 lb/in^2 (10,341 kN/m^2); for steel collars on babbitted rings, 200 lb/in^2 (1,378.8 kN/m^2). In multicollar thrust bearings, the values are reduced considerably because of the difficulty in distributing the load evenly between the several collars.

The performance of the bearing thrust rings is much improved by the introduction of **grooves** with tapered lands as shown in Fig. 8.4.39. The lands extend on either side of the groove. The taper angle of the lands is very slight, so that a pressure oil film is formed between the bearing ring

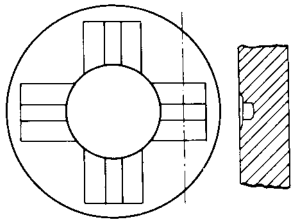


Fig. 8.4.39 Thrust collar with grooves fitted with tapered lands.

and the collar of the shaft. It is generally known that slightly tapered radial grooves will develop a hydrodynamic load-carrying film, when formed in the manner of Fig. 8.4.39. The taper angle should be on the order of 0.5°. Alternatively, a shallow recessed area that is a couple of film thicknesses deep can be used in place of the taper.

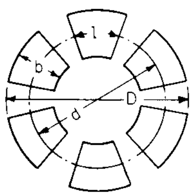


Fig. 8.4.40 Kingsbury thrust bearing with six shoes.

For high speeds or where low friction losses and a low wear rate are essential, **pivoted segmental thrust bearings** are used (Kingsbury thrust bearing, or Michell bearing in Europe). The bearing members in this type are tiltable shoes which rest on hard steel buttons mounted on the bearing housing. The shoes are free to form automatically a wedge-shaped oil film between the shoe surface and the collar of the shaft (Figs. 8.4.40 to 8.4.42).

The **minimum oil-film thickness** h_0, in, between the shoe and the collar, at the trailing edge of the shoe, is approximately

$$h_0 = 0.26\sqrt{\mu u l / P_{\text{avg}}}$$

where μ is the absolute viscosity; u is the velocity of the collar, on the mean diam; l is the length of a shoe, at the mean diam of the collar, in the direction of sliding motion; P_{avg} is the average load on the shoes. As indicated in Fig. 8.4.40, $b = l$, approximately. The standard thrust bearings have six shoes. Load-carrying capacities of Kingsbury thrust bearings are given in Table 8.4.3.

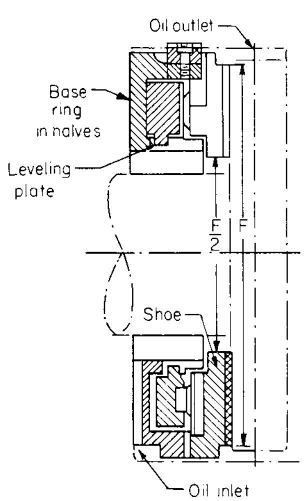


Fig. 8.4.41 Left half of six-shoe self-aligning equalizing horizontal thrust bearing for load in either axial direction.

The coefficient of friction in Kingsbury thrust bearings, referred to the mean diameter of the shoes, is approximately $f = 11.7h_0/l$, where h_0 is computed as shown above. Figures 8.4.41 and 8.4.42 show typical pivoted segmental thrust bearings. They usually embody a system of

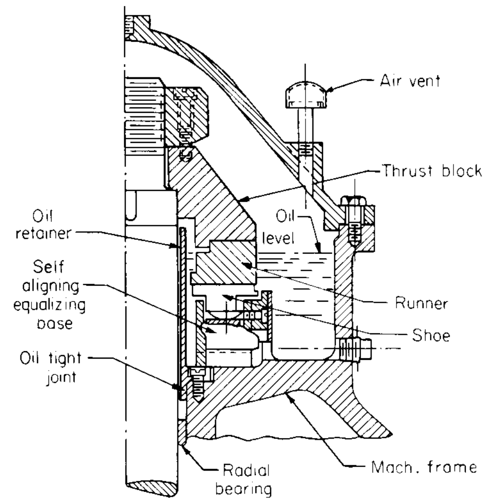


Fig. 8.4.42 Half section of mounting for vertical thrust bearing.

rocking levers which are used for alignment and equalization of load on the several shoes (Fig. 8.4.43).

Thrust may be carried on a hydrostatic step bearing as shown schematically in Fig. 8.4.44, where high-pressure oil at P_o is supplied at the

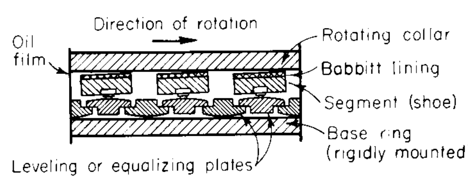


Fig. 8.4.43 Kingsbury thrust bearings. (Developed cylindrical sections.)

center of the bearing from an external pump. The lubricant flows radially outward through the annulus of depth h_0 and escapes at the periphery of the shaft at some pressure P_1 which is usually at atmospheric pressure. An oil film will be present whether the shaft rotates or not. Friction in these bearings can be made to approach zero, depending

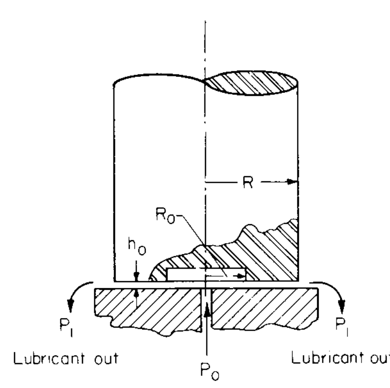


Fig. 8.4.44 Hydrostatic step bearing.

Table 8.4.3 Capacities of Six-Shoe Standard-Duty Horizontal and Vertical Thrust Bearings
(Based on viscosity of 150 s Saybolt at operating temperatures. Capacities given may be increased from 10 to 25% if viscosity is increased in same proportion)

Bearing size, in	Area, in²	Speed, r/min						Bearing size, in	Area, in²	Speed, r/min					
		100	200	400	800	1,800	3,600			100	150	200	300	500	700
		Safe load, 10^3 lb								Safe load, 10^3 lb					
5	12.5	1.44	1.7	2.0	2.4	2.9	3.5	19	180	40.00	44.0	48.0	53.0	60.0	65.0
6	18.0	2.30	2.7	3.2	3.8	4.6	5.5	21	220	51.00	57.0	61.0	68.0	77.0	84.0
7	24.5	3.30	3.9	4.7	5.6	6.8	8.0	23	264	65.00	72.0	77.0	85.0	97.0	105.0
8	32.0	4.60	5.5	6.6	7.8	9.6	11.4	25	312	80.00	88.0	95.0	105.0	119.0	123.0
9	40.5	6.20	7.4	8.8	10.4	13.0	15.0	27	364	97.00	107.0	115.0	127.0	144.0	146.0
10½	55.1	9.20	10.8	13.0	15.4	19.0	22.0	29	420	116.00	128.0	137.0	152.0	168.0	168.0
12	72.0	12.80	15.2	18.0	21.0	26.0	29.0	31	480	137.00	151.0	162.0	180.0	192.0	192.0
13½	91.1	17.20	20.0	24.0	29.0	35.0	36.0	33	544	160.00	177.0	189.0	210.0	220.0	220.0
15	112.5	22.00	26.0	32.0	37.0	45.0		37	684	215.00	235.0	250.0	275.0	275.0	
17	144.5	30.00	36.0	43.0	51.0	58.0		41	840	275.00	305.0	325.0	335.0	335.0	
								45	1012	345.00	385.0	405.0	405.0		

upon the rotational velocity and the viscosity of the lubricant film. Figure 8.4.45 shows the step bearing of a vertical turbogenerator. The load-carrying capacity is

$$W = \frac{P_o\pi}{2}\,\frac{R^2 - R_o^2}{\ln (R/R_o)}$$

This equation is valid even when the recess is eliminated, in which case R_o becomes the radius of the inlet oil supply pipe. The volume flow rate of lubricant and the clearance are related thus:

$$Q = P_o\pi h_0^3[6\mu \ln(R/R_o)]$$

The friction power loss in the bearing is

$$H_f = \frac{\pi\mu\omega^2(R^4 - R_o^4)}{2h_0}$$

The pumping power loss in forcing the lubricant through the bearing is $H_p = Q(P_o - P_1)/\eta$, where η is the efficiency of the pump.

EXAMPLE 10. A typical 5,000-kW vertical turbogenerator has a thrust load of about 101,000 lb; outside diameter of bearing, 16 in; diam of recess, 10 in; pump efficiency, 0.5; speed, 750 r/min. Substituting these values,

$$101{,}000 = \frac{P_o\pi}{2}\left[\frac{8^2 - 5^2}{\ln (8/5)}\right]$$

or

$$P_o = 774 \text{ lb/in}^2$$

In practice, about 825 lb/in² is used on this step bearing to provide some margin of safety. Film thickness in the bearing should be from 0.001 to 0.01 in to protect the surfaces from metal-to-metal contact and allow passage of harmful grit that may

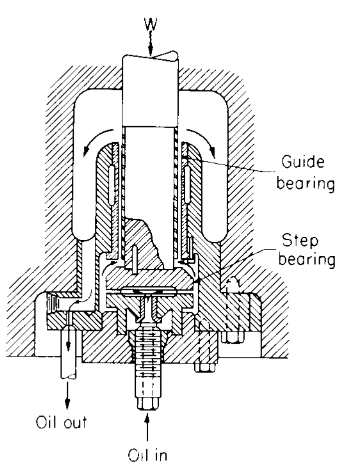


Fig. 8.4.45 Step bearing of a vertical turbogenerator.

find its way into the system. The film thickness determines the oil flow for a given viscosity and pressure. With $h_0 = 0.006$ in (0.1524 mm) and SAE 20 oil at 130°F (29 centipoises), $Q = 8.25 \times \pi \times (0.006)^3/6 \times 29 \times 1.45 \times 10^{-7} \times 0.470 = 46.8$ in³/s (766.91 cm³/s). Flow $= 46.8 \times 60/231 = 12.15$ gal/min (45.99 L/min). The horsepower lost due to friction, $H_f = 750^2 \times 29 \times 1.45(8^4 - 5^4)/383{,}000 \times 0.006 = 3.58$ hp (2.669 kW). The horsepower lost due to pumping with pump efficiency of 0.5, $H_p = 46.8 \times 825/6{,}600 \times 0.5 = 11.7$ hp (8.725 kW). The total energy lost $= 11.7 + 3.58 = 15.28$ hp (11.39 kW).

Evaluation of these equations for other film thicknesses will show that the minimum lost energy will occur between $h_0 = 0.004$ and $h_0 = 0.006$ in (0.1016 and 0.1524 mm). The coefficient of friction corresponding to an energy loss of 15.28 hp in the above example is 0.002.

Other configurations for hydrostatically lubricated thrust bearings are shown in Fig. 8.4.46 from Loeb. They may be used directly to obtain the value of load-carrying capacity W and flow rate Q.

LINEAR SLIDING BEARINGS

All sliding bearings (Fig. 8.4.47), to wear true, must have the sliding parts of nearly equal lengths. Bearings made in this way will be found not to wear out of true. Oiling is accomplished in several ways, an acceptable method being that shown in Fig. 8.4.48. Short slides in many machine tools are lubricated by oil pads or direct oil application. The weight of the table and work and thrust of the tool cause wear on the bottom and sides of the guides. To compensate for the wear in both directions, bearings are sometimes made V-shaped, as shown in Fig. 8.4.49.

Simpler sliding bearings in machine tools are made with provision for adjustment (as shown in Fig. 8.4.50) of which there are many modifications. Recent applications involving hydrostatic lubrication on machine-tool ways have been very successful.

GAS-LUBRICATED BEARINGS

The fluid-film calculations included in Examples 1 through 10 have assumed that oil (or, in one case, water) was the lubricant. Actually, almost any **process fluid** may be used if proper recognition is given to the viscosity, corrosive action, change in state (where a liquid is close to its boiling point), toxicity, and in the case of a gas, its compressibility. Fluid-film journal and thrust bearings have run successfully, for example, on water, kerosene, gasoline, acid, liquid refrigerants, mercury, molten metals, and a wide variety of gases.

The previous equations for load-carrying capacity, film thickness, friction, and flow may be used for process liquids, but for gases, proper recognition must be made of the compressibility effects.

Because of the great value of gas-lubricated bearings for special applications, and to demonstrate the methods for handling the compressibility action, an introduction to the **design** of **gas-lubricated bearings** follows.

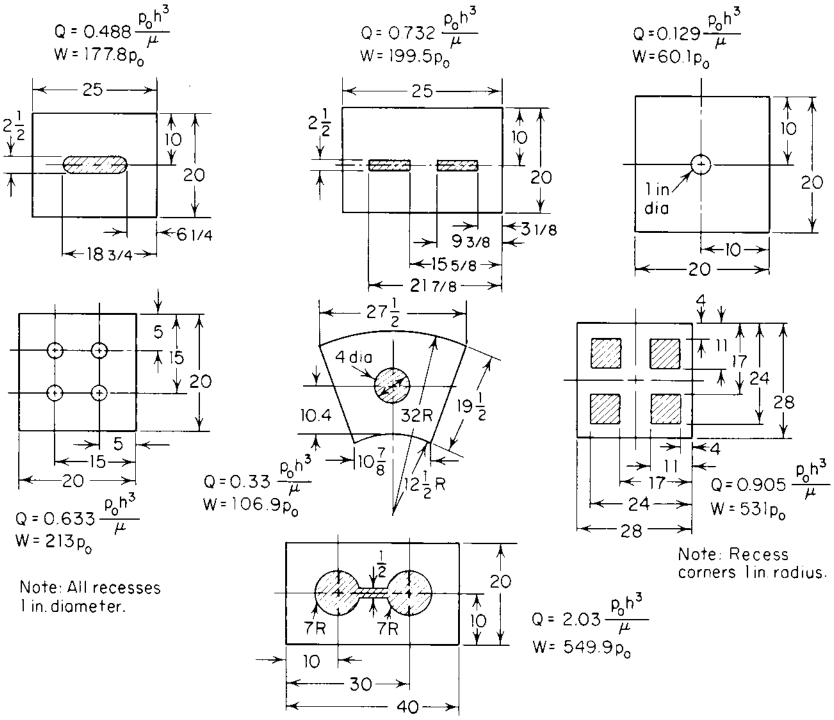


Fig. 8.4.46 Load-carrying capacity and flow for several flat thrust bearings *(Loeb).* Lengths in inches.

Naturally, if the change in pressure within the bearing clearance is small compared to ambient pressure, the compressibility effect will be likewise small, and lubrication equations based on liquids may be used. A **compressibility parameter Λ** indicates the extent of this action. For hydrodynamic journal bearings it has the form $\Lambda = 6\mu\omega/(P_a m^2)$. For values of Λ less than one, the previous equations of this section for journal bearings may be used. For values of Λ greater than one, compressibility effects are included through the use of Figs. 8.4.51 to 8.4.54. (Data from Elrod and Burgdorfer, Proceedings First International Symposium on Gas-lubricated Bearings, 1959, and Raimondi, *Trans. ASLE,* vol. IV, 1961.)

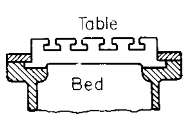


Fig. 8.4.47

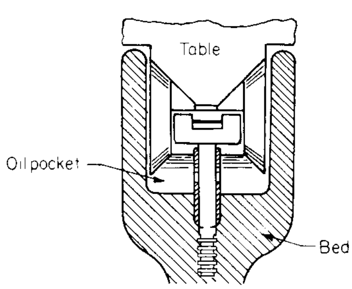


Fig. 8.4.48

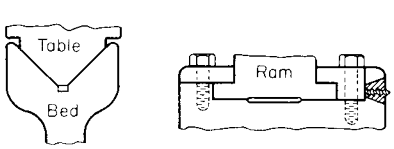


Fig. 8.4.49

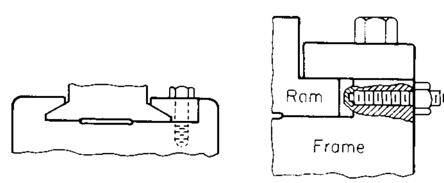


Fig. 8.4.50

EXAMPLE 11. Determine the minimum film thickness for a journal bearing 0.5 in (1.27 cm) diameter by 0.5 in long. Ambient pressure 14.7 lb/in² abs (101.34 kN/m² abs). Speed 12,000 r/min. Load 0.4 lb (0.88 kg). Diametral clearance 0.0005 in (0.0127 mm). Lubricant, air at 100°F and 14.7 lb/in² abs (2.68×10^{-9} lb·s/in² from Fig. 8.4.55). $m = 0.0005/0.5 = 0.001$ in/in. $\omega = 12{,}000 \times 2\pi/60 = 1{,}256$ rad/s, $\Lambda = (6 \times 2.68 \times 10^{-9} \times 1{,}256)/14.7 \times 0.001^2 = 1.37$, and $W/(dlP_a) = 0.4/0.5 \times 0.5 \times 14.7 = 0.109$. Then, in Fig. 8.4.53 ($l/d = 1$), we find that $\varepsilon = 0.22$, and the minimum film thickness $h_0 = 0.00025(1 - 0.22) = 0.000195$ in (0.00495 mm).

Gas-lubricated journal bearings should be checked for **whirl stability.** Figure 8.4.56 is applicable with sufficient accuracy to bearings where l/d is equal to or greater than one. It is used in conjunction with Fig. 8.4.51 for $l/d = \infty$. The stability parameter is ω_1^* which, for a bearing having only gravity loading, has the value $\omega_1^* = \omega\sqrt{mr/g}$.

EXAMPLE 12. To determine whether the bearing of Example 11 is stable at the running speed of 12,000 r/min, we compute ω_1^* as $1{,}256\sqrt{0.00025/386} = 1.015$. The value of eccentricity ratio ε_0 for $l/d = \infty$ is computed from Fig. 8.4.51

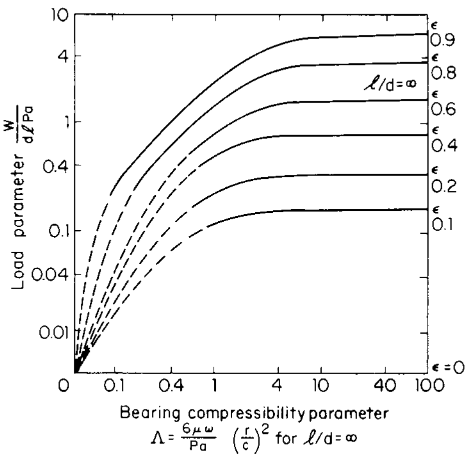


Fig. 8.4.51

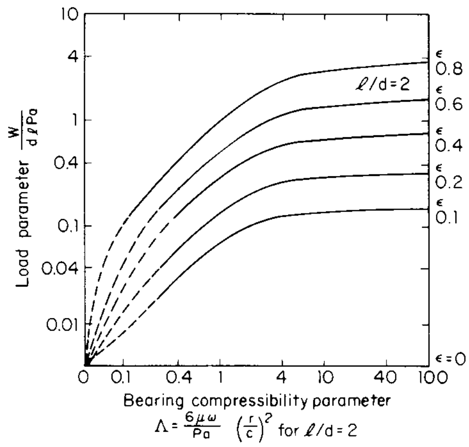


Fig. 8.4.52

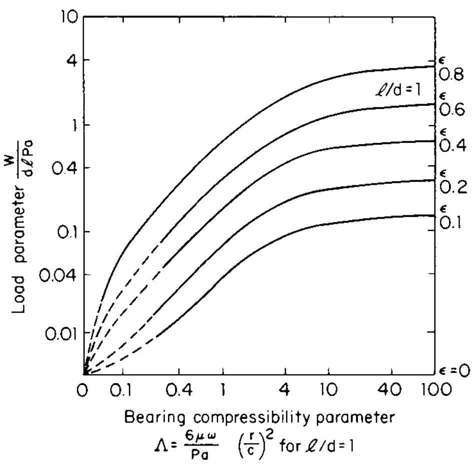


Fig. 8.4.53

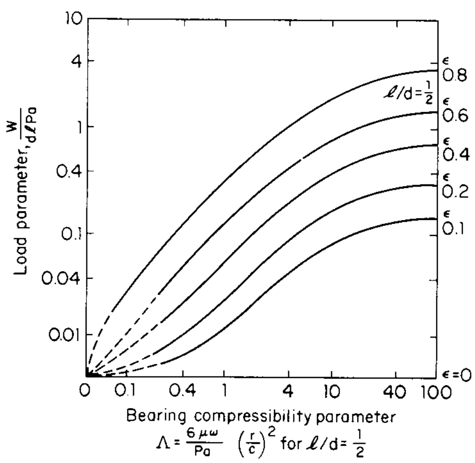


Fig. 8.4.54

Figs. 8.4.51–8.4.54 Theoretical load-carrying parameter vs. compressibility parameter for a full journal bearing: $l/d = \infty$. (Fig. 8.4.51), $l/d = 2$ (Fig. 8.4.52), $l/d = 1$ (Fig. 8.4.53), and $l/d = 0.5$ (Fig. 8.4.54). *(Elrod and Raimondi.)*

on the basis of W being the load per inch of bearing length. Thus W(lb/in) $= 0.4/0.5 = 0.8$ lb/in. For Fig. 8.4.51, $W/(dlP_a) = 0.218$ and in Fig. 8.4.51, we determine $\varepsilon_0 = 0.18$. Then (in Fig. 8.4.56), for $\omega_1^* = 1.015$ and $\Lambda = 1.37$, we find the intersection at about where a curve for $\varepsilon_0 = 0.18$ would be found. The bearing should just be stable. An intersection point on the ε_0 line or to the left should represent a stable condition. An intersection point to the right of the appropriate ε_0 line would predict an unstable condition.

The plain cylindrical journal bearing (360°) is the least stable of possible bearing designs. Control of ''half-frequency'' whirl has been achieved and the threshold of instability has been raised through modification of the geometry of the plain bearing. The simplest modification is the insertion of axial grooves. Bearings with three or four such grooves have been successful, but lose much stiffness.

A typical three-groove (three-sector) bearing is shown in Fig. 8.4.57.

Half-frequency whirl indicates a dynamic instability in which the journal orbits at approximately one-half of the shaft rotational speed, which coincides with the average speed of the lubricant in the film. If one looked at the bearing geometry from a coordinate system rotating at this whirl speed, one would see the journal attempting to pump lubricant

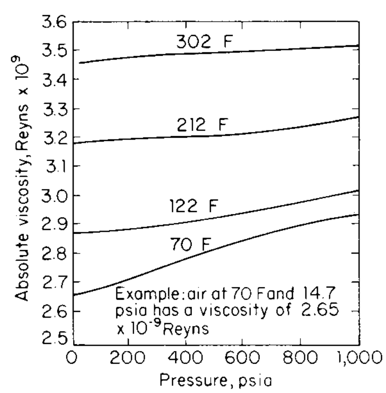


Fig. 8.4.55 Absolute viscosity of air. *(Iwaski, Sci. Rpts., Research Inst., Tohuku Univ., Ser. A; Kestin and Pliarczyk, Trans. ASME, **56,** 1954.)* Reyns $= 1.45 \times 10^{-7}$ cP.

in one direction and the bearing attempting to do the opposite. The capacity of the film to sustain any load is thus greatly diminished, and failure often occurs. Half-frequency whirl can occur when the dynamic system in which the bearing operates has a natural frequency at approximately one-half the speed of rotation. If the energy dissipation rate is

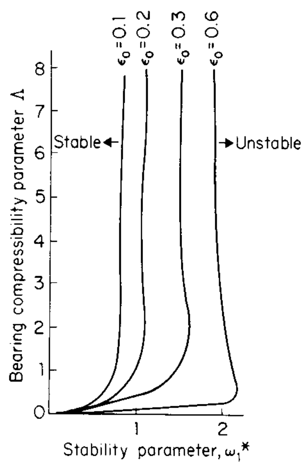


Fig. 8.4.56 Half-frequency translatory whirl threshold for infinite length, 360° journal bearing. *(Castelli and Elrod, Solution for the Stability Problem for 360°, Self Acting Gas Lubricated Bearings, Trans. ASME,* ***87,*** *Mar. 1965.)*

not sufficiently large, instability occurs. In the dynamic system mentioned above, the stiffness and damping characteristics of the bearing, or bearings, play a major role. Damping arises from squeezing the lubricant in and out of the bearing by the action of the journal vibrations against the viscous resistance. When the journal moves, gas can compress and act as a capacitance rather than flow and act as a damper. Therefore, gas bearings are much more prone to instability than are liquid-lubricated ones.

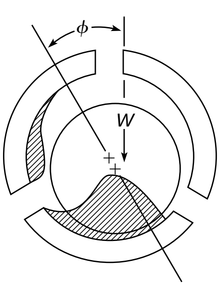


Fig. 8.4.57 Sector sleeve bearing.

Aside from avoiding resonance conditions, two often successful techniques for whirl prevention are shown in Figs. 8.4.58 and 8.4.59. The first depicts a complete journal bearing with a rotating geometric artifact causing a synchronous disturbance. This artifact can be a variation in the clearance or an asymmetric mass leading to dynamic unbalance. The second depicts a bearing geometry with shallow, inward-pumping spiral grooves. The depth of the grooves is approximately twice the radial clearance, their angle is from 25° to 30° to the axis, the land-to-groove ratio is 0.5, and the axial extent of the grooved area is one-half the length of the bearing. The grooves can be on either the stator or the rotor, depending on manufacturing convenience. Etching is a common method for production of the grooves. Design data for this excellent type of self-acting gas bearing can be found in Sec. 4 of Gross's book. (See References.)

The tilting-pad journal bearing in Fig. 8.4.60 is considered to be one of the most stable of all possible designs. The shoes, or pads, are supported on rounded pins and are free to pitch and roll through very small angles. Analysis shows that this freedom to move achieves the stability characteristics of these bearings, and also, of course, permits them to be self-aligning. Three-, four-, and five-shoe configurations are often used. Figure 8.4.60 shows an early design used for machine tool grinding spindles.

When applied to gas lubrication, pressurized gas may be supplied through a drilled pivot (hydrostatic lubrication), as an aid in starting and to provide a reserve of load-carrying capacity (Fig. 8.4.61). See Gunter, Hinkle, and Fuller, Design Guide for Gas-Lubricated, Tilting-Pad Journal and Thrust Bearings, NYO-2512-1, U.S. AEC, I-A 2392-3-1, Nov. 1964, Contract AT 30-1-2512.

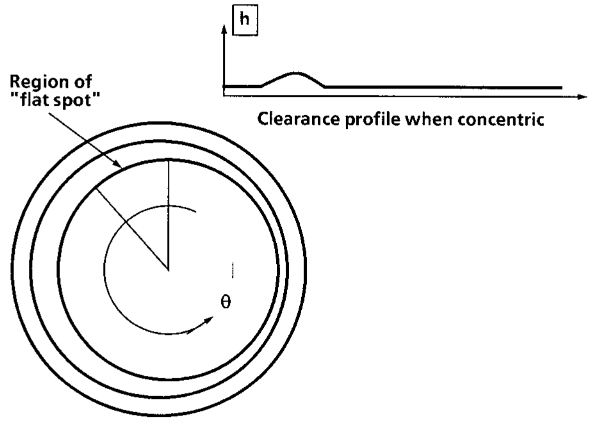


Fig. 8.4.58 Stabilizing rotating geometry. (Clearance greatly exaggerated.)

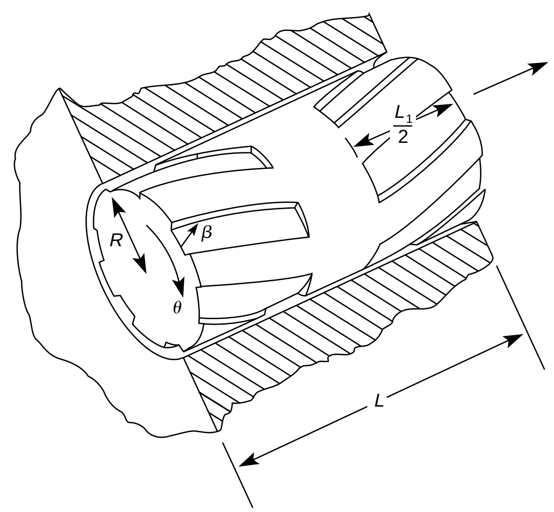


Fig. 8.4.59 Spiral groove bearing shown as outward-pumping. (Clearance is greatly exaggerated.)

A bearing design development that is simple, inexpensive, and very stable is the foil bearing. It is also very tolerant of thermal distortions and possible loss of clearance resulting from elevated temperatures. Probably the most widely used of several designs is shown in Fig. 8.4.62. It consists of overlapping metal shims, anchored at the base end like a cantilever beam. The ''free'' ends deflect and are able to automatically form their own operating clearance. These bearings are widely used in aircraft cabin cooling and for auxiliary power supply systems (Suriano, Dayton, and Woessner, Test Experience with Turbine-end Foil Bearing Equipped Gas Turbine Engines, ASME Paper 83-GT-73, 1983).

Thrust bearings of the tilting-pad variety are less susceptible to compressibility effects and may be considered as liquid-lubricated for values of Λ (suitable for thrust bearings) less than about 30. $\Lambda = 6u\mu l/(h_0^2 P_a)$ where l is the length of the shoe in the direction of sliding and U is the linear velocity at the mean radius. However, the shoes

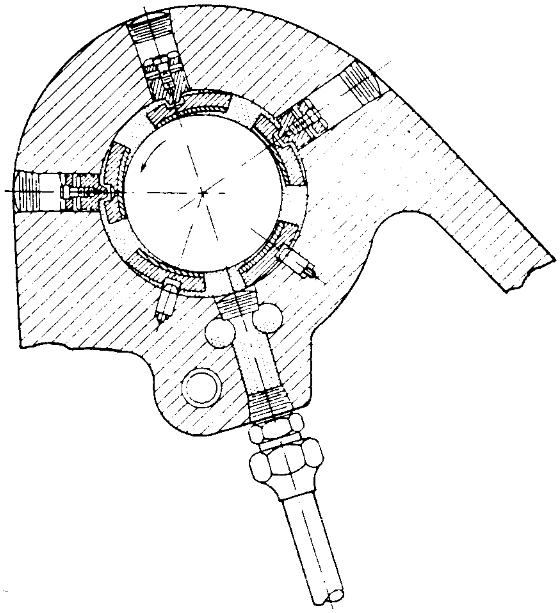


Fig. 8.4.60 Filmatic bearing. *(Courtesy Cincinnati Milacron Corp.)*

should not be made flat for gas operation but should have a crowned contour (see Fig. 8.4.63). (Gross, ''Gas Film Lubrication,'' Wiley.) An approximate value for the crown is to make $\delta = \frac{3}{4} h_0$. The tilting-pad bearing design is probably the most common gas bearing presently in existence. Every hard-disk computer memory since the early 1960s has had its read-write heads supported by self-acting tilting-pad sliders. Hundreds of millions of such units, called *flying heads,* have been manufactured to date. Some designs employ the crown geometry while,

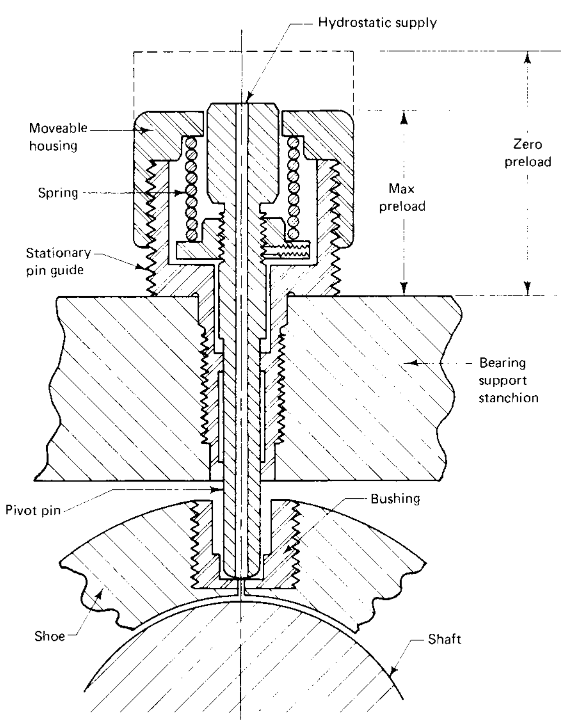


Fig. 8.4.61 Cross-sectional view, spring-mounted pivot assembly. *(Courtesy of The Franklin Institute Research Labs.)*

most commonly, heads with flat multiple sliders with straight ramps in their forward sections are used. The reason for the multiple thin sliders is the achievement of maximum damping possible. The typical minimum film heights have decreased steadily through the years from 1 μm

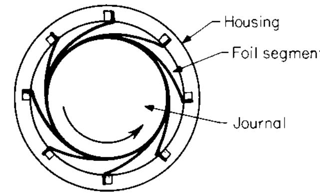


Fig. 8.4.62 Bending-dominated segments foil bearing.

(40 millionths of an inch) 25 years ago to less than 0.2 μm (8 millionths of an inch) currently (1995). This trend is driven by the achievement of the higher and higher recording densities possible at lower flying heights. Design of these devices is done rather precisely from first principles by means of special simulation programs. At these low clearances, allowance must be made for the finiteness of the **molecular mean free path,** which represents the mean distance that a gas molecule must travel between collisions. This effect manifests itself in a lowering of viscosity and wall shear resistance.

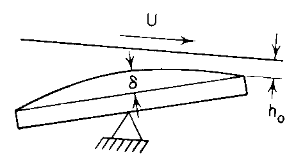


Fig. 8.4.63 Schematic of tilting-pad shoe, showing crown height δ.

Gas-lubricated hydrostatic bearings, unlike liquid-lubricated bearings, cannot be designed on the basis of fixed flow rate. They are designed instead to have a pressure loss produced by an **orifice restrictor** in the supply line. Such throttling enables the bearing to have load-carrying capacity and stiffness. For maximum stiffness the pressure drop in the orifice may be about one-half of the manifold supply pressure. For a circular thrust bearing with a single circular orifice, the load-carrying capacity is given with sufficient accuracy by the equation previously used for liquids (see Fig. 8.4.44). $W = (P_R - P_a/2)[R^2 - R_0^2/\ln (R/R_0)]$, where P_R is the recess pressure, lb/in^2 abs. The flow volume, however, is given by $Q_0 = \pi h_0^3/[6\mu \ln (R/R_0)](P_0^2 - P_1^2)/2P_0$. Q_0 and P_0 refer to recess conditions, and Q_1 and P_1 refer to ambient conditions. Pressures are absolute.

EXAMPLE 13. A circular thrust bearing 6 in (15.24 cm) diameter with a recess 2 in (5.08 cm) diameter has a film thickness of $h_0 = 0.0015$ in (0.0381 mm). $P_0 =$ 30 lb/in^2 gage or 44.7 lb/in^2 abs (308.16 kN/m^2). P_1 is room pressure, 14.7 lb/in^2 abs (101.34 kN/m^2 abs). Depth of recess is 0.02 in. Applied load is 375 lb. $Q_0 = (\pi \times 0.0015^3)/(6 \times 2.68 \times 10^{-9} \ln 3)(44.7^2 - 14.7^2)/(2 \times 44.7)$, $Q_0 = 12.3$ in^3/s (201.6 cm^3/s) at recess pressure. Converted to free air, $Q_1 = Q_0(P_0/P_1)$ with isothermal expansion, $Q_1 = 12.3(44.7/14.7) = 37.4$ in^3/s (612.87 cm^3/s), or $Q_1 = 37.4 \times 60 = 2{,}244$ in^3/min (36.77 L/min). Actual measured flow $= 2{,}440$ in^3/min (39.98 L/min).

Externally pressurized gas bearings are not as easily designed as liquid-lubricated ones. Whenever a volume larger than approximately that of the film is present between the restrictor and the film, a phenomenon known as **air hammer** or **pneumatic instability** can take place. Therefore, in practical terms, recesses cannot be used and orifice restrictors must be obtained by the smallest flow cross-section at the very entrance to the film; this area is equal to the perimeter of the inlet holes multiplied by the local height of the film. This technique is called **inherent compensation.** Unfortunately, as one can readily see, the area of the restrictors is smaller where the film is smaller; thus, the stiffness is lower than that obtainable by incompressible lubrication. Design data are available in Sec. 5 of Gross's book (see References).

8.5 BEARINGS WITH ROLLING CONTACT

by Michael W. Washo

REFERENCES: Anti-Friction Bearing Manufacturers Association, Inc. (AFBMA), Method of Evaluating Load Ratings. American National Standards Institute (ANSI), Load Ratings for Ball and Roller Bearings. AFBMA, ''Mounting Ball and Roller Bearings.'' Tedric A. Harris, ''Rolling Bearing Analysis.''

COMPONENTS AND SPECIFICATIONS

Rolling-contact bearings are designed to support and locate rotating shafts or parts in machines. They transfer loads between rotating and stationary members and permit relatively free rotation with a minimum of friction. They consist of **rolling elements (balls or rollers)** between an **outer** and **inner ring. Cages** are used to space the rolling elements from each other. Figure 8.5.1 illustrates the common terminology used in describing rolling-contact bearings.

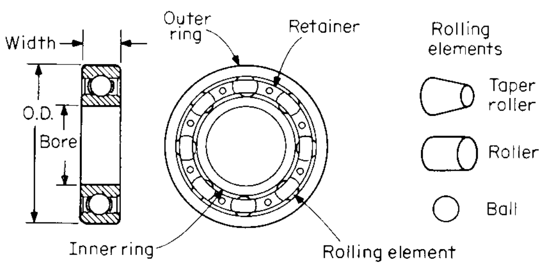


Fig. 8.5.1 Radial contact bearing terminology.

Rings The inner and outer rings of a rolling-contact bearing are normally made of SAE 52100 steel, hardened to Rockwell C 60 to 67. The rolling-element raceways are accurately ground in the rings to a very fine finish (16 μin or less).

Rings are available for special purposes in such materials as stainless steel, ceramics, and plastic. These materials are used in applications where corrosion is a problem.

Rolling Elements Normally the rolling elements, balls or rollers, are made of the same material and finished like the rings. Other rolling-element materials, such as stainless steel, ceramics, Monel, and plastics, are used in conjunction with various ring materials where corrosion is a factor.

Cages Cages, sometimes called separators or retainers, are used to space the rolling elements from each other. Cages are furnished in a wide variety of materials and construction. Pressed-steel cages, riveted or clinched and filled nylon, are most common. Solid machined cages are used where greater strength or higher speeds are required. They are fabricated from bronze or phenolic-type materials. At high speeds, the phenolic type operates more quietly with a minimum amount of friction. Bearings without cages are referred to as full-complement.

A wide variety of rolling-contact bearings are normally manufactured to standard boundary dimensions (bore, outside diameter, width) and tolerances which have been standardized by the AFBMA. All bearing manufacturers conform to these standards, thereby permitting interchangeability. ANSI has for the most part adopted these and published them jointly as AFBMA/ANSI standards as follows:

Title	*Standard*	*Title*	*Standard*
Terminology	1	Ball Standards	10
Gaging Practice	4	Roller Load Ratings	11
Mounting Dimensions	7	Instrument Bearings	12
Mounting Accessories	8.2	Vibration and Noise	13
Ball Load Ratings	9	Basic Boundary Dimensions	20

The Annular Bearing Engineers Committee (ABEC) of the AFBMA has established progressive levels of precision for ball bearings. Designated as ABEC-1, ABEC-5, ABEC-7, and ABEC-9, these standards specify tolerances for bore, outside diameter, width, and radial runout. Similarly, roller bearings have established precision levels as RBEC-1 and RBEC-5.

PRINCIPAL STANDARD BEARING TYPES

The selection of the type of rolling-contact bearing depends upon many considerations, as evidenced by the numerous types available. Furthermore, each basic type of bearing is furnished in several **standard ''series''** as illustrated in Fig. 8.5.2. Although the bore is the same, the outside diameter, width, and ball size are progressively larger. The result is that a wide range of load-carrying capacity is available for a given size shaft, thus giving designers considerable flexibility in selecting standard-size interchangeable bearings. Some of the more common bearings are illustrated below and their characteristics described briefly.

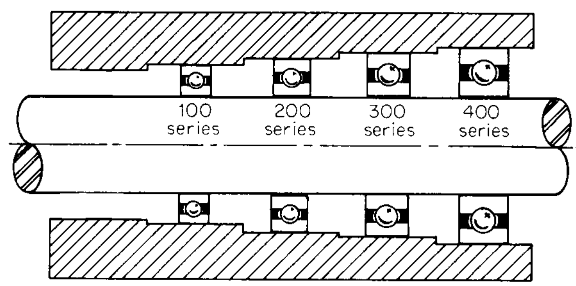


Fig. 8.5.2 Bearing standard series.

Ball Bearings

Single-Row Radial (Fig. 8.5.3) This bearing is often referred to as the **deep groove** or conrad bearing. Available in many variations—single or double shields or seals. Normally used for radial and thrust loads (maximum two-thirds of radial).

Maximum Capacity (Fig. 8.5.4) The geometry is similar to that of a deep-groove bearing except for a **filling slot.** This slot allows more balls in the complement and thus will carry heavier radial loads. However, because of the filling slot, the thrust capacity in both directions is reduced drastically.

Double-Row (Fig. 8.5.5) This bearing provides for heavy radial and light thrust loads without increasing the OD of the bearing. It is approximately 60 to 80 percent wider than a comparable single-row bearing. Because of the filling slot, thrust loads must be light.

Internal Self-Aligning Double-Row (Fig. 8.5.6) This bearing may be used for primarily radial loads where **self-alignment** ($\pm$ 4°) is required. The self-aligning feature should not be abused, as excessive misalignment or thrust load (10 percent of radial) causes early failure.

Angular-Contact Bearings (Fig. 8.5.7) These bearings are designed to support **combined radial and thrust** loads or heavy thrust loads depending on the contact-angle magnitude. Bearings having large contact angles can support heavier thrust loads. They may be mounted in pairs (Fig. 8.5.8) which are referred to as **duplex bearings:** back-to-back, tandem, or face-to-face. These bearings (ABEC-7 or ABEC-9) may be preloaded to minimize axial movement and deflection of the shaft.

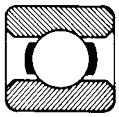


Fig. 8.5.3

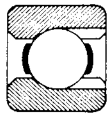


Fig. 8.5.4

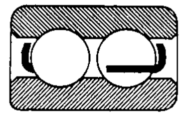


Fig. 8.5.5

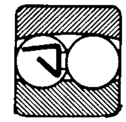


Fig. 8.5.6

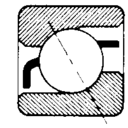


Fig. 8.5.7

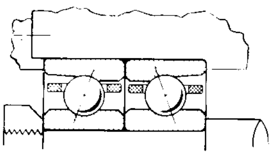


Fig. 8.5.8

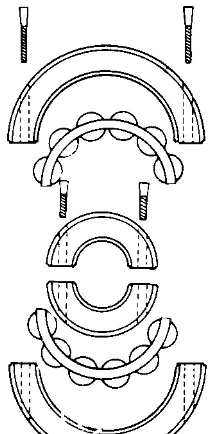


Fig. 8.5.10

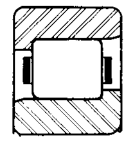


Fig. 8.5.11

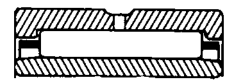


Fig. 8.5.12

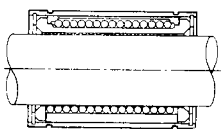


Fig. 8.5.9

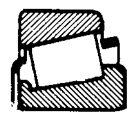


Fig. 8.5.13

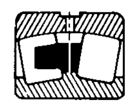


Fig. 8.5.14

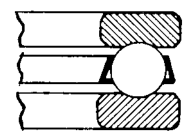


Fig. 8.5.15

Ball Bushings (Fig. 8.5.9) This type of bearing is used for linear motions on hardened shafts (Rockwell C 58 to 64). Some types can be used for linear and rotary motion.

Split-Type Ball Bearing (Fig. 8.5.10) This type of ball or roller bearing has split inner ring, outer ring, and cage. They are assembled by screws. This feature is expensive but useful where it is difficult to install or remove a solid bearing.

Roller Bearings

Cylindrical Roller (Fig. 8.5.11) These bearings utilize cylinders with approximate length/diameter ratio ranging from 1 : 1 to 1 : 3 as rolling elements. Normally used for heavy radial loads. Especially useful for free axial movement of the shaft. Highest speed limits for roller bearings.

Needle Bearings (Fig. 8.5.12) These bearings have rollers whose length is at least 4 times their diameter. They are most useful where space is a factor and are available with or without inner race. If shaft is used as inner race, it must be hardened and ground. Full-complement type is used for high loads, oscillating, or slow speeds. Cage type should be used for rotational motion. They cannot support thrust loads.

Tapered-Roller (Fig. 8.5.13) These bearings are used for heavy radial and thrust loads. The bearing is designed so that all elements in the rolling surface and the raceways intersect at a common point on the axis: thus **true rolling** is obtained. Where maximum system rigidity is required, the bearings can be adjusted for a preload. They are available in double row.

Spherical-Roller (Fig. 8.5.14) These bearings are excellent for heavy radial loads and moderate thrust. Their internal self-aligning feature is useful in many applications such as HVAC fans.

Thrust Bearings

Ball Thrust Bearing (Fig. 8.5.15) It may be used for low-speed applications where other bearings carry the radial load. These bearings are made with shields, as well as the open type.

Straight-Roller Thrust Bearing (Fig. 8.5.16) These bearings are made of a series of short rollers to minimize the skidding, which causes twisting, of the rollers. They may be used for moderate speeds and loads.

Tapered-Roller Thrust (Fig. 8.5.17) It eliminates the skidding that takes place with straight rollers but causes a thrust load between the ends of the rollers and the shoulder on the race. Thus speeds are limited because the roller end and race flange are in sliding contact.

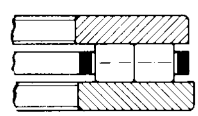


Fig. 8.5.16

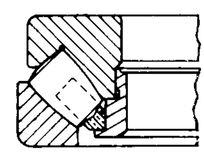


Fig. 8.5.17

Selection of Ball or Roller Bearing

Selection of the type of rolling-element bearing is a function of many factors, such as load, speed, misalignment sensitivity, space limitations, and desire for precise shaft positioning. However, to determine if a ball or roller bearing should be selected, the following **general rules** apply:

1. Ball bearings function on theoretical point contact. Thus they are suited for higher speeds and lighter loads than roller bearings.

2. Roller bearings are generally more expensive except in larger sizes. Since they function theoretically on line contact, they will carry heavy loads, including shock, more satisfactorily, but are limited in speed.

Use Fig. 8.5.18 as a general guide to determine if a ball or roller bearing should be selected. This figure is based on a rated life of 30,000 h.

ROLLING-CONTACT BEARINGS' LIFE, LOAD, AND SPEED RELATIONSHIPS

An accurate knowledge of the load-carrying capacity and expected life is essential in the proper selection of ball and roller bearings. Bearings that are subject to millions of different stress applications fail owing to

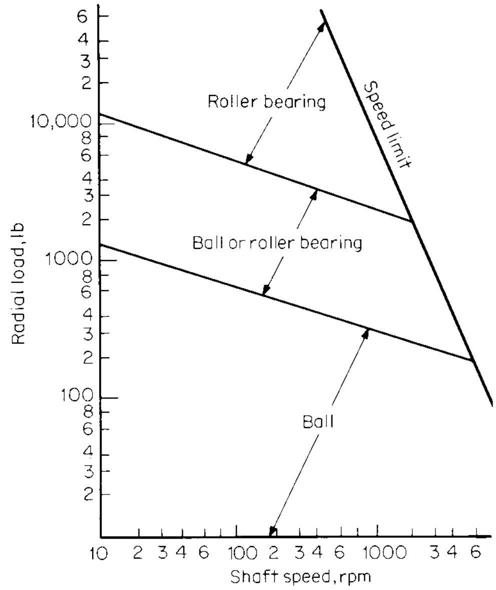


Fig. 8.5.18 Guide to selection of ball or roller bearings.

fatigue. In fact, **fatigue** is the only cause of **failure** if the bearing is properly lubricated, mounted, and sealed against the entrance of dust or dirt and is maintained in this condition. For this reason, the **life** of an individual bearing is defined as the total number of revolutions or hours at a given constant speed at which a bearing runs before the first evidence of fatigue develops.

Definitions

Rated Life L_{10} The number of revolutions or hours at a given constant speed that 90 percent of an apparently identical group of bearings will complete or exceed before the first evidence of fatigue develops; i.e., 10 out of 100 bearings will fail before rated life. The names **Minimum life** and **L_{10} life** are also used to mean rated life.

Basic Load Rating C The radial load that a ball bearing can withstand for one million revolutions of the inner ring. Its value depends on bearing type, bearing geometry, accuracy of fabrication, and bearing material. The basic load rating is also called the **specific dynamic capacity,** the **basic dynamic capacity,** or the **dynamic load rating**.

Equivalent Radial Load P Constant stationary radial load which, if applied to a bearing with rotating inner ring and stationary outer ring, would give the same life as that which the bearing will attain under the actual conditions of load and rotation.

Static Load Rating C_0 Static radial load which produces a maximum contact stress of 580,000 lb/in² (4,000 MPa).

Static Equivalent Load P_0 Static radial load, if applied, which produces a maximum contact stress equal in magnitude to the maximum contact stress in the actual condition of loading.

Bearing Rated Life

Standard formulas have been developed to predict the statistical rated life of a bearing under any given set of conditions. These formulas are based on an exponential relationship of load to life which has been established from extensive research and testing.

$$L_{10} = \left(\frac{C}{P}\right)^K \times 10^6 \qquad (8.5.1)$$

where L_{10} = rated life, r; C = basic load rating, lb; P = equivalent radial load, lb; K = constant, 3 for ball bearings, 10/3 for roller bearings.

To convert to hours of life L_{10}, this formula becomes

$$L_{10} = \frac{16{,}700}{N}\left(\frac{C}{P}\right)^K \qquad (8.5.2)$$

where N = rotational speed, r/min. Table 8.5.1 lists some common design lives vs. the type of application. These may be altered to suit unusual circumstances.

Load Rating

The **load rating** is a function of many parameters, such as number of balls, ball diameter, and contact angle. Two load ratings are associated with a rolling-contact bearing: **basic** and **static** load rating.

Basic Load Rating C This rating is *always* used in determining bearing life for all speeds and load conditions [see Eqs. (8.5.1) and (8.5.2)].

Static Load Rating C_0 This rating is used only as a check to determine if the maximum allowable stress of the rolling elements will be exceeded. It is *never* used to calculate bearing life.

Values for C and C_0 are readily attainable in any bearing manufacturer's catalog as a function of size and bearing type. Table 8.5.2 lists the basic and static load ratings for some common sizes and types of bearings.

Equivalent Load

There are two **equivalent-load** formulas. Bearings operating with some finite speed use the equivalent radial load P in conjunction with C [Eq. (8.5.1)] to calculate bearing life. The static equivalent load is used in comparison with C_0 in applications when a bearing is highly loaded in a static mode.

Equivalent Radial Load P All bearing loads are converted to an equivalent radial load. Equation (8.5.3) is the general formula used for both ball and roller bearings.

$$P = XR + YT \qquad (8.5.3)$$

where P = equivalent radial loads, lb; R = radial load, lb; T = thrust (axial) load, lb; X and Y = radial and thrust factors (Table 8.5.3). The empirical X and Y factors in Eq. (8.5.3) depend upon the geometry, loads, and bearing type. Average X and Y factors can be obtained from Table 8.5.3. Two values of X and Y are listed. The set X_1 Y_1 or X_2 Y_2 giving the largest equivalent load should always be used.

Static Equivalent Load P_0 The static equivalent load may be compared directly to the static load rating C_0. If P_0 is greater than the C_0

Table 8.5.1 Design-Life Guide

Application	Design life, h, L_{10}	Application	Design life, h, L_{10}
Agricultural equipment	3,000–6,000	Domestic appliances	1,000–2,000
Aircraft engines	1,000–3,000	Electric motors:	
Aircraft jet engines	1,500–4,000	Domestic	1,000–2,000
Automotive:		Industrial	20,000–30,000
Bus, car	2,000–5,000	Elevator	8,000–15,000
Trucks	1,500–2,500	Fans:	
Blowers:	20,000–30,000	Industrial	8,000–15,000
Continuous 8-h service	20,000–40,000	Mine ventilation	40,000–50,000
Continuous 24-h service	40,000–60,000	Gearing units (multipurpose)	8,000–15,000
Continuous 24-h service (extreme reliability)	100,000–200,000	Intermittent service	8,000–15,000
Compressors	40,000–60,000	Paper machines	50,000–60,000
Conveyors	20,000–40,000	Pumps	40,000–60,000

Table 8.5.2 Approximate Basic and Static Load Ratings vs. Types and Sizes
(Ratings are in pounds) 1 lb = 4,448 N

Bearing bore, mm	Ball single-row 200 series		Ball single-row 300 series		Ball double-row 200 series		Roller cylindrical 300 series		Roller spherical 22200 series	
	C_0	C	C_0	C	C_0	C	C_0	C	C_0	C
10	600	1,040	850	1,430	800	1,210	1,020	1,960		
12	680	1,180	1,040	1,650	1,250	1,820	1,350	2,540		
15	780	1,330	1,220	1,970	1,430	2,030	1,520	2,820		
17	1,000	1,660	1,470	2,340	1,840	2,510	2,070	3,700		
20	1,390	2,220	1,760	2,730	2,540	3,480	2,560	4,490		
25	1,560	2,420	2,350	3,550	2,858	3,780	3,720	6,360		
30	2,250	3,360	3,120	4,600	4,110	5,140	5,070	8,460		
35	3,070	4,430	4,020	5,770	5,600	6,700	6,400	10,400		
40	3,520	5,040	5,020	7,060	6,430	7,680	7,930	12,500	11,800	15,200
45	4,000	5,660	6,130	8,430	7,320	8,620	9,310	14,700	12,600	15,900
50	4,450	6,070	8,010	10,750	8,130	9,220	11,600	17,900	13,600	16,800
55	5,630	7,500	9,400	12,410	10,300	11,400	12,600	19,100	16,500	20,300
60	6,950	9,070	10,902	14,179	12,700	13,800	15,200	22,800	20,800	25,200
65	7,660	9,900	12,516	16,051	14,000	15,000	19,900	29,000	25,500	30,200
70	8,410	10,714	14,240	18,030	15,400	16,300	21,400	30,800	27,500	31,900
75	9,190	11,610	16,080	19,600	16,900	17,300	23,200	32,900	29,100	33,100
80	10,010	12,550	18,020	21,230	18,300	19,100	27,000	38,100	32,100	36,800
85	11,750	14,490	20,080	22,880	19,500	19,700	30,900	43,300	38,200	43,200
90	13,630	16,540	22,250	24,580	22,100	22,600	35,200	48,800	44,500	49,800
95	15,650	18,740	24,530	26,300	28,600	28,600	39,500	54,200	48,800	54,700
100	17,800	21,130	29,430	29,940	32,500	32,100	44,700	60,800	55,700	61,900
110	20,100	23,000	32,040	31,800	30,500	30,700	53,200	70,500	72,000	78,400

rating, permanent deformation of the rolling element will occur. Calculate P_0 as follows:

$$P_0 = X_0 R = Y_0 T \tag{8.5.4}$$

where P_0 = static equivalent load, lb; X_0 = radial factor (see Table 8.5.4); Y_0 = thrust factor (see Table 8.5.4); R = radial load, lb; T = thrust (axial) load, lb. If a load higher than the basic static load rating is imposed while rotating, the deformation is distributed evenly and no practical impairment occurs until the deformation becomes quite large. Some equipment operates with loads greatly exceeding the static capacity, such as bearings supporting artillery (twice static capacity), or aircraft control pulleys (four times static capacity). The load which will fracture a bearing is approximately eight times the static load rating.

Table 8.5.3 Radial and Thrust Factors

Bearing type	X_1	Y_1	X_2	Y_2
Single-row ball	1.0	0.0	0.56	1.40
Double-row ball	1.0	0.75	0.63	1.25
Cylindrical roller	1.0	0.0	1.0	0.0
Spherical roller	1.0	2.5	0.67	3.7

Table 8.5.4 Radial and Thrust Factors

Type of bearing	X_0	Y_0
Single-row ball	0.6	0.5
Double-row ball	0.6	0.5
Spherical roller, 22200 series	1.0	2.9
Cylindrical roller	1.0	0.0

Oscillating loads, where the motion is such that the rolling element rotates less than half a revolution, approach static load conditions. This type of load is conducive to rapid **false brinelling** and requires special lubrication techniques.

Required Capacity

The basic load rating C is very useful in the selection of the type and size of bearing. By calculating the required capacity needed for a bearing in a certain application and comparing this with known capacities, a bearing can be selected. To calculate the required capacity, the following formula can be used:

$$C_r = \frac{P(L_{10}N)^{1/K}}{Z} \tag{8.5.5}$$

where C = required capacity, lb; L_{10} = rated life, h; P = equivalent radial load, lb; K = constant, 3 for ball bearings, 10/3 for roller bearings; Z = constant, 25.6 for ball bearings, 18.5 for roller bearings; N = rotation speed, r/min.

LIFE ADJUSTMENT FACTORS

Modifications to Eq. (8.5.2) can be made, based on a better understanding of causes of fatigue. Influencing factors include

1. Reliability factors for survival rates greater than 90 percent
2. Improved raw materials and manufacturing processes for ball bearing rings and balls.
3. The beneficial effects of elastrohydrodynamic lubricant films

Equation (8.5.2) can be rewritten to reflect these influencing factors:

$$L_{10} \text{ modified} = A_1 A_2 A_3 \frac{16{,}700}{N}\left(\frac{C}{P}\right)^K \tag{8.5.6}$$

where A_1 = statistical life reliability factor for a chosen survival rate, A_2 = life-modifying factor reflecting bearing material type and condition, and A_3 = elastohydrodynamic lubricant film factor.

Factor A_1

Reliability factors listed in Table 8.5.5 represent rates from 90 to 99 percent. Using rates other than 90 percent will yield larger bearings for equivalent loads and, therefore, will increase costs. Rates higher than 90 percent should be used only when absolutely necessary.

Table 8.5.5 Reliability Factor A_1 for Various Survival Rates

Survival rate, %	Bearing life notation	Reliability factor A_1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

Factor A_2

While not formally recognized by AFBMA, estimated A_2 factors are commonly used as represented by the values in Table 8.5.6. The main considerations in establishing A_2 values are the material type, melting procedure, mechanical working and grain orientation, and hardness.

Table 8.5.6 Life-Modifying Factor A_2

Material	Factor A_2
A1S1 440C, Air Melted	0.025
SAE 52100, Vacuum Processed	1.0
M50 VIM-VAR Melted	2.0

Factor A_3

This factor is based on elastohydrodynamic lubricant film calculations which relate film thickness and surface finish to fatigue life. A factor of 1 to 3 indicates adequate lubrication, with 1 being the minimum value for which the fatigue formula can still be applied. As A_3 goes from 1 to 3, the life expectancy will increase proportionately, with 3 being the largest value for A_3 that is meaningful. If A_3 is less than 1, poor lubrication conditions are presumed. Calculations for A_3 are beyond the scope of this section.

Speed Limits

Many factors combine to determine the limiting speeds of ball and roller bearings. It depends on several factors, like bearing size, inner- or outer-ring rotation, contacting seals, radial clearance and tolerances, operating loads, type of cage and cage material, temperature, and type of lubrication. A convenient check on speed limits can be made from a *dn* value. The *dn* value is a direct function of size and speed and is dependent on type of lubrication. It is calculated by multiplying the bore in millimeters (mm) by the speed in r/min.

$$dn = \text{bore (mm)} \times \text{speed (r/min)} \tag{8.5.7}$$

A guide for *dn* values is listed in Table 8.5.7. When these values are exceeded, bearing life is shortened. The values are only a guide for approaching difficulties and can be exceeded by special bearings, lubrication, and application.

Table 8.5.7 *dn* Values vs. Bearing Types

Bearing type	Series	Max *dn* value: Grease	Max *dn* value: Oil
Single-row ball	100, 200, 300, 400, 30, in	200,000	300,000
Double-row ball	200, 300	160,000	220,000
Cylindrical roller	200, 300	150,000	200,000
Spherical roller	22200	120,000	170,000

Friction

One of the assets of rolling-contact bearings is their low friction. The **coefficient of friction** varies appreciably with the type of bearing, load, speed, lubrication, and sealing element. For rough calculations the following coefficients can be used for normal operating conditions and favorable lubrication:

Single-row ball bearings	0.0015
Roller bearings	0.0018

Excess grease, contact seals, etc., will increase these values, and allowances should be made.

PROCEDURE FOR DETERMINING SIZE, LIFE, AND BEARING TYPE

Basically, three common situations may be encountered in the analysis of a bearing system; bearing-size selection, bearing-type selection, and bearing-life determination. Each of these problems requires the following conditions to be known; radial load, thrust load, and speed. The static load capacity is not considered in the following procedures but should be analyzed if the bearing rotational speed is slow or if the bearing is idle for a period of time.

Bearing Size Selection

Known type and series:

1. Select desired design life (Table 8.5.1).
2. Calculate equivalent radial load P [Eq. (8.5.3)].
3. Calculate required capacity C_r [Eq. (8.5.5)].
4. Compare C_r with capacities C in Table 8.5.2. Select first bore size having a capacity C greater than C_r.
5. Check bearing speed limit [Eq. (8.5.7)].

Bearing-Type Selection

Known bore size and life:

1. Select ball or roller bearing (Fig. 8.5.18).
2. Calculate equivalent load P [Eq. (8.5.3)] for various bearing types (conrad, spherical, etc.).
3. Calculate C_r [(Eq. (8.5.5)].
4. Compare C_r with capacities C in Table 8.5.3, and select the type that has a capacity equal to or greater than C_r.
5. Check bearing speed limit [Eq. (8.5.7)].

Bearing-Life Determination

Known bearing size:

1. Select ball or roller bearing (Fig. 8.5.18).
2. Calculate equivalent radial load P [Eq. (8.5.3)].
3. Select basic load rating C from Table 8.5.3.
4. Calculate rated life L_{10} [Eq. (8.5.1) or (8.5.2)].
5. Check calculated life with design life.

BEARING CLOSURES

Rolling-element bearings are made with a wide variety of **closures**. Basically, they are open, shielded, or sealed (Figs. 8.5.19 and 8.5.20). **Shielded bearings** have a small clearance between the stationary shield and rotating ring. This provides reasonable exclusion of dirt without an increase in friction. **Sealed bearings** have a flexible lip (usually synthetic rubber) in contact with the inner ring. Friction is increased, but more effective retention of lubricant and exclusion of dirt is obtained. Seals should not be used to seal a fluid head or at high speeds.

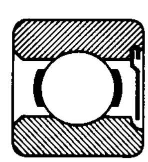


Fig. 8.5.19

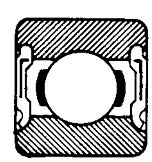


Fig. 8.5.20

BEARING MOUNTING

Correct **mounting** of a rolling-contact bearing is essential to obtain its rated life. Many types of mounting methods are available. The selection of the proper method is a function of the accuracy, speed, load, and cost of the application. The most common and best method of bearing retention is a press fit against a shaft shoulder secured with a locknut. End caps are used to secure the bearing against the housing shoulder (Fig. 8.5.21). Retaining rings are also used to fix a bearing on a shaft or in a housing (Fig. 8.5.22). Each shaft assembly normally must provide for expansion by allowing one end to float. This can be accomplished by

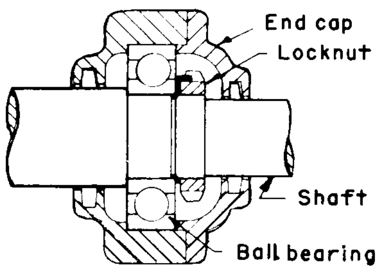


Fig. 8.5.21

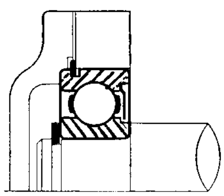


Fig. 8.5.22

allowing the bearing to expand linearly in the housing or by using a straight roller bearing on one end. Care must be exercised when designing a **floating installation** because it requires a slip fit. An excessively loose fit will cause the bearing to spin on the shaft or in the housing.

Table 8.5.8 lists shaft and housing tolerances for press fits with ABEC 1 precision applications (pumps, gear reducers, electric motors, etc.) and ABEC 7 precision applications (grinding spindles, etc.).

Table 8.5.8 Shaft and Housing Tolerances for Press Fit

Bearing bore, mm	Shaft tolerances, in, ABEC 1 precision	Bearing bore, mm	Shaft tolerances, in, ABEC 7 precision
4–6	+ 0.0000 − 0.0002	4–30	+ 0.00000 − 0.00015
7–17	+ 0.0000 − 0.0003	35–50	+ 0.0000 − 0.0002
20–50	+ 0.0000 − 0.0004	55–80	+ 0.0000 − 0.0003
55–80	+ 0.0000 − 0.0005	85–120	+ 0.00000 − 0.00035
85–120	+ 0.0000 − 0.0006		
Bearing OD, mm	**Housing tolerances, in, ABEC 1 precision**	**Bearing OD, mm**	**Housing tolerances, in, ABEC 7 precision**
16–30	+ 0.0008 − 0.0000	16–80	+ 0.0002 − 0.0000
32–47	+ 0.0010 − 0.0000	85–120	+ 0.0003 − 0.0000
52–80	+ 0.0012 − 0.0000	125–225	+ 0.0004 − 0.0000
85–120	+ 0.0014 − 0.0000		
125–180	+ 0.0016 − 0.0000		

Mounted units such as **ball-bearing pillow blocks** (Fig. 8.5.23) are frequently used for fans and conveyors. Three common methods are used to attach the bearing to the shaft; setscrew, eccentric locking collar, and taper-sleeve adapter.

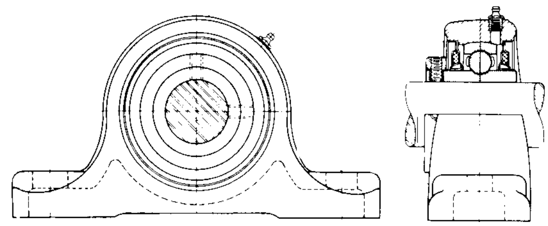


Fig. 8.5.23

Setscrew Figure 8.5.24 illustrates the use of an extended inner-ring bearing held to the shaft with a setscrew. This is a simple method and is suitable only for lightly loaded bearings.

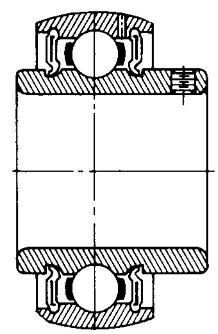


Fig. 8.5.24

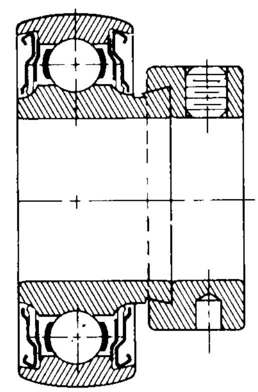


Fig. 8.5.25

Eccentric Locking Collar Figure 8.5.25 illustrates the use of an extended inner-ring bearing held to the shaft with an eccentric collar. This method tends to keep the shaft centered in the bearing more concentrically than the setscrew method. It is suitable for light to moderate loads.

Taper-Sleeve Adapter Figure 8.5.26 illustrates the use of a taper-sleeve adapter to mount the bearing on the shaft. It provides uniform concentric contact between the shaft and bearing bore. However, skill is required to tighten the locking nut enough to keep the sleeve from spinning on the shaft and yet not so tight that the inner race of the bearing is expanded to the point where the clearance is removed from the bearing. It is very difficult to obtain the correct setting with light-series bearings. They are excellent for heavy-duty spherical roller bearings.

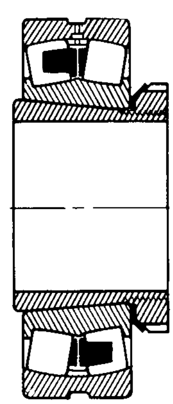


Fig. 8.5.26

LUBRICATION

Rolling-contact bearings need a **fluid lubricant** to obtain or exceed their rated life. In the absence of high-temperature environment, only a small amount of lubricant is required for excellent performance. Excess lubricant will cause heating of the bearing and accelerate the deterioration of the lubricant. Optimum lubrication of rolling-contact bearings can be predicted by elastohydrodynamic theory (EHD). It has been shown that film thickness is sensitive to bearing speed of operation and lubricant viscosity properties and, moreover, that the film thickness is virtually insensitive to load.

Grease is commonly used for lubrication of rolling-contact bearings because of its convenience and minimum maintenance. A high-quality lithium-based NLGI 2 grease should be used for temperatures up to 180°F (82°C), or polyurea-based grease for temperatures up to 300°F (150°C). In applications involving high speed, oil lubrication is often necessary. Table 8.5.9 can be used as a general guide in selecting oil of the proper viscosity for rolling-contact bearings.

Table 8.5.9 Oil-Lubrication Viscosity
(Viscosity in ISO identification numbers*)

Bearing bore, mm	Bearing speed, r/min				
	10,000	3,600	1,800	600	50
4–7	68	150	220		
10–20	32	68	150	220	460
25–45	10	32	68	150	320
50–70	7	22	68	150	320
75–90	3	10	22	68	220
100	3	7	22	68	220

* ISO identification number = midpoint viscosity in centistokes at 40°C.

Table 8.5.10 Ball-Bearing Grease Relubrication Intervals
(Hours of operation)

Bearing bore, mm	Bearing speed, r/min				
	5,000	3,600	1,750	1,000	200
10	8,700	12,000	25,000	44,000	220,000
20	5,500	8,000	17,000	30,000	150,000
30	4,000	6,000	13,000	24,000	127,000
40	2,800	4,500	11,000	20,000	111,000
50		3,500	9,300	18,000	97,000
60		2,600	8,000	16,000	88,000
70			6,700	14,000	81,000
80			5,700	12,000	75,000
90			4,800	11,000	70,000
100			4,000	10,000	66,000

In applications using grease, it is necessary to replenish the lubricant. Relubrication intervals in hours of operation are dependent on temperature, speed, and bearing size. Table 8.5.10 is a general guide which represents the time after which it is advisable to add a small amount of grease in order to safeguard the bearings. The intervals are valid up to 160°F (71°C) and should be divided by 2 for cylindrical roller bearings and by 10 for spherical roller bearings.

8.6 PACKINGS AND SEALS

by John W. Wood, Jr.

REFERENCES: Staniar, ''Plant Engineering Handbook,'' McGraw-Hill. Thorn, Rubber and Plastic Packings, *Rubber Age,* Jan. 1956. Roberts, Gaskets and Bolted Joints, *Jour. Applied Mechanics,* June 1950. Nonmetallic Gaskets, *Mach. Des.,* Nov. 1954. Elonka, Basic Data on Seals, a *Power* reprint, McGraw-Hill. Fluidtec Engineered Products, Training Manuals.

Packings are materials used to control or stop leakage of fluids (liquids and/or gases) or solid dry products through mechanical clearances when the contained material is under static or dynamic pressure.

Gaskets are compressible materials installed in static clearances which normally exist between parallel flanges or concentric cylinders. Sealing of flat flange gaskets is effected by compressive loading achieved through bolting or other mechanical means. The full face gasket (Fig. 8.6.1) is not recommended because the material outside the bolt holes is ineffective. The simple ring gasket (Fig. 8.6.2) is more efficient and economical. With irregularly contoured flanges, bolt holes may serve to locate the gasket, in which case they should be placed in lobes with full sealing flange width maintained between the inner edge of the holes and the inside of the gasket. **Metal-to-metal** fits require a recess whose volume is greater than that of the gasket to be used. The gasket, such as an O ring (Fig. 8.6.13), either rectangular or round cross section, extends above the groove sufficiently to provide a minimum cross-sectional compression of 15 percent for initial seating. In service, the fluid load automatically provides additional sealing force. **Warped, wavy,** or irregular flanges, often resulting from welding, other fabrication, or as found in glass-lined equipment, require gaskets that are softer or thicker than normal in order to compensate for surface imperfections. Excessive thickness or volume of gasket material, even though the gasket is installed in a groove, must be avoided to prevent distortion or ''mushrooming,'' which will result in inadequate loading. Tongue and groove joints (Fig. 8.6.4) confine the gasket material and may adapt to the extra thickness, within limits.

In addition to the types (Figs. 8.6.5 to 8.6.7) shown, as defined in the table (Fig. 8.6.37), there are the machined metal profile gasket (Fig. 8.6.8) and solid metal designs in flat, round, and either octagonal or oval API ring joint gaskets for extreme pressures and temperatures to seal against steam, oil, and gases. These types have very low compressibilities, and their behavior depends on their cross sections. The envelope gasket (Fig. 8.6.3), usually polytetrafluoroethylene with a variety of cores, is particularly useful for extremely corrosive or noncontaminating service under average pressure.

Cylindrical or **concentric** gasketing uses a retaining gland follower and is mechanically loaded, e.g., the standard mechanical joint for cast-iron pipe (Fig. 8.6.10) or the condenser tube-sheet ferrule (Fig. 8.6.11). Cup-shaped gaskets are designed to be self-tightening under pressure (Fig. 8.6.12). The O ring (Fig. 8.6.13) located in an annular groove and precompressed as in the grooved flange, is a self-energized gasket. A cylindrical ring with internal single lip or double lips, also automatic in action, is quite common in pipe joints.

Beyond these types are many specialty gaskets designed for specific or proprietary use, e.g., a seal for a removable drumhead.

The compressibility of various gasketing materials is shown in Fig. 8.6.37, and their common usage is listed in Table 8.6.1. Beyond rubber are many elastomeric materials generally similar in mechanical behavior but varying as to temperature limits and fluid compatibility (see Sec. 6).

The **proper design** of a gasketed joint requires flange rigidity to avoid distortion, surface finish commensurate with gasket type and good sealing pressure, and adequate bolt loading. The load must seat the gasket, i.e., cause the material to flow into and fill flange irregularities. It must seal sufficiently that the residual fluid pressure on the gasket exceeds the pressure of the fluid being contained. These values, known respectively as the *seating load y* in lb/in^2 and the *gasket factor m*, vary with gasket material and thickness. The ASME Code for Unfired Pressure Vessels, section VIII, gives sufficient detail for typical joint design and tabulates values for y and m for various gasketing materials.

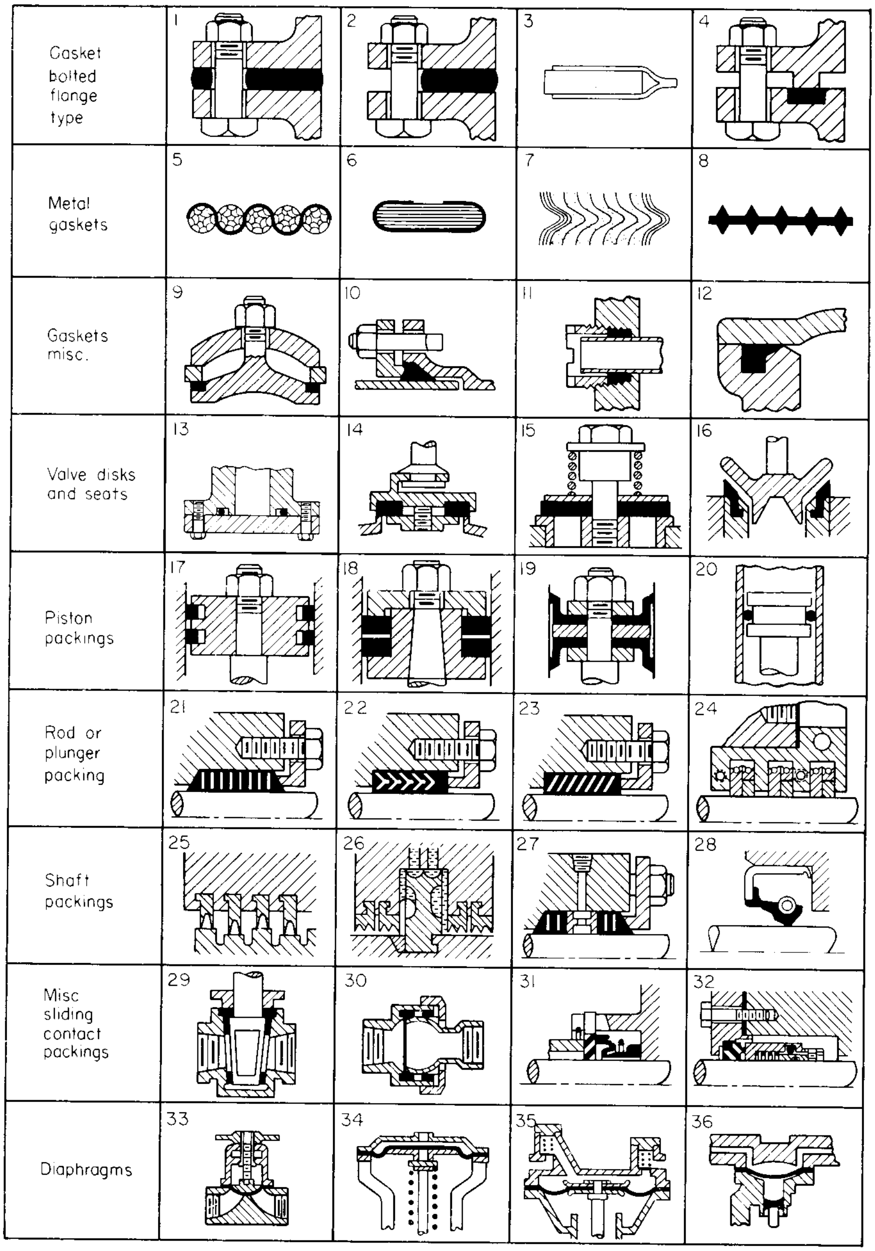


Fig. 8.6.1–8.6.36 Packings.

High bolt loading is desirable for tight and enduring gasket joints, but it must not crush the gasket material. Crushing-strength values, which will vary with thickness and temperature, can be obtained from the gasket manufacturers. Consistent with the condition of the flanges, the thinner the gasket, the more efficient the joint.

Data on the design of **O-ring joints** are available from suppliers. The nominal pressure limit for O rings, based on typical mechanical clearances, is 1,500 lb/in^2 (10 MN/m^2) without backup rings and 3,000 lb/in^2 (20 MN/m^2) with backup rings. If clearances can be eliminated, as in a flanged joint with close metal-to-metal contact, no limit can be set. Other self-energizing joints, such as the boiler hand-hole plate (Fig. 8.6.9), need only sufficient load to effect an initial seal.

Valve disks are specialized gaskets designed for joints that are frequently broken and reseated. Disks for globe valves (Fig. 8.6.14) are usually encased in a disk holder with a swivel mounting, which ensures precise reseating without abrasion during the closing and opening cycles. They are made of firm rubber for bib washers, hard rubber and phenolics for more severe service, and plastics such as nylon and polytetrafluoroethylene. Pump valves (Fig. 8.6.15) are described in Sec. 14. Rubber **valve seats** are used with metal valve disks on some pumps, e.g., the rotary drilling pump valve (Fig. 8.6.16). Plastics are also used for seats, notably in ball valves.

Dynamic packings include all packings that operate on moving surfaces. To retain fluid under pressure, they are subjected to the hydraulic load. When no pressure exists, as in many oil-seal applications, the packing is mechanically loaded by a spring (Fig. 8.6.28) or by its own resiliency. Dynamic packings operate like bearings, wherein the lubricant serves as both a separating film and a coolant. The film is vital for satisfactory service life, but some leakage will occur. Low-viscosity fluids and high pressures add to leakage problems, as both require thin films to minimize leakage. This causes higher friction and generates heat, which is the single most detrimental factor in packing life. Deep packings reduce leakage but increase frictional heat, particularly at high speeds. Normally the fluid being sealed serves as the lubricant. Maximum efficiency is attained when oil is the fluid being sealed; in decreasing order of efficiency are clean water, solvents, and fluids containing solids. These are progressively more unsatisfactory unless supplemental lubrication is provided. Lubrication may be provided by using a lantern ring in the center of the packing set through which lubricant is fed to the packings (Fig. 8.6.27). The preferred method of introducing the lubricant is to supply it at a pressure slightly higher than that of the fluid being sealed, say, 5 to 10 lb/in^2 (3.5 to 7 kN/m^2) higher. The choice of lubricant is governed by the fluid being sealed, since the two should be compatible. In cases of extreme contamination, the lantern ring is moved to the bottom of the packing set to introduce clean lubricant and to prevent abrasives from migrating along the dynamic sealing surface. Use of a lantern ring also seals against air being drawn into the system when equipment is operating at negative head or starts under vacuum. Centrifugal pumps equipped in this manner are said to have a **water seal**.

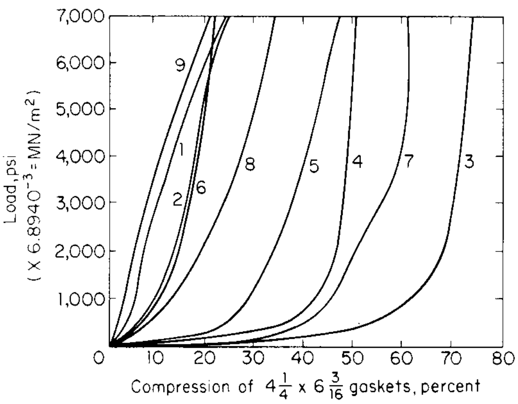


Fig. 8.6.37 Compressibility of gaskets. (See Table 8.6.1.)

Dynamic packings are classified in three ways:

1. On the basis of shape of the surfaces: cylindrical, conical, spherical, or flat. Cylindrical packings are in turn classified according to whether they pack on the outside perimeter, as in piston packings (Fig. 8.6.17 to 8.6.20) or inside perimeter, as on rods or shafts (Figs. 8.6.21 to 8.6.28). Other examples are: conical, the plug cock lining (Fig. 8.6.29); spherical, the ball joint (Fig. 8.6.30); and flat, the mechanical seals (Figs. 8.6.31 and 8.6.32).

2. On the basis of the type of motion: rotary, oscillating, reciprocating, or helical (as in a rising-stem valve packing).

3. On the basis of being nonautomatic soft or jamb packings/compression packings (tightened by external means, usually a gland follower); or automatic preformed, molded shapes (self-tightening under pressure).

The **selection of a packing** is a matter of economics. In most cases several types are available, some of which, though expensive in the first place, yield exceptional service. A cheaper packing could yield degraded service. Service requirements often dictate the final choice of a packing material, and they must reflect and balance the fluid being sealed, compatibility between fluid and gasket material, operating pressure in the system, and ease of maintenance and replacement.

For **reciprocating elements,** the **O ring** (Fig. 8.6.20) is extremely simple. It is a precision part manufactured to close tolerances, as must be the seat into which it is placed. As an elastomeric material completely exposed to the operating fluid, it is subject to chemical degradation. The O-ring material must be carefully chosen to ensure compatibility with the fluid being sealed; the wrong choice will lead to either shrinkage or swelling, with premature failure of the O ring. It is best suited to medium-pressure service from 1,500 to 3,000 lb/in^2 (10 to 20 MN/m^2) with backup rings and intermittent movement, as in hydraulic cylinder or valve stem service. It is not recommended for pump service. Backup rings are preferably of heavy blocklike cross section in either tetrafluoroethylene or similar material, avoiding the thin spiral type. The split **piston ring** (Fig. 8.6.17), usually cast iron, is widely used in gas, oil, and steam engines and compressors. Large pistons frequently employ segmental rings similar to floating metal rod packings (Fig. 8.6.24) but facing outward. **Floating metal packing rings** are made of numerous radial or tangential segments, making it possible for them to contract on the shaft; they are assembled in sets of two to break the joints and are held together with garter springs. They are used for steam, gas, or air, in either engines or compressors under the most severe operating conditions and at pressures up to 35,000 lb/in^2 (241 MPa). Normally oil lubrication is provided; for less severe service, filled polytetrafluoroethylene (PTFE) rings perform very well in dry gases without auxiliary lubrication. Step-, scarf-, or butt-cut rings of laminated cotton fabric, bonded with an elastomer or phenolic resin, are employed in water pumps, gasoline pumps, etc. They may float similar to cast iron piston rings, or be retained by a gland follower, as in Fig. 8.6.18. **Cups** (Fig. 8.6.19) are fully automatic and very tight; cups in their inverted form, with the lip on the ID, are known as **flange packings** and are also fully automatic and very tight. They are used principally for slow-speed applications. **Nested V and conical rings** (Figs. 8.6.22 and 8.6.23) are automatic, though often provided with a gland follower to effect initial fit. They are made of a wide range of materials from homogeneous elastomers and polymers, through reinforced woven fibers (cotton, aramid, or fiberglass) for severe duty. They range in hardness from soft and

Table 8.6.1 Common Usage of Gasketing Materials*

No.	Type	Service principally for:	Thickness tested, in (× 25.4 = mm)
1	Sheet rubber	Water	1⁄16
2	Cloth inserted sheet	Water	1⁄16
3	Cork composition	Oil, low-pressure	1⁄8
4	Gasket paper	Oil, low-pressure	1⁄16
5	Rubberized asbestos cloth (Fig. 8.6.9)	Hot water (boiler manholes, etc.)	1⁄4
6	Compressed asbestos sheet	All services up to 750°F (400°C)	1⁄16
7	Corrugated sheet metal with filling (Fig. 8.6.5)	Steam, oil at high temperatures	1⁄4
8	Metal jacket over asbestos center (Fig. 8.6.6)	Steam, oil at high temperatures	1⁄8
9	Spirally wound steel strip with intervening asbestos layers (Fig. 8.6.7)	Steam, oil at high temperatures	3⁄16

* Asbestos bearing material is found generally in older equipment; current, new, and/or replacement parts are compounded of other materials suitable to the service application. Fiberglass is a common substitute for asbestos in these applications.

flexible to semirigid. Use of multiple rings allows them to be of the cut or split type for ease of installation and replacement. **Soft or jamb packings** are best suited for rod or plunger service, since an adjustable gland follower (Fig. 8.6.21) is required. They are normally formed in rectangular section with a butt joint staggered from ring to ring at installation. Many materials are employed, such as braided flax saturated with wax or viscous lubricants for water and aqueous solutions; braided fiberglass similarly treated or often impregnated with PTFE/graphite suspensoid for more severe service; laminated rubberized cotton fabric for hot water, low-pressure steam, and ammonia; rolled rubberized fiberglass or aramid fabric for steam; and rolled or twisted metal foil for high-temperature and high-pressure conditions. Packings containing woven or braided fibers are also made from wire-inserted yarns to gain additional strength. For pipe expansion joints, see Sec. 8.

Rotary shafts are generally packed with adjustable soft packings, with the notable exception of the mechanical seals (Figs. 8.6.31 and 8.6.32); where pressures are low, nested V or conical styles may be used. At zero or negligible pressures, the oil seal, a spring-loaded flange packing (Fig. 8.6.28), is very widely used. Where some leakage can be tolerated, the labyrinth (Fig. 8.6.25) and controlled-gap seals are used, particularly on high-speed equipment such as steam and gas turbines. **Soft packings** are of the same general type as those used for reciprocating service, with the fiber braid lubricated with grease and graphite or with polytetrafluoroethylene fibers and suspensoid. Aramid, carbon, and graphite fibers filled with various lubricants and reinforcements are used at higher speeds and fluid pressures. Fiber braid with PTFE suspensoid is widely applied on valve stems operating below 500°F (260°C) and on centrifugal pumps. This material is an insulator, however, and results in high heat buildup on the dynamic surface; a better choice lies in use of a packing with better heat-transfer characteristics, such as one containing carbon or graphite. For continuous rotary service, **automatic packings** are best restricted to low pressure because their tightness under high pressure results in overheating. For intermittent service, as on valve stems, they are excellent.

Oil seals (Fig. 8.6.28) are unique flange packings having an elastomer lip generally bonded to a metal cup which is press-fitted into a smooth cylindrical bore. Basically, an oil seal is a flange packing with a flexible lip and a narrow contact area about 1⁄16 in (1.6 mm) wide which, under pressure, causes extreme local heating and wear. They are recommended only for nonpressure service and perform best in good lubricating media. To accommodate shaft runout up to 0.020 in (0.5 mm) depending on the rotating speed, the lip is spring-loaded with a coil spring or a finger spring. Coil springs are safer inasmuch as they are molded into the elastomer and are less likely to become dislodged and cause shaft damage. Since the lip is completely exposed to the sealed fluid, particular care should be taken to ensure compatibility between the elastomer and the fluid. Temperature is another operating condition which must be taken into consideration when one is using oil seals.

Mechanical, Rotary, or End Face Seals

The greatest advancements in the design of end face mechanical seals have come about in response to environmental regulations; requirements to minimize energy consumption and operating costs; safety; and concerns over loss of the product which is being sealed. The application of seals to replace packing in rotary equipment has increased dramatically and continues.

All end face mechanical seals (Figs. 8.6.31 and 8.6.32) consist of four parts: a stationary flat face, a rotating flat face, secondary sealing elements (usually elastomeric), and a flexible loading device. The assembled seal is placed and effects proper leak control. The two flat-face seal rings (one stationary, one rotating) rub and create the primary seal. Normally, the flat seal rings have different hardness values, and the soft one is narrower than the hard one. Secondary sealing elements prevent leakage between the rotating shaft and the rotating seal ring, and they block the leakage path around the outside of the stationary seal face. They also serve as gaskets between the assembled parts (i.e., gland plate and housing). The flexible loading device usually consists of one or more springs which press the flat seal rings together. Spring loading ensures a seal when there is little or no hydraulic pressure available to press the faces together and helps maintain constant pressure between the faces as the soft (sacrificial) face wears down. The springs also act as vibration dampers to mitigate against the intrusion of transmitted vibrations, which may affect the efficient operation of the seal assembly.

Types of End Face Mechanical Seals

1. *Inside-mounted.* The seal head is mounted inside the stuffing box (Fig. 8.6.38*a*).

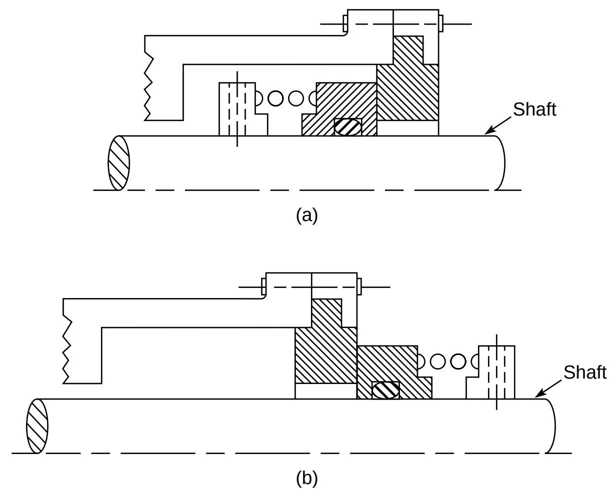


Fig. 8.6.38 Rotary end face seal. (*a*) Inside the seal chamber/stuffing box; (*b*) outside the seal chamber/stuffing box.

2. *Outside-mounted.* The seal head is mounted outside the stuffing box (Fig. 8.6.38*b*).

3. *Unbalanced seal.* The full hydraulic pressure in the seal chamber is transmitted to the seal faces (Fig. 8.6.39*a*).

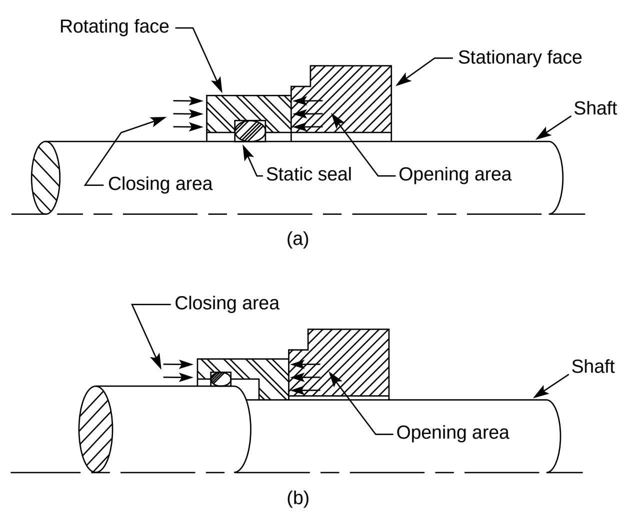


Fig. 8.6.39 Rotary end face seals showing (*a*) unbalanced and (*b*) balanced configurations.

4. *Balanced.* The seal elements are designed to reduce the hydraulic forces transmitted to the seal faces (Fig. 8.6.39*b*). A complete balance is not practical.

5. *Rotary seal.* In this design, the springs rotate with the shaft.

6. *Stationary seal.* In this design, the springs do not rotate with the shaft.

7. *Metal bellows.* Welded or formed metal bellows exert a spring load; there is no dynamic secondary seal element (Fig. 8.6.40).

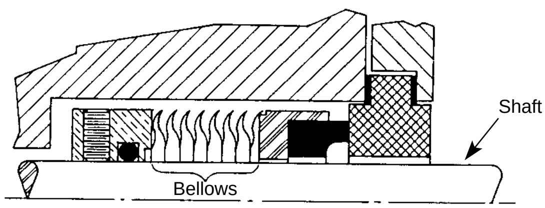


Fig. 8.6.40 Rotary end face seal with metal bellows and "t" clamp stationary.

8. *Double seal.* Two mechanical seals are mounted back to back, face to face, or in tandem, between which a barrier fluid (liquid or gas) can be introduced for environmental control (Fig. 8.6.41).

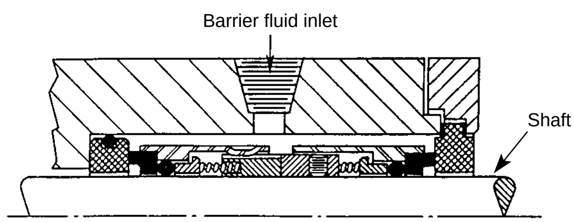


Fig. 8.6.41 Back-to-back double seal. Barrier fluid must have an inlet and an outlet.

End face mechanical seal materials must satisfy a number of design requirements, including chemical compatibility between the sealed fluid and the seal materials, ability of the seal materials to remain serviceable under the worst operating conditions, and ability to provide a reasonably long life in service at the operating conditions. Mating faces of the seals can be made from ordinary materials like bronze and PTFE

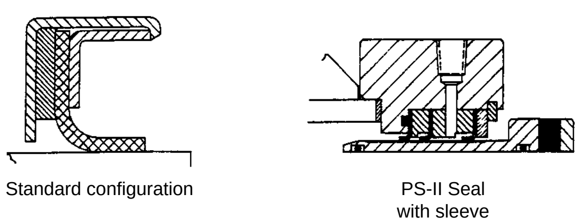


Fig. 8.6.42 High-performance lip seal with modified PTFE elastomer. Illustration shows single and staged elements.

for mild service on up to carbon, carbides, stainless steels, and other exotic alloys as service conditions become more severe. Hard faces can utilize ceramics, tungsten and silicon carbides, and hard coatings over base metals (chromium oxide over stainless steel 316SS, subsequently lapped flat).

Secondary seal materials are usually elastomeric and include these:

Elastomer	*Temperature range*
Nitrile	−75 to 250°F (−59 to 121°C)
Neoprene	−65 to 250°F (−54 to 121°C)
Viton	−40 to 450°F (−40 to 232°C)
Ethylene propylene	−65 to 300°F (−59 to 149°C)
Kalrez	32 to 550°F (−0 to 288°C)

Environmental controls as applied to seals are special techniques used to control the environment in which the seal operates. Some examples include discharge return recirculation, suction return, flush from an outside source, flush, and quench and drain. The controlling techniques often require supplementary equipment such as heaters, coolers, pumping rings, external circulation devices, various special solenoids and valves compatible with the working environment, and so on. Sealed products that solidify, vaporize, abrade, or carry contaminants are successfully sealed with end face seals augmented by environmental controls.

High Performance Lip Seals The nature of some sealed products is such that end face mechanical seals are not applicable. In many difficult instances of that type, sealing can be achieved with high-performance modified PTFE lip seals (Fig. 8.6.42). Gylon is such a material which can serve in seals operating over a wide range of pressures, temperatures, and rotating speeds. It is particularly useful to seal against dry products, viscous resins, heavy slurries, salting solutions, and products which tend to solidify on seal faces. Dry running is possible under some circumstances. Unlike conventional lip seal material, modified PTFE lip seals in multiples can operate from high vacuums (10^{-3} inHg) up to 10 bar (150 lb/in²), within a temperature range of −130 to +500°F (−90 to +260°C), and exhibit excellent compatibility with a wide range of sealed fluids. Manufacturers' literature will provide data showing the effect of temperature and rotating speed on the permissible operating pressure.

For extremely high speeds, where it is desirable to eliminate all rubbing contact, the **labyrinth** seal (Fig. 8.6.25) is chosen. This seal is not fluid-tight but restricts serious flow by means of a torturous path and induced turbulence. It is widely used on steam turbines (Sec. 9.4). Where no leakage is permissible, a liquid seal based on the U-tube principle (Fig. 8.6.26) may be used. The natural weight of the liquid is amplified by centrifugal force so that under high rotating speed a fair pressure differential can be sealed. Another noncontacting seal is the **controlled gap seal** which is being used on gas turbines where pressure differentials are not excessive and a small amount of leakage can be tolerated. The seal consists of a ring with a shaft clearance in the range of 0.0005 to 0.0015 in (0.013 to 0.038 mm) and is made of exotic heat-resisting materials capable of maintaining that clearance at all operating temperatures. Usually one end of the ring is faced to form an axial seal against the inside of its housing.

Diaphragms are a form of dynamic packing but include the requirements of a gasket where they are gripped or held in position. In service they are leakless, although generally limited in travel. By literally rolling one cylinder inside another, considerable increase in travel is possible. This type is often called a bellows, and a simple application is the mechanical seal suspension shown in Fig. 8.6.31. In the diaphragm valve (Fig. 8.6.33) the diaphragm replaces both the conventional stem packing and valve disk. **Diaphragms** of fabric such as cotton or nylon (except friable materials such as glass) covered with an elastomer suitable for the fluids and temperatures involved are used in pumps (fuel pump, Fig. 8.6.35) and in motors (Fig. 8.6.34) to operate valves, switches, and other controls. Correctly designed diaphragms are made with slack to permit a natural rolling action. Flat sheet stock should be used only where limited travel is desired. An unusual application is shown in Fig. 8.6.36, where the diaphragm is under balanced fluid pressure on both sides and is unstressed. Thin sheet metal, usually with concentric corrugations, is used where movement is limited and long life is desired. Where considerable movement is involved, the possibility of fatigue must be considered.

PTFE and Glyon diaphragms are used with chemically aggressive fluids. Experience shows that PTFE has a tendency toward cold flow, which leads to leaking at the clamp areas; Gylon has proved more dimensionally stable and serviceable.

8.7 PIPE, PIPE FITTINGS, AND VALVES

by Helmut Thielsch

REFERENCES: M. L. Nayyar, ''Piping Handbook,'' McGraw-Hill. ANSI Code for Power Piping. ASTM Specifications. Tube Turns Division, Natural Cylinder Gas Co., catalogs. Crane Co., catalogs and bulletins. Grinnell Co., Inc., ''Piping Design and Engineering.'' M. W. Kellogg Co., ''Design of Piping Systems,'' Wiley. United States Steel Co., catalogs and bulletins.

EDITOR'S NOTE: The several piping standards listed in this section are subject to continuing periodic review and/or modification. It is suggested that the reader make inquiry to the issuing organizations (see Table 8.7.1) as to the currency of a given standard as listed.

PIPING STANDARDS

Codes for various piping services have been developed by nationally recognized engineering societies, standardization bodies, and trade associations. The sound engineering practices incorporated in these codes generally cover minimum safety requirements for the selection of materials, dimensions, design, fabrication, erection, and testing of piping systems. By means of interpretation and revision these codes continually reflect the knowledge gained through experience, testing, and research.

Generally, piping codes form the basis for many state and municipal safety laws. Compliance with a code which has attained this status is mandatory for all systems included within the jurisdiction. Although some of today's piping installations are not within the scope of any mandatory code, it is advisable to comply with the applicable code in the interests of safety and as a basis for contract negotiations. Contracts with various agencies of the federal government are regulated by federal specifications or rules. These often do not have a direct connection with the codes enumerated below.

The reader is cautioned that the **piping standards** are changing more often than in previous years. Although the formulas and other data provided are in accordance with the code rules in effect at the time of publication, it must be recognized that code rules may change, and piping engineering and design work performed in accordance with information contained herein does not provide complete assurance that all extant code requirements have been met. The reader is urged to become familiar with the specific code edition and addenda applicable in a particular project, for they may contain mandatory requirements applicable to the particular project.

The **ASME Boiler and Pressure Vessel Code** is mandatory in many cities, states, and provinces in the United States and Canada. Local application of this code into law is not uniform, making it necessary to investigate the city or state laws which have jurisdiction over the installation in question. Compliance with this code is required in all locations to qualify for insurance approval.

Section I: ''Power Boilers'' concerns all piping connections to power boilers or superheaters including the first stop valve on single boilers, or including the second stop valve for cross-connected multiple-boiler installations. Section I refers to ASME B31.1 which contains rules for design and construction of ''boiler external piping.'' ''Boiler external piping'' is under the jurisdiction of Section I and requires inspection and code stamping in accordance with Section I even though the rules for its design and construction are contained in the ASME Code for Pressure Piping, section B31.1.

Section II ''Material Specifications'' provides detailed specifications of the materials which are acceptable under this code. (These specifications generally are identical to the corresponding ASTM Standards.)

Section III: ''Nuclear Components'' includes all nuclear piping. It is the responsibility of the designer to determine whether or not a particular piping system is ''nuclear'' piping, since Section III makes this determination the responsibility of the designer. In general, piping whose failure could result in the release of radiation which would endanger the public or plant personnel is considered ''nuclear'' piping.

Section VIII: ''Unfired Pressure Vessels'' concerns piping only to the extent of the flanged or threaded connections to the pressure vessel, except that the entire section will apply in those special cases where unfired pressure vessels are made from pipe and fittings.

Section IX: ''Welding and Brazing Qualifications'' establishes the minimum requirements for ASME Code welding.

Section XI: ''Rules for Inservice Inspection of Nuclear Power Plant Components'' contains rules for the examination and repair of components throughout the life of the plant.

The ASME Code for Pressure Piping B31 is, at present, a nonmandatory code in the United States except where U.S. state legislative bodies and Canadian provinces have adopted this code as a legal requirement. The minimum safety requirements of these codes have been accepted by the industry as a standard for all piping outside the jurisdiction of other codes. The piping systems covered by the separate sections of this code are listed below:

Power Piping	B31.1
Fuel Gas Piping	B31.2
Chemical Plant and Petroleum Refinery Piping	B31.3
Liquid Petroleum Transportation Piping Systems	B31.4
Refrigeration Piping	B31.5
Gas Transmission and Distribution Piping Systems	B31.8
Building Service Piping	B31.9

Several other engineering societies and trade associations have also issued standards covering piping. Foremost among these is the American Society for Testing and Materials (ASTM), the American National Standards Institute (ANSI), the American Water Works Association (AWWA), the American Petroleum Institute (API), and the Manufacturers Standardization Society of the Valve and Fitting Industry (MSS).

Additional piping specifications have been issued by the American Welding Society (AWS), the Pipe Fabrication Institute (PFI), the National Fire Protection Association (NFPA), the Copper Development Association (CDA), the Plastics Pipe Institute (PPI), and several others.

The piping standards issued by the ASTM are most commonly referred to in specifications covering piping for power plants, chemical plants, refineries, pulp and paper mills, and other industrial plants. The large majority of ASTM Standards has also been issued by the ASME in Section II of the ASME Boiler and Pressure Vessel Code. The same specification numbers are applied by the ASME as were originally assigned by the ASTM.

The ANSI formerly prepared the various standards of the B31 Code for Pressure Piping. These standards are now issued by the ASME. The ANSI, however, continues to prepare and issue various standards covering pipe fittings, flanges, and other piping components. Note that ASME B16 prepares and issues standards for fittings, flanges, etc.

The AWWA has issued various standards for waterworks applications. The majority of these involve ductile iron pipe, ductile iron and cast iron pipe fittings, etc.

The MSS has prepared various standards for valves, hangers, and fittings, generally involving the lower range of pressures and temperatures.

Table 8.7.1 gives the most **commonly used piping standards** and the organizations from which the standards are available.

Table 8.7.1 Commonly Used Piping Standards

ASTM Specifications	ASTM Specifications (*Cont.*)	ASTM Specifications (*Cont.*)	ASTM Specifications (*Cont.*)
*A 20-82	*A 516-82	*C 361-78	*F 492-77
*A 36-81a	*A 522-81	*C 582-68 (R1974)	*F 493-80
*A 47-77	*A 524-80	*C 599-70 (R1977)	*F 546-77
*A 48-76	*A 530-82	*D 1503-73 (R1978)	*F 599-78
*A 53-81a	*A 537-82	*D 1527-77	SAE Specifications
*A 105-82	*A 553-82	D 1600-80	*J 513f-1977
*A 106-82	*A 579-79	D 1694-79	*J 514j-1980
*A 120-82	*A 571-82	*D 1785-76	*J 518c-1972
*A 126-73 (R1980)	*A 587-78	*D 2104-74	ASNT Standard
*A 134-80	*A 645-82	*D 2235-81	SNT TC-1A-1980
*A 135-79	*A 658-82	*D 2239-74	*A3.0-1980
*A 139-74 (R1980)	*A 671-80	*D 2241-80	*A5.1-1981
*A 167-81a	*A 672-81	*D 2282-77	*A5.4-1981
*A 179-79b	*A 675-82	*D 2310-80	*A5.5-1981
*A 181-81	*A 691-81	*D 2412-77	Copper Development Assn.
*A 182-82	*B 12-76a	*D 2446-73 (R1978)	Copper Tube Handbook
*A 193-82	*B 21-81	*D 2447-74	EJMA Standards, Current
*A 194-82	*B 26-82b	*D 2464-76	*American National Standards
*A 197-79	*B 42-82	*D 2465-73 (R1979)	A21.14-1979
*A 202-82	*B 43-80	*D 2466-78	A21.52-1981
*A 203-82	*B 61-82a	*D 2467-76a	A58.1-1972
*A 204-82	*B 62-82a	*D 2468-80	B1.1-1974
*A 211-75a (R1980)	*B 68-80	*D 2469-76	B1.20.3-1976
*A 216-82	*B 75-81a	*D 2513-80a	B1.20.1-1983
*A 217-81	*B 88-81	D 2517-81	B1.20.7-1966 (R1983)
*A 225-82	*B 96-82	*D 2560-80	B16.1-1975
*A 234-82	*B 98-82	*D 2564-80	B16.3-1977
*A 240-82b	*B 127-80a	*D 2609-74	B16.4-1977
*A 263-82	*B 148-82	*D 2657-79	B16.5-1981
*A 264-82	*B 150-82a	*D 2662-81	B16.9-1978
*A 265-81a	*B 152-82	*D 2666-81a	B16.9a-1981
*A 268-82a	*B 160-81	*D 2672-80	B16.10-1973
*A 269-82	*B 161-81	*D 2683-80	B16.11-1980
*A 276-82a	*B 162-80	D 2737-74	B16.14-1977
*A 278-75 (R1980)	*B 164-81	*D 2740-80	B16.15-1978
*A 283-81	*B 165-81	*D 2837-76	B16.18-1978
*A 285-82	*B 166-81	*D 2846-80	B16.20-1973
*A 299-82	*B 167-81	*D 2855-80	B16.21-1978
*A 302-82	*B 168-80a	*D 2992-71 (R1977)	B16.22-1980
*A 307-82a	*B 169-82	*D 2996-81	B16.24-1979
*A 312-82	*B 209-82b	*D 2997-71 (R1977)	B16.25-1979
*A 320-82	*B 210-82a	D 3000-73	B16.26-1975
*A 325-82	*B 211-82b	D 3035-74	B16.28-1978
*A 333-82	*B 221-82a	D 3036-73	B16.33-1981
*A 334-79	*B 241-82a	D 3139-77	B16.34-1981
*A 335-81a	*B 247-82a	*D 3140-72 (R1977)	B16.36-1975 (A1979)
*A 338-61 (R1977)	*B 283-81	D 3197-73	B16.38-1978
*A 350-81a	*B 333-77	*D 3261-81	B16.39-1977
*A 351-82	*B 335-77	D 3287-73	B16.42-1979
*A 352-82	*B 337-78	*D 3309-80a	B18.2.1-1981
*A 353-82	*B 345-82	*E 112-82	B18.2.2-1972
*A 354-82b	*B 361-81	*E 114-75 (R1981)	B36.10-1979
*A 358-81	*B 366-81	*E 125-63 (R1980)	B36.19-1976
*A 369-79a	*B 402-81	*E 142-77	B46.1-1978
*A 370-82	*B 407-77	*E 155-79	B93.11-1981
*A 376-81	*B 443-82	*E 165-80	PPI Technical Reports
A 377-79	*B 444-82	*E 186-81	TR-21-1973
*A 381-81	*B 446-82	*E 272-75 (R1979)	Pipe Fabrication Institute
*A 387-82	*B 466-82a	*E 280-81	ES-7-1962 (R1980)
*A 395-80	*B 467-82	*E 310-75 (R1979)	Federal Specification
*A 403-82	*B 574-77	*E 446-81	WW-P-421D. Sept. 1976
*A 409-81a	*B 575-77	*E 709-80	
*A 420-81a	*B 581-81	*F 336-78	
*A 426-80	*B 582-81	F 423-75	
*A 430-79	*B 584-82	*F 437-77	
*A 437-82	*B 619-81	*F 438-77	
*A 451-80	*B 620-77	*F 439-77	
*A 452-79	*B 621-77	*F 441-77	
*A 453-80	*C 14-78	*F 442-77	
*A 494-81	*C 296-78	*F 443-77	
*A 515-82	*C 301-79	*F 491-77	

NOTE: Footnotes appear at the end of the table.

Table 8.7.1 Commonly Used Piping Standards (*Continued*)

API Standards†

5B, 10th ed., 1979 (A1982)
5L, 32d ed., 1982
5LE, 2d ed., 1976
5LP, 4th ed., 1976
5LR, 4th ed., 1976
5LS, 12th ed., 1982
5LX, 24th ed., 1982
*526, 2d ed., 1969 (R1977)
593, 2d ed., 1981
594, 2d ed., 1977
595, 2d ed., 1979
597, 3d ed., 1981
599, 2d ed., 1978
600, 8th ed., 1981
601, 5th ed., 1982
602, 4th ed., 1978
603, 3d ed., 1977
604, 4th ed., 1981
*605, 3d ed., 1980
606, 1st ed., 1976
609, 2d ed., 1978
*C101-1967 (R1977)
*C110-1977
*C111-1980
*C115-1975
*C150-1976

API Standards (*Cont.*)

*C151-1976
*C200-1980
*C207-1978
C208-1959
C300-1974
*C301-1972 (A1974)
C302-1974
C400-1977
*C402-1977
*C500-1980
*C504-1980
*C600-1982
C900-1975

ASME Codes

*ASME Boiler and Pressure Vessel Code, 1980 ed.
*Section V, incl. addenda through W82
*Section VIII, Division 1
*Section VIII, Division 2
*Section IX, incl. addenda through W82

MSS Standard Practices

SP-6-1985

MSS Standard Practices (*Cont.*)

SP-9-1984
SP-25-1978 (R83)
SP-42-1985
SP-43-1982
SP-44-1985
SP-45-1982
SP-51-1982
SP-53-1985
SP-55-1985
*SP-58-1983
SP-60-1982
SP-61-1985
SP-65-1983
SP-67-1985
SP-68-1984
SP-69-1983
SP-70-1984
SP-71-1984
SP-72-1970
SP-73-1982
SP-75-1983
SP-77-1984
SP-78-1977
SP-79-1980
SP-80-1979
SP-81-1981

MSS Standard Practices (*Cont.*)

SP-82-1976 (R81)
SP-83-1976
SP-85-1985
SP-86-1981
SP-87-1987
SP-88-1978
SP-89-1985
SP-90-1980
SP-91-1984
SP-92-1982
SP-93-1982
SP-94-1983

CGA

G-4.1-1977

NACE

Corrosion Data Survey

NBS

PS 15-69

NFPA Specifications, current

Uniform Building Code, current

Aluminum Assn.

The referenced standards are available from the listed organizations:

Standards sources

Alum. Assn.	Aluminum Association 900 19th St., NW, Washington, DC 20006 202 862-5100
ANSI	American National Standards Institute, Inc. 11 West 42d St., New York, NY 10036 212 642-4900
API	American Petroleum Institute 1220 L Street, NW, Washington, DC 20005-8029 202 682-8000
ASME	The American Society of Mechanical Engineers 345 East 47th Street, New York, NY 10017 212 705-7722
ASNT	American Society for Nondestructive Testing 3200 Riverside Drive, Columbus, OH 43221 614 488-7921
ASTM	American Society for Testing and Materials 1916 Race Street, Philadelphia, PA 19103 215 299-5400
AWWA	American Water Works Association 6666 W. Quincy Avenue, Denver, CO 80235 303 794-7711
AWS	American Welding Society 2501 N.W. 7th Street, Miami, FL 33125 305 642-7090
CDA	Copper Development Association 260 Madison Avenue, New York, NY 10016 212 251-7234
(a) CGA	Compressed Gas Association 1235 Jefferson Davis Highway Arlington, VA 22202
(a) EJMA	Expansion Joint Manufacturers Association 25 North Broadway, North Tarrytown, NY 10591 914 382-0040
Fed. Spec.	Federal Specification: Superintendent of Documents United States Government Printing Office Washington, DC 20402 202 541-3000
MSS	Manufacturers Standardization Society of the Valve and Fittings Industry 127 Park Street, NE, Vienna, VA 22180 703 281-6613
NACE	National Association of Corrosion Engineers P.O. Box 986 Katy, TX 77450 713 492-0535
NIST	National Institute of Standards and Technology (U.S. Dept. of Commerce): Publications available from Superintendent of Documents United States Government Printing Office Washington, DC 20402 202 541-3000
NFPA	National Fire Protection Association P.O. Box 9101 1 Batterymarch Park, Quincy, MA 02269-9101 617 770-3000
PFI	Pipe Fabrication Institute Box 173, Lenore Avenue, Springdale, PA 15144-1518 412 274-4722
PPI	Plastics Pipe Institute 65 Madison Avenue, Morristown, NJ 07960-6078 No telephone listed
SAE	Society of Automotive Engineers 400 Commonwealth Drive Warrendale, PA 15096 412 776-4841
UBC	Uniform Building Code International Conference of Building Officials 5360 South Workman Mill Road Whittier, CA 90601 213 699-0541

* Indicates that the standard has been approved as an American National Standard by the American National Standards Institute.
† Including supplements to these API Standards through spring 1981.
NOTE: The issue date shown immediately following the hyphen after the number of the standard (e.g., B16.9-1978, C207-1978, and A 47-77) is the effective date of the issue (edition) of the Standard. Any additional number shown following the issue date and prefixed by the letter R is the latest date of reaffirmation [e.g., C101-1967 (R1977)]. Any edition number prefixed by the letter A is the date of the latest addenda accepted [e.g., B16.36-1975 (A1979)].

PIPING, PIPE, AND TUBING

The term **piping** generally is broadly applied to pipe, fittings, valves, and other components that convey liquids, gases, slurries, etc.

The term **pipe** is applied to tubular products of dimensions and materials commonly used for pipelines and connections, formerly designated as **iron pipe size (IPS)**. The outside diameter of all weights and kinds of IPS pipe is of necessity the same for a given pipe size on account of threading. Nevertheless, the large majority of pipe is furnished unthreaded with butt-weld ends.

The word **tube** (or **tubing**) is generally applied to tubular products as utilized in boilers, heat exchangers, instrumentation, and in the machine, aircraft, automotive, and related industries.

Pipe and Tube Products—General

Commercial pipe and tube products are grouped into various classifications generally based on the application or use and not on the manufacturing method. Most tubular products fall into one of three very broad classifications: (1) pipe, (2) pressure tubes, and (3) mechanical tubes. Each classification falls into various subgroupings, which may have been defined and standardized differently by the different trade or user groups. The same standard materials specifications may apply to several of the (user) classifications. For example, ASTM A120 or A53 pipe may be used for applications representing refrigeration, pressure, and nipple service.

Cost considerations enter into the selection of specific piping materials. In some sizes, prices of pipe made to different materials specifications may vary, whereas in other sizes, they may be identical.

Within the broad **use classifications** listed above, the **production method classifications** are also recognized. These are primarily (1) seamless wrought pipe, (2) seamless cast pipe, and (3) seam-welded pipe or tubes. The large variety of single and combination pipe- or tube-forming methods can produce different characteristics and properties in essentially identical pipe materials. In addition, the final finishing can result in hot-finished or cold-finished products. Cold-finishing may be accomplished by reducing or by expanding. Heat treatments may also affect the properties of the finished product.

Piping

On the basis of user classification, the more commonly used types of pipe are tabulated in Table 8.7.2. This listing ignores method of manufacture, size range, wall thickness, and finish, for which the different user groups may have developed different standard requirements.

Table 8.7.2 Major Pipe Classification and Examples of Applications

Identification of pipe	Uses
Standard	Mechanical (structural) service pipe, low-pressure service pipe, refrigeration (ice-machine) pipe, ice-rink pipe, dry-kiln pipe
Pressure	Liquid, gas, or vapor service pipe, service for elevated temperature or pressure, or both
Line	Threaded or plain end, gas, oil, and steam pipe
Water well	Reamed and drifted, water-well casing, drive pipe, driven well pipe, pump pipe, turbine-pump pipe
Oil country tubular goods	Casing, well tubing, drill pipe
Other pipe	Conduit, piles, nipple pipe, sprinkler pipe, bedstead tubing

Standard Pipe Mechanical service pipe is produced in three classes of wall thickness—standard weight, extra strong, and double extra strong. It is available as welded or seamless pipe of ordinary finish and dimensional tolerances, produced in sizes up to 12-in nominal OD. This pipe is used for structural and mechanical purposes. Certain applications have other requirements for size, surface finish, or straightness.

Refrigeration Pipe This pipe is also known as ice-machine pipe or ammonia pipe. It may be butt-welded, lap-welded, electric-resistance-welded, or seamless and is intended for use as a conveyor of refrigerants. This pipe is suitable for coiling, bending, and welding. The sizes commonly used range from ¾ to 2 in. The piping is produced in random and double random lengths in standard line pipe sizes and weights. Double random lengths are used as ice-rink pipe. It can be produced with plain ends, with threaded ends only, or with threaded ends and line pipe couplings, as desired.

Dry-Kiln Pipe This pipe is butt-welded, electric-resistance-welded, or seamless pipe for use in the lumber industry. It is produced in standard-weight pipe sizes of ¾, 1, and 1¼ in. Joints are designed to permit subsequent ''makeup'' after expansion has occurred. Dry-kiln pipe is commonly produced with threaded ends and couplings and in random lengths.

Pressure Pipe Pressure pipe is used for conveying fluids or gases at normal, subzero, or elevated temperatures and/or pressures. It generally is not subjected to external heat application. The range of sizes is ⅛-in nominal size to 36-in actual OD. It is produced in various wall thicknesses. Pressure piping is furnished in random lengths, with threaded or plain ends, as required. Pressure pipe generally is hydrostatically tested at the mill.

Line Pipe Line pipe is seamless or welded pipe produced in sizes from ⅛-in nominal OD to 48-in actual OD. It is used principally for conveying gas, oil, or water. Line pipe is produced with ends which are plain, threaded, beveled, grooved, flanged, or expanded, as required for various types of mechanical couplers, or for welded joints. When threaded ends and couplings are required, recessed couplings are normally supplied.

Water-Well Pipe Water-well pipe is welded or seamless steel pipe used for conveying water for municipal and industrial applications. Pipelines for such purposes involve flow mains, transmission mains, force mains, water mains, or distribution mains. The mains are generally laid underground. Sizes range from ⅛- to 106-in OD in a variety of wall thicknesses. Pipe is produced with ends suitably prepared for mechanical couplers, with plain ends beveled for welding, with ends fitted with butt straps for field welding, or with bell-and-spigot joints with rubber gaskets for field joining. Pipe is produced in double random lengths of about 40 ft, single random lengths of about 20 ft, or in definite cut lengths, as specified. Wall thicknesses vary from 0.068 in for ⅛-in nominal OD to 1.00 in for 106-in actual OD.

When required, water-well pipe is produced with a specified coating or lining or both. For example, cement-mortar lining and coatings are extensively used.

Oil Country Goods Casing is used as a structural retainer for the walls of oil or gas wells. It is also used to exclude undesirable fluids, and to confine and conduct oil or gas from productive subsurface strata to the ground level. Casing is produced in sizes 4½- to 20-in OD. Size designations refer to actual outside diameter and weight per foot. Ends are commonly threaded and furnished with couplings. When required, the ends are prepared to accommodate other types of joints.

Drill Pipe Drill pipe is used to transmit power by rotary motion from ground level to a rotary drilling tool below the surface and also to convey flushing media to the cutting face of the tool. Drill pipe is produced in sizes 2⅜- to 6⅝-in OD. Size designations refer to actual outside diameter and weight per foot. Drill pipe is generally upset, either internally or externally, or both, and is furnished with threaded ends and couplings, threaded only, or prepared to accommodate other types of joints.

Tubing is used within the casing of oil wells to conduct oil to ground level. It is produced in sizes 1.050- to 4.500-in OD in several weights per foot. Ends are threaded and fitted with couplings and may or may not be upset externally.

Other Pipe Classifications **Rigid conduit pipe** is welded or seamless pipe intended especially for the protection of electrical wiring systems. Conduit pipe is not subjected to hydrostatic tests unless so specified. It is furnished in standard-weight pipe sizes from ¼- to 6-in OD in 10-ft lengths,* with plain ends or with threaded ends and couplings, as specified.

Piling pipe is welded or seamless pipe for use as piles, where the cylinder section acts as a permanent load-carrying member or where it acts as a shell to form cast-in-place concrete piles. Specifications provide for the choice of three grades by minimum tensile strength, in which the sizes listed are 8⅝- to 24-in OD in a variety of wall thicknesses and in two length ranges. Ends are plain or beveled for welding.

Nipple pipe is standard-weight, extra-strong, or double-extra-strong welded or seamless pipe produced for the manufacture of pipe nipples. Standard-weight pipe with threaded ends is also used in sprinkler systems. Nipple pipe is commonly produced in random lengths with plain ends in nominal sizes ⅛- to 12-in OD. Close OD tolerances, sound welds, good threading properties, and surface cleanliness are essential in this product. It is commonly coated with oil or zinc and well protected in shipment. When reference is made to ASTM Specifications for this application, Specification A120 is generally used for diameters to 5-in OD and A53 for diameters of 5 in and over.

Standard Pipe Sizes Standard pressure, line, and other pipe with plain ends for welding or with threaded ends is standardized in two ranges. Diameters of 12 in and less have a nominal size which represents approximately that of the inside diameter of standard-weight pipe. The nominal outside diameter is standard, regardless of weight. Increase in wall thickness results in a decrease of the inside diameter.

The standardization of pipe sizes over 12 in is based on the actual outside diameter, the wall thickness, and the weight per foot.

The principal dimensions, weights, and characteristics of commercial piping materials are summarized in Table 8.7.3.

The weights of butt-welding elbows, tees, and laterals and flanges are given in Tables 8.7.4 to 8.7.9 for several common pipe sizes. The weights of reducing fittings are approximately the same as for full-size fittings.

The weights of welding reducers are for one size reduction and are thus only approximately correct for other reductions.

Hot-finished or cold-drawn seamless low-alloy steel tubes generally are process-annealed at temperatures between 1,200 and 1,350°F.

Austenitic stainless-steel tubes are usually annealed at temperatures between 1,800 and 2,100°F, with specific temperatures varying somewhat with each grade. This is generally followed by pickling, unless bright-annealing was done.

Mechanical Tubing

Unlike pipe and pressure tubes, mechanical tubing is generally classified by the method of manufacture and the degree of finish. Examples of classifications are ''seamless hot-finished,'' ''cold-drawn welded,'' ''flash-in-grade,'' etc.

Seamless Tubes Seamless tubes are available as either hot- or cold-finished. They are normally made in sizes from 0.187-in OD to 10.750-in OD.

Dimensions for hot-finished mechanical tubes are provided in Table 8.7.11. Dimensions for cold-finished tubes are listed in Table 8.7.12.

Welded Tubes Welded tubes generally are produced by electric resistance methods. Where required, the welding flash is removed with a cutting tool. Industry practice normally recognizes a number of finish conditions which are summarized in Table 8.7.13.

Flash-in Type Tubing This tubing is generally limited to applications where nothing is inserted in the tube.

Flash-Controlled Tubing This tubing is used where moderate control of the inside diameter is required. Generally, the outside and inside diameters are specified.

For special materials, the equations listed below for weights of tubes and weights of contents of tubes are helpful.

$$\text{Weight of tube, lb/ft} = F \times 10.68 \times T \times D - T$$

where T = wall thickness, in; D = outside diameter, in; F = relative weight factor.

The weight of tube calculation is based on low-carbon steel weighing 0.2833 lb/in^3 and is extended to other materials through the factor F.

Relative weight factor F

Aluminum	0.35
Brass	1.12
Cast-iron	0.91
Copper	1.14
Ferritic stainless steel	1.02
Steel	1.00
Wrought iron	0.98

$$\text{Weight of contents of tube, lb/ft} = G \times 0.3405 \times (D - 2T)^2$$

where G = specific gravity of contents; T = tube wall thickness, in; D = tube outside diameter, in.

The weight per foot of steel pipe is subject to the tolerances listed in Table 8.7.10.

The designation *sink-draw tubes* is specified where close control over the outer diameter is required with normal tolerance applying to the wall thickness. Smoothness of the inside surface is not controlled, except that the flash is generally controlled to a height of 0.005 or 0.010 in maximum.

Mandrel-drawn tubes usually are normalized after welding by passing the tubes through a continuous atmosphere-controlled furnace. After descaling, the tubes are cold-drawn through a die with a mandrel on the inside of the tube. These tubes provide maximum control over surface finish, outside or inside diameters, and wall thickness. The normalizing heat treatment removes the effects of welding and provides a uniform microstructure around the tube circumference.

The different finish classifications may result in substantial differences in the mechanical properties of the steel material.

Typical examples for low-carbon steel material are given in Table 8.7.14. Differences in carbon content and other chemistry, heat treatment, etc., may significantly change these typical values.

Other Tubing Types Among other tube classifications are sanitary tubing usually made of 18% Cr-8% Ni stainless steel and available as seamless or welded tubing. This tubing is used extensively in the dairy, beverage, and food industries. Sanitary tubing is generally available in sizes from 1- to 4-in OD. It may be furnished either hot- or cold-finished. The tubes are normally annealed at temperatures above 1,900°F.

Some welded tube is also produced by fusion-welding methods utilizing either the inert-gas tungsten-arc-welding or gas-shielded consumable metal-arc-welding process. This tubing is generally more expensive than the resistance-welded types.

The butt-welded cold-finished tubes are made from hot-rolled or cold-rolled strip and fusion-welded. This tubing is usually furnished as sink-drawn or mandrel-drawn.

Butt-welded tubing is made in heavier wall thicknesses than the resistance-welded tube.

Several tubing materials used in the automobile industry are covered by specifications of the Society of Automotive Engineers, ''SAE Handbook.''

Pressure Tubing

Pressure-tube applications commonly involve external heat applications, as in boilers or superheaters.

Pressure tubing is produced to the actual outside diameter and minimum wall or average wall thickness specified by the purchaser. Pressure tubing may be hot- or cold-finished.

The wall thickness is normally given in decimal parts of an inch rather than as a fraction or gage number. When gage numbers are given without reference to a gage system, Birmingham wire gage (BWG) is implied.

Pressure tubing is usually made from steel produced by the open-hearth, basic oxygen, or electric-furnace processes.

* Although some specifications of rigid conduit pipe list lengths to 20 ft, the National Electric Code, 1965, limits lengths to 10 ft.

Table 8.7.3 Properties of Commercial Steel Pipe

Nominal pipe size, outside diameter, in	Schedule number*			Wall thick-ness, in	Inside diameter, in	Inside area, in²	Metal area, in²	Outside surface, ft²/ft	Inside surface, ft²/ft	Weight, lb/ft†	Weight of water, lb/ft	Moment of inertia, in⁴	Section modulus, in³	Radius of gyration, in
	a	b	c											
⅛			10S	0.049	0.307	0.0740	0.0548	0.106	0.0804	0.186	0.0321	0.00088	0.00437	0.1271
0.405	40	Std	40S	0.068	0.269	0.0568	0.0720	0.106	0.0705	0.245	0.0246	0.00106	0.00525	0.1215
	80	XS	80S	0.095	0.215	0.0364	0.0925	0.106	0.0563	0.315	0.0157	0.00122	0.00600	0.1146
¼			10S	0.065	0.410	0.1320	0.0970	0.141	0.1073	0.330	0.0572	0.00279	0.01032	0.1694
0.540	40	Std	40S	0.088	0.364	0.1041	0.1250	0.141	0.0955	0.425	0.0451	0.00331	0.01230	0.1628
	80	XS	80S	0.119	0.302	0.0716	0.1574	0.141	0.0794	0.535	0.0310	0.00378	0.01395	0.1547
⅜			5S	0.065	0.710	0.396	0.1582	0.220	0.1859	0.538	0.1716	0.01197	0.0285	0.2750
0.675			10S	0.065	0.545	0.2333	0.1246	0.177	0.1427	0.423	0.1011	0.00586	0.01737	0.2169
	40	Std	40S	0.091	0.493	0.1910	0.1670	0.177	0.1295	0.568	0.0827	0.00730	0.02160	0.2090
	80	XS	80S	0.126	0.423	0.1405	0.2173	0.177	0.1106	0.739	0.0609	0.00862	0.02554	0.1991
½			5S	0.065	0.710	0.3959	0.1583	0.220	0.1859	0.538	0.171	0.0120	0.0285	0.2750
0.840			10S	0.083	0.674	0.357	0.1974	0.220	0.1765	0.671	0.1547	0.01431	0.0341	0.2692
	40	Std	40S	0.109	0.622	0.304	0.2503	0.220	0.1628	0.851	0.1316	0.01710	0.0407	0.2613
	80	XS	80S	0.147	0.546	0.2340	0.320	0.220	0.1433	1.088	0.1013	0.02010	0.0478	0.2505
	160			0.187	0.466	0.1706	0.383	0.220	0.1220	1.304	0.0740	0.02213	0.0527	0.2402
		XXS		0.294	0.252	0.0499	0.504	0.220	0.0660	1.714	0.0216	0.02425	0.0577	0.2192
¾			5S	0.065	0.920	0.655	0.2011	0.275	0.2409	0.684	0.2882	0.02451	0.0467	0.349
1.050			10S	0.083	0.884	0.614	0.2521	0.275	0.2314	0.857	0.2661	0.02970	0.0566	0.343
	40	Std	40S	0.113	0.824	0.533	0.333	0.275	0.2157	1.131	0.2301	0.0370	0.0706	0.334
	80	XS	80S	0.154	0.742	0.432	0.435	0.275	0.1943	1.474	0.1875	0.0448	0.0853	0.321
	160			0.218	0.614	0.2961	0.570	0.275	0.1607	1.937	0.1284	0.0527	0.1004	0.304
		XXS		0.308	0.434	0.1479	0.718	0.275	0.1137	2.441	0.0641	0.0579	0.1104	0.2840
1			5S	0.065	1.185	1.103	0.2553	0.344	0.310	0.868	0.478	0.0500	0.0760	0.443
1.315			10S	0.109	1.097	0.945	0.413	0.344	0.2872	1.404	0.409	0.0757	0.1151	0.428
	40	Std	40S	0.133	1.049	0.864	0.494	0.344	0.2746	1.679	0.374	0.0874	0.1329	0.421
	80	XS	80S	0.179	0.957	0.719	0.639	0.344	0.2520	2.172	0.311	0.1056	0.1606	0.407
	160			0.250	0.815	0.522	0.836	0.344	0.2134	2.844	0.2261	0.1252	0.1903	0.387
		XXS		0.358	0.599	0.2818	1.076	0.344	0.1570	3.659	0.1221	0.1405	0.2137	0.361
1¼			5S	0.065	1.530	1.839	0.326	0.434	0.401	1.107	0.797	0.1038	0.1250	0.564
1.660			10S	0.109	1.442	1.633	0.531	0.434	0.378	1.805	0.707	0.1605	0.1934	0.550
	40	Std	40S	0.140	1.380	1.496	0.669	0.434	0.361	2.273	0.648	0.1948	0.2346	0.540
	80	XS	80S	0.191	1.278	1.283	0.881	0.434	0.335	2.997	0.555	0.2418	0.2913	0.524
	160			0.250	1.160	1.057	1.107	0.434	0.304	3.765	0.458	0.2839	0.342	0.506
		XXS		0.382	0.896	0.631	1.534	0.434	0.2346	5.214	0.2732	0.341	0.411	0.472
1½			5S	0.065	1.770	2.461	0.375	0.497	0.463	1.274	1.067	0.1580	0.1663	0.649
1.900			10S	0.109	1.682	2.222	0.613	0.497	0.440	2.085	0.962	0.2469	0.2599	0.634
	40	Std	40S	0.145	1.610	2.036	0.799	0.497	0.421	2.718	0.882	0.320	0.326	0.623
	80	XS	80S	0.200	1.500	1.767	1.068	0.497	0.393	3.631	0.765	0.391	0.412	0.605
	160			0.281	1.338	1.406	1.429	0.497	0.350	4.859	0.608	0.483	0.508	0.581
		XXS		0.400	1.100	0.950	1.855	0.497	0.288	6.408	0.412	0.568	0.598	0.549
				0.525	0.850	0.567	2.267	0.497	0.223	7.710	0.246	0.6140	0.6470	0.5200
				0.650	0.600	0.283	2.551	0.497	0.157	8.678	0.123	0.6340	0.6670	0.4980

2			5S	0.065	2.245	3.96	0.472	0.622	0.588	1.604	1.716	0.315	0.2652	0.817
2.375			10S	0.109	2.157	3.65	0.776	0.622	0.565	2.638	1.582	0.499	0.420	0.802
	40	Std	40S	0.154	2.067	3.36	1.075	0.622	0.541	3.653	1.455	0.666	0.561	0.787
	80	XS	80S	0.218	1.939	2.953	1.477	0.622	0.508	5.022	1.280	0.868	0.731	0.766
	160			0.343	1.689	2.240	2.190	0.622	0.442	7.444	0.971	1.163	0.979	0.729
		XXS		0.436	1.503	1.774	2.656	0.622	0.393	9.029	0.769	1.312	1.104	0.703
				0.562	1.251	1.229	3.199	0.622	0.328	10.882	0.533	1.442	1.2140	0.6710
				0.687	1.001	0.787	3.641	0.622	0.262	12.385	0.341	1.5130	1.2740	0.6440
2½			5S	0.083	2.709	5.76	0.728	0.753	0.709	2.475	2.499	0.710	0.494	0.988
2.875			10S	0.120	2.635	5.45	1.039	0.753	0.690	3.531	2.361	0.988	0.687	0.975
	40	Std	40S	0.203	2.469	4.79	1.704	0.753	0.646	5.793	2.076	1.530	1.064	0.947
	80	XS	80S	0.276	2.323	4.24	2.254	0.753	0.608	7.661	1.837	1.925	1.339	0.924
	160			0.375	2.125	3.55	2.945	0.753	0.556	10.01	1.535	2.353	1.637	0.894
		XXS		0.552	1.771	2.464	4.03	0.753	0.464	13.70	1.067	2.872	1.998	0.844
				0.675	1.525	1.826	4.663	0.753	0.399	15.860	0.792	3.0890	2.1490	0.8140
				0.800	1.275	1.276	5.212	0.753	0.334	17.729	0.554	3.2250	2.2430	0.7860
3			5S	0.083	3.334	8.73	0.891	0.916	0.873	3.03	3.78	1.301	0.744	1.208
3.500			10S	0.120	3.260	8.35	1.274	0.916	0.853	4.33	3.61	1.822	1.041	1.196
	40	Std	40S	0.216	3.068	7.39	2.228	0.916	0.803	7.58	3.20	3.02	1.724	1.164
	80	XS	80S	0.300	2.900	6.61	3.02	0.916	0.759	10.25	2.864	3.90	2.226	1.136
	160			0.437	2.626	5.42	4.21	0.916	0.687	14.32	2.348	5.03	2.876	1.094
		XXS		0.600	2.300	4.15	5.47	0.916	0.602	18.58	1.801	5.99	3.43	1.047
				0.725	2.050	3.299	6.317	0.916	0.537	21.487	1.431	6.5010	3.7150	1.0140
				0.850	1.800	2.543	7.073	0.916	0.471	24.057	1.103	6.8530	3.9160	0.9840
3½			5S	0.083	3.834	11.55	1.021	1.047	1.004	3.47	5.01	1.960	0.980	1.385
4.000			10S	0.120	3.760	11.10	1.463	1.047	0.984	4.97	4.81	2.756	1.378	1.372
	40	Std	40S	0.226	3.548	9.89	2.680	1.047	0.929	9.11	4.28	4.79	2.394	1.337
	80	XS	80S	0.318	3.364	8.89	3.68	1.047	0.881	12.51	3.85	6.28	3.14	1.307
		XXS		0.636	2.728	5.845	6.721	1.047	0.716	22.850	2.530	9.8480	4.9240	1.2100
4			5S	0.083	4.334	14.75	1.152	1.178	1.135	3.92	6.40	2.811	1.249	1.562
4.500			10S	0.120	4.260	14.25	1.651	1.178	1.115	5.61	6.17	3.96	1.762	1.549
				0.188	4.124	13.357	2.547	1.178	1.082	8.560	5.800	5.8500	2.6000	1.5250
	40	Std	40S	0.237	4.026	12.73	3.17	1.178	1.054	10.79	5.51	7.23	3.21	1.510
	80	XS	80S	0.337	3.826	11.50	4.41	1.178	1.002	14.98	4.98	9.61	4.27	1.477
	120			0.437	3.626	10.33	5.58	1.178	0.949	18.96	4.48	11.65	5.18	1.445
				0.500	3.500	9.621	6.283	1.178	0.916	21.360	4.160	12.7710	5.6760	1.4250
	160			0.531	3.438	9.28	6.62	1.178	0.900	22.51	4.02	13.27	5.90	1.416
		XXS		0.674	3.152	7.80	8.10	1.178	0.825	27.54	3.38	15.29	6.79	1.374
				0.800	2.900	6.602	9.294	1.178	0.759	31.613	2.864	16.6610	7.4050	1.3380
				0.925	2.650	5.513	10.384	1.178	0.694	35.318	2.391	17.7130	7.8720	1.3060
5			5S	0.109	5.345	22.44	1.868	1.456	1.399	6.35	9.73	6.95	2.498	1.929
5.563			10S	0.134	5.295	22.02	2.285	1.456	1.386	7.77	9.53	8.43	3.03	1.920
	40	Std	40S	0.258	5.047	20.01	4.30	1.456	1.321	14.62	8.66	15.17	5.45	1.878
	80	XS	80S	0.375	4.813	18.19	6.11	1.456	1.260	20.78	7.89	20.68	7.43	1.839
	120			0.500	4.563	16.35	7.95	1.456	1.195	27.04	7.09	25.74	9.25	1.799
	160			0.625	4.313	14.61	9.70	1.456	1.129	32.96	6.33	30.0	10.80	1.760
		XXS		0.750	4.063	12.97	11.34	1.456	1.064	38.55	5.62	33.6	12.10	1.722
				0.875	3.813	11.413	12.880	1.456	0.998	43.810	4.951	36.6450	13.1750	1.6860
				1.000	3.563	9.966	14.328	1.456	0.933	47.734	4.232	39.1110	14.0610	1.6520

NOTE: See footnotes at end of table.

Table 8.7.3 Properties of Commercial Steel Pipe (*Continued*)

Nominal pipe size, outside diameter, in	Schedule number* a	b	c	Wall thick-ness, in	Inside diameter, in	Inside area, in²	Metal area, in²	Outside surface, ft²/ft	Inside surface, ft²/ft	Weight, lb/ft†	Weight of water, lb/ft	Moment of inertia, in⁴	Section modulus, in³	Radius of gyration, in
6			5S	0.109	6.407	32.2	2.231	1.734	1.677	5.37	13.98	11.85	3.58	2.304
6.625			10S	0.134	6.357	31.7	2.733	1.734	1.664	9.29	13.74	14.40	4.35	2.295
				0.219	6.187	30.100	4.410	1.734	1.620	15.020	13.100	22.6600	6.8400	2.2700
	40	Std	40S	0.280	6.065	28.89	5.58	1.734	1.588	18.97	12.51	28.14	8.50	2.245
	80	XS	80S	0.432	5.761	26.07	8.40	1.734	1.508	28.57	11.29	40.5	12.23	2.195
	120			0.562	5.501	23.77	10.70	1.734	1.440	36.39	10.30	49.6	14.98	2.153
	160			0.718	5.189	21.15	13.33	1.734	1.358	45.30	9.16	59.0	17.81	2.104
		XXS		0.864	4.897	18.83	15.64	1.734	1.282	53.16	8.17	66.3	20.03	2.060
				1.000	4.625	16.792	17.662	1.734	1.211	60.076	7.284	72.1190	21.7720	2.0200
				1.125	4.375	15.025	19.429	1.734	1.145	66.084	6.517	76.5970	23.1240	1.9850
8			5S	0.109	8.407	55.5	2.916	2.258	2.201	9.91	24.07	26.45	6.13	3.01
8.625			10S	0.148	8.329	54.5	3.94	2.258	2.180	13.40	23.59	35.4	8.21	3.00
				0.219	8.187	52.630	5.800	2.258	2.150	19.640	22.900	51.3200	11.9000	2.9700
	20			0.250	8.125	51.8	6.58	2.258	2.127	22.36	22.48	57.7	13.39	2.962
	30			0.277	8.071	51.2	7.26	2.258	2.113	24.70	22.18	63.4	14.69	2.953
	40	Std	40S	0.322	7.981	50.0	8.40	2.258	2.089	28.55	21.69	72.5	16.81	2.938
	60			0.406	7.813	47.9	10.48	2.258	2.045	35.64	20.79	88.8	20.58	2.909
	80	XS	80S	0.500	7.625	45.7	12.76	2.258	1.996	43.39	19.80	105.7	24.52	2.878
	100			0.593	7.439	43.5	14.96	2.258	1.948	50.87	18.84	121.4	28.14	2.847
	120			0.718	7.189	40.6	17.84	2.258	1.882	60.63	17.60	140.6	32.6	2.807
	140			0.812	7.001	38.5	19.93	2.258	1.833	67.76	16.69	153.8	35.7	2.777
		XXS		0.875	6.875	37.1	21.30	2.258	1.800	72.42	16.09	162.0	37.6	2.757
	160			0.906	6.813	36.5	21.97	2.258	1.784	74.69	15.80	165.9	38.5	2.748
				1.000	6.625	34.454	23.942	2.258	1.734	81.437	14.945	177.1320	41.0740	2.7190
				1.125	6.375	31.903	26.494	2.258	1.669	90.114	13.838	190.6210	44.2020	2.6810
10			5S	0.134	10.482	86.3	4.52	2.815	2.744	15.15	37.4	63.7	11.85	3.75
10.750			10S	0.165	10.420	85.3	5.49	2.815	2.728	18.70	36.9	76.9	14.30	3.74
				0.219	10.312	83.52	7.24	2.815	2.70	24.63	36.2	100.46	18.69	3.72
	20			0.250	10.250	82.5	8.26	2.815	2.683	28.04	35.8	113.7	21.16	3.71
	30			0.307	10.136	80.7	10.07	2.815	2.654	34.24	35.0	137.5	25.57	3.69
	40	Std	40S	0.365	10.020	78.9	11.91	2.185	2.623	40.48	34.1	160.8	29.90	3.67
	60	XS	80S	0.500	9.750	74.7	16.10	2.815	2.553	54.74	32.3	212.0	39.4	3.63
	80			0.593	9.564	71.8	18.92	2.815	2.504	64.33	31.1	244.9	45.6	3.60
	100			0.718	9.314	68.1	22.63	2.815	2.438	76.93	29.5	286.2	53.2	3.56
	120			0.843	0.064	64.5	26.24	2.815	2.373	89.20	28.0	324	60.3	3.52
				0.875	9.000	63.62	27.14	2/815	2.36	92.28	27.6	333.46	62.04	3.50
	140			1.000	8.750	60.1	30.6	2.815	2.091	104.13	26.1	368	68.4	3.47
	160			1.125	8.500	56.7	34.0	2.815	2.225	115.65	24.6	399	74.3	3.43
				1.250	8.250	53.45	37.31	2.815	2.16	126.832	23.2	428.17	79.66	3.39
				1.500	7.750	47.15	43.57	2.815	2.03	148.19	20.5	478.59	89.04	3.31

12			5S	0.156	12.438	121.4	6.17	3.34	3.26	20.99	52.7	122.2	19.20	4.45
12.750			10S	0.180	12.390	120.6	7.11	3.34	3.24	24.20	52.2	140.5	22.03	4.44
	20			0.250	12.250	117.9	9.84	3.34	3.21	33.38	51.1	191.9	30.1	4.42
	30			0.330	12.090	114.8	12.88	3.34	3.17	43.77	49.7	248.5	39.0	4.39
		Std	40S	0.375	12.000	113.1	14.58	3.34	3.14	49.56	49.0	279.3	43.8	4.38
	40			0.406	11.938	111.9	15.74	3.34	3.13	53.53	48.5	300	47.1	4.37
		XS	80S	0.500	11.750	108.4	19.24	3.34	3.08	65.42	47.0	362	56.7	4.33
	60			0.562	11.626	106.2	21.52	3.34	3.04	73.16	46.0	401	62.8	4.31
	80			0.687	11.376	101.6	26.04	3.34	2.978	88.51	44.0	475	74.5	4.27
				0.750	11.250	99.40	28.27	3.34	2.94	96.2	43.1	510.7	80.1	4.25
	100			0.843	11.064	96.1	31.5	3.34	2.897	107.20	41.6	562	88.1	4.22
				0.875	11.000	95.00	32.64	3.34	2.88	110.9	41.1	578.5	90.7	4.21
	120			1.000	10.750	90.8	36.9	3.34	2.814	125.49	39.3	642	100.7	4.17
	140			1.125	10.500	86.6	41.1	3.34	2.749	139.68	37.5	701	109.9	4.13
				1.250	10.250	82.50	45.16	3.34	2.68	153.6	35.8	755.5	118.5	4.09
	160			1.312	10.126	80.5	47.1	3.34	2.651	160.27	34.9	781	122.6	4.07
14			5S	0.156	13.688	147.20	6.78	3.67	3.58	23.0	63.7	162.6	23.2	4.90
14.000			10S	0.188	13.624	145.80	8.16	3.67	3.57	27.7	63.1	194.6	27.8	4.88
				0.210	13.580	144.80	9.10	3.67	3.55	30.9	62.8	216.2	30.9	4.87
				0.219	13.562	144.50	9.48	3.67	3.55	32.2	62.6	225.1	32.2	4.87
	10			0.250	13.500	143.1	10.80	3.67	3.53	36.71	62.1	255.4	36.5	4.86
				0.281	13.438	141.80	12.11	3.67	3.52	41.2	61.5	285.2	40.7	4.85
	20			0.312	13.376	140.5	13.42	3.67	3.50	45.68	60.9	314	44.9	4.84
				0.344	13.312	139.20	14.76	3.67	3.48	50.2	60.3	344.3	49.2	4.83
	30	Std		0.375	13.250	137.9	16.05	3.67	3.47	54.57	59.7	373	53.3	4.82
	40			0.437	13.126	135.3	18.62	3.67	3.44	63.37	58.7	429	61.2	4.80
				0.469	13.062	134.00	19.94	3.67	3.42	67.8	58.0	456.8	64.3	4.79
		XS		0.500	13.000	132.7	21.21	3.67	3.40	72.09	57.5	484	69.1	4.78
	60			0.593	12.814	129.0	24.98	3.67	3.35	84.91	55.9	562	80.3	4.74
		XXS		0.625	12.750	127.7	26.26	3.67	3.34	89.28	55.3	589	84.1	4.73
	80			0.750	12.500	122.7	31.2	3.67	3.27	106.13	53.2	687	98.2	4.69
	100			0.937	12.126	115.5	38.5	3.67	3.17	130.73	50.0	825	117.8	4.63
	120			1.093	11.814	109.6	44.3	3.67	3.09	150.67	47.5	930	132.8	4.58
	140			1.250	11.500	103.9	50.1	3.67	3.01	170.22	45.0	1017	146.8	4.53
	160			1.406	11.188	98.3	55.6	3.67	2.929	189.12	42.6	1127	159.6	4.48
16			5S	0.165	15.670	192.90	8.21	4.19	4.10	28	83.5	257	32.2	5.60
16.000			10S	0.188	15.624	191.70	9.34	4.19	4.09	32	83.0	292	36.5	5.59
	10			0.250	15.500	188.7	12.37	4.19	4.06	42.05	81.8	384	48.0	5.57
	20			0.312	15.376	185.7	15.38	4.19	4.03	52.36	80.5	473	59.2	5.55
	30	Std		0.375	15.250	182.6	18.41	4.19	3.99	62.58	79.1	562	70.3	5.53
	40	XS		0.500	15.000	176.7	24.35	4.19	3.93	82.77	76.5	732	91.5	5.48
	60			0.656	14.688	169.4	31.6	4.19	3.85	107.50	73.4	933	116.6	5.43
	80			0.843	14.314	160.9	40.1	4.19	3.75	136.46	69.7	1157	144.6	5.37
	100			1.031	13.938	152.6	48.5	4.19	3.65	164.83	66.1	1365	170.6	5.30
	120			1.218	13.564	144.5	56.6	4.19	3.55	192.29	62.6	1556	194.5	5.24
	140			1.437	13.126	135.3	65.7	4.19	3.44	223.64	58.6	1760	220.0	5.17
	160			1.593	12.814	129.0	72.1	4.19	3.35	245.11	55.9	1894	236.7	5.12

Table 8.7.3 Properties of Commercial Steel Pipe (*Continued*)

Nominal pipe size, outside diameter, in	Schedule number* a	Schedule number* b	Schedule number* c	Wall thick-ness, in	Inside diameter, in	Inside area, in²	Metal area, in²	Outside surface, ft²/ft	Inside surface, ft²/ft	Weight, lb/ft†	Weight of water, lb/ft	Moment of inertia, in⁴	Section modulus, in³	Radius of gyration, in
18			5S	0.165	17.670	245.20	9.24	4.71	4.63	31	106.2	368	40.8	6.31
18.000			10S	0.188	17.624	243.90	10.52	4.71	4.61	36	105.7	417	46.4	6.30
	10			0.250	17.500	240.5	13.94	4.71	4.58	47.39	104.3	549	61.0	6.28
	20			0.312	17.376	237.1	17.34	4.71	4.55	59.03	102.8	678	75.5	6.25
		Std		0.375	17.250	233.7	20.76	4.71	4.52	70.59	101.2	807	89.6	6.23
	30			0.437	17.126	230.4	24.11	4.71	4.48	82.06	99.9	931	103.4	6.21
		XS		0.500	17.00	227.0	27.49	4.71	4.45	93.45	98.4	1053	117.0	6.19
	40			0.562	16.876	223.7	30.8	4.71	4.42	104.75	97.0	1172	130.2	6.17
	60			0.750	16.500	213.8	40.6	4.71	4.32	138.17	92.7	1515	168.3	6.10
	80			0.937	16.126	204.2	50.2	4.71	4.22	170.75	88.5	1834	203.8	6.04
	100			1.156	15.688	193.3	61.2	4.71	4.11	207.96	83.7	2180	242.2	5.97
	120			1.375	15.250	182.6	71.8	4.71	3.99	244.14	79.2	2499	277.6	5.90
	140			1.562	14.876	173.8	80.7	4.71	3.89	274.23	75.3	2750	306	5.84
	160			1.781	14.438	163.7	90.7	4.71	3.78	308.51	71.0	3020	336	5.77
20			5S	0.188	19.634	302.40	11.70	5.24	5.14	40	131.0	574	57.4	7.00
20.000			10S	0.218	19.564	300.60	13.55	5.24	5.12	46	130.2	663	66.3	6.99
	10			0.250	19.500	298.6	15.51	5.24	5.11	52.73	129.5	757	75.7	6.98
	20	Std		0.375	19.250	291.0	23.12	5.24	5.04	78.60	126.0	1114	111.4	6.94
	30	XS		0.500	19.000	283.5	30.6	5.24	4.97	104.13	122.8	1457	145.7	6.90
	40			0.593	18.814	278.0	36.2	5.24	4.93	122.91	120.4	1704	170.4	6.86
	60			0.812	18.376	265.2	48.9	5.24	4.81	166.40	115.0	2257	225.7	6.79
				0.875	18.250	261.6	52.6	5.24	4.78	178.73	113.4	2409	240.9	6.77
	80			1.031	17.938	252.7	61.4	5.24	4.70	208.87	109.4	2772	277.2	6.72
	100			1.281	17.438	238.8	75.3	5.24	4.57	256.10	103.4	3320	332	6.63
	120			1.500	17.00	227.0	87.2	5.24	4.45	296.37	98.3	3760	376	6.56
	140			1.750	16.500	213.8	100.3	5.24	4.32	341	92.6	4200	422	6.48
	160			1.968	16.064	202.7	111.5	5.24	4.21	379.01	87.9	1490	459	6.41
22			5S	0.188	21.624	367.3	12.88	5.76	5.66	44	159.1	766	69.7	7.71
22.000			10S	0.218	21.564	365.2	14.92	5.76	5.65	51	158.2	885	80.4	7.70
	10			0.250	21.500	363.1	17.18	5.76	5.63	58	157.4	1010	91.8	7.69
	20	Std		0.375	21.250	354.7	25.48	5.76	5.56	87	153.7	1490	135.4	7.65
	30	XS		0.500	21.000	364.4	33.77	5.76	5.50	115	150.2	1953	177.5	7.61
				0.625	20.750	338.2	41.97	5.76	5.43	143	146.6	2400	218.2	7.56
				0.750	20.500	330.1	50.07	5.76	5.37	170	143.1	2829	257.2	7.52
	60			0.875	20.250	322.1	58.07	5.76	5.30	197	139.6	3245	295.0	7.47
	80			1.125	19.750	306.4	73.78	5.76	5.17	251	132.8	4029	366.3	7.39
	100			1.375	19.250	291.0	89.09	5.76	5.04	303	126.2	4758	432.6	7.31
	120			1.625	18.750	276.1	104.02	5.76	4.91	354	119.6	5432	493.8	7.23
	140			1.875	18.250	261.6	118.55	5.76	4.78	403	113.3	6054	550.3	7.15
	160			2.125	17.750	247.4	132.68	5.76	4.65	451	107.2	6626	602.4	7.07

24			5S	0.218	23.564	436.1	16.29	6.28	6.17	55	188.9	1152	96.0	8.41
24.000	10			0.250	23.500	434	18.65	6.28	6.15	63.41	188.0	1316	109.6	8.40
	20	Std		0.375	23.250	425	27.83	6.28	6.09	94.62	183.8	1943	161.9	8.35
		XS		0.500	23.000	415	36.9	6.28	6.02	125.49	180.1	2550	212.5	8.31
	30			0.562	22.876	411	41.4	6.28	5.99	140.80	178.1	2840	237.0	8.29
				0.625	22.750	406	45.9	6.28	5.96	156.03	176.2	3140	261.4	8.27
	40			0.687	22.626	402	50.3	6.28	5.92	171.17	174.3	3420	285.2	8.25
				0.750	22.500	398	54.8	6.28	5.89	186.24	172.4	3710	309	8.22
				0.875	22.250	388.6	63.54	6.28	5.83	216	168.6	4256	354.7	8.18
	60			0.968	22.064	382	70.0	6.28	5.78	238.11	165.8	4650	388	8.15
	80			1.218	21.564	365	87.2	6.28	5.65	296.36	158.3	5670	473	8.07
	100			1.531	20.938	344	108.1	6.28	5.48	367.40	149.3	6850	571	7.96
	120			1.812	20.376	326	126.3	6.28	5.33	429.39	141.4	7830	652	7.87
	140			2.062	19.876	310	142.1	6.28	5.20	483.13	134.5	8630	719	7.79
	160			2.343	19.314	293	159.4	6.28	5.06	541.94	127.0	9460	788	7.70
26				0.250	25.500	510.7	19.85	6.81	6.68	67	221.4	1646	126.6	9.10
26.000	10			0.312	25.376	505.8	25.18	6.81	6.64	86	219.2	2076	159.7	9.08
		Std		0.375	25.250	500.7	30.19	6.81	6.61	103	217.1	2478	190.6	9.06
	20	XS		0.500	25.000	490.9	40.06	6.81	6.54	136	212.8	3259	250.7	9.02
				0.625	24.750	481.1	49.82	6.81	6.48	169	208.6	4013	308.7	8.98
				0.750	24.500	471.4	59.49	6.81	6.41	202	204.4	4744	364.9	8.93
				0.875	24.250	461.9	69.07	6.81	6.35	235	200.2	5458	419.9	8.89
				1.000	24.000	452.4	78.54	6.81	6.28	267	196.1	6149	473.0	8.85
				1.125	23.750	443.0	87.91	6.81	6.22	299	192.1	6813	524.1	8.80
28				0.250	27.500	594.0	21.80	7.33	7.20	74	257.3	2098	149.8	9.81
28.000	10			0.312	27.376	588.6	27.14	7.33	7.17	92	255.0	2601	185.8	9.79
		Std		0.375	27.250	583.2	32.54	7.33	7.13	111	252.6	3105	221.8	9.77
	20	XS		0.500	27.000	572.6	43.20	7.33	7.07	147	248.0	4085	291.8	9.72
	30			0.625	26.750	562.0	53.75	7.33	7.00	183	243.4	5038	359.8	9.68
				0.750	26.500	551.6	64.21	7.33	6.94	218	238.9	5964	426.0	9.64
				0.875	26.250	541.2	74.56	7.33	6.87	253	234.4	6865	490.3	9.60
				1.000	26.000	530.9	84.82	7.33	6.81	288	230.0	7740	552.8	9.55
				1.125	25.750	520.8	94.98	7.33	6.74	323	225.6	8590	613.6	9.51
30			5S	0.250	29.500	683.4	23.37	7.85	7.72	79	296.3	2585	172.3	10.52
30.000	10		10S	0.312	29.376	677.8	29.19	7.85	7.69	99	293.7	3201	213.4	10.50
		Std		0.375	29.250	672.0	34.90	7.85	7.66	119	291.2	3823	254.8	10.48
	20	XS		0.500	29.000	660.5	46.34	7.85	7.59	158	286.2	5033	335.5	10.43
	30			0.625	28.750	649.2	57.68	7.85	7.53	196	281.3	6213	414.2	10.39
	40			0.750	28.500	637.9	68.92	7.85	7.46	234	276.6	7371	491.4	10.34
				0.875	28.250	620.7	80.06	7.85	7.39	272	271.8	8494	466.2	40.30
				1.000	28.000	615.7	91.11	7.85	7.33	310	267.0	9591	639.4	10.26
				1.125	27.750	604.7	102.05	7.85	7.26	347	262.2	10653	710.2	10.22
32				0.250	31.500	779.2	24.93	8.38	8.25	85	337.8	3141	196.3	11.22
32.000	10			0.312	31.376	773.2	31.02	8.38	8.21	106	335.2	3891	243.2	11.20
		Std		0.375	31.250	766.9	37.25	8.38	8.18	127	332.5	4656	291.0	11.18
	20	XS		0.500	31.000	754.7	49.48	8.38	8.11	168	327.2	6140	383.8	11.14
	30			0.625	30.750	742.5	61.59	8.38	8.05	209	321.9	7578	473.6	11.09
	40			0.688	30.624	736.6	67.68	8.38	8.02	230	319.0	8298	518.6	11.07
				0.750	30.500	730.5	73.63	8.38	7.98	250	316.7	8990	561.9	11.05
				0.875	30.250	718.3	85.52	8.38	7.92	291	311.6	10372	648.2	11.01
				1.000	30.000	706.8	97.38	8.38	7.85	331	306.4	11680	730.0	10.95
				1.125	29.750	694.7	109.0	8.38	7.79	371	301.3	13023	814.0	10.92

NOTE: See footnotes at end of table.

Table 8.7.3 Properties of Commercial Steel Pipe (*Continued*)

Nominal pipe size, outside diameter, in	Schedule number* a	Schedule number* b	Schedule number* c	Wall thick-ness, in	Inside diameter, in	Inside area, in^2	Metal area, in^2	Outside surface, ft^2/ft	Inside surface, ft^2/ft	Weight, lb/ft†	Weight of water, lb/ft	Moment of inertia, in^4	Section modulus, in^3	Radius of gyration, in
34				0.250	33.500	881.2	26.50	8.90	8.77	90	382.0	3773	221.9	11.93
34.000	10			0.312	33.376	874.9	32.99	8.90	8.74	112	379.3	4680	275.3	11.91
		Std		0.375	33.250	867.8	39.61	8.90	8.70	135	376.2	5597	329.2	11.89
	20	XS		0.500	33.000	855.3	52.62	8.90	8.64	179	370.8	7385	434.4	11.85
	30			0.625	32.750	841.9	65.53	8.90	8.57	223	365.0	9124	536.7	11.80
	40			0.688	32.624	835.9	72.00	8.90	8.54	245	362.1	9992	587.8	11.78
				0.750	32.500	829.3	78.34	8.90	8.51	266	359.5	10829	637.0	11.76
				0.875	32.250	816.4	91.01	8.90	8.44	310	354.1	12501	735.4	11.72
				1.000	32.000	804.2	103.67	8.90	8.38	353	348.6	14114	830.2	11.67
				1.125	31.750	791.3	116.13	8.90	8.31	395	343.2	15719	924.7	11.63
36				0.250	35.500	989.7	28.11	9.42	9.29	96	429.1	4491	249.5	12.64
36.000	10			0.312	35.376	982.2	34.95	9.42	9.26	119	426.1	5565	309.1	12.62
		Std		0.375	35.250	975.8	42.01	9.42	9.23	143	423.1	6664	370.2	12.59
	20	XS		0.500	35.000	962.1	55.76	9.42	9.16	190	417.1	8785	488.1	12.55
	30			0.625	34.750	948.3	69.50	9.42	9.10	236	411.1	10872	604.0	12.51
	40			0.750	34.500	934.7	83.01	9.42	9.03	282	405.3	12898	716.5	12.46
				0.875	34.250	920.6	96.50	9.42	8.97	328	399.4	14903	827.9	12.42
				1.000	34.000	907.9	109.96	9.42	8.90	374	393.6	16851	936.2	12.38
				1.125	33.750	894.2	123.19	9.42	8.89	419	387.9	18763	1042.4	12.34
42				0.250	41.500	1352.6	32.82	10.99	10.86	112	586.4	7126	339.3	14.73
42.000		Std		0.375	41.250	1336.3	49.08	10.99	10.80	167	579.3	10627	506.1	14.71
	20	XS		0.500	41.000	1320.2	65.18	10.99	10.73	222	572.3	14037	668.4	14.67
	30			0.625	40.750	1304.1	81.28	10.99	10.67	276	565.4	17373	827.3	14.62
	40			0.750	40.500	1288.2	97.23	10.99	10.60	330	558.4	20689	985.2	14.59
				1.000	40.000	1256.6	128.81	10.99	10.47	438	544.8	27080	1289.5	14.50
				1.250	39.500	1225.3	160.03	10.99	10.34	544	531.2	33233	1582.5	14.41
				1.500	39.000	1194.5	190.85	10.99	10.21	649	517.9	39181	1865.7	14.33

NOTE: The following formulas are used in the computation of the values shown in the table:

Weight of pipe per foot (pounds) $= 10.6802t(D - t)$

Weight of water per foot (pounds) $= 0.3405d^2$

Square feet outside surface per foot $= 0.2618D$

Square feet inside surface per foot $= 0.2618d$

Inside area (square inches) $= 0.785d^2$

Area of metal (square inches) $= 0.785(D^2 - d^2)$

Moment of inertia (inches⁴) $= 0.0491(D^4 - d^4) = A_m R_g^2$

Section modulus (inches³) $= \dfrac{0.0982(D^4 - d^4)}{D}$

Radius of gyration (inches) $= 0.25\sqrt{D^2 + d^2}$

where A_m = area of metal, in^2; d = inside diameter, in; D = outside diameter, in; R_g = radius of gyration, in; t = pipe wall thickness, in.

* Schedule numbers: Standard-weight pipe and schedule 40 are the same in all sizes through 10 in; from 12 in through 24 in, standard weight pipe has a wall thickness of ⅜ in. Extra-strong-weight pipe and schedule 80 are the same in all sizes through 8 in; from 8 in through 24 in, extra-strong-weight pipe has a wall thickness of ½ in. Double-extra-strong-weight pipe has no corresponding schedule number.

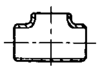
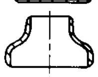
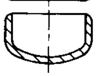
a: ANSI B36.10 steel pipe schedule numbers

b: ANSI B36.10 steel pipe nominal wall thickness designation

c: ANSI B36.19 stainless steel pipe schedule numbers

† The ferritic steels may be about 5% less and the austenitic stainless steels about 2% greater than the values shown in this table, which are based on weights for carbon steel.

Table 8.7.4 Weight of Standard Pipe Fittings and Materials, 3-in Size (3.500-in OD)

		Schedule no.	40	80	160	
		Wall designation	Std	XS		XXS
		Thickness, in	0.216	0.300	0.438	0.600
		Pipe, lb/ft	7.58	10.25	14.32	18.58
		Water, lb/ft	3.20	2.86	2.35	1.80
Welding fittings		L.R. 90° elbow	**4.6**	**6.1**	**8.4**	**10.7**
			0.8	0.8	0.8	0.8
		S.R. 90° elbow	**3**	**4**		
			0.5	0.5		
		L.R. 45° elbow	**2.4**	**3.2**	**4.4**	**5.4**
			0.3	0.3	0.3	0.3
		Tee	**7.4**	**9.5**	**12.2**	**14.8**
			0.8	0.8	0.8	0.8
		Lateral	**13**	**19**		
			1.8	1.8		
		Reducer	**2.2**	**2.9**	**3.7**	**4.7**
			0.3	0.3	0.3	0.3
		Cap	**1.4**	**1.8**	**3.5**	**3.7**
			0.5	0.5	0.5	0.5

	Temperature range, °F		100–199	200–299	300–399	400–499	500–599	600–699	700–799	800–899	900–999	1,000–1,099	1,100–1,200
Insulation	85% magnesia calcium silicate	Nom. thick., in	1	1	1½	2	2	2½	3	3	3	3½	3½
		lb/ft	1.25	1.25	2.08	3.01	3.01	4.07	5.24	5.24	5.24	6.65	6.65
	Combination	Nom. thick., in						2½	3	3	3	3½	3½
		lb/ft						5.07	6.94	6.94	6.94	9.17	9.17
	*Asbestos fiber-sodium silicate	Nom. thick., in	1	1	1	1½	1½	2	2	3	3	3½	3½
		lb/ft	1.61	1.61	1.61	2.74	2.74	3.98	3.98	6.99	6.99	8.99	8.99

Table 8.7.4 Weight of Standard Pipe Fittings and Materials, 3-in Size (3.500-in OD) (*Continued*)

		Pressure rating, lb/in²								
		Cast iron		Steel						
		125	250	150	300	400	600	900	1,500	2,500
Flanges	Screwed or slip-on	**9** 1.5	**17** 1.5	**9** 1.5	**17** 1.5	**20** 1.5	**20** 1.5	**37** 1.5	**61** 1.5	**102** 1.5
	Welding neck			**11** 1.5	**19** 1.5	**27** 1.5	**27** 1.5	**38** 1.5	**61** 1.5	**113** 1.5
	Lap joint			**9** 1.5	**17** 1.5	**19** 1.5	**19** 1.5	**36** 1.5	**60** 1.5	**99** 1.5
	Blind	**10** 1.5	**19** 1.5	**10** 1.5	**20** 1.5	**24** 1.5	**24** 1.5	**38** 1.5	**61** 1.5	**105** 1.5
Flanged fittings	S.R. 90° elbow	**26** 3.9	**46** 4	**32** 3.9	**53** 4		**67** 4.1	**98** 4.3	**150** 4.6	
	L.R. 90° elbow	**30** 4.3	**50** 4.3	**40** 4.3	**63** 4.3					
	45° elbow	**22** 3.5	**41** 3.6	**28** 3.5	**46** 3.6		**60** 3.8	**93** 3.9	**135** 4	
	Tee	**39** 5.9	**67** 6	**52** 5.9	**81** 6		**102** 6.2	**151** 6.5	**238** 6.9	
Valves	Flanged bonnet gate	**66** 7	**112** 7.4	**70** 4	**125** 4.4		**155** 4.8	**260** 5	**410** 5.5	
	Flanged bonnet globe or angle	**56** 7.2	**121** 7.6	**60** 4.3	**95** 4.5		**155** 4.8	**225** 5	**495** 5.5	
	Flanged bonnet check	**46** 7.2	**100** 7.6	**60** 4.3	**70** 4.4		**120** 4.8	**150** 4.9	**440** 5.8	
	Pressure seal bonnet–gate							**208** 3	**235** 3.2	
	Pressure seal bonnet–globe							**135** 2.5	**180** 3	

NOTES: Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.
Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lb/ft³. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lb/ft³ and the outer layer at 11 lb/ft³.
Insulation weights include allowances for wire, cement, canvas, bands and paint, but no special surface finishes.
To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.
Valve weights are approximate. When possible, obtain weights from the manufacturer.
Cast-iron valve weights are for flanged end valves; steel weights for welding end valves.
All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.
* 16 lb/ft³ density. (Existing installations only.)

Table 8.7.5 Weights of Standard Pipe Fittings and Materials, 4-in Size (4.500-in OD)

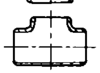
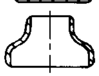
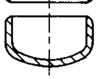
		Schedule no.	40	80	120	160							
		Wall designation	Std	XS			XXS						
		Thickness, in	0.237	0.337	0.438	0.531	0.674						
		Pipe, lb/ft	10.79	14.98	18.96	22.51	27.54						
		Water, lb/ft	5.51	4.98	4.48	4.02	3.38						
Welding fittings		L.R. 90° elbow	**8.7** 1	**11.9** 1		**17.6** 1	**21** 1						
Welding fittings		S.R. 90° elbow	**5.8** 0.7	**7.9** 0.7									
Welding fittings		L.R. 45° elbow	**4.3** 0.4	**5.9** 0.4		**8.5** 0.4	**10.1** 0.4						
Welding fittings		Tee	**12.6** 1	**16.4** 1		**23** 1	**27** 1						
Welding fittings		Lateral	**21** 2.1	**33** 2.1									
Welding fittings		Reducer	**3.6** 0.3	**4.9** .3		**6.6** .3	**8.2** 0.3						
Welding fittings		Cap	**2.6** 0.6	**3.4** 0.6		**6.5** 0.6	**6.7** 0.6						
	Temperature range, °F		100–199	200–299	300–399	400–499	500–599	600–699	700–799	800–899	900–999	1,000–1,099	1,100–1,200
Insulation	85% magnesia calcium silicate	Nom. thick., in	1	1	1½	2	2½	2½	3	3	3½	3½	4
		lb/ft	1.62	1.62	2.55	3.61	4.66	4.66	6.07	6.07	7.48	7.48	9.10
Insulation	Combination	Nom. thick., in						2½	3	3	3½	3½	3½
		lb/ft						6.07	8.30	8.30	10.6	10.6	10.6
Insulation	Asbestos fiber-sodium silicate	Nom. thick., in	1	1	1	1½	1½	2	2	3	3	3½	3½
		lb/ft	2.04	2.04	2.04	3.28	3.28	4.70	4.70	8.29	8.29	10.25	10.25

Table 8.7.5 Weights of Standard Pipe Fittings and Materials, 4-in Size (4.500-in OD) (*Continued*)

		Pressure rating, lb/in²								
		Cast iron		Steel						
		125	250	150	300	400	600	900	1,500	2,500
Flanges	Screwed or slip-on	**16** 1.5	**26** 1.5	**15** 1.5	**26** 1.5	**32** 1.5	**43** 1.5	**66** 1.5	**90** 1.5	**158** 1.5
	Welding neck			**17** 1.5	**29** 1.5	**41** 1.5	**48** 1.5	**64** 1.5	**90** 1.5	**177** 1.5
	Lap joint			**15** 1.5	**26** 1.5	**31** 1.5	**42** 1.5	**64** 1.5	**92** 1.5	**153** 1.5
	Blind	**18** 1.5	**29** 1.5	**19** 1.5	**31** 1.5	**39** 1.5	**47** 1.5	**67** 1.5	**90** 1.5	**164** 1.5
Flanged fittings	S.R. 90° elbow	**45** 4.1	**72** 4.2	**59** 4.1	**85** 4.2	**99** 4.3	**128** 4.4	**185** 4.5	**254** 4.8	
	L.R. 90° elbow	**52** 4.5	**79** 4.5	**72** 4.5	**98** 4.5					
	45° elbow	**40** 3.7	**65** 3.8	**51** 3.7	**78** 3.8	**82** 3.9	**119** 4	**170** 4.1	**214** 4.2	
	Tee	**70** 6.1	**109** 6.2	**86** 6.1	**121** 6.3	**153** 6.4	**187** 6.6	**262** 6.8	**386** 7.2	
Valves	Flanged bonnet gate	**109** 7.2	**188** 7.5	**100** 4.2	**175** 4.5	**195** 5	**255** 5.1	**455** 5.4	**735** 6	
	Flanged bonnet globe or angle	**97** 7.4	**177** 7.8	**95** 4.3	**145** 4.8	**215** 5	**230** 5.1	**415** 5.5	**800** 6	
	Flanged bonnet check	**80** 7.4	**146** 7.8	**80** 4.3	**105** 4.5	**160** 4.8	**195** 5	**320** 5.6	**780** 6	
	Pressure seal bonnet–gate						**215** 2.8	**380** 3	**520** 4	
	Pressure seal bonnet–globe							**240** 2.7	**290** 3	

NOTE: See footnotes to Table 8.7.4.

Table 8.7.6 Weights of Standard Pipe Fittings and Materials, 8-in Size (8.625-in OD)

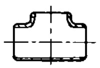
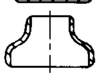
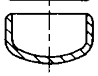
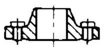
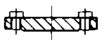
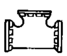
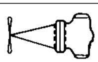
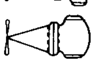
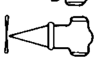
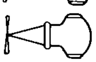
		Schedule no.	20	30	40	60	80	100	120	140		160	
		Wall designation			Std		XS				XXS		
		Thickness, in	0.250	0.277	0.322	0.406	0.500	0.598	0.718	0.812	0.875	0.906	
		Pipe, lb/ft	22.36	24.70	28.55	35.64	43.4	50.9	60.6	67.8	72.4	74.7	
		Water, lb/ft	22.48	22.18	21.69	20.79	19.8	18.8	17.6	16.7	16.1	15.8	
Welding fittings		L.R. 90° elbow			**46**		**69**				**114**	**117**	
					2		2				2	2	
		S.R. 90° elbow			**31**		**46**						
					1.3		1.3						
		L.R. 45° elbow			**23**		**34**				**55**	**56**	
					0.8		0.8				0.8	0.8	
		Tee			**54**		**76**				**118**	**120**	
					1.8		1.8				1.8	1.8	
		Lateral			**76**		**140**						
					3.8		3.8						
		Reducer			**13.9**		**20**				**32**	**33**	
					0.5		0.5				0.5	0.5	
		Cap			**11.3**		**16.3**				**31**	**32**	
					1		1				1	1	
	Temperature range, °F		100–199	200–299	300–399	400–499	500–599	600–699	700–799	800–899	900–999	1,000–1,099	1,100–1,200
Insulation	85% magnesia calcium silicate	Nom. thick., in	1½	1½	2	2	2½	3	3½	3½	4	4	4½
		lb/ft	4.13	4.13	5.64	5.64	7.85	9.48	11.5	11.5	13.8	13.8	16.0
	Combination	Nom. thick., in						3	3½	3½	4	4	4½
		lb/ft						12.9	16.2	16.2	20.4	20.4	23.8
	Asbestos fiber-sodium silicate	Nom. thick., in	1½	1½	1½	1½	1½	2½	2½	3½	3½	4½	4½
		lb/ft	5.38	5.38	5.38	5.38	5.38	10.60	10.60	15.85	15.85	20.85	20.85

Table 8.7.6 Weights of Standard Pipe Fittings and Materials, 8-in Size (8.625-in OD) (*Continued*)

			Pressure rating, lb/in²								
			Cast iron		Steel						
			125	250	150	300	400	600	900	1,500	2,500
Flanges		Screwed or slip-on	**34** 1.5	**64** 1.5	**33** 1.5	**67** 1.5	**82** 1.5	**135** 1.5	**207** 1.5	**319** 1.5	**601** 1.5
		Welding neck			**42** 1.5	**76** 1.5	**104** 1.5	**137** 1.5	**222** 1.5	**334** 1.5	**692** 1.5
		Lap joint	1.5	1.5	**33** 1.5	**67** 1.5	**79** 1.5	**132** 1.5	**223** 1.5	**347** 1.5	**587** 1.5
		Blind	**45** 1.5	**83** 1.5	**48** 1.5	**90** 1.5	**115** 1.5	**159** 1.5	**232** 1.5	**363** 1.5	**649** 1.5
Flanged fittings		S.R. 90° elbow	**117** 4.5	**201** 4.7	**157** 4.5	**238** 4.7	**310** 5	**435** 5.2	**639** 5.4	**995** 5.7	
		L.R. 90° elbow	**152** 5.3	**236** 5.3	**202** 5.3	**283** 5.3					
		45° elbow	**101** 3.9	**171** 4	**127** 3.9	**203** 4	**215** 4.1	**360** 4.4	**507** 4.5	**870** 4.8	
		Tee	**175** 6.8	**304** 7.1	**230** 6.8	**337** 7.1	**445** 7.5	**610** 7.8	**978** 8.1	**1,465** 8.6	
Valves		Flanged bonnet gate	**251** 7.5	**583** 8.1	**305** 4.5	**505** 5.1	**730** 6	**960** 6.3	**1,180** 6.6	**2,740** 7	
		Flanged bonnet globe or angle	**317** 8.4	**554** 8.6	**475** 5.4	**505** 5.5	**610** 5.9	**1,130** 6.3	**1,160** 6.3	**2,865** 7	
		Flanged bonnet check	**302** 8.4	**454** 8.6	**235** 5.2	**310** 5.3	**475** 5.6	**725** 6	**1,140** 6.4	**2,075** 7	
		Pressure seal bonnet–gate						**925** 4.5	**1,185** 4.7	**2,345** 5.5	
		Pressure seal bonnet–globe							**1,550** 4	**1,680** 5	

NOTE: See footnotes to Table 8.7.4.

Table 8.7.7 Weights of Standard Pipe Fittings and Materials, 12-in Size (12.750-in OD)

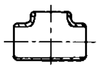
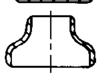
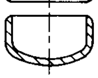
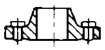
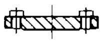
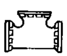
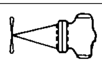
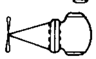
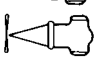
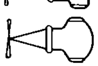
		Schedule no.	20	30		40		60	80	100	120	140	160
		Wall designation			Std		XS						
		Thickness, in	0.250	0.330	0.375	0.406	0.500	0.562	0.687	0.843	1.000	1.125	1.312
		Pipe, lb/ft	33.38	43.8	49.6	53.5	65.4	73.2	88.5	107.2	125.5	139.7	160.3
		Water, lb/ft	51.10	49.7	49.0	48.5	47.0	46.0	44.0	41.6	39.3	37.5	34.9
Welding fittings		L.R. 90° elbow			**119**		**157**						**375**
					3		3						3
		S.R. 90° elbow			**80**		**104**						
					2		2						
		L.R. 45° elbow			**60**		**78**						**181**
					1.3		1.3						1.3
		Tee			**132**		**167**						**360**
					2.5		2.5						2.5
		Lateral			**180**		**273**						
					5.4		5.4						
		Reducer			**33**		**44**						**94**
					0.7		0.7						0.7
		Cap			**30**		**38**						**89**
					1.5		1.5						1.5
	Temperature range, °F		100–199	200–299	300–399	400–499	500–599	600–699	700–799	800–899	900–999	1,000–1,099	1,100–1,200
Insulation	85% magnesia calcium silicate	Nom. thick., in	1½	1½	2	2½	3	3	3½	4	4	4½	4½
		lb/ft	6.04	6.04	8.13	10.5	12.7	12.7	15.1	17.9	17.9	20.4	20.4
	Combination	Nom. thick., in						3	3½	4	4	4½	4½
		lb/ft						17.7	21.9	26.7	26.7	31.1	31.1
	Asbestos fiber-sodium silicate	Nom. thick., in	1½	1½	1½	1½	1½	2½	2½	4	4	5	5
		lb/ft	5.22	5.22	5.22	5.22	5.22	14.20	14.20	24.64	24.64	32.40	32.40

Table 8.7.7 Weights of Standard Pipe Fittings and Materials, 12-in Size (12.750-in OD) (*Continued*)

			Pressure rating, lb/in²								
			Cast iron		Steel						
			125	250	150	300	400	600	900	1,500	2,500
Flanges		Screwed or slip-on	**71** 1.5	**137** 1.5	**72** 1.5	**140** 1.5	**164** 1.5	**261** 1.5	**388** 1.5	**820** 1.5	**1,611** 1.5
		Welding neck			**88** 1.5	**163** 1.5	**212** 1.5	**272** 1.5	**434** 1.5	**843** 1.5	**1,919** 1.5
		Lap joint			**72** 1.5	**164** 1.5	**187** 1.5	**286** 1.5	**433** 1.5	**902** 1.5	**1,573** 1.5
		Blind	**96** 1.5	**177** 1.5	**118** 1.5	**209** 1.5	**261** 1.5	**341** 1.5	**475** 1.5	**928** 1.5	**1,755** 1.5
Flanged fittings		S.R. 90° elbow	**265** 5	**453** 5.2	**345** 5	**509** 5.2	**669** 5.5	**815** 5.8	**1,474** 6.2		
		L.R. 90° elbow	**375** 6.2	**553** 6.2	**485** 6.2	**624** 6.2			**1,598** 6.2		
		45° elbow	**235** 4.3	**383** 4.3	**282** 4.3	**414** 4.3	**469** 4.5	**705** 4.7	**1,124** 4.8		
		Tee	**403** 7.5	**684** 7.8	**513** 7.5	**754** 7.8	**943** 8.3	**1,361** 8.7	**1,928** 9.3		
Valves		Flanged bonnet gate	**687** 7.8	**1,298** 8.5	**635** 4	**1,015** 5	**1,420** 5.5	**2,155** 7	**2,770** 7.2	**4,650** 8	
		Flanged bonnet globe or angle	**808** 9.4	**1,200** 9.5	**710** 5	**1,410** 5.5					
		Flanged bonnet check	**674** 9.4	**1,160** 9.5	**560** 6	**720** 6.5		**1,410** 7.2	**2,600** 8	**3,370** 8	
		Pressure seal bonnet–gate						**1,975** 5.5	**2,560** 6	**4,515** 7	
		Pressure seal bonnet–globe									

NOTE: See footnotes to Table 8.7.4.

Table 8.7.8 Weights of Standard Pipe Fittings and Materials, 24-in Size (24-in OD)

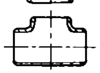
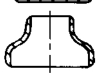
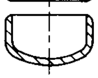
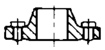
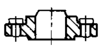
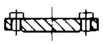
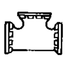
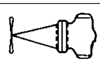
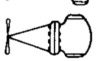
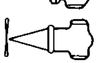
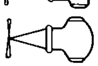
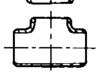
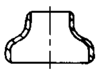
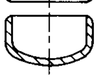
		Schedule no.	10	20		30	40	60	80	100	120	140	160
		Wall designation		Std	XS								
		Thickness, in	0.250	0.375	0.500	0.562	0.687	0.968	1.218	1.531	1.812	2.062	2.343
		Pipe, lb/ft	63.4	94.6	125.5	140.8	171.2	238.1	296.4	267.4	429.4	483.1	541.9
		Water, lb/ft	188.0	183.8	180.1	178.1	174.3	165.8	158.3	149.3	141.4	134.5	127.0
Welding fittings		L.R. 90° elbow		**458** 6	**606** 6								
Welding fittings		S.R. 90° elbow		**305** 3.7	**404** 3.7								
Welding fittings		L.R. 45° elbow		**229** 2.5	**302** 2.5								
Welding fittings		Tee		**445** 4.9	**563** 4.9								
Welding fittings		Lateral		**544** 10	**882** 10								
Welding fittings		Reducer		**167** 1.7	**220** 1.7								
Welding fittings		Cap		**102** 2.8	**134** 2.8								
Temperature range, °F			100–199	200–299	300–399	400–499	500–599	600–699	700–799	800–899	900–999	1,000–1,099	1,100–1,200
Insulation	85% magnesia calcium silicate	Nom. thick., in	1½	1½	2	2½	3	3	3½	4	4	4½	4½
Insulation		lb/ft	10.0	10.0	13.4	17.0	21.0	21.0	24.8	28.7	28.7	32.9	32.9
Insulation	Combination	Nom. thick., in						30	3½	4	4	4½	4½
Insulation		lb/ft						29.2	36.0	43.1	43.1	50.6	50.6
Insulation	Asbestos fiber-sodium silicate	Nom. thick., in	1½	1½	1½	2	2	3	3	4½	4½	5	5
Insulation		lb/ft	13.55	13.55	13.55	18.44	18.44	28.38	28.38	45.06	45.06	50.97	50.97

Table 8.7.8 Weights of Standard Pipe Fittings and Materials, 24-in Size (24-in OD) (*Continued*)

			Pressure rating, lb/in²								
			Cast iron		Steel						
			125	250	150	300	400	600	900	1,500	2,500
Flanges		Screwed or slip-on	**255** 1.5		**245** 1.5	**577** 1.5	**676** 1.5	**1,056** 1.5	**1,823** 1.5	**3,378** 1.5	
		Welding neck			**295** 1.5	**632** 1.5	**777** 1.5	**1,157** 1.5	**2,450** 1.5	**4,153** 1.5	
		Lap joint			**295** 1.5	**617** 1.5	**752** 1.5	**1,046** 1.5	**2,002** 1.5	**3,478** 1.5	
		Blind	**405** 1.5	**757** 1.5	**446** 1.5	**841** 1.5	**1,073** 1.5	**1,355** 1.5	**2,442** 1.5		
Flanged fittings		S.R. 90° elbow	**1,231** 6.7	**2,014** 6.8	**1,671** 6.7	**2,174** 6.8	**2,474** 7.1	**3,506** 7.6	**6,155** 8.1		
		L.R. 90° elbow	**1,711** 8.7	**2,644** 8.7	**1,821** 8.7	**2,874** 8.7					
		45° elbow	**871** 4.8	**1,604** 5	**1,121** 4.8	**1,634** 5	**1,974** 5.1	**2,831** 5.5	**5,124** 6		
		Tee	**1,836** 10	**3,061** 10.2	**2,276** 10	**3,161** 10.2	**3,811** 10.6	**5,184** 11.4	**9,387** 12.1		
Valves		Flanged bonnet gate	**3,062** 8.5	**6,484** 9.8	**2,500** 5	**4,675** 7	**6,995** 8.7	**8,020** 9.5			
		Flanged bonnet globe or angle									
		Flanged bonnet check	**2,956** 12								
		Pressure seal bonnet–gate									
		Pressure seal bonnet–globe									

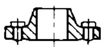
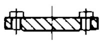
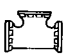
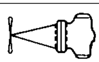
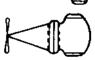
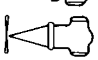
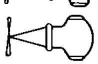
NOTE: See footnotes to Table 8.7.4.

Table 8.7.9 Weights of Standard Pipe Fittings and Materials, 36-in Size (36-in OD)

		Schedule no.	10		20	30	40
		Wall designation		Std	XS		
		Thickness, in	0.312	0.375	0.500	0.625	0.750
		Pipe, lb/ft	119.1	142.7	189.6	236.1	282.4
		Water, lb/ft	425.9	422.6	416.6	411.0	405.1
Welding fittings		L.R. 90° elbow		**1,040** 12	**1,380** 12		
		S.R. 90° elbow		**692** 5	**913** 5		
		L.R. 45° elbow		**518** 4.8	**686** 4.8		
		Tee		**1,294** 9.5	**1,610** 9.5		
		Lateral					
		Reducer		**340** 3.6	**360** 3.6		
		Cap		**175** 6	**235** 6		

			100–199	200–299	300–399	400–499	500–599	600–699	700–799	800–899	900–999	1,000–1,099	1,100–1,200
	Temperature range, °F												
Insulation	85% magnesia calcium silicate	Nom. thick., in	1½	1½	2	2½	3	3½	4	4½	5	5	6
		lb/ft	14.2	14.2	19.2	24.2	29.5	34.8	40.3	45.9	51.7	51.7	63.5
	Combination	Nom. thick., in						3½	4½	5½	6	6½	7
		lb/ft						49.8	69.3	89.7	100.2	111.0	122.0
	Asbestos fiber-sodium silicate	Nom. thick., in	3	3	3	3	3	3	3	4½	4½	5	5
		lb/ft	40.84	40.84	40.84	40.84	40.84	40.84	40.84	55.83	55.83	71.48	71.48

Table 8.7.9 Weights of Standard Pipe Fittings and Materials, 36-in Size (36-in OD) (*Continued*)

			Pressure rating, lb/in^2								
			Cast iron		Steel						
			125	250	150	300	400	600	900	1,500	2,500
Flanges		Screwed or slip-on			**480** 1.5	**1,200** 1.5	**1,325** 1.5	**1,699** 1.5	**3,350** 1.5		
		Welding neck			**520** 1.5	**1,300** 1.5	**1,475** 1.5	**1,750** 1.5	**3,450** 1.5		
		Lap joint									
		Blind			**1,125** 1.5	**2,275** 1.5	**2,525** 1.5	**2,950** 1.5	**4,900** 1.5		
Flanged fittings		S.R. 90° elbow									
		L.R. 90° elbow									
		45° elbow									
		Tee									
Valves		Flanged bonnet gate									
		Flanged bonnet globe or angle									
		Flanged bonnet check									
		Pressure seal bonnet–gate									
		Pressure seal bonnet–globe									

NOTE: See footnotes to Table 8.7.4.

Table 8.7.10 Weight Tolerances of Steel Piping

Specification	Size	Tolerance, %	
ASTM A53	Std wt	+5	−5
	XS wt	+5	−5
ASTM A120	XXS wt	+10	−10
ASTM A106	Sch 10–120	+6.5	−3.5
	Sch 140–160	+10	−3.5
ASTM A335	12 in and under	+6.5	−3.5
	Over 12 in	+10	−5
ASTM A312 ASTM A376	12 in and under	+6.5	−3.5
API 5L	Std. wt Reg wt XS wt XXS wt	+10	−3.5
	Spec. plain end pipe	+10	−5

Seamless pressure tubing may be either hot-finished or cold-drawn. Cold-drawn steel tubing is frequently process-annealed at temperatures above 1,200°F. To ensure quality, maximum hardness values are frequently specified. For example, in ASME Specification SA192,* Specification for Seamless Carbon Steel Boiler Tubes for High-Pressure Service, the following maximum hardness values are given.

Boiler tubes	Brinell hardness no. (tubes 0.200 in and over in wall thickness)	Rockwell hardness no. (tubes less than 0.200 in in wall thickness)
Hot-finished tubes	137	B77
Cold-drawn (normalized) tubes	125	B72

Piping Fabrication Methods Commercial steel pipe is furnished most commonly as seamless pipe. Seamless pipe can be produced by (1) piercing, (2) hollow forging, and (3) forging, turning, and boring. Commercial pipe is also produced by welding involving (1) electric resistance welding, (2) electric fusion welding, or (3) submerged arc welding. Some mills are prepared to extrude small-diameter pipe and tubing in a variety of geometric shapes or to cast large-diameter steel pipe.

Electric-Fusion Welding Flat plate, known as **skelp,** is prepared in proper width and thickness for the desired pipe inside and outside diameters. It is then charged into an electric furnace and, when the proper welding temperature has been reached, is drawn through a funnel-shaped die so shaped that the plate is gradually formed into the shape of a tube, with the edges of the plate being forced squarely together and fused. The formed pipe then passes through a series of rolls in which it is sized or drawn to final dimensions.

* Specifications for boiler tubes generally reference the ASME Standards such as SA178, SA192, SA210, etc.

Electric-Resistance Welding For pipes or tubes sized 4-in (10.2-cm) OD and under, strip is fed onto a set of forming rolls which consists of horizontal and vertical rollers so placed as to gradually form the flat strip into a tube. The tube form then passes to the welding electrodes. The electrodes are copper disks connected to the secondary of a revolving transformer assembly. The copper-disk electrodes make contact on each side of the seam of the tube form; a flow of current takes place across the seam, and temperature is raised to the welding point. Outside flash is removed by a cutting tool as the tube leaves the electrodes; inside flash is removed either by an air hammer or by passing a mandrel through the welded tube after the tube has been cooled.

Submerged-Arc Electric Welding This process is used for pipes from 24-in to 36-in (61.0- to 9.1.4-cm) OD. Flat plate is first pressed into a U and later into an O shape. The O shape is placed in an automatic welder and backed up on the inside by a water-cooled copper shoe. Two electrodes in close proximity are used. The electrodes are not in actual contact with the pipe. The current passes from one electrode through a granular flux and across the gap in the pipe to the second electrode. The high temperature of the arc heats the edges of the plate; a welding rod placed just over the seam is thereby melted and metal is deposited in the groove. After the outside weld has been made, the pipe is conveyed to an inside welder where a similar operation is carried on, except that no backup shoe is needed.

Seamless Tubing and Pipe A heated billet is brought into contact with tapered revolving rolls in such a way that the billet is pulled into the space allowed between the rolls. A piercing mandrel is placed in this space; the soft center of the billet makes it possible for the rolls to draw the billet over the mandrel, producing a hollow shell. When the billet has entirely passed over the mandrel, it is in the form of a thick-walled seamless tube. The heavy-walled tube is then passed to a rolling mill which reduces the tube to pipe of proper outside diameter and wall thickness.

The method of fabrication described above is limited as to diameter and thickness. For seamless alloy tubes and for heavy-wall carbon-steel tubes or pipe, a process known as **cupping and drawing** is frequently used. A circular flat plate of proper diameter and thickness is heated, placed in a hydraulic press, and pressed by a ram through a die. The cup so formed is reheated and pressed through a smaller die, thus elongating the cup so that it becomes a short cylinder with one closed end. This short cylinder is then placed in a horizontal drawbench, and with reheating as necessary, is pushed by a ram through dies of successively smaller diameters until the desired outer diameter is reached.

Forged, Turned, and Bored Tubing In this process, the ingot is heated and forged to a rough cylindrical shape, oversize in both diameter and length. The forging is then placed in a lathe and the outside turned down to the desired outer diameter. Rough ends are then removed so that the finally desired length is obtained. The cylinder is then placed in a boring mill, and the inside bored out until the desired wall

Table 8.7.11 Diameter and Wall-Thickness Tolerances for Seamless Hot-finished Mechanical Tubing of Carbon and Alloy Steel (AISI)*

Specified size, OD, in	Ratio of wall thickness to OD	OD tolerance		Wall thicknesses tolerance, %: 0.109 in and under		Over 0.109 to 0.172 in, incl.		Over 0.172 to 0.203 in, incl.		Over 0.203 in	
		Over	Under	Over	Under	Over	Under	Over	Under	Over	Under
Under 3	All wall thicknesses	0.023	0.023	16.5	16.5	15	15	14	14	12.5	12.5
3–5½, excl.	All wall thicknesses	0.031	0.031	16.5	16.5	15	15	14	14	12.5	12.5
5½–8 excl.	All wall thicknesses	0.047	0.047					14	14	12.5	12.5
8–10¾, incl	5% and over	0.047	0.047							12.5	12.5
8–10¾, incl.	Under 5%	0.063	0.063							12.5	12.5

* The common range of sizes of hot-finished tubes is 1½ to and including 10¾ in outside diameter with wall thickness not less than 0.095 in (no. 13 BWG) or 3% or more of the outside diameter. For sizes under 1½ or over 10¾ in outside diameter, the tolerances are commonly negotiated between the purchaser and producer.
SOURCE: AISI, ''Steel Products Manual.''

Table 8.7.12 Diameter and Wall-Thickness Tolerances for Seamless Cold-Worked Mechanical Tubing of Carbon and Alloy Steel (AISI)*

	Unannealed or finish-annealed				Soft annealed or normalized				Quenched and tempered				Wall thickness all conditions, %	
	OD, in		ID, in		OD, in		ID, in		OD, in		ID, in			
Size, OD, in	Over	Under	Over	Under	Over	Under	Over	Under	Over	Under	Over	Under	Over	Under
3/16–½, excl.†‡	0.004	0			0.006	0.002			0.010	0.010			15	15
½–1½, excl.†‡§¶	0.005	0	0	0.005	0.008	0.002	0.002	0.008	0.015	0.015	0.015	0.015	10	10
1½–3½, excl.†‡§¶	0.010	0	0	0.010	0.015	0.005	0.005	0.015	0.030	0.030	0.030	0.030	10	10
3½–5½, excl.§¶	0.015	0	0.005	0.015	0.023	0.007	0.015	0.025	0.045	0.045	0.045	0.045	10	10
5½–8, excl.,¶ wall less than 5% OD	0.030	0.030	0.035	0.035	0.060	0.060	0.070	0.070					10	10
5½–8, excl., wall from 5 to 7.5% OD	0.020	0.020	0.025	0.025	0.040	0.040	0.050	0.050					10	10
5½–8, excl.,§ wall over 7.5% OD	0.030	0	0.015	0.030	0.045	0.015	0.037	0.053					10	10
8–10¾, incl.,¶ wall less than 5% OD	0.045	0.045	0.050	0.050									10	10
8–10¾, incl., wall from 5 to 75% OD	0.035	0.035	0.040	0.040									10	10
8–10¾, incl.,§ wall over 7.5% OD	0.045	0	0.015	0.040									10	10

* For tolerances closer than those indicated, availability, and applicable tolerances for tubing less than 3/16 in OD or larger than 10¾-in OD, the producer should be consulted.

† For those tubes with inside diameter less than ½ in (or less than ⅝ in when the wall thickness is more than 20% of the outside diameter), which are not commonly drawn over a mandrel. Note § is not applicable. Unless otherwise agreed upon by the purchaser and producer, the wall thickness may vary 15% over and under that specified, and the inside diameter is governed by the outside diameter and wall-thickness tolerances shown.

‡ For tubes with inside diameter less than ½ in (or less than ⅝ in when the wall thickness is more than 20% of the outside diameter), which can be produced by the rod or bar mandrel process, the tolerances are as shown in the above table except that the wall-thickness tolerances are 10% over and under the specified wall thickness.

§ Many tubes with inside diameter less than 50% of outside diameter, or with wall thickness more than 25% of outside diameter, or with wall thickness over 1¼ in, or weighing more than 90 lb/ft are difficult to draw over a mandrel. Unless otherwise agreed upon by the purchaser and producer the inside diameter may vary over or under by an amount equal to 10% of the wall thickness and the wall thickness may vary 12½% over and under that specified.

¶ Tubing having a wall thickness less than 3% of the outside diameter cannot be straightened properly without a certain amount of distortion. Consequently, such tubes, while having an average outside diameter and inside diameter within the tolerances shown in the above table, require an ovality tolerance of 0.5% over and under nominal outside diameter, this being in addition to the tolerances indicated in the above table.

SOURCE: AISI, ''Steel Products Manual.''

Table 8.7.13 Finish Classifications Normally Recognized as Welded Mechanical Tubing

	Type	Condition
A	Flash-in, hot-rolled	Made by longitudinally forming and welding hot-rolled strip; the interior welding flash remains in place
B	Flash-in, cold-rolled	Made by longitudinally forming and welding cold-rolled strip; the interior flash remains in place
C	Flash controlled – 0.010 in max, hot-rolled	Same as A, except that the interior flash is partially removed so that the height remaining does not exceed 0.010 in
D	Flash controlled – 0.010 in max, cold-rolled	Same as C, except that cold-rolled strip has been used
E	Flash controlled – 0.005 in max, hot-rolled	Same as C, except that flash height on tube inside does not exceed 0.005 in
F	Flash controlled – 0.005 in max, cold-rolled	Same as E, except that cold-rolled strip has been used
G	Sink-drawn, hot-rolled	Tubing with flash controlled to 0.010 in max, which has been descaled and cold-drawn through a die to cold-finish the exterior surface
H	Sink-drawn, cold-rolled	Tubing with flash controlled to 0.010 in max, which has been cold-drawn through a die to cold-finish the exterior surface
I	Mandrel-drawn	Cold-rolled tubing with the interior flash removed and drawn through a die and over a mandrel to cold-finish the exterior and interior surfaces

thickness is secured. Because of the relatively high cost, this process is now rarely used.

Hollow-Forged Pipe and Tubing In this process, ingots are cast and their ends cropped; then they are placed in a furnace and heated to a specified temperature. The heated ingot is placed in a press where it is pierced. This hollow cylinder, open at one end, is then descaled and drawn over a mandrel on a horizontal drawbench. The closed end is then burned off, and the hollow forging is chemically descaled. Following this, the forging is straightened, placed in a lathe, and the outer diameter machined to a true dimension. The inside is dressed to remove scale, but no machining is done on the inside.

Carbon-steel piping is most frequently used as manufactured in accordance with ASTM Specifications A106 and A53 (or ASME Specifications SA106 and SA53). The chemical compositions of these two materials are identical except for the deoxidation practice which applies to the A106 pipe. Both are subjected to physical tests, but those for A106 are more rigorous. A53 and A106 are made in grades A and B; grade B has higher strength properties but is less ductile and, for this reason, grade A is permitted only for cold bending or close coiling. When carbon steel is intended for use in welded construction at temperatures in excess of 775°F (413°C), consideration should be given to the possibility of graphite formation.

Chromium-molybdenum steel has been used for temperatures up to 1,100°F (593°C). In the small diameters, the material is usually available in the seamless construction; because of the inability of the seamless mills to fabricate large-diameter and heavy-walled pipe, it may be necessary to resort to the more expensive hollow-forged or forged-and-bored piping for higher pressures and temperatures. The material for a high-temperature piping system should be selected after a careful review of technical and economic considerations; the following is intended only as being indicative of recent and current practice.

For temperatures up to 1,000°F, 1¼% Cr-½% Mo (A335, grade Pll) is used. For temperatures from 950 to 1050°F, 2¼% Cr-1% Mo (A335, P22) generally is used. Where there is a combination of high temperatures and erosive action, 5% Cr-½% Mo (A335, grade 5) or other more highly chromium-molybdenum or chromium stainless steels have been used.

Stainless-steel piping is available in a variety of compositions, most popular of which are ASTM A213, grade TP304 (16% Cr – 8% Ni), and ASTM A213, grade TP316 (18% Cr – 12% Ni and 3% Mo). For high-temperature service, type 34 stainless steels are used (18% Cr – 8% Ni and stabilized with columbium). This material may be used up to 1,200°F (649°C); particular care must be given to choice of welding filler metal to avoid brittleness in the welds.

The permissible stress values for a large variety of piping materials at low and elevated temperatures are provided in Table 8.7.15.

PIPE FITTINGS

The various major piping materials are also produced in the form of standard fittings. Among the more widely used are wrought-steel fittings, welded-steel fittings, cast-steel fittings, cast-iron fittings, ductile-iron fittings, malleable-iron fittings, brass and copper fittings, aluminum fittings, etc. Other major nonferrous piping materials are also produced in the form of cast and wrought fittings.

Cast-iron, ductile-iron, and malleable-iron fittings are made by conventional founding methods for a variety of joints including bell-and-spigot, flanged, and mechanical (gland-type), or other proprietary joint designs.

Schedule Designations Over 100 years ago piping was designated as standard, extra-strong, and double extra-strong. There was no provision for thin-walled pipe, and no intervening standard thicknesses between the three schedules, which covered too great a spread to be economical without intermediate weights. Table 8.7.3 lists piping as a function of the schedule number which is given, approximately, by the following relationship: Schedule no. = 1,000 $P/(SE)$, where P is operating pressure, lb/in^2 gage, S is allowable stress, lb/in^2 (Table 8.7.15), and E is the quality factor (Tables 8.7.16 and 8.7.17).

Commercial sizes of steel pipe are known by their nominal inside diameter (ID) from ⅛ in (0.3175 cm) to 12 in (30.5 cm). Above 12-in ID, pipe is usually known by its outside diameter (OD). All classes of pipe of a given nominal size have the same OD, the extra thickness for different weights being on the inside.

Thickness of Pipe The following notes, covering power piping systems, have been abstracted from Part 2 of the Code for Power Piping (ASME B31.1).

For inspection purposes, the minimum thickness of pipe wall to be used for piping at different pressures and for temperatures not exceeding those for the various materials listed in Table 8.7.15 shall be determined by the formula

$$t_m = \frac{PD}{2(SE + Py)} + A \qquad (8.7.1)$$

where t_m = minimum pipe-wall thickness, in, allowable on inspection; P = maximum internal service pressure, lb/in^2 gage (plus water-

Table 8.7.14 Typical Mechanical Properties of Resistance-Welded Mechanical Carbon-Steel Tubing

Type	Yield strength, lb/in^2	Tensile strength, lb/in^2	Elongation, %	Hardness, Rockwell B
Flash-in or flash-controlled, hot-rolled	48,000	58,000	29	68
Flash-in or flash-controlled, cold-rolled	68,000	76,000	17	84
Normalized	34,000	52,000	39	61
Sink-drawn	73,000	76,000	20	84
Mandrel-drawn	80,000	83,000	15	86

Table 8.7.15 Basic Allowable Stresses in Tension for Metals

Material	Spec. no.	P no. (5)†	Grade	Notes	Min temp. (6)
Iron					
Centrifugally cast pipe	A 377			(8) (9) (48)	− 20
Castings					
Gray	A 48		20	(8) (9) (48)	− 20
Gray	A 48		25	(8) (9) (48)	− 20
Gray	A 48		30	(8) (9) (48)	− 20
Gray	A 48		35	(8) (9) (48)	− 20
Gray	A 48		40	(8) (9) (48)	− 20
Gray	A 48		45	(8) (9) (48)	− 20
Gray	A 48		50	(8) (9) (48)	− 20
Gray	A 48		55	(8) (9) (48)	− 20
Gray	A 48		60	(8) (9) (48)	− 20
Gray	A 278		70	(8) (9) (53)	− 20
Gray	A 278		80	(8) (9) (53)	− 20
Cupola malleable	A 197			(8) (9)	− 20
Malleable	A 47		32510	(8) (9)	− 20
Malleable	A 47		35018	(8) (9)	− 20
Ductile	A 395			(8) (9)	− 20
Ferritic ductile	A 395			(8) (9)	− 20
Austenitic ductile	A 571		Type D-2M	(9)	− 20

Materials	Spec. no.	P no. (5)	Grade	Notes	Min temp. (6)	SMTS, ksi	SMYS, ksi
Carbon steel							
Pipes and tubes							
	A 120	1		(8)	− 20		
A 285 gr. A	A 672	1	A45	(57) (59) (67)	− 20	45	24
	A53	1	Type F	(8)	− 20	45	25
Butt weld	API 5L	1	A25				
Smls & ERW	API 5L	1	A25	(57) (59)	− 20	45	25
	A 179	1		(57) (59)	− 20	47	26
	A 135	1	A	(57) (59)	− 20	48	30
A 285 gr. B	A 672	1	A50	(57) (59) (67)	− 20	50	27
A 285 gr. C	A 134	1		(8) (57)	− 20	55	30
A 285 gr. C	A 671	1	CA55	(59) (67)	− 20	55	30
A 516 gr. 60	A 671	1	CC60	(57) (67)	− 20	60	32
A 515 gr. 60	A 672	1	B60	(57) (67)	− 20	60	32
	A 135	1	B	(57) (59)	− 20	60	35
	A 53	1	B	(57) (59)	− 20		
	A 106	1	B	(57)	− 20	60	35
	A 139	1	D	(8)	− 20	60	46
(> ⅜ in thick)	A 381	SP3	Y48	(51)	− 20	62	48
(> ⅜ in thick)	A 381	SP3	Y50	(51)	− 20	64	50
A 515 gr. 65	A 672	1	B65	(57) (67)	− 20	65	35
(> ⅜ in thick)	A 381	SP3	Y52	(51)	− 20	66	52
(≤ ⅜ in thick)	A 381	SP3	Y48	(51)	− 20	67	48
A 516 gr. 70	A 671	1	CC70	(57) (67)	− 20	70	38
	A 106	1	C	(57)	− 20	70	40
(≤ ⅜ in thick)	A 381	SP3	Y52	(51)	− 20	72	52
A 299 (> 1 in thick)	A 672	1	N75	(57) (67)	− 20	75	40
A 299 (≤ 1 in thick)	A 672	1	N75	(57) (67)	− 20	75	42

NOTE: Footnotes appear at the end of the table.

SMTS,‡ ksi	SMYS,‡ ksi	Basic allowable stress *S*, ksi (1), at metal temperature, °F (7): Min temp. to 100	200	300	400	500	600	650
		4.0	4.0	4.0	4.0			
20		2.0	2.0	2.0	2.0			
25		2.5	2.5	2.5	2.5			
30		3.0	3.0	3.0	3.0			
35		3.5	3.5	3.5	3.5			
40		4.0	4.0	4.0	4.0			
45		4.5	4.5	4.5	4.5			
50		5.0	5.0	5.0	5.0			
55		5.5	5.5	5.5	5.5			
60		6.0	6.0	6.0	6.0			
70		7.0	7.0	7.0	7.0	7.0	7.0	7.0
80		8.0	8.0	8.0	8.0	8.0	8.0	8.0
40	30	8.0	8.0	8.0	8.0	8.0	8.0	8.0
50	32	10.0	10.0	10.0	10.0	10.0	10.0	10.0
53	35	10.6	10.6	10.6	10.6	10.6	10.6	10.6
60	40	20.0	19.0	17.9	16.9	15.9	14.9	14.1
65	30	20.0						

Basic allowable stress *S*, ksi (1), at metal temperature, °F (7)

Min temp. to 100	200	300	400	500	600	650	700	750	800	850	900	950	1,000	1,050	1,100
12.0	11.4														
15.0	14.6	14.2	13.7	13.0	11.8	11.6	11.5	10.3	9.0	7.8	6.5	4.5	2.5	1.6	1.0
15.0	15.0	14.5	13.8												
15.0	15.0	14.5	13.8												
15.7	15.0	14.2	13.5	12.8	12.1	11.8	11.5	10.6	9.2	7.9	6.5	4.5	2.5	1.6	1.0
16.0	16.0	16.0	16.0	16.0	14.8	14.5	14.4	10.7	9.3	7.9	6.5	4.5	2.5	1.6	1.0
16.7	16.4	16.0	15.4	14.6	13.3	13.1	13.0	11.2	9.6	8.1	6.5	4.5	2.5	1.6	1.0
18.3	18.3	17.7	17.2	16.2	14.8	14.5	14.4	12.0	10.2	8.3	6.5				
18.3	18.3	17.7	17.2	16.2	14.8	14.5	14.4	12.1	10.2	8.4	6.5	4.5	2.5	1.6	1.0
20.0	19.5	18.9	18.3	17.3	15.8	15.5	15.4	13.0	10.8	8.7	6.5	4.5	2.5		
20.0	19.5	18.9	18.3	17.3	15.8	15.5	15.4	13.0	10.8	8.7	6.5	4.5	2.5	1.6	1.0
20.0	20.0	20.0	20.0	18.9	17.3	17.0	16.5	13.0	10.8	8.7	6.5	4.5	2.5		
20.0	20.0	20.0	20.0	18.9	17.3	17.0	16.5	13.0	10.8	8.7	6.5	4.5	2.5	1.6	1.0
20.0	20.0	20.0													
20.6	19.7	18.7	17.8	16.9	16.0	15.5									
21.3	20.3	19.3	18.4	17.4	16.5	16.0									
21.7	21.3	20.7	20.0	18.9	17.3	17.0	16.8	13.9	11.4	9.0	6.5	4.5	2.5	1.6	1.0
22.0	21.0	19.9	18.9	17.9	17.0	16.5									
22.3	21.3	20.2	19.2	18.2	17.3	16.7									
23.3	23.1	22.5	21.7	20.5	18.7	18.4	18.3	14.8	12.0	9.3	6.5	4.5	2.5		
23.3	23.3	23.3	22.9	21.6	19.7	19.4	19.2	14.8	12.0						
24.0	22.9	21.7	20.7	19.6	18.6	18.0									
25.0	24.4	23.7	22.9	21.6	19.7	19.4	19.2	15.7	12.6	9.5	6.5	4.5	2.5	1.6	1.0
25.0	25.0	24.8	24.0	22.7	20.7	20.4	20.2								

Table 8.7.15 Basic Allowable Stresses in Tension for Metals (*Continued*)

Materials	Spec. no.	P no. (5)	Grade	Notes	Min temp. (6)	SMTS, ksi	SMYS, ksi
Carbon steel (*Cont.*)							
Pipes (structural grade)							
A 283 gr. A	A 134	1		(8)	− 20	45	24
A 570 gr. 30	A 134	1		(8)	− 20	49	30
A 283 gr. B	A 134	1		(8)	− 20	50	27
A 570 gr. 33	A 134	1		(8)	− 20	52	33
A 570 gr. 36	A 134	1		(8)	− 20	53	36
A 570 gr. 40	A 134	1		(8)	− 20	55	40
A 36	A 134	1		(8)	− 20	58	36
A 570 gr. 45	A 134	1		(8)	− 20	60	45
A 570 gr. 50	A 134	1		(8)	− 20	65	50
Forgings and fittings							
	A 350	1	LF-1	(9) (57) (59)	− 20	60	30
	A 105	1		(9) (57) (59)	− 20	70	36
Castings							
	A 216	1	WCA	(9) (57)	− 20	60	30
	A 352	1	LCB	(9) (57)	− 50	65	35
	A 216	1	WCC	(9) (57)	− 20	70	40

Material	Spec. no.	P no. (5)	Grade	Notes	Min temp. (6)
Low and intermediate alloy steel					
Pipes					
C−½Mo	A 335	3	P1	(58)	− 20
½Cr−½Mo	A 691	3	½Cr	(11) (67)	− 20
A 387 gr. 2 cl. 1					
1Cr−½Mo	A 691	4	1Cr	(11) (67)	− 20
A 387 gr. 12 cl. 1					
1½Si−½Mo	A 335	3	P15		− 20
7Cr−½Mo	A 426	5	CP7	(10)	− 20
3Cr−Mo	A 426	5	CP21	(10)	− 20
¾C−¾Ni−Cu−Al	A 333	4	4		− 150
2Cr−½Mo	A 369	4	FP3b		− 20
7Cr−½Mo	A 335	5	P7		− 20
1¼Cr−½Mo	A 335	4	P11		− 20
5Cr−½Mo	A 691	5	5Cr	(11) (67)	
A 387 gr. 5 cl. 1					
5Cr−½Mo−Si	A 335	5	P5b		− 20
9Cr−1Mo	A 335	5	F9		− 20
3Cr−1Mo	A 335	5	P21		− 20
2¼Cr−1Mo	A 691	5	2¼Cr	(11) (67)	− 20
A 387 gr. 22 cl. 1					
2¼Cr−1Mo	A 335	5	P22		− 20
2Ni−1Cu	A 333	9A	9		− 100
C−Mo A 204 gr. B	A 672	3	L65	(11) (58) (67)	− 20
2¼Ni	A 333	9A	7		− 100
3½Ni	A 333	9B	3		− 150
2¼Ni A 203 gr. B	A 671	9A	CF70	(11) (65) (67)	− 20
C−Mo A 204 gr. B	A 672	3	L70	(11) (58) (67)	− 20
1¼Cr−½Mo	A 426	4	CP11	(10)	− 20
C−Mo A 204 gr. C	A 672	3	L75	(11) (58) (67)	− 20
12¾Cr	A 426	6	CPCA-15	(10)	− 20
5Cr−½Mo	A 426	5	CP5	(10)	− 20
9Ni	A 333	11A-SG1	8	(47)	− 320
Forgings and fittings					
C−½Mo	A 234	3	WP1	(58)	− 20
1Cr−½Mo	A 234	4	WP12		− 20
1¼Cr−½Mo	A 234	4	WP11		− 20

Basic allowable stress *S*, ksi (1), at metal temperature, °F (7)															
Min temp. to 100	200	300	400	500	600	650	700	750	800	850	900	950	1,000	1,050	1,100
13.7	13.0	12.4													
15.0	15.0	15.0	15.0												
15.3	14.4	13.9													
15.9	15.9	15.9	15.9												
16.3	16.3	16.3	16.3												
16.9	16.9	16.9	16.9												
17.6	16.8	16.8	16.8												
18.4	18.4	18.4	18.4												
19.9	19.9	19.9	19.9												
20.0	18.3	17.7	17.2	16.2	14.8	14.5	14.4	12.9	10.8	8.6	6.5	4.5	2.5	1.6	1.0
23.3	21.9	21.3	20.6	19.4	17.8	17.4	17.3	14.8	12.0	9.3	6.5	4.5	2.5	1.6	1.0
20.0	18.3	17.7	17.2	16.2	14.8	14.5	14.4	13.0	10.8	8.6	6.5	4.5	2.5	1.6	1.0
21.7	21.3	20.7	20.0	18.9	17.3	17.0	16.8	13.8	11.4	8.9	6.5	4.5	2.5	1.6	1.0
23.3	23.3	23.3	22.9	21.6	19.7	19.4	19.2	14.8	12.0	9.3	6.5	4.5	2.5		

		Basic allowable stress *S*, ksi (1), at metal temperature, °F (7)							
SMTS, ksi	SMYS ksi	Min temp. to 100	200	400	600	800	1,000	1,100	1,200
55	30	18.3	18.3	16.9	15.7	13.5	4.8		
55	33	18.5	18.3	18.3	17.3	13.8	5.9		
55	33	18.3	18.3	18.3	17.3	15.9	6.6	2.6	1.0
60	30	18.8	18.2	17.0	15.9	14.4	6.3	2.4	
60	30	18.8	17.9	16.2	14.5	12.5	5.0	2.5	1.2
60	30	18.8	18.1	16.8	15.5	13.9	7.0	4.0	1.5
60	35	20.0	19.1	17.3	15.5				
60	30	20.0	18.5	16.4	15.7	13.5	6.2	2.6	1.0
60	30	20.0	18.1	17.2	16.8	12.8	5.8	2.9	1.2
60	30	20.0	18.7	17.5	16.7	15.0	7.8	4.0	1.2
60	30	20.0	18.1	17.2	16.8	12.8	5.8	2.9	1.3
60	30	20.0	18.1	17.2	16.8	12.8	7.4	3.3	1.5
60	30	20.0	18.7	17.5	16.7	15.0	7.0	4.0	1.5
60	30	20.0	18.5	17.9	17.9	15.2	7.8	4.2	1.6
60	30	20.0	18.5	17.9	17.9	15.2	7.8	4.2	2.0
63	46	21.0							
65	37	21.7	21.7	20.7	19.3	15.8	4.8		
65	35	21.7	19.6	18.7	16.8	11.4	2.5	1.0	
65	35	21.7	19.6	18.7	16.8	11.4	2.5	1.0	
70	40	23.3							
70	40	23.3	23.3	22.5	20.9	17.5	4.8		
70	40	23.3	23.3	23.3	22.3	15.0	7.8	4.0	1.2
75	43	25.0	25.0	24.1	22.5	18.8	4.8		
90	65	30.0							
90	60	30.0	28.0	24.1	20.1	14.5	5.6	3.1	1.3
100	75	31.7	31.7						
55	30	18.3	18.3	16.9	15.7	13.5	4.8		
60	30	20.0	18.7	17.5	16.7	15.0	7.5	2.8	1.0
60	30	20.0	18.7	17.5	16.7	15.0	7.8	4.0	1.2

Table 8.7.15 Basic Allowable Stresses in Tension for Metals (*Continued*)

Material	Spec. no.	P no. (5)†	Grade	Notes	Min temp. (6)
Low and intermediate alloy steel (*Cont.*)					
Forgings and fittings (*Cont.*)					
5Cr–½Mo	A 234	5	WP5		−20
9Cr–½Mo	A 234	5	WP9		−20
3½Ni	A 350	9B	LF3	(9)	−150
C–½Mo	A 182	3	F1	(9) (58)	−20
½Cr–½Mo	A 182	3	F2	(9)	−20
1¼Cr–½Mo	A 182	4	F11	(9)	−20
5Cr–½Mo	A 182	5	F5	(9)	−20
13Cr	A 182	6	F6a Cl.1		−20
3Cr–1Mo	A 182	5	F21	(9)	−20
13Cr	A 182		F6a Cl.2		−20
5Cr–½Mo	A 182	5	F5a	(9)	−20
9Ni	A 420	11A-SG1	WPL8	(47)	−320
13Cr	A 182		F6a Cl.3		−20
13Cr–½Mo	A 182		F6b		
Castings					
C–½Mo	A 217	3	WC1	(9) (58)	−20
2½ Ni	A 352	9A	LC2	(9)	−100
Ni–Cr–1Mo	A 217	4	WC5	(9)	−20
2¼Cr–1Mo	A 217	5	WC9	(9)	−20
12Cr	A 217	6	CA15	(9) (35)	−20
5Cr–½Mo	A 217	5	C5	(9)	−20

Material	Spec. no.	P no. (5)	Grade	Notes	Min temp. (6)	SMTS, ksi	SMYS, ksi
Stainless steel (4) (40)							
Pipes and tubes							
18CR–8Ni pipe	A 312	8	TP304L		−425	70	25
Type 304L A 240	A 358	8	304L	(36)	−425		
16Cr–12Ni–2Mo pipe	A 312	8	TP316L		−325	70	25
23Cr–13Ni	A 451	8	CPH8	(26) (28) (35)	−325	65	28
18Cr–Ti tube	A 268	7	TP430TI	(35) (49)	−20	60	40
16Cr–8Ni–2Mo pipe	A 376	8	16–8–2H	(26) (31) (35)	−325	75	30
12Cr–Al tube	A 268	7	TP405	(35)	−20	60	30
16Cr tube	A 268	7	TP430	(35) (49)	−20	60	35
Type 310S A 240	A 358	8	310S	(28) (31) (35) (36)	−325	75	30
18Cr–10Ni–Ti	A 312	8	TP321	(28)	−325	75	30
Type 321 A 240	A 358	8	321	(28) (30) (36)	−325		
Type 309S A 240	A 358	8	309S	(28) (31) (35) (36)	−325	75	30
18Cr–8Ni	A 451	8	CPF8	(26) (28)	−425	70	30
Type 347 A 240	A 358	8	347	(28) (30) (36)	−425	75	30
Type 348 A 240	A 358	8	348	(28) (30) (36)	−325		
18 Cr–10Ni–Cb pipe	A 409	8	TP348	(28) (30) (36)	−325		
Type 310S A 240	A 358	8	310S	(28) (29) (31) (35) (36)	−325	75	30
18Cr–10Ni–Ti pipe	A 312	8	TP321H		−325	75	30
Type 316 A 240	A 358	8	316	(26) (28) (31) (36)	−325		
16Cr–12Ni–2Mo pipe	A 376	8	TP316	(26) (28) (31) (35)	−325	75	30
18Cr–13Ni–3Mo pipe	A 312	8	TP317	(26) (28)	−325		
16Cr–12Ni–2Mo pipe	A 430	8	FP316H	(26) (31) (36)	−325	70	30
16Cr–12Ni–2Mo pipe	A 312	8	TP316H	(26)	−325	75	30
18Cr–10Ni–Cb pipe	A 430	8	FP347H	(30) (36)	−325	70	30
18Cr–10Ni–Cb pipe	A 312	8	TP348H		−325	75	30
18Cr–8Ni pipe	A 430	8	FP304	(26) (31) (36)	−425	70	30
Type 304 A 240	A 358	8	304	(26) (28) (31) (36)	−425	75	30
						70	30
18Cr–8Ni pipe	A 312	8	TP304H	(26)	−325	75	30
18Cr–8Ni	A 452	8	TP304H	(26)	−325	75	30

SMTS, ksi	SMYS, ksi	Basic allowable stress S, ksi (1), at metal temperature, °F (7) Min temp. to 100	200	400	600	800	1,000	1,100	1,200
60	30	20.0	18.1	17.2	16.8	12.8	5.8	2.9	1.3
60	30	20.0	18.1	17.2	16.8	12.8	7.4	3.3	1.5
70	37.5	23.3							
70	40	23.3	23.3	22.5	20.9	17.5	4.8		
70	40	23.3	23.3	22.5	20.9	17.5	5.9		
70	40	23.3	23.3	22.5	20.9	19.2	6.9	2.8	1.2
70	40	23.3	23.3	22.4	22.0	14.8	5.8	2.9	1.3
70	40								
75	45	25.0	25.0	24.1	23.8	22.5	6.8	3.2	1.3
85	55	28.3	28.3	27.2	26.1	23.2			
90	65	30.0	29.9	28.9	28.3	19.1	5.8	2.9	1.3
110	75	31.7	31.7						
110	85								
110–135	90								
65	35	21.7	21.5	19.7	18.3	15.8	4.8		
70	40	23.3	17.5	17.5	17.5				
70	40	23.3	23.3	22.5	20.9	17.5	6.9	2.8	
70	40	23.3	23.3	22.5	22.4	21.0	7.6	4.4	1.3
90	65	30.0	21.5	20.0	18.8	16.8	5.0	2.3	1.0
90	60	30.0	29.9	28.9	28.3	19.1	5.8	2.9	1.3

Basic allowable stress S, ksi (1), at metal temperature, °F (7) Min temp. to 100	200	400	600	800	1,000	1,200	1,400	1,500
16.7	16.7	15.8	14.0	13.0	7.8	3.2	1.1	0.9
16.7	16.7	15.5	13.5	12.4	11.2	6.4	1.8	1.0
18.7	18	18.7	18.0	16.3	10.4	3.7	1.3	0.8
20.0								
20.0								
20.0	18.4	17.4	16.8	11.1	4.0			
20.0	20.0	19.2	18.5	11.1	6.5	1.7		
20.0	20.0	20.0	19.2	17.5	11.0	2.5	0.4	0.2
20.0	20.0	18.6	16.4	15.5	13.8	3.6	0.8	0.3
20.0	20.0	20.0	19.2	17.5	10.5	3.8	1.3	0.7
20.0	20.0	17.5	15.7	14.8	10.8	4.4	1.3	0.8
20.0	20.0	20.0	19.3	18.3	14.0	4.4	1.2	0.8
20.0	20.0	20.0	19.2	17.5	11.0	6.0	1.6	0.8
20.0	20.0	18.6	16.4	15.5	14.0	5.4	1.9	1.1
20.0	20.0	19.3	17.0	15.9	15.3	7.4	2.3	1.3
20.0	20.0	19.3	17.0	15.9	15.3	7.4	2.3	1.3
20.0	20.0	19.2	18.3	18.2	18.0	7.9	2.5	1.3
20.0	20.0	20.0	19.3	18.3	18.0	7.9	2.5	1.3
20.0	20.0	18.7	16.4	15.2	13.8	6.0	2.3	1.4
20.0	20.0	18.7	16.4	15.2	13.8	6.0	2.3	1.4
20.0	20.0	18.7	16.5	15.1	13.8	6.0	2.2	1.4

Table 8.7.15 Basic Allowable Stresses in Tension for Metals (*Continued*)

Material	Spec. no.	P no. (5)	Grade	Notes	Min temp. (6)	SMTS, ksi	SMYS, ksi
Stainless steel (4) (40) (*Cont.*)							
Pipes and tubes (*Cont.*)							
18Cr–10Ni–Mo	A 451	8	CPF8M	(26) (28)	−425	70	30
27Cr tube	A 268	10E	TP446	(35)	−20	70	40
26Cr–3Ni–1Mo tube	A 268	10E	TP329	(35)	−20	90	70
Forgings and Fittings							
18Cr–8Ni	A 182	8	F304L	(9)	−425	65	25
16Cr–12Ni–2Mo	A 182	8	F316L	(9)	−325	65	25
20Ni–8Cr	A 182	8	F10	(9) (26) (28) (39)	−325	80	30
25Cr–20Ni	A 182	8	F310	(9) (28) (35) (39)	−425	75	30
18Cr–10Ni–Ti	A 182	8	F321	(9) (21) (28)	−325	75	30
25Cr–20Ni	A 182	8	F310	(9) (28) (29) (35) (39)	−425	75	30
18Cr–10Ni–Cb	A 182	8	F347	(9) (21) (28)	−425	75	30
18Cr–10Ni–Ti	A 182	8	F321H	(9) (21)	−325	75	30
16Cr–12Ni–2Mo	A 182	8	F316H	(9) (21) (26)	−325	75	30
18Cr–10Ni–Cb	A 182	8	F347H	(9) (21)	−325	75	30
16Cr–12Ni–2Mo	A 403	8	WP316	(26) (28) (31) (32) (37)	−325	75	30
18Cr–8Ni	A 182	8	F304	(9) (21) (26) (28)	−425	75	30
18Cr–8Ni	A 403	8	WP304H	(26) (31) (32) (37)	−325	75	30
Castings							
28Ni–20Cr–2Mo–3Cb	A 351	8	CN7M	(9) (30)	−325	62	25
35Ni–15Cr–Mo	A 351	8	HT30	(36) (39)	−325	65	28
15Cr–15Ni–2Mo–Cb	A 351	8	CF10MC	(9) (30)	−325	70	30
18Cr–8Ni	A 351	8	CF3	(9)	−425	70	30
18Cr–8Ni	A 351	8	CF8	(9) (26) (27) (31)	−425	70	30
25Cr–13Ni	A 351	8	CH10	(9) (27) (31) (35)	−325	70	30
18Cr–10Ni–2Mo	A 351	8	CF8M	(9) (26) (27) (30)	−425	70	30
25Cr–20Ni	A 351	8	HK40	(35) (36) (39)	−325	62	35
18Cr–8Ni	A 351	8	CF3A	(9) (26)	−425	70	35

Material	Spec. no.	P no. (5)	Grade	Class§	Notes	Size range, in	Min temp. (6)
Nickel and nickel alloy (4)							
Pipes and tubes							
Low C Ni	B 161	41	201 (N02201)	Ann.		> 5 OD	−325
Ni	B 161	41	200 (N02200)	Ann.		> 5 OD	−325
Ni–Cu	B 165	42	400 (N04400)	Ann.		> 5 OD	−325
Ni–Cr–Fe	B 167	43	600 (N06600)	H.F. or H.F. Ann.		> 5 OD	−325
Ni–Cu	B 165	42	400 (N04400)	Ann.		≤ 5 OD	−325
Ni–Cr–Fe	B 167	43	600 (N06600)	H.F. or H.F. Ann.		≤ 5 OD	−325
Ni–Fe–Cr	B 407	45	800 (N08800)	C.D. Ann.	(61)		−325
Ni–Cr–Fe–Mo–Cu	B 619	45	G1 (N06007)	Sol. Ann.			−325
Cr–Ni–Fe–Mo–Cu–Cb	B 464	45	20Cb (N08020)	Ann.			−325
Ni–Cr–Fe–Mo–Cu	B 622	45	G (N06007)	Sol. Ann.			−325
Ni–Cr–Mo–Fe	B 619		X (N06002)	Sol. Ann.			−325
Ni–Mo–Cr	B 619	44	C-276 (N10276)	Sol. Ann.			−325
Ni–Mo–Cr	B 622	44	C-276 (N10276)	Sol. Ann.			−325
Ni–Mo	B 619	44	B-2 (N10665)	Sol. Ann.			−325
Ni–Mo	B 619		B (N10001)	Sol. Ann.			−325
Ni–Cr–Mo–Cb	B 444	43	625 (N06625)	Ann.	(64)		−325
Ni–Mo	B 622	44	B-2 (N10655)	Sol. Ann.			−325
Forgings and fittings							
Low C Ni	B 160	41	201 (N02201)	Ann.	(9)	All	−325
Ni	B 160	41	200 (N02200)	H.F.	(9)	All	−325
Ni–Cu	B 164	42	400 (N04400)	Ann. Forg.	(9) (13)	All	−325
	B 366	43	WPNC1 (N06600)		(32)	All	−325

Basic allowable stress *S*, ksi (1), at metal temperature, °F (7)								
Min temp. to 100	200	400	600	800	1,000	1,200	1,400	1,500
20.0	20.0	19.4	17.1	15.5	15.4	6.8	2.3	1.4
23.3	23.3	20.4	18.4	16.2	4.5			
30.0								
16.7	16.7	15.8	14.0	13.0	7.8	3.2	1.1	0.9
16.7	16.7	15.5	13.5	12.4	11.2	6.4	1.8	1.0
20.0								
20.0	20.0	20.0	19.2	17.5	11.0	2.5	0.4	0.2
20.0	20.0	18.6	16.4	15.5	13.8	3.6	0.8	0.3
20.0	20.0	20.0	19.2	17.5	11.0	6.0	1.6	0.8
20.0	20.0	20.0	19.3	18.3	14.0	4.4	1.2	0.8
20.0	20.0	18.6	16.4	15.5	14.0	5.4	1.9	1.1
20.0	20.0	19.3	17.0	15.9	15.3	7.4	2.3	1.3
20.0	20.0	20.0	19.3	18.3	18.0	7.9	2.5	1.3
20.0	20.0	19.3	17.0	15.9	15.3	7.4	2.3	1.3
20.0	20.0	18.7	16.4	15.2	13.8	6.0	2.3	1.4
20.0	20.0	18.7	16.4	15.2	13.8	6.0	2.3	1.4
16.6								
18.6								
20.0								
20.0	20.0	17.6	15.6	14.7				
20.0	20.0	17.6	15.6	14.7	10.7	4.5	1.4	0.7
20.0	20.0	20.0	19.2	17.5	10.5	3.7	1.2	0.7
20.0	20.0	19.4	17.1	15.6	13.1	6.7	2.4	1.5
20.6								
23.2								

		Basic allowable stress *S*, ksi (1), at metal temperature, °F (7)								
SMTS, ksi	SMYS, ksi	Min temp. to 100	200	400	600	800	1,000	1,200	1,400	1,500
50	10	6.7	6.4	6.2	6.2	5.9	3.0	1.2		
55	12	8.0	8.0	8.0	8.0					
70	25	16.7	14.7	13.2	13.2	12.7				
75	25	16.7	16.7	16.7	16.7	16.7	7.0	2.0		
70	28	18.7	16.4	14.8	14.8	14.2				
80	30	20.0	20.0	20.0	20.0	20.0	7.0	2.0		
75	30	20.0	20.0	20.0	20.0	20.0	17.6	6.6	1.1	0.8
90	35	22.5	22.5	21.9	21.1	20.5	18.9			
85	35	23.3	21.3	20.6	20.3	19.2				
90	35	23.5	23.3	23.3	22.7	21.8				
100	40	26.6	24.1	22.9	21.1	19.8	18.6	11.3		
100	41	27.3	27.3	27.3	25.4	22.9	21.8			
100	41	27.5	27.3	27.3	25.4	23.0				
110	51	29.1	28.9	28.9	28.9	28.9				
100	45	30.0	30.0	30.0	30.0	27.6				
120	60	30.0	30.0	28.2	26.4	26.0	26.0	13.2		
110	51	34.2	34.0	34.0	34.0	34.0				
50	10	6.6	6.4	6.2	6.2	5.9	3.0	1.2		
60	15	10.0	10.0	10.0	8.3					
70	25	16.6	14.6	13.2	13.1	12.7				
75	25	16.7	16.7	16.7	16.7	16.7	7.0	2.0		

Table 8.7.15 Basic Allowable Stresses in Tension for Metals (*Continued*)

Material	Spec. no.	P no. (5)	Grade	Class§	Notes	Size range, in	Min temp. (6)
Nickel and nickel alloy (4) (*Cont.*)							
Forgings and fittings (*Cont.*)							
Ni–Cr–Fe	B 166	43	600 (N06600)	H.F.	(9) (13)	All	−325
Ni–Cr–Mo–Fe	B 366		WPHX (N06002)		(32)		−325
Castings							
Ni–Mo–Cr	A 494		CW-12M-1		(9) (44)		−325
Ni–Mo–Cr	A 494		CW-12M-2		(9) (44)		−325

Material	Spec. no.	P no. (5)	Grade	Notes	Min temp. (6)
Unalloyed titanium					
Pipes and tubes					
C.P.	B 337	51	1	(17)	−75
C.P.	B 337	51	2	(17)	−75
C.P.	B 337	52	3	(17)	−75

Material	Spec. no.	P no. (5) (46)	Class	Temper	Size range, in	Notes
Copper and copper alloy						
Pipes and tubes						
Cu tube	B 75	31	C10200, C12000, C12200, C14200	Annealed		(14)
Cu tube	B 88	31	C10200, C12000, C12200	Annealed		(14) (24)
Red brass	B 43	32	C23000	Annealed		(14)
Cu–Ni 90/10	B 467	34	C70600	Annealed	> 4.5 OD	(14)
Cu–Ni 90/10	B 467	34	C70600	Annealed	≤ 4.5 OD	(14)
Cu–Ni 70/30	B 467	34	C71500	Annealed	> 4.5 OD	(14)
Cu	B 42	31	C10200, C1200, C12200	Drawn	NPS 2½ thru 12	(14) (34)
Cu tube	B 88	31	C10200, C12000, C12200	Drawn		(14) (24) (34)
Cu–Ni 70/30	B 466	34	C71500	Annealed		(14)
Forgings						
Cu	B 283	a	C11000			(9) (14)
High Si bronze (A)	B 283	33	C65500			(9) (14)
Forging brass	B 283	a	C37700			(9) (14)
Leaded naval brass	B 283	a	C48500			(9) (14)
Naval brass	B 283	32	C46400			(9) (14)
Mn–bronze (A)	B 283	32	C67500			(9) (14)
Al–Si bronze	B 283	35	C63900			(9) (14)
Castings						
Composition bronze	B 62	a	C83600			(9)
Leaded Ni–bronze	B 584	a	C97600			(9)
Leaded Sn–bronze	B 584	a	C92200			(9)
Steam bronze	B 61	a	C92200			(9)
Sn–bronze	B 584	b	C90300			(9)
Leaded Mn–bronze	B 584	a	C86400			(9)
No. 1 Mn–bronze	B 584	b	C86500			(9)
Al–bronze	B 584	b	C95200			(9)
Si–Al–bronze	B 584	b	C95600			(9)

SMTS, ksi	SMYS, ksi	Basic allowable stress *S*, ksi (1), at metal temperature, °F (7): Min temp. to 100	200	400	600	800	1,000	1,200	1,400	1,500
85	35	23.3	21.2	21.2	21.2	20.4	14.5	5.5		
95	35	23.3	23.3	22.9	21.1	19.8	19.3	11.3		
72	46	24.0	16.4	16.4	16.4					
72	46	24.0	16.4	16.4	16.4	15.5	14.6			

SMTS, ksi	SMYS, ksi	Basic allowable stress *S*, ksi (1), at metal temperature, °F (7): Min temp. to 100	200	300	400	500	600
35	25	11.7	9.7	7.7	6.4	5.3	4.2
50	40	16.7	16.7	12.3	9.8	8.0	7.3
65	55	21.7	19.0	15.6	12.3	9.9	8.0

Min temp. (6)	Specified min strength, ksi: Tensile	Yield	Basic allowable stress *S*, ksi (1), at metal temperature, °F (7): Min temp. to 100	200	300	400	500	600	700
−325	30	9	6.0	4.8	4.7	3.0	0.8		
−325	30	9	6.0	5.9	5.0	2.5	0.8		
−325	40	12	8.0	8.0	8.0	5.0			
−325	38	13	8.7	8.1	7.8	7.5	7.2	6.0	
−325	40	15	10.0	9.5	8.9	8.5	8.0	6.0	
−325	45	15	10.0	9.5	9.1	8.6	8.2	8.0	7.8
−325	36	30	12.0	9.0	8.7	8.2			
−325	36	30	12.0	8.7	8.0	2.5	0.8		
−325	50	18	12.0	11.3	10.8	10.3	9.9	9.6	9.4
−325	33	11	7.3	6.5	5.0	2.5	0.8		
−325	52	18	12.0	10.0	10.0	2.0			
−325	58	23	15.3	12.0	10.5	2.0			
−325	62	24	16.0	15.0	13.0	2.0			
−325	64	26	17.3	15.3	13.0	2.0			
−325	72	34	22.7	12.0	10.5	2.0			
−325	83	41	27.3	17.3	17.1	16.8			
−325	30	14	9.4	9.4	9.1	8.6			
−325	40	17	10.0	7.3	6.3				
−325	34	16	10.6	10.6	10.6	10.3			
−325	34	16	10.6	10.6	10.6	10.3	9.0		
−325	40	18	12.0	9.5	8.5	7.0			
−325	60	20	13.3	12.0	10.5				
−325	65	25	16.6	13.4	10.5				
−325	65	25	16.6	16.1	15.5	14.5	10.0		
−325	60	28	18.8						

Table 8.7.15 Basic Allowable Stresses in Tension for Metals (*Continued*)

Material	Spec. no.	P no. (5) (46)	Class	Temper	Size range, in	Notes
Copper and copper alloy (*Cont.*)						
Al–bronze	B 584	b	C95400			(9)
Mn–bronze	B 584	a	C86700			(9)
High-strength Mn–bronze	B 584	b	C86300			(9)

Material	Spec. no.	P no. (5)	Grade	Temper	Notes	Size or thickness limitations, in
Aluminum alloy						
Seamless pipes and tubes						
	B 210, B 241, B 345	21	1060	0, H112	(14) (33)	
	B 241	21	1100	0, H112	(14) (33)	
	B 210, B 241, B 345	21	3003	H18	(14) (33)	
	B 210, B 241, B 345	21	Alclad 3002	0, H112	(14) (33)	
	B 210	21	Alclad 3003	H18	(14) (33)	
	B 210, B 241	22	5052	0	(14)	
	B 210, B 241, B 345	25	5083	0, H112	(33)	
	B 210, B 241, B 345	25	5086	0, H112	(33)	
	B 210	25	5086	H34	(33)	
	B 210	22	5154	H34	(33)	
	B 241	22	5454	0, H112	(33)	
	B 210, B 241	25	5456	0, H112	(33)	
	B 241	22	5652	0, H112	(33)	
	B 210	23	6061	T4	(33)	
	B 241, B 345	23	6061	T4	(33) (63)	
	B 241, B 345	23	6061	T6	(33)	Pipe < NPS 1
	B 241, B 345	23	6061	T6	(33) (63)	Pipe ≥ NPS 1 all tube
	B 241, B 345	23	6063	T4	(33)	≤ 0.500
	B 241, B 345	23	6063	T5	(33)	≤ 0.500
	B 241, B 345	23	6063	T6	(33)	
	B 210, B 241, B 345	23	6063	T4, T5, T6 Wld.		
Structural tubes						
	B 221	21	1100	0, H112	(14) (33)	
	B 221	21	3003	0, H112	(14) (33)	
	B 221	22	5052	0	(14)	
	B 221	25	5083	0		
	B 221	25	5086	0		
	B 221	22	5454	0		
	B 221	23	6061	T4	(33) (63)	
	B 221	23	6061	T4, T6 Wld.	(22) (63)	
	B 221	23	6063	T4	(33)	≤ 0.500
	B 221	23	6063	T5	(33)	≤ 0.500
	B 221	23	6063	T6	(33)	
	B 221	23	6063	T4, T5, T6 Wld.		
Forgings and fittings						
	B 247	25	5083	0, H112	(9) (33)	
	B 247	23	6061	T6	(9) (33) (45)	
	B 361	21	WP3003	0, H112	(13) (14) (23) (32) (33)	
	B 361	22	WP5154	0, H112	(14) (23) (32) (33)	
	B 361	23	WP6061	T4	(13) (14) (23) (32) (33)	
	B 361	23	WP6061	T4, T6 Wld.	(22) (23) (32)	
	B 361	23	WP6063	T4	(13) (14) (23) (32) (33)	
	B 361	23	WP6063	T4, T6 Wld.	(23) (32)	
Castings						
	B 26		443.0	F	(9) (43)	
	B 26		356.0	T6	(9) (43)	

SOURCE: Adapted from ASME B31.3-1984 with permission.
† Numbers in parentheses refer to notes at end of table. All specifications are ASTM unless noted otherwise.
‡ In the table, SMTS = standard minimum tensile stress, SMYS = standard minimum yield stress.
§ Abbreviations in class column: Ann. = annealed, C.D. = cold drawn, Forg. = forged, H.F. = hot-finished, H.R. = hot-rolled, Plt. = plate, R. = Rolled, Rel. = relieved, Sol. = solution, and Str. = stress.

Min temp. (6)	Specified min strength, ksi		Basic allowable stress *S*, ksi (1), at metal temperature, °F (7)						
	Tensile	Yield	Min temp. to 100	200	300	400	500	600	700
−325	75	30	20.0	18.0	16.3	14.8	11.0		
−325	80	32	21.3	15.3	10.5				
−325	110	60	36.6	19.0	10.5				

Min temp. (6)	SMTS, ksi	SMYS, ksi	Basic allowable stress *S*, ksi (1), at metal temperature, °F (7)						
			Min temp. to 100	150	200	250	300	350	400
−452	8.5	2	1.7	1.7	1.6	1.5	1.3	1.1	0.8
−452	11	3	2.0	2.0	2.0	1.9	1.7	1.3	1.0
−452	27	24	9.0	9.0	8.9	6.3	5.4	3.5	2.5
−452	13.5	4.5	3.0	3.0	3.0	2.8	2.2	1.6	1.3
−452	26	23	8.1	8.1	8.0	5.7	4.9	3.2	2.2
−452	25	10	6.7	6.7	6.7	6.2	5.6	4.1	2.3
−452	39	16	10.7	10.7					
−452	35	14	9.3	9.3					
−452	44	34	14.7	14.7					
−452	39	29	13.0	13.0					
−452	31	12	8.0	8.0	8.0	7.4	5.5	4.1	3.0
−452	41	19	12.7	12.7					
−452	25	10	6.7	6.7	6.7	6.2	5.6	4.1	2.3
−452	30	16	10.0	10.0	10.0	9.8	9.2	7.9	5.6
−452	26	16	8.7	8.7	8.7	8.5	8.0	7.9	5.6
−452	42	35							
−452	38	35	12.7	12.7	12.7	12.1	10.6	7.9	5.6
−452	19	10	6.7	6.7	6.7	6.7	6.7	3.4	2.0
−452	22	16	7.3	7.3	7.2	6.8	6.1	3.4	2.0
−452	30	25	10.0	10.0	9.8	9.0	6.6	3.4	2.0
−452	17		5.7	5.7	5.7	5.6	5.2	3.0	2.0
−452	11	3	2.0	2.0	2.0	1.9	1.7	1.3	1.0
−452	14	5	3.3	3.3	3.3	3.1	2.4	1.8	1.4
−452	25	10	6.7	6.7	6.7	6.2	5.6	4.1	2.3
−452	39	16	10.7	10.7					
−452	35	14	9.3	9.3					
−452	31	12	8.0	8.0	8.0	7.4	5.5	4.1	3.0
−452	26	16	8.7	8.7	8.7	8.5	8.0	7.7	5.3
−452	24		8.0	8.0	8.0	7.9	7.4	6.1	4.3
−452	19	10	6.4	6.4	6.4	6.4	6.4	3.4	2.0
−452	22	16	7.3	7.3	7.2	6.8	6.1	3.4	2.0
−452	30	25	10.0	10.0	9.8	9.0	6.6	3.4	2.0
−452	17		5.7	5.7	5.7	5.6	5.2	3.0	2.0
−452	39	16	10.7	10.7					
−452	38	35	12.7	12.7	12.7	12.1	10.6	7.9	5.6
−452	14	5	3.3	3.3	3.3	3.1	2.4	1.8	1.4
−452	30	11	7.3	7.3					
−452	26	16	8.7	8.7	8.7	8.5	8.0	7.7	5.6
−452	24		8.0	8.0	8.0	7.9	7.4	6.1	4.3
−452	18	9	6.0	6.0	6.0	6.0	6.0	3.4	2.0
−452	17		5.7	5.7	5.7	5.6	5.2	3.0	2.0
−452	17	6	4.0	4.0	4.0	4.0	4.0	4.0	3.0
−452	30	20	10.0	10.0	10.0	8.4			

Table 8.7.15 Basic Allowable Stresses in Tension for Metals (*Continued*)

NOTES: These notes are requirements of the Code. Those marked with an asterisk (*) restate requirements found in the text of the Code. The other notes are limitations or special requirements applicable to particular materials. At this time, metric equivalents have not been provided in the stress tables for metals. (P-number groupings, tables, and appendixes cited in these notes for Table 8.7.15 will be found in the basic reference ASME B31.3.)

(1)* The stress values in Table A-1 and the design stress values in Table A-2 are basic allowable stresses in tension in accordance with 302.3.1(a). For pressure design, the stress values from Table A-1 are multiplied by the appropriate quality factor E (E_c from Table A-1A, or E_j from Table A-1B). Stress values in shear and bearing are stated in 302.3.1(b); those in compression in 302.3.1(c).

(2)* The quality factors for castings E_c in Table A-1A are basic factors in accordance with 302.3.3(b). The quality factors for longitudinal weld joints E_j in Table A-1B are basic factors in accordance with 302.3.4(a). See 302.3.3(c) and 302.3.4(b) for enhancement of quality factors. See also 302.3.1(a), footnote 9.

(3)* This casting quality factor can be enhanced by supplementary examination in accordance with 302.3.3(c) and Table 302.3.3C. The higher factor from Table 302.3.3C may be substituted for this factor in pressure design equations.

(4)* In shaded areas, stress values printed in *italics* exceed two-thirds of the expected yield strength at temperature. All other stress values in shaded areas are equal to 90% of expected yield strength at temperature. See 302.3.2(d)(4) and 302.3.2(d) [Note (3)].

(5)* See 327.5.2 for description of P-number groupings.

(6)* The minimum temperature shown is that design minimum temperature for which the material is normally suitable without impact testing other than that required by the material specification. However, the use of a material at a design minimum temperature below − 20°F (− 29°C) is established by rules elsewhere in this Code, including any necessary impact test requirements.

(7)* A single bar (|) in these stress tables indicates there are conditions other than stress which affect usage above or below the temperature, as described in other referenced notes. A double bar (‖) after a tabled stress indicates that use of the material is prohibited above that temperature. A double bar (‖) before the stress value for ''Min. temp. to 100°F'' indicates that the use of the material is prohibited below the listed minimum temperature. At temperatures where there are no stress values, the material may be used in accordance with 323.2 unless prohibited by a double bar (‖).

(8)* There are restrictions on the use of this material in the text of the Code.

(9)* Pressure-temperature ratings of cast and forged parts as published in standards referenced in this Code section may be used for parts meeting requirements of these standards. Allowable stresses for castings and forgings, where listed, are for use in design of special components not furnished in accordance with such standards.

(10)* These casting quality factors are applicable only when proper supplementary examination has been specified (see 302.3.3).

(11) For use under this Code, radiography shall be performed after heat treatment.

(12)* Certain forms of this material, as stated in Table 323.2.2, must be impact-tested to qualify for service below − 20°F (− 29°C). Alternatively, if provisions for impact testing are included in the material specification as supplementary requirements and are invoked, the material may be used down to the temperature at which the test was conducted in accordance with the specification.

(13) Properties of this material vary with thickness or size. Stresses are based on minimum properties for the thickness listed.

(14) For use in Code piping at the stated stress values, the required minimum tensile and yield properties must be verified by tensile test at the mill. If such tests are not required by the material specification, they shall be specified in the purchase order.

(15) These stress values are established from a consideration of strength only and will be satisfactory for average service. For bolted joints where freedom from leakage over a long period of time without retightening is required, lower stress values may be necessary as determined from the flexibility of the flange and bolts and corresponding relaxation properties.

(16) This joint factor shall be applied to stress values in Table A-1 unless the longitudinal welded joint has been 100% radiographed as required by the specification for fabricated welds.

(17)* Filler metal shall not be used in the manufacture of this pipe or tube.

(18)* This specification does not include requirements for 100% radiographic inspection. If this higher joint factor is to be used, the material shall be purchased to the special requirements of Table 327.4.1A for longitudinal butt welds with 100% radiography in accordance with Table 302.3.4.

(19)* This specification includes requirements for random radiographic inspection for mill quality control. If the 0.90 joint factor is to be used, the welds shall meet the requirements of Table 327.4.1A for longitudinal butt welds with spot radiography in accordance with Table 302.3.4. This shall be a matter of special agreement between purchaser and manufacturer.

(20) For pipe sizes NPS 8 and larger and for wall thicknesses of Schedule 140 or heavier, the minimum specified tensile strength is 70.0 ksi (483 MPa).

(21) For material thickness greater than 5 in (125 mm), the minimum specified tensile strength shall be 70.0 ksi (483 MPa).

(22) The minimum tensile strength for weld (qualification) and stress values shown shall be multiplied by 0.90 for pipe having an outside diameter less than 2 in (51 mm) and a D/t value less than 15. This requirement may be waived if it can be shown that the welding procedure to be used will consistently produce welds that meet the listed minimum tensile strengths of 24.0 ksi (165 MPa).

(23) Stress values apply only to fittings made from seamless material conforming to ASTM B 210 or B 241. Otherwise, the value of factor E_j shall be selected from Table 302.3.4 for the appropriate construction.

(24) Yield strengths listed are not included in the material specifications. The value shown is based on yield strengths of materials with similar characteristics.

(26) These unstabilized grades of stainless steel have increasing tendency to intergranular carbide precipitation as the carbon content increases above 0.03%.

(27) For temperatures above 800°F (425°C), these stress values apply only when the carbon content is 0.04% or higher.

(28) For temperatures above 1,000°F (538°C), these stress values apply only when the carbon content is 0.04% or higher.

(29) The higher stress values at 1,050°F (566°C) and above for this material shall be used only when the steel has an austenitic micrograin size no. 6 or less (coarser grain) as defined in ASTM E 112. Otherwise, the lower stress values shall be used.

(30) For temperatures above 1,000°F (538°C), these stress values may be used only if the material has been heat-treated at a temperature of 2,000°F (1,090°C) minimum.

(31) For temperatures above 1,000°F (538°C), these stress values may be used only if the material has been heat-treated by heating to a minimum temperature of 1,000°F (1,040°C) and quenching in water or rapidly cooling by other means.

(32) Stress values shown are for the lowest-strength base material permitted by the specification to be used in the manufacture of this grade of fitting. If a higher strength base material is used, the higher stress values for that material may be used in design.

(33) For welded construction with work-hardened grades, use the stress values for annealed material; for welded construction with precipitation hardened grades, use the special stress values for welded construction given in the table.

(34) After use above the temperature indicated by a single bar (|), use at a lower temperature shall be based on the stress values allowed for the annealed condition of the material.

(35) These steels are intended for use at high temperatures; however, they may have low ductility and/or low impact properties at room temperature after being used above the temperature indicated by the single bar (|).

(36) The specification permits this material to be furnished without solution heat treatment or with other than a solution heat treatment. When the material has not been solution heat-treated, the minimum temperature shall be − 20°F (− 29°C) unless the material is impact tested per 323.3.

(37) Impact requirements for seamless fittings shall be governed by those listed in this table for the particular basic material specification in the grades permitted (A 312, A 240, and A 182). When A 276 materials are used in the manufacture of these fittings, the notes, minimum temperatures, and allowable stresses for comparable grades of A 240 materials shall apply.

(38) For use at temperatures below − 20°F through − 50°F (− 29°C through − 45°C), this material must be quenched and tempered.

(39) This material when used below − 20°F (− 29°C) requires impact testing if the carbon content is above 0.10%.

(40) The stress values for austenitic stainless steels in this table may not be applicable if the material has been given a final heat treatment other than that required by the material specification and any overriding requirements of this Code called for by note (30) or (31).

(41) Design stresses for the cold-drawn temper are based on hot-rolled properties until required data on cold-drawn are submitted.

(42) This is a product specification. No design stresses are necessary. Limitations on metal temperature for materials covered by this specification are:

	°F	°C		°F	°C
Grades 1 and 2	− 20 to 900	− 29 to 480	Grade 6	− 20 to 800	− 29 to 425
Grade 2H	− 50 to 1,100	− 45 to 595	Grade 8FA [see note (39)]	− 20 to 800	− 29 to 425
Grade 3	− 20 to 1,100	− 29 to 595	Grades 8MA and 8TA	− 325 to 1,500	− 198 to 815
Grade 4	− 150 to 1,100	− 100 to 595	Grades 8A and 8CA	− 425 to 1,500	− 254 to 815

(43)* The stress values given for this material are not applicable when either welding or thermal cutting is employed [see 323.4.2(c)].

(44) This material shall not be welded.

(45) Stress values shown are applicable for ''die'' forgings only.

(46) The letter ''a'' indicates alloys which are not recommended for welding and which, if welded, must be individually qualified. The letter ''b'' indicates copper-base alloys which must be individually qualified.

(47) If no welding is employed in fabrication of piping from these materials, the stress values may be increased to 33.3 ksi (230 MPa).

(48) The stress value to be used for this gray cast iron material at its upper temperature limit of 450°F (232°C) is the same as that shown in the 400°F (204°C) column.

(49) If the chemical composition of this grade is such as to render it hardenable, qualification under P no. 6 is required.

(50) This material is grouped in P no. 7 because its hardenability is low.

(51) Special P-numbers SP-1, SP-2, and SP-3 of carbon steels are not included in P no. 1 because of possible high-carbon, high-manganese combination which would require special consideration in qualification. Qualification of any high carbon, high manganese grade may be extended to other grades in its group.

(52) Copper-silicon alloys are not always suitable when exposed to certain media and high temperature, particularly above 212°F (100°C). The user should satisfy himself that the alloy selected is satisfactory for the service for which it is to be used.

(53) Stress relief heat treatment is required for service above 450°F (232°C).

Table 8.7.15 Basic Allowable Stresses in Tension for Metals (*Continued*)

(54) The maximum operating temperature is arbitrarily set at 500°F (260°C) because harder temper adversely affects design stress in the creep rupture temperature ranges.
(55) Pipe produced to this specification is not intended for high-temperature service. The stress values apply to either nonexpanded or cold-expanded material in the as-rolled, normalized, or normalized and tempered condition.
(56) Because of thermal instability, this material is not recommended for service above 800°F (425°C).
(57)* Conversion of carbides to graphite may occur after prolonged exposure to temperatures over 800°F (425°C) (see App. F).
(58)* Conversion of carbides to graphite may occur after prolonged exposure to temperatures over 875°F (468°C) (see App. F).
(59)* For temperatures above 900°F (480°C), consider the advantages of killed steel (see App. F).
(60) For all design temperatures, the maximum hardness shall be Rockwell C35 immediately under the thread roots. The hardness shall be taken on a flat area at least ⅛ in (3 mm) across, prepared by removing threads. No more material than necessary shall be removed to prepare the area. Hardness determination shall be made at the same frequency as tensile tests.
(61) Annealed at approximately 1,800°F (980°C).
(62) Annealed at approximately 2,000°F (1,150°C).
(63) For stress-relieved tempers (T351, T3510, T3511, T451, T4510, T4511, T651, T6510, T6511), stress values for material in the listed temper shall be used.
(64) The minimum tensile strength of the reduced section tensile specimen in accordance with QW-462.1 of BPV Code, Section IX, shall not be less than 110.0 ksi (758 MPa).
(65) The minimum temperature shown is for the heaviest wall permissible by the specification. The minimum temperature for lighter walls shall be as shown in the following tabulation:

Spec. and grade	Temp. for plate thicknesses shown					
	°F			°C		
	1 in max	2 in max	Over 2 to 3 in	25 mm max	50 mm max	Over 50 to 76 mm
A 203, A	−90	−90	−75	−68	−68	−60
A 203, B	−90	−90	−75	−68	−68	−60
A 203, D	−150	−150	−125	−101	−101	−87
A 203, E	−150	−150	−125	−101	−101	−87

(66) Stress values shown are 90% of those for the corresponding core material.
(67) For use under this Code, the heat treatment requirements for pipe manufactured to ASTM A 671, A 672, and A 691 shall be as required by 331 for the particular material being used. In some cases, 331 does not require heat treatment. In these cases, if the user requires no radiography, the designation shall be class 13. If 100% radiography is required, it shall be class 12. If heat treatment is required by the plate thickness or by the engineering design, the designation shall be class 23 (no radiography) or class 22 (100% radiography).
(68) The tension test specimen from plate 0.500 in (12.7 mm) and thicker is machined from the core and does not include the cladding alloy; therefore, the stress values listed are those for materials less than 0.500 in (12.7 mm).

Table 8.7.16 Basic Casting Quality Factors E_c
Abstracted from ASME B31.3 (1984) with permission

Spec. no.	Description	E_c (2)*	Notes*
Iron			
FS-WW-P421c	Centrifugally cast pipe	1.00	
A 377	Centrifugally cast pipe	1.00	
A 47	Malleable iron castings	1.00	
A 48	Gray iron castings	1.00	
A 126	Gray iron castings	1.00	
A 197	Cupola malleable iron castings	1.00	
A 278	Gray iron castings	1.00	
A 338	Malleable iron castings	1.00	
A 395	Ductile and ferritic ductile iron castings	0.80	(3)
A 571	Austenitic ductile iron castings	0.80	(3)
Carbon steel			
A 216	Carbon steel castings	0.80	(3)
A 352	Ferritic steel castings	0.80	(3)
Low and intermediate alloy steel			
A 426	Centrifugally cast pipe	1.00	(10)
A 217	Martensitic stainless and alloy castings	0.80	(3)
A 352	Ferritic steel castings	0.80	(3)
Stainless steel			
A 451	Centrifugally cast pipe	0.90	(3) (10)
A 452	Centrifugally cast pipe	0.85	(3)
A 351	Austenitic steel castings	0.80	(3)
Copper and copper alloy			
B 61	Steam bronze castings	0.80	(3)
B 62	Composition bronze castings	0.80	(3)
B 148	Al–bronze and Si–Al–bronze castings	0.80	(3)
B 584	Copper alloy castings	0.80	(3)
Nickel and nickel alloy			
A 494	Nickel and nickel alloy castings	0.80	(3)
Aluminum alloy			
B 26, temper F	Aluminum alloy castings	1.00	(10)
B 26, temper T6, T71	Aluminum alloy castings	0.80	(3)

* See Notes at end of Table 8.7.15.

Table 8.7.17 Basic Quality Factors for Longitudinal Weld Joints in Pipes, Tubes, and Fittings, E_j

Spec. no.	Class material	Description	E_j (2)*	Notes*
Carbon steel				
API 5L		Seamless	1.00	
		Electric resistance welded	0.85	
		Electric fusion welded, double butt, straight or spiral	0.85	
		Furnace butt welded	0.60	
A 53	Type S	Seamless	1.00	
	Type E	Electric resistance welded	0.85	
	Type F	Furnace butt welded	0.60	
A 105		Forgings and fittings	1.00	
A 106		Seamless	1.00	
A 120		Seamless	1.00	
		Electric resistance welded	0.85	
		Furnace butt welded	0.60	
A 134		Electric fusion welded, single butt, straight or spiral	0.80	
A 135		Electric resistance welded	0.85	
A 139		Electric fusion welded, straight or spiral	0.80	
A 179		Seamless	1.00	
A 181		Forgings and fittings	1.00	
A 211		Spiral welded	0.75	
A 234		Seamless and welded fittings	1.00	
A 333		Seamless	1.00	
		Electric resistance welded	0.85	
A 334		Seamless	1.00	
A 350		Forgings and fittings	1.00	
A 369		Seamless	1.00	
A 381		Electric fusion welded, 100% radiograph	1.00	(18)
		Electric fusion welded, spot radiograph	0.90	(19)
		Electric fusion welded, as manufactured	0.85	
A 420		Welded fittings, 100% radiograph	1.00	
A 524		Seamless	1.00	
A 587		Electric resistance welded	0.85	
A 671	12, 22	Electric fusion welded, 100% radiograph	1.00	
	13, 23	Electric fusion welded, double butt	0.85	

Table 8.7.17 Basic Quality Factors for Longitudinal Weld Joints in Pipes, Tubes, and Fittings, E_j (*Continued*)

Spec. no.	Class material	Description	E_j (2)*	Notes*
A 672	12, 22	Electric fusion welded, 100% radiograph	1.00	
	13, 23	Electric fusion welded, double butt	0.85	
A 691	12, 22	Electric fusion welded, 100% radiograph	1.00	
	13, 23	Electric fusion welded, double butt	0.85	
Low and intermediate alloy steel				
A 182		Forgings and fittings	1.00	
A 234		Seamless and welded fittings	1.00	
A 333		Seamless	1.00	
		Electric resistance welded	0.85	
A 334		Seamless	1.00	
A 335		Seamless	1.00	
A 350		Forgings and fittings	1.00	
A 369		Seamless	1.00	
A 420		Welded fittings, 100% radiograph	1.00	
A 671	12, 22	Electric fusion welded, 100% radiograph	1.00	
	13, 23	Electric fusion welded, double butt	0.85	
A 672	12, 22	Electric fusion welded, 100% radiograph	1.00	
	13, 23	Electric fusion welded, double butt	0.85	
A 691	12, 22	Electric fusion welded, 100% radiograph	1.00	
	13, 23	Electric fusion welded, double butt	0.85	
Stainless steel				
A 182		Forgings and fittings	1.00	
A 268		Seamless	1.00	
		Electric fusion welded, double butt	0.85	
		Electric fusion welded, single butt	0.80	
A 269		Seamless	1.00	
		Electric fusion welded, double butt	0.85	
		Electric fusion welded, single butt	0.80	
A 312		Seamless	1.00	
		Electric fusion welded, double butt	0.85	
		Electric fusion welded, single butt	0.80	
A 358	1, 3, 4	Electric fusion welded, 100% radiograph	1.00	
	5	Electric fusion welded, spot radiograph	0.90	
	2	Electric fusion welded, double butt	0.85	
A 376		Seamless	1.00	
A 403		Seamless fitting	1.00	
		Welded fitting, 100% radiograph	1.00	(16)
		Welded fitting, double butt	0.85	
		Welded fitting, single butt	0.80	
A 409		Electric fusion welded, double butt	0.85	
		Electric fusion welded, single butt	0.80	
A 430		Seamless	1.00	
Copper and copper alloy				
B 42		Seamless	1.00	
B 43		Seamless	1.00	
B 68		Seamless	1.00	
B 75		Seamless	1.00	
B 88		Seamless	1.00	
B 466		Seamless	1.00	
Copper and copper alloy (*Cont.*)				
B 467		Electric resistance welded	0.85	
		Electric fusion welded, double butt	0.85	(16)
		Electric fusion welded, single butt	0.80	(16)
Nickel and nickel alloy				
B 160		Forgings and fittings	1.00	
B 161		Seamless	1.00	
B 164		Forgings and fittings	1.00	
B 165		Seamless	1.00	
B 166		Forgings and fittings	1.00	
B 167		Seamless	1.00	
B 366		Seamless and welded fittings	1.00	(16)
B 407		Seamless	1.00	
B 444		Seamless	1.00	
B 619		Electric resistance welded	0.85	
		Electric fusion welded, double butt	0.85	
		Electric fusion welded, single butt	0.80	
Unalloyed titanium				
B 337		Seamless	1.00	
		Electric fusion welded, double butt	0.85	
Aluminum alloy				
B 210		Seamless	1.00	
B 241		Seamless	1.00	
B 247		Forgings and fittings	1.00	
B 345		Seamless	1.00	
B 361		Seamless fittings	1.00	

* See Notes at end of Table 8.7.15.
SOURCE: Abstracted from ASME B31.3 (1984) with permission

hammer allowance in case of cast-iron conveying liquids); D = OD of pipe, in; S = maximum allowable stress in material due to internal pressure, lb/in²; E = quality factor, y = a coefficient, values for which are listed in Table 8.7.18; A = allowance for threading, mechanical strength, and corrosion, in, with values of A listed in Table 8.7.19.

The thickness of ductile-iron pipe conveying liquid may be taken from Table 8.7.24, using the pressure class next higher than the maximum internal service pressure in pounds per square inch. Where

Table 8.7.18 Values of *y*
(Interpolate for intermediate values) (ASME B31.1)

	Temp, °F (°C)					
	900 (482) and below	950 (510)	1,000 (538)	1,050 (566)	1,100 (593)	1,150 (621) and above
Ferritic steels	0.4	0.5	0.7	0.7	0.7	0.7
Austenitic steels	0.4	0.4	0.4	0.4	0.5	0.7

Table 8.7.19 Values of *A*
(ASME B31.1)

Type of pipe	A, in
Cast-iron pipe, centrifugally cast	0.14
Cast-iron, pit-cast	0.18
Threaded-steel, wrought-iron, or nonferrous pipe:	
⅜ in and smaller	0.05
½ in and larger	Depth of thread
Grooved-steel, wrought-iron or nonferrous pipe	Depth of groove
Plain-end steel, wrought-iron or tube:	
1 in and smaller	0.05
1¼ in and larger	0.065
Plain-end nonferrous pipe or tube	0.000

ductile-iron pipe is used for steam service, the thickness should be calculated by Eq. (8.7.1).

Plain-end pipe includes pipe joined by flared compression couplings, lapped joints, and by welding, i.e., by any method that does not reduce the wall thickness of the pipe at the joint.

Physical and Chemical Properties of Pipes, Tubes, etc. The design of piping for operation above 750°F (399°C) presents many problems not encountered at lower temperatures. For the properties of steel applicable to high-temperature service (as well as to ordinary service) for pipes, tubes, fittings, bolting material, etc., see Sec. 6. For a discussion of creep properties, see Sec. 5.

Piping of thickness designed in accordance with Eq. (8.7.1) may be used for any combination of pressure and temperature for which *S* and *E* values are listed in Tables 8.7.15 to 8.7.17. The following summarizes piping industry practice.

Steam Pressures above 250 lb/in² (1,724 kPa), and Not above 2,500 lb/in² (17,238 kPa), Temperatures Not above 1,100°F (593°C) For pressures in excess of 100 lb/in² (690 kPa), the pipe may be seamless steel (A106), (A312), (A335), or (A376); or electric-fusion-welded steel (A691); or forged-and-bored steel (A369); or automatic-welded steel (A312). For pressures between 250 and 600 lb/in² (9,224 and 22,137 N/m²) the pipe may be seamless steel (A106) or (A53); electric-fusion-welded steel (A155); electric-resistance-welded steel (A135) or (A53). For pressures of 250 lb/in² (1,724 kPa) and lower and for service up to 750°F (399°C), any of the following may be used: electric-fusion-welded steel (A134) or (A139); electric-resistance-welded steel (A135); seamless or welded steel (A53). Grade A seamless pipe (A106) or (A53); or grade A electric-welded pipe (A53), (A135), or (A139) is used for close coiling or cold bending. Pipe permissible for services specified may be used for temperatures higher than 750°F (399°C), unless otherwise prohibited, if the *S* and *E* values of Tables 8.7.15 to 8.7.17 are used when calculating the required wall thickness.

Because of several failures in seam-welded 1¼% Cr–½% Mo or 2¼% Cr–1% Mo piping produced to ASTM Specification A155 operating at temperatures above 950°F, a preference has developed for seamless piping in these applications. Nevertheless, in general, seam-welded piping has provided entirely satisfactory service in high-temperature applications above 950°F.

Valves and fittings must have flange openings or welded ends, and valves must have external stem threads. Valves must be of cast or forged steel or may be fabricated from plate and pipe. Valves of nonferrous materials are generally cast or forged. Forged and cast-steel threaded valves and fittings may be used up to 300 lb/in² and 500°F for 3 (2) [1½] in pipe, and pressure from 250 to 400 (400 to 600) [600 to 2,500] lb/in². Malleable-iron threaded fittings (300 lb/in²) may be used for pressures not greater than 300 lb/in² and temperatures not over 500°F. Valves 8 in and larger should have the bypass of at least ¾ in, commercial size.* Welded fittings may be used of the same material and thickness as the pipe to which they are to be connected.

Steam Pressures from 125 to 250 lb/in² (862 to 1,724 kPa), Temperature Not above 450°F (232°C) Pipe may be electric-fusion-welded steel (A134 or A139). Copper and brass may be used if the temperature does not exceed 406°F. Cast iron may also be used. For close coiling or cold bending, grade A seamless steel (A53); or grade A electric-welded steel (A53), (A135), or (A139) is suitable. Pipe permissible for this service may be used for temperatures above 450°F (232°C) if the proper *S* and *E* are used in calculating the pipe-wall thickness.

Valves below 3 in may have inside stem screws. Stop valves 8 in and over must be bypassed. Bodies, bonnets, and yokes are of cast iron, malleable iron, steel, bronze, brass, or Monel. Flanged-steel fittings must conform to the class 300 ANSI Standard B16.5; if of cast iron, to the class 250 ANSI Standard B16.1; or, for threaded fittings, to the ANSI Standard B16.4. Malleable-iron threaded fittings must conform to the class 300 ANSI B16.3 standard, except that the class 150 ANSI Standard B16.3 may be used for pressures not greater than 150 lb/in². Welded fittings may be used.

* See Manufacturers Standardization Society SP-45 for recommended size of bypass valves.

Steam Pressures from 25 to 125 lb/in² (172 to 802 kPa) Temperatures Not above 450°F Pipe may be of steel, ductile iron, copper, or brass; valve bodies of cast iron, malleable iron, ductile iron, steel, or brass. Fittings are of class 125 or class 150 American Standard cast iron with screwed or flanged ends, or of ductile or malleable iron with screwed ends.

Steam Pressures 25 lb/in² (172 kPa) and Less, Temperature up to 450°F Pipe may be of steel, brass, copper, or cast iron. Flanged fittings conform to the class 25 ANSI Standard B16.1. Screwed fittings are of the class 125 ANSI Standard B16.4 or of the class 150 ANSI Standard B16.3 for malleable iron, or conform to B16.15 for cast bronze. Welded-steel fittings are extensively used.

Pipe coils are made from any of the commercial sizes of iron, steel, brass, and copper pipe and tubing. Limiting center-to-center dimensions, to which pipe coils can be fabricated in sizes ¾ to 2 in, are given in Table 8.7.20. Steel tubing cannot be bent to the absolute limits of brass or copper.

Table 8.7.20 Center-to-Center Dimensions of Pipe Coils

Nominal pipe size, in	Recommended and advisable minimum, in	
	Schedule 40	Schedule 80
¾	3½	2½
1	4	3
1¼	5	4
1½	6	5
2	8	6

Seamless mechanical tubing is obtainable in outside diameters ranging from ¼ to 10¾ in and in wall thickness from 20 gage to 2 in (0.091 to 5.08 cm). Oval, square, rectangular, and other special shapes can be obtained in various sizes and wall thicknesses. Mechanical tubing is available either hot-finished or cold-drawn, but is furnished principally cold-drawn. It is readily adaptable to varied treatment by expansion, cupping, tapering, swaging, flanging, coiling, welding, and similar manipulations. Typical of the many uses are aircraft tubing, automobile axle housings, driveshafts, drive-shaft housings, tie rods, steering columns, piston rods and pins, gear rings, roller-bearing cases and cones, cylinders for various purposes, machine parts, sleeves, bushings, spacers, surgical instruments, and hypodermic needles. Table 8.7.21 lists weights and dimensions of round seamless-steel tubing for sizes that have by common usage become standard. Detailed information on mechanical tubing for any particular applications can be obtained from manufacturers.

Dimensions and weights of **condenser** and **heat-exchanger tubes** are given in Table 8.7.22 and of **boiler** tubes in Table 8.7.23.

Spiral Pipe Spiral pipe is strong lightweight steel pipe with a single continuous welded helical seam from end to end stiffening it throughout. It is listed in sizes 6- to 42-in ID (15.24- to 106.7-cm), in various thicknesses, and in lengths up to 40 ft (12.19 m). It is used for high- and low-pressure water lines, vacuum lines, exhaust-steam lines, low-pressure air lines, sand and gravel slurry conveying and similar services. It is also used extensively by the petroleum industry, for oil and gas lines, for low-pressure steam lines, etc.

Spiral pipe may be asphalt-coated or galvanized. The pipe is designed for special joints, flanges, and lightweight fittings, but the ANSI flanges and fittings can be furnished, if desired.

The **sleeve-type coupling** illustrated in Fig. 8.7.1 is particularly suitable for plain-end pipe and is widely used. A gasket is used to make a tight joint. Advantages of this coupling are low cost, the use of unskilled labor in making the connections, and the fact that small changes in alignment and grade can be made with regular straight lengths of pipe by a movement in the coupling. This type of coupling is used extensively in long oil lines.

Table 8.7.21 Approx Weight of Round Seamless Cold-Finished Carbon-Steel Mechanical Tubing, lb/ft*
(Carbon 0.25% max. Standard sizes for warehouse stocks random lengths. United States Steel Corporation)

Wall thickness		OD, in																	
in	g or in	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2
0.035	20 g	0.127	0.174	0.221	0.267	0.314	0.361	0.407	0.454	—	0.548								
0.049	18 g	0.171	0.236	0.301	0.367	0.432	0.498	0.563	0.629	0.694	0.759	0.825	0.890	—	1.02				
0.058	17 g	—	0.274	0.351	0.429	—	0.584												
0.065	16 g	0.215	0.302	0.389	0.476	0.562	0.649	0.736	0.823	0.909	0.996	1.08	1.17	1.26	1.34	1.43	1.52	1.60	1.69
0.083	14 g	—	0.370	0.480	0.591	0.702	0.813	0.924	1.03	1.15	1.26	—	—	—	1.70				
0.095	13 g	—	0.411	0.538	0.665	0.791	0.918	1.05	1.17	1.30	1.43	1.55	1.68	1.81	1.93	2.06	2.19	2.31	2.44
0.109	12 g	—	0.455	0.601	0.746	0.892	1.04	1.18	1.33	—	1.62								
0.120	11 g	—	0.487	0.647	0.807	0.968	1.13	1.29	1.45	1.61	1.77	1.93	2.09	2.25	2.41	2.57	2.73	2.89	3.05
0.134	10 g	—	—	0.703	0.882	—	1.24	1.42	1.60	1.78	1.96	2.13	2.31	—	2.67				
0.156	5/32	—	—	0.781	0.990	1.20	1.41	1.61	1.82	2.03	2.24	2.45	2.66	2.86	3.07	3.28	3.49	3.70	3.91
0.188	3/16	—	—	—	1.13	1.38	1.63	1.88	2.13	2.38	2.63	2.89	3.14	3.39	3.64	3.89	4.14	4.39	4.64
0.219	7/32	—	—	—	—	1.53	1.83	2.12	2.41	2.70	3.00	3.29	3.58	3.87	4.17	4.46	4.75	5.04	5.34
0.250	1/4	—	—	—	1.34	1.67	2.00	2.34	2.67	3.00	3.34	3.67	4.01	4.34	4.67	5.01	5.34	5.67	6.01
0.281	9/32	—	—	—	—	—	—	—	2.91	—	3.66	4.03	4.41	4.78	5.16	—	5.91	6.28	6.66
0.313	5/16	—	—	—	—	—	—	—	3.13	3.55	3.97	4.39	4.80	5.22	5.64	6.06	6.48	6.89	7.31
0.375	3/8	—	—	—	—	—	—	—	3.50	4.01	4.51	5.01	5.51	6.01	6.51	7.01	7.51	8.01	8.51
0.438	7/16	—	—	—	—	—	—	—	—	—	4.97	5.53	6.14	6.72	7.31	7.89	8.48	9.06	9.65
0.500	1/2	—	—	—	—	—	—	—	—	—	5.34	6.01	6.68	7.34	8.01	8.68	9.35	10.0	10.7
0.625	5/8	—	—	—	—	—	—	—	—	—	—	—	—	—	9.18	—	10.8	11.7	12.5

* Other standard sizes, in certain standard wall thicknesses, vary by 1/8-in increments for 2 1/2 to 3 1/2 in; by 1/4-in increments from 3 1/2 to 7 1/2 in; 1/2-in increments from 7 1/2 to 10 1/2 in OD. There are also standard sizes for every 1/16 in from 3/8 to 1 5/8 in OD.
To obtain weights in kg/m, multiply tabular values shown by 1.42.

Table 8.7.22 Steel Condenser and Heat-Exchanger Tubes
(Dimensions and weights. United States Steel Corporation)

		Avg wall			Min wall		
OD, in	Thickness, in	ID, in	Area of metal, in^2*	Weight per ft, lb†	ID, in	Area of metal, in^2*	Weight per ft, lb†
1/2	0.035	0.430	0.0511	0.1738	0.423	0.0558	0.1898
	0.050	0.400	0.0707	0.2403	0.390	0.0769	0.2614
	0.065	0.370	0.0888	0.3020	0.357	0.0963	0.3272
5/8	0.035	0.555	0.0649	0.2205	0.548	0.0709	0.2412
	0.050	0.525	0.0903	0.3071	0.515	0.0985	0.3348
	0.065	0.495	0.1144	0.3888	0.482	0.1243	0.4227
	0.085	0.455	0.1442	0.4902	0.438	0.1561	0.5308
3/4	0.050	0.650	0.1100	0.3738	0.640	0.1201	0.4082
	0.065	0.620	0.1399	0.4755	0.607	0.1524	0.5181
	0.085	0.580	0.1776	0.6037	0.563	0.1928	0.6556
	0.095	0.560	0.1955	0.6646	0.541	0.2119	0.7204
7/8	0.050	0.775	0.1296	0.4406	0.765	0.1417	0.4817
	0.065	0.745	0.1654	0.5623	0.732	0.1805	0.6136
	0.085	0.705	0.2110	0.7172	0.688	0.2296	0.7804
	0.095	0.685	0.2328	0.7914	0.666	0.2530	0.8599
1	0.050	0.900	0.1492	0.5073	0.890	0.1633	0.5551
	0.065	0.870	0.1909	0.6491	0.857	0.2086	0.7090
	0.085	0.830	0.2443	0.8306	0.813	0.2663	0.9052
	0.095	0.810	0.2701	0.9182	0.791	0.2940	0.9994
1 1/4	0.050	1.150	0.1885	0.6408	1.140	0.2065	0.7020
	0.065	1.120	0.2420	0.8226	1.107	0.2647	0.8999
	0.085	1.080	0.3111	1.058	1.163	0.3397	1.155
	0.095	1.060	0.3447	1.172	1.041	0.3761	1.278
	0.105	1.040	0.3777	1.284	1.019	0.4117	1.399
1 1/2	0.050	1.400	0.2278	0.7743	1.390	0.2497	0.8488
	0.065	1.370	0.2930	0.9962	1.357	0.3209	1.091
	0.085	1.330	0.3779	1.285	1.313	0.4053	1.378
	0.095	1.310	0.4193	1.426	1.291	0.4581	1.557
	0.105	1.290	0.4602	1.564	1.269	0.5024	1.708
1 3/4	0.065	1.620	0.3441	1.170	1.606	0.3803	1.293
	0.085	1.580	0.4446	1.512	1.561	0.4907	1.668
	0.095	1.560	0.4939	1.679	1.539	0.5448	1.852
	0.105	1.540	0.5426	1.845	1.517	0.5902	2.007
	0.120	1.510	0.6145	2.089	1.484	0.6766	2.300
2	0.065	1.870	0.3951	1.343	1.856	0.4370	1.486
	0.085	1.830	0.5114	1.738	1.811	0.5649	1.920
	0.095	1.810	0.5685	1.933	1.789	0.6275	2.133
	0.105	1.790	0.6251	2.125	1.767	0.6896	2.344
	0.120	1.760	0.7087	2.409	1.734	0.7812	2.656

* Multiply values shown by 0.0645 to obtain areas in cm^2.
† Multiply values shown by 1.42 to obtain weights in kg/m.

Table 8.7.23 Seamless-Steel Boiler Tubes

Outside diam, in	Thickness BWG	Thickness in	Mfg.* wt, lb/ft	Outside diam, in	Thickness BWG	Thickness in	Mfg.* wt, lb/ft	Outside diam, in	Thickness BWG	Thickness in	Mfg.* wt, lb/ft
1	13	0.095	1.037	2½	12	0.109	3.171	4½	10	0.134	7.103
	12	0.109	1.168		11	0.120	3.457		9	0.148	7.817
	11	0.120	1.263		10	0.134	3.835		8	0.165	8.702
	10	0.134	1.384		9	0.148	4.207		7	0.180	9.447
1¼	13	0.095	1.323	2¾	12	0.109	3.504	5	9	0.148	8.720
	12	0.109	1.502		11	0.120	3.823		8	0.165	9.711
	11	0.120	1.628		10	0.134	4.244		7	0.180	10.550
	10	0.134	1.793		9	0.148	4.658		6	0.203	11.810
1½	13	0.095	1.619	3	12	0.109	3.838	5½	9	0.148	9.622
	12	0.109	1.836		11	0.120	4.189		8	0.165	10.720
	11	0.120	1.994		10	0.134	4.652		7	0.180	11.650
	10	0.134	2.201		9	0.148	5.110		6	0.203	13.050
1¾	13	0.095	1.910	3¼	11	0.120	4.555	6	7	0.180	12.750
	12	0.109	2.169		10	0.134	5.061		6	0.203	14.290
	11	0.120	2.360		9	0.148	5.061		5	0.220	15.410
	10	0.134	2.610		8	0.165	6.179		4	0.238	16.640
2	13	0.095	2.201	3½	11	0.120	4.921				
	12	0.109	2.503		10	0.134	5.469				
	11	0.120	2.726		9	0.148	6.012				
	10	0.034	3.018		8	0.065	6.683				
2¼	13	0.095	2.492	4	10	0.134	6.286				
	12	0.109	2.837		9	0.148	6.915				
	11	0.120	3.092		8	0.165	7.693				
	10	0.134	3.427		7	0.180	8.347				

* Multiply values shown by 1.42 to obtain weights in kg/m.
SOURCE: United States Steel Corporation.

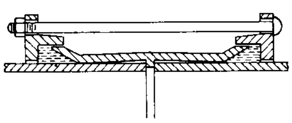


Fig. 8.7.1 Sleeve type, plain end coupling.

CAST-IRON AND DUCTILE-IRON PIPE

Cast-iron pipe was extensively produced since before the turn of the century until approximately 1960. Even though cast-iron pipe generally has given excellent service, the manufacture of cast-iron pipe was discontinued at that time.

Since then, **ductile-iron pipe** has replaced cast-iron pipe because of its improved ductility. Ductile-iron pipe is now extensively utilized for water, gas, sewage, culverts, drains, etc. It is produced in a wide range of sizes for varying pressures. Ductile-iron pipe is particularly adaptable to underground and submerged service because of its comparatively superior corrosion resistance compared to steel pipe. Nevertheless, steel pipe, when properly coated and wrapped, can also provide adequate resistance to corrosion when placed in certain soils.

Pipe fittings are available as cast-iron pipe fittings and ductile-iron pipe fittings. Both types of fittings are used with ductile-iron pipe.

Ductile-iron pipe may be obtained in various thicknesses and weights with (1) flanges cast on, (2) ends threaded for screwed-on flanges, (3) ends prepared for mechanical joint, (4) ends grooved or shouldered for patented coupling, (5) one end bell, other end spigot, and (6) one end hub, other end spigot. **Bell-and-spigot** ends are most popular for underground work; hub-and-spigot ends are most frequently used for sewage systems in enclosed spaces. Spigot-end joints are prepared by tightly tamping in hemp or jute at the bottom of the recess with a yarning iron and then pouring in molten lead; the lead, when cooled, is caulked in tightly with a caulking iron and makes a gastight joint. For exposed piping, flanged ends are used, the joints being made up with gaskets. Flanged pipe has superior strength and tightness of the joint and is used where pipelines can be well supported. The bell-and-spigot joint possesses greater flexibility and provides for expansion and contraction. It is therefore suitable for water pipe and is largely used for that purpose. Figure 8.7.2 shows a typical form of this joint for ordinary pressures. Figure 8.7.3 shows one form of this joint for ordinary pressures. Figure 8.7.3 shows one form of **mechanical joint** suitable for water, gas, or oil. Other forms of joint, plain-end pipe with couplings, and threaded pipe also are manufactured. Cast-iron and ductile-iron pipe, fittings, and valves have been found unsuitable for superheated steam service. The Code for Pressure Piping, B31.1 (Power Piping), states that cast iron or ductile-iron pipe may be used for steam service not over 250 lb/in² or 406°F (1,724 kPa or 208°C) provided that it meets the requirements as dictated by Eq. (8.7.1).

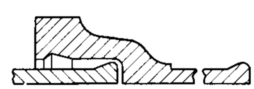


Fig. 8.7.2 Standard bell-and-spigot joint.

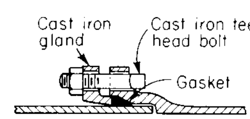


Fig. 8.7.3 Mechanical joint.

Wall thicknesses for the various conditions which ductile-iron pipe is designed to meet are determined in accordance with the requirements of ANSI 21.50 (AWWA C150).

Ductile-iron pipe is made by **centrifugal casting,** in which molten iron is admitted to the interior of a sand-lined or metal-lined mold, the mold being rotated at high speeds so that the molten metal is thrown by centrifugal force against the lining. ANSI specifications have been prepared for the various combinations of fabrication procedure and intended end use.

Table 8.7.24 lists thicknesses and weight data for centrifugally cast ductile-iron pipe intended for use with water or other liquids.

The employment of ductile-iron pipe for gas supply and distribution is second in importance only to its use for carrying water. Bell-and-spigot gas pipe is similar in design to bell-and-spigot water pipe (Fig. 8.7.2). For flanged gas pipe, the class 25 ANSI B16.1 Standard flanges are approved for maximum gas pressures of 25 lb/in² (172 kPa). The class 125 ANSI B16.1 Standard flanges are approved for gas pressures of 125 lb/in² (862 kPa), up to 4 in nominal pipe size; 100 lb/in² (689 kPa), 6 to 12 in; and 80 lb/in² (552 kPa), 16 to 48 in. The type of joint shown in Fig. 8.7.2 is also widely used for gas.

Table 8.7.24 Standard Weights and Thicknesses of Ductile-Iron Bell-and-Spigot Pipe for Water

	Class 50, 50 lb/in² (345 kPa) 115-ft (35.05-m) head			Class 100, 100 lb/in² (690 kPa) 231-ft (70.41-m) head			Class 150, 150 lb/in² (1,034 kPa) 346-ft (105.5-m) head				
Nominal size, in	Thickness, in	OD, in	Wt, lb per avg ft	Thickness, in	OD, in	Wt, lb per avg ft	Thickness, in	OD, in	Wt, lb per avg ft	Approx lb lead per joint 2 in thick	Approx lb hemp or jute per joint
3	0.32	3.96	12.4	0.32	3.96	12.4	0.32	3.96	12.4	6.2	0.17
4	0.35	4.80	16.5	0.35	4.80	16.5	0.35	4.80	16.5	7.5	0.21
6	0.38	6.90	25.9	0.38	6.90	25.9	0.38	6.90	25.9	10.3	0.31
8	0.41	9.05	37.0	0.41	9.05	37.0	0.41	9.05	37.0	13.3	0.44
10	0.44	11.10	49.1	0.44	11.10	49.1	0.44	11.10	49.1	16.0	0.53
12	0.48	13.20	63.7	0.48	13.20	63.7	0.48	13.20	63.7	19.0	0.61
14	0.48	15.30	74.6	0.51	15.30	78.8	0.51	15.65	80.7	22.0	0.81
16	0.54	17.40	95.2	0.54	17.40	95.2	0.54	17.80	97.5	30.0	0.94
18	0.54	19.50	107.6	0.58	19.50	114.8	0.58	19.92	117.2	33.8	1.00
20	0.57	21.60	125.9	0.62	21.60	135.9	0.62	22.06	138.9	37.0	1.25
24	0.63	25.80	166.0	0.68	25.80	178.1	0.73	26.32	194.0	44.0	1.50
30	0.79	32.00	257.6	0.79	32.00	257.6	0.85	32.00	275.4	54.3	2.06
36	0.87	38.30	340.9	0.87	38.30	340.9	0.94	38.30	365.9	64.8	3.00
42	0.97	44.50	442.0	0.97	44.50	442.0	1.05	44.50	475.3	75.3	3.62
48	1.06	50.80	551.6	1.06	50.80	551.6	1.14	50.80	589.6	85.5	4.37

Pipe weights indicated are approximate and include allowance for bell based on a 16-ft laying length. Calculations are for pipe laid without blocks, on flat-bottom trench, with tamped backfill under 5 ft of cover. Thicknesses given above include allowance for water hammer and factory tolerance.
To obtain weights in kg/m, multiply values shown in lb per avg ft by 1.42.
SOURCE: Condensed from Table 8.2 of Specification AWWA C 108-70.

Flexible-Joint Pipe The necessity for crossing streams and other waterways and of laying pipelines into them has led to the development of various forms of flexible-joint pipe adapted to laying under water, which are capable of motion through several degrees without leakage. Figure 8.7.4 shows one style of such joint which has an adjustment of about 15° in standard sizes.

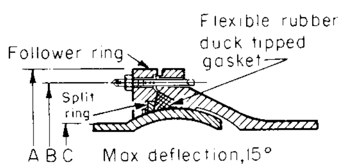


Fig. 8.7.4 Flexible joint. (See Table 8.7.25 for dimensions.)

In selecting the thickness of a pipe for a submerged line, the internal-service pressure is seldom the determining factor, as ample allowance should be made to minimize the risk of breakage in laying and to withstand external shocks from floating ice or other objects. The dimensions and weights given in Table 8.7.25 are typical of those listed by several manufacturers.

''Universal'' pipe (Fig. 8.7.5) is ductile-iron pipe with hub-and-spigot ends, the contact surfaces of which are machined on a taper, giving an iron-to-iron joint, By making the tapers of slightly different pitch, the joint provides for flexibility while remaining tight. Two bolts to the

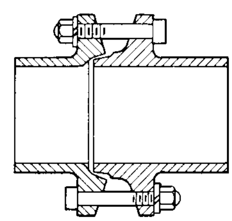


Fig. 8.7.5 Universal ductile-iron pipe and joint.

Table 8.7.25 Dimensions and Weights of Flexible-Joint Pipe*
(Dimensions refer to Fig. 8.7.4)

		Dimensions, in			Bolts				
Nominal diam, in	Class	*A*	*B*	*C*	No. required	Size, in	Length, in	Average metal thickness, in	Weight† of pipe, incl. bell, lb per 12-ft length
4	B	12.13	9.75	5.00	8	0.75	4.50	0.45	290
4	C	12.13	9.75	5.00	8	0.75	4.50	0.48	305
4	D	12.13	9.75	5.00	8	0.75	4.50	0.52	325
6	B	14.25	11.75	7.10	12	0.75	4.50	0.48	440
6	C	14.25	11.75	7.10	12	0.75	4.50	0.51	460
6	D	14.25	11.75	7.10	12	0.75	4.50	0.55	490
8	B	17.25	14.75	9.30	12	0.75	5.25	0.51	635
8	C	17.25	14.75	9.30	12	0.75	5.25	0.56	680
8	D	17.25	14.75	9.30	12	0.75	5.25	0.60	720
10	B	20.56	18.00	11.40	16	0.75	5.25	0.57	905
10	C	20.56	18.00	11.40	16	0.75	5.25	0.62	965
10	D	20.56	18.00	11.40	16	0.75	5.25	0.68	1,035
12	B	23.75	21.00	13.50	16	0.75	6.25	0.62	1,200
12	C	23.75	21.00	13.50	16	0.75	6.25	0.68	1,280
12	D	23.75	21.00	13.50	16	0.75	6.25	0.75	1,375

* United States Pipe and Foundry Co.
† Weights do not include follower rings, bolts, or gaskets. For sizes above 12 in, see manufacturers' catalogs. Multiply weights shown by 38.7 to obtain weight in kg per m or by 0.4536 to obtain weight in kg per 12-ft length.

Table 8.7.26 Standard Weights and Thicknesses of Universal Ductile-Iron Pipe
(Central Foundry Co.)

Nominal ID, in	Class 150, 150 lb/in² (1,034 kPa) Approx thickness, in	Class 150 Estimated wt, lb per* ft	Class 150 Estimated wt, lb per* 6-ft length	Class 250, 250 lb/in² (1,724 kPa) Approx thickness, in	Class 250 Estimated wt, lb per* ft	Class 250 Estimated wt, lb per* 6-ft length
2	0.25	7	42	0.31	8	48
3	0.30	11¼	67½	0.34	12½	75
4	0.32	16½	99	0.37	18	108
6	0.36	26.6	160	0.43	30	180
8	0.39	38½	231	0.47	44¼	265½
10	0.43	53½	321	0.50	60½	363
12	0.47	69	414	0.53	77½	465
14	0.50	87	522	0.565	98½	591
16	0.53	106	636	0.60	121	726

ID, in	2	3	4	6	8	10	12	14	16
Bolt sizes, in	½ × 4	½ × 4¼	⅝ × 5	¾ × 6	⅞ × 6¾	1 × 7½	1 × 8	1⅛ × 9	1¼ × 9½

* Multiply wts. by 0.4536 to obtain wt. in kilograms.

joint are sufficient, except for pressures above 175 lb/in² (1,207 kPa). Universal ductile-iron pipe is used largely for carrying gas and water and is suitable for all pressures and services. The pipe is tested with hydrostatic pressure of 300 to 500 lb/in² (2,067 to 3,448 kPa). All universal pipe and special castings of a given diameter and of any class are interchangeable with those of a different class. Standard laying lengths are 6 ft (1.83 m). Thicknesses and weights of standard types up to 16 in are given in Table 8.7.26. Information on other types and sizes and on fittings may be obtained from pipe producers.

Fittings for Ductile-Iron Water Pipe Flanged fittings of the dimensions of the ANSI standard for steam are not often used with ductile-iron water pipe. The longer fittings of the AWWA are generally preferred because of low friction loss. The dimensions of the flanged fittings of this class conform very closely to the dimensions of the bell-and-spigot fittings of the AWWA. The flange thicknesses and drillings conform to those of the ANSI standards. These fittings, both flange and bell-and-spigot type, are made in a great variety of forms known as ''standard special fittings.'' For dimensions and weights, see manufacturers' catalogs or standard specifications of the AWWA.

Cast-iron soil pipe and fittings are of the hub-and-spigot form, similar in design to the pipe shown in Fig. 8.7.2. Tapped openings and pipe plugs are threaded in accordance with the taper pipe thread requirements of ANSI B1.20.1-1983.

The ANSI standard, Threaded Cast-Iron Pipe for Drainage, Vent, and Waste Services, ANSI A40.5-1943, covers two types of pipes having threaded joints in nominal pipe sizes 1¼ to 12 in and in lengths 5 to 27 ft. One type has external threads on both ends; the other type has external threads on one end and an internal threaded drainage hub on the other end.

PIPES AND TUBES OF NONFERROUS MATERIALS

Brass tubing is commercially available in the form known as yellow brass, an alloy consisting of approximately 65 percent copper and 35

Table 8.7.27 Sizes and Weights of SPS Copper and 85 Red Brass Pipe*

Standard size, in	Nominal dimensions, in: Outside diameter	Inside diameter	Wall thickness	Theoretical areas based on nominal dimensions: Cross-sectional area of bore, in²	External surface, ft²/lin. ft	Internal surface ft²/lin. ft	Theoretical weight, lb/ft	Allowable internal pressure, lb/in²: 100°F, $S = 6{,}000$	200°F, $S = 4{,}800$	300°F, $S = 4{,}700$	400°F, $S = 3{,}000$
¼	0.540	0.410	0.065	0.132	0.141	0.107	0.376	1,500	1,200	1,180	740
⅜	0.675	0.545	0.065	0.233	0.177	0.143	0.483	1,170	940	910	580
½	0.840	0.710	0.065	0.396	0.220	0.186	0.613	920	740	720	460
¾	1.050	0.920	0.065	0.665	0.275	0.241	0.780	730	580	560	360
1	1.315	1.185	0.065	1.10	0.344	0.310	0.989	580	460	450	290
1¼	1.660	1.530	0.065	1.84	0.435	0.401	1.26	450	360	350	220
1½	1.900	1.770	0.065	2.46	0.497	0.464	1.45	400	310	300	190
2	2.375	2.245	0.065	3.96	0.622	0.588	1.83	300	240	240	140
2½	2.875	2.745	0.065	5.92	0.753	0.719	2.22	250	200	190	120
3	3.500	3.334	0.083	8.73	0.916	0.874	3.45	260	210	210	130
3½	4.000	3.810	0.095	11.4	1.05	0.998	4.52	270	210	210	130
4	4.500	4.286	0.107	14.4	1.18	1.12	5.72	270	210	210	130
5	5.562	5.298	0.132	22.0	1.46	1.39	8.73	270	210	210	130
6	6.625	6.309	0.158	31.3	1.73	1.65	12.4	270	210	210	130
8	8.625	8.215	0.205	53.0	2.26	2.15	21.0	270	210	210	130
10	10.750	10.238	0.256	82.3	2.81	2.68	32.7	270	210	210	130
12	12.750	12.124	0.313	115.	3.34	3.18	47.4	280	220	220	130

* The values in the table above are based on the formula in The Code for Pressure Piping, ANSI B31:

$$t_m = \frac{PD}{2S + 0.8P} + C \qquad \text{or when } C \text{ is } O \qquad P = \frac{2St_m}{D - 0.8t_m}$$

where t_m = minimum pipe wall thickness, in; P = maximum rated internal working pressure, lb/in²; D = outside diameter of pipe, in; S = allowable stress in material due to internal pressure, at operating temperature, lb/in²; C = allowance for threading, mechanical strength and/or corrosion, in. The allowable internal pressures apply to the pipe itself after brazing.

SOURCE: Copper Development Association.

Table 8.7.28 Allowable Internal Pressures, lb/in², for Temperatures up to 400°F for Red Brass and Copper Pipe*

	Pipe with threaded ends								Pipe with plain ends for use with welded, brazed, or soldered fittings							
	Red brass				Copper				Red brass				Copper			
Standard size, in	100°F $S = 8{,}000$	200°F $S = 8{,}000$	300°F $S = 8{,}000$	400°F $S = 5{,}000$	100°F $S = 6{,}000$	200°F $S = 4{,}800$	300°F $S = 4{,}700$	400°F $S = 3{,}000$	100°F $S = 8{,}000$	200°F $S = 8{,}000$	300°F $S = 8{,}000$	400°F $S = 5{,}000$	100°F $S = 6{,}000$	200°F $S = 4{,}800$	300°F $S = 4{,}700$	400°F $S = 3{,}000$
Regular																
⅛	370	370	370	230	280	210	210	130	2,640	2,640	2,640	1,650	1,980	1,570	1,540	980
¼	870	870	870	550	650	510	510	320	2,610	2,610	2,610	1,640	1,960	1,560	1,530	970
⅜	890	890	890	570	670	530	510	320	2,280	2,280	2,280	1,440	1,710	1,350	1,330	840
½	900	900	900	570	670	530	520	320	2,160	2,160	2,160	1,350	1,620	1,280	1,260	800
¾	810	810	810	520	610	480	470	300	1,800	1,800	1,800	1,130	1,350	1,070	1,050	670
1	630	630	630	400	480	370	370	230	1,580	1,580	1,580	1,000	1,190	940	920	590
1¼	690	690	690	430	520	410	400	250	1,440	1,440	1,440	900	1,080	890	840	530
1½	630	630	630	400	480	370	370	230	1,280	1,280	1,280	800	960	750	740	470
2	540	540	540	350	410	320	310	190	1,050	1,050	1,050	670	790	620	610	380
2½	450	450	450	280	340	260	250	160	1,040	1,040	1,040	650	780	620	610	380
3	510	510	510	320	380	300	290	180	1,000	1,000	1,000	630	750	590	580	370
3½	570	570	570	370	430	330	330	200	1,000	1,000	1,000	630	750	590	580	370
4	510	510	510	320	380	300	290	180	880	880	880	550	660	530	520	320
5	410	410	410	270	310	230	240	140	710	710	710	450	540	420	410	260
6	340	340	340	220	260	190	200	120	600	600	600	380	450	350	340	220
8	360	360	360	230	270	270	210	130	560	560	560	350	420	320	320	200
10	360	360	360	230	270	210	210	130	520	520	520	330	390	300	300	190
12	320	320	320	200	240	190	200	120	450	450	450	280	340	260	250	160
Extra-Strong																
⅛	1,960	1,960	1,960	1,240	1,470	1,160	1,140	720	4,630	4,630	4,630	2,900	3,470	2,750	2,710	1,730
¼	2,210	2,210	2,210	1,340	1,660	1,310	1,290	820	4,200	4,200	4,200	2,640	3,150	2,500	2,460	1,570
⅜	1,840	1,840	1,840	1,150	1,380	1,090	1,070	680	3,360	3,360	3,360	2,100	2,520	2,000	1,960	1,250
½	1,760	1,760	1,760	1,100	1,320	1,040	1,030	660	3,130	3,130	3,130	1,970	2,350	1,860	1,830	1,160
¾	1,510	1,510	1,510	950	1,130	900	880	560	2,560	2,560	2,560	1,600	1,920	1,520	1,500	960
1	1,340	1,340	1,340	850	1,010	790	780	490	2,360	2,360	2,360	1,490	1,770	1,400	1,370	880
1¼	1,160	1,160	1,160	730	880	730	680	430	1,950	1,950	1,950	1,220	1,460	1,160	1,140	720
1½	1,090	1,090	1,090	680	820	660	640	410	1,770	1,770	1,770	1,120	1,330	1,050	1,030	660
2	1,000	1,000	1,000	630	750	590	570	360	1,520	1,520	1,520	950	1,140	910	890	560
2½	970	970	970	620	730	580	560	360	1,600	1,600	1,600	1,000	1,200	950	930	590
3	910	910	910	570	680	530	530	340	1,420	1,420	1,420	900	1,070	840	830	530
3½	860	860	860	550	650	510	500	310	1,300	1,300	1,300	820	980	790	760	480
4	840	840	840	530	630	490	480	300	1,230	1,230	1,230	770	920	730	710	460
5	770	770	770	480	580	450	440	340	1,080	1,080	1,080	680	810	640	630	400
6	800	800	800	500	600	470	460	290	1,060	1,060	1,060	670	800	620	610	380
8	710	710	710	450	530	430	410	260	910	910	910	570	680	540	540	340
10	550	550	550	350	420	340	330	200	710	710	710	450	540	440	420	240

* The values above are based on the formula in The Code for Pressure Piping, ANSI B31:

$$t_m = \frac{PD}{2S + 0.8P} + C \quad \text{or} \quad P = \frac{2S(t_m - C)}{D - 0.8(t_m - C)} \quad \text{or when } C \text{ is O} \quad P = \frac{2St_m}{D - 0.8t_m}$$

where t_m = minimum pipe wall thickness, in; P = maximum rated internal working pressure, lb/in²; D = outside diameter of pipe, in; S = allowable stress in annealed material due to internal pressure, at operating temperature, lb/in²; C = allowance for threading, mechanical strength and/or corrosion, in. C equals 0.05 for sizes ⅜ in and smaller and the depth of the thread for all other sizes, and equals O for pipe with plain ends. These allowable internal pressures apply to the pipe itself and do not take into account the limitations which may be imposed by the type of joint and the joining material.

SOURCE: Copper Development Association.

percent zinc, and is used principally for ornamental work and hand railings, It has a density of approximately 0.3 lb/in³ (8.31 g/cm³), the exact density being dependent upon the specific chemical composition. **Brass piping** is most frequently furnished as red brass, an alloy consisting of approximately 85 percent copper and 15 percent zinc. This alloy, having a density of about 0.32 lb/in³ (8.86 g/cm³), has been found to be structurally superior to the yellow brasses and is used where the fluid being conveyed has corrosive properties.

Copper is available either as pipe or as tubing. In the form of piping, it has the same outer diameter as that of standard steel pipe (Tables 8.7.27 and 8.7.28). As tubing, it is used for a variety of purposes, such as for compressed-air instrumentation lines, hydraulic control lines around machinery, domestic oil-burner and heating systems, and for general plumbing purposes. **Copper tubing** (Table 8.7.29) is furnished in 12-ft (3.7-m) and 20-ft (6.1-m) straight lengths or in coils of 100-ft (30.5-m) length. **Type K** tubing, in coils, is used for underground work where the minimum number of joints, combined with greater thickness of type K tubing, is of distinct advantage. **Type L** tubing, usually in straight lengths, is used as a principal piping material for plumbing systems in homes and buildings; this is largely due to the economy of installation made possible by the use of soldered fittings. Copper deteriorates rapidly under high temperatures and repeated stresses. At a temperature of 360°F (182°C) its strength is reduced 15 percent, and on this account it should never be used for high steam pressures and temperatures.

Commercial sizes of **aluminum tubing** are listed by the manufacturers in even outside diameters and in wall thicknesses conforming to Stubs gage. Aluminum pipe is available as listed in Table 8.7.30. To obtain the approximate weight per foot of aluminum pipe or tubing, a weight of 0.098 lb/in³ (2.71 g/cm³) may be used.

Lead pipe is supplied in straight lengths, in coils, or in reels.

Block tin is a term used in the metal trade to refer to products made wholly from strictly pure high-grade tin. While tin pipe has many and varied uses, its most important applications are in types of equipment handling liquids intended for human consumption. Tin pipe does not corrode, and therefore does not contaminate most of the liquids passing through it.

Plastic pipes and **tubes** are available in a wide range of diameters and thicknesses, with Table 8.7.31 generally representative. The plastic used is resistant to attack by many chemicals, light in weight, flexible, and available in coiled form so that installation time is low. It is used for a variety of purposes including drainage, irrigation, sewage, and for conveying chemical solutions or waters that would attack metal piping. Plastic pipe used in gas service is listed in Table 8.7.32. Caution should be used in selection of plastic piping insofar as service temperature is concerned: e.g., polyethylene is suitable for a maximum temperature of 120°F (49°C). Table 8.7.33 lists corrosion-resistance data for polyethylene plastic piping.

Pipes with Special Linings For use in lines through which are passed solutions containing more or less free acid or other corrosive agents, standard pipe, valves, and fittings may be **lead-lined, tin-lined,** or **rubber-lined,** to resist corrosive action. This lining prolongs the life of the pipe and also gives it additional strength. For mine service in coal districts where the drainage water is more or less impregnated with sulfur or free sulfuric acid, **wood-lined pipe** and fittings are sometimes used. For special service, **seamless-copper-lined pipe** is also used. The **cement lining** of ductile-iron and steel pipe for water and other services is advantageous because of its protection against unusual destructive agencies and its ability to prevent tuberculation. Standard **hard-rubber pipe** and fittings have been developed for working pressures of 50 lb/in² (345 kPa) at normal temperatures. Standard sizes run from ¼- to 4-in diam (0.635- to 10.2-cm), in 10-ft (3.05-m) lengths. For temperature above 120°F (49°C), the use of hard-rubber-lined steel pipe is recommended. This pipe is suitable for conveying strong acids and chemicals.

VITRIFIED, WOODEN-STAVE, AND CONCRETE PIPE

Vitrified pipe is used extensively for drains and sewerage systems. Burnt-clay tile, being rendered impervious to water by glazing, is by far the best material for sewage purposes as it is not attacked by acids. Dimensions are given in Table 8.7.34 (see also Fig. 8.7.6). For sizes larger than 36 in (91.4 cm) and other data, refer to the publications of the Clay Products Assoc., Chicago (see ASTM Standard C-700).

Wood-stave pipe (Fig. 8.7.7) is used to a large extent for municipal water supply, outfall sewers, mining, irrigation, and various other uses providing for the transportation of water. The water carried may be hot, cold, or acid. It is made either untreated or creosoted by a vacuum and pressure process. This process uses 8 lb of creosote per cubic foot of wood treated. The untreated pipe is most used where the pipe is constantly full of water, and the wood therefore completely saturated, although in many such instances the creosoted wood is used to give assurance of permanence. (See also Sec. 6.)

Wood-stave pipe is made in two types: machine-banded pipe and continuous-stave pipe. **Machine-banded pipe** is banded with wire and is

Table 8.7.29 Sizes and Weights of Copper Tubes

Nominal size, in	OD, in, types* K, L	ID, in		Wall thickness, in		Permissible variation of mean OD, in — Types K, L		Weight,† lb/ft	
		Type K	Type L	Type K	Type L	Annealed	Hard-drawn	Type K	Type L
⅜	0.500	0.402	0.430	0.049	0.035	0.0025	0.001	0.269	0.198
½	0.625	0.527	0.545	0.049	0.040	0.0025	0.001	0.344	0.285
⅝	0.750	0.652	0.666	0.049	0.042	0.0025	0.001	0.418	0.362
¾	0.875	0.745	0.785	0.065	0.045	0.003	0.001	0.641	0.455
1	1.125	0.995	1.025	0.065	0.050	0.0035	0.0015	0.839	0.655
1¼	1.375	1.245	1.265	0.065	0.055	0.004	0.0015	1.04	0.884
1½	1.625	1.481	1.505	0.072	0.060	0.0045	0.002	1.36	1.14
2	2.125	1.959	1.985	0.083	0.070	0.005	0.002	2.06	1.75
2½	2.625	2.435	2.465	0.095	0.080	0.005	0.002	2.93	2.48
3	3.125	2.907	2.945	0.109	0.090	0.005	0.002	4.00	3.33
3½	3.625	3.385	3.425	0.120	0.100	0.005	0.002	5.12	4.29
4	4.125	3.857	3.905	0.134	0.110	0.005	0.002	6.51	5.38
5	5.125	4.805	4.875	0.160	0.125	0.005	0.002	9.67	7.61
6	6.125	5.741	5.845	0.192	0.140	0.005	0.002	13.9	10.2
8	8.125	7.583	7.725	0.271	0.200	0.006	0.003	25.9	19.3
10	10.125	9.449	9.625	0.338	0.250	0.008	0.004	40.3	30.1
12	12.125	11.315	11.565	0.405	0.280	0.008	0.004	57.8	40.4

* Type K recommended for underground service and general plumbing. Type L suitable for interior plumbing and other services.
† Multiply these values by 1.48 to obtain weight in kg/m.
SOURCE: American Brass Co.

Table 8.7.30 Aluminum Piping

Nominal pipe size, in	Schedule no.	OD, in	ID, in	Wall thickness, in	Weight per linear foot, lb, plain ends†	Cross-sectional wall area, in^2	Inside cross-sectional area, in^2	Moment of inertia, in^4	Section modulus, in^3	Radius of gyration, in
⅛	40‡	0.405	0.269	0.068	0.085	0.0720	0.0568	0.0011	0.0053	0.1215
	80§	0.405	0.215	0.095	0.109	0.0925	0.0363	0.0012	0.0060	0.1146
¼	40‡	0.540	0.364	0.088	0.147	0.1250	0.1041	0.0033	0.0123	0.1628
	80§	0.540	0.302	0.119	0.185	0.1574	0.0716	0.0038	0.0139	0.1547
⅜	40‡	0.675	0.493	0.091	0.196	0.1670	0.1909	0.0073	0.0216	0.2090
	80§	0.675	0.423	0.126	0.256	0.2173	0.1405	0.0086	0.0255	0.1991
½	40‡	0.840	0.622	0.109	0.294	0.2503	0.3039	0.0171	0.0407	0.2613
	80§	0.840	0.546	0.147	0.376	0.3200	0.2341	0.0201	0.0478	0.2505
¾	10	1.050	0.884	0.083	0.297	0.2521	0.6138	0.0297	0.0566	0.3432
	40‡	1.050	0.824	0.113	0.391	0.3326	0.5333	0.0370	0.0705	0.3337
	80§	1.050	0.742	0.154	0.510	0.4335	0.4324	0.0448	0.0853	0.3214
1	5	1.315	1.185	0.065	0.300	0.2553	1.103	0.0500	0.0760	0.4425
	10	1.315	1.097	0.109	0.486	0.4130	0.9452	0.0757	0.1151	0.4382
	40‡	1.315	1.049	0.133	0.581	0.4939	0.8643	0.0873	0.1328	0.4205
	80§	1.315	0.957	0.179	0.751	0.6388	0.7193	0.1056	0.1606	0.4066
1¼	5	1.660	1.530	0.065	0.383	0.3257	1.839	0.1037	0.1250	0.5644
	10	1.660	1.442	0.109	0.625	0.5311	1.633	0.1605	0.1934	0.5497
	40‡	1.660	1.380	0.140	0.786	0.6685	1.496	0.1947	0.2346	0.5397
	80§	1.660	1.278	0.191	1.037	0.8815	1.283	0.2418	0.2913	0.5238
1½	5	1.900	1.770	0.065	0.441	0.3747	2.461	0.1579	0.1662	0.6492
	10	1.900	1.682	0.109	0.721	0.6133	2.222	0.2468	0.2598	0.6344
	40‡	1.900	1.610	0.145	0.940	0.7995	2.036	0.3099	0.3262	0.6226
	80§	1.900	1.500	0.200	1.256	1.068	1.767	0.3912	0.4118	0.6052
2	5	2.375	2.245	0.065	0.555	0.4717	3.958	0.3149	0.2652	0.8170
	10	2.375	2.157	0.109	0.913	0.7760	3.654	0.4992	0.4204	0.8021
	40‡	2.375	2.067	0.154	1.264	1.074	3.356	0.6657	0.5606	0.7871
	80§	2.375	1.939	0.218	1.737	1.477	2.953	0.8679	0.7309	0.7665
2½	5	2.875	2.709	0.083	0.856	0.7280	5.764	0.7100	0.4939	0.9876
	10	2.875	2.635	0.120	1.221	1.039	5.453	0.9873	0.6868	0.9750
	40‡	2.875	2.469	0.203	2.004	1.704	4.788	1.530	1.064	0.9474
	80§	2.875	2.323	0.276	2.650	2.254	4.238	1.924	1.339	0.9241
3	5	3.500	3.334	0.083	1.048	0.8910	8.730	1.301	0.7435	1.208
	10	3.500	3.260	0.120	1.498	1.274	8.346	1.822	1.041	1.196
	40‡	3.500	3.068	0.216	2.621	2.228	7.393	3.017	1.724	1.164
	80§	3.500	2.900	0.300	3.547	3.016	6.605	3.894	2.225	1.136
3½	5	4.000	3.834	0.083	1.201	1.021	11.55	1.960	0.9799	1.385
	10	4.000	3.760	0.120	1.720	1.463	11.10	2.755	1.378	1.372
	40‡	4.000	3.548	0.226	3.151	2.680	9.887	4.788	2.394	1.337
	80§	4.000	3.364	0.318	4.326	3.678	8.888	6.281	3.140	1.307
4	5	4.500	4.334	0.083	1.354	1.152	14.75	2.810	1.249	1.562
	10	4.500	4.260	0.120	1.942	1.651	14.25	3.963	1.761	1.549
	40‡	4.500	4.026	0.237	3.733	3.174	12.73	7.232	3.214	1.510
	80§	4.500	3.826	0.337	5.183	4.407	11.50	9.611	4.272	1.477
5	40‡	5.563	5.047	0.258	5.057	4.300	20.01	15.16	5.451	1.878
	80§	5.563	4.813	0.375	7.188	6.112	18.19	20.67	7.432	1.839
6	40‡	6.625	6.065	0.280	6.564	5.581	28.89	28.14	8.496	2.246
	80§	6.625	5.761	0.432	9.884	8.405	26.07	40.49	12.22	2.195
8	30	8.625	8.071	0.277	8.543	7.265	51.16	63.35	14.69	2.953
	40‡	8.625	7.981	0.322	9.878	8.399	50.03	72.49	16.81	2.938
	80§	8.625	7.625	0.500	15.01	12.76	45.66	105.7	24.51	2.878
10		10.750	10.192	0.279	10.79	9.178	81.59	125.9	23.42	3.704
	30	10.750	10.136	0.307	11.84	10.07	80.69	137.4	25.57	3.694
	40‡	10.750	10.020	0.365	14.00	11.91	78.85	160.7	29.90	3.674
	80§	10.750	9.750	0.500	18.93	16.10	74.66	211.9	39.43	3.628
12	30	12.750	12.090	0.330	15.14	12.88	114.8	248.5	38.97	4.393
	‡	12.750	12.000	0.375	17.14	14.58	113.1	279.3	43.81	4.377
	§	12.750	11.750	0.500	22.63	19.24	108.4	361.5	56.71	4.335
					Construction pipe					
		2.00	1.900	0.050	0.3602	0.3063	2.835	0.1457	0.1457	0.6897
		3.00	2.900	0.050	0.5449	0.4634	6.605	0.5042	0.3361	1.043
		4.00	3.900	0.050	0.7297	0.6205	11.95	1.210	0.6051	1.397
		5.00	4.896	0.052	0.9506	0.8083	18.83	2.474	0.9896	1.749
		6.00	5.876	0.062	1.360	1.157	2.712	5.098	1.699	2.100
		7.00	6.856	0.072	1.843	1.567	36.92	9.403	2.687	2.450
		8.00	7.812	0.094	2.745	2.335	47.93	18.24	4.561	2.795

* Aluminum Co. of America.
† Weights calculated for 6061 and 6063. For 3003 multiply by 1.010.
‡ Also designated as standard pipe.
§ Also designated as extra-heavy or extra-strong pipe. All calculations based on nominal dimensions.

Table 8.7.31 Commerical Sizes (IPS) and Weights of Polyvinyl Chloride (PVC) Pipe*

Nominal size, in	Schedule	Wall thickness,* in	OD, in	ID, in	Theoretical weight,† lb/ft	Calculated min bursting pressure, lb/in²	
						Note 1	Note 2
¼	40	0.088	0.540	0.364	0.076	2,490	1,950
	80	0.119	0.540	0.302	0.096	3,620	2,830
½	40	0.109	0.840	0.622	0.153	1,910	1,490
	80	0.147	0.840	0.546	0.185	2,720	2,120
¾	40	0.113	1.050	0.824	0.203	1,540	1,210
	80	0.154	1.050	0.742	0.265	2,200	1,720
1	40	0.133	1.315	1.049	0.305	1,440	1,130
	80	0.179	1.315	0.957	0.385	2,020	1,580
1¼	40	0.140	1.660	1.380	0.409	1,180	920
	80	0.191	1.660	1.278	0.550	1,660	1,300
1½	40	0.145	1.900	1.610	0.489	1,060	830
	80	0.200	1.900	1.500	0.653	1,510	1,180
2	40	0.154	2.375	2.067	0.640	890	690
	80	0.218	2.375	1.939	0.910	1,290	1,010
3	40	0.216	3.500	3.068	1.380	840	660
	80	0.300	3.500	2.900	1.845	1,200	940
4	40	0.237	4.500	4.026	1.965	710	560
	80	0.337	4.500	3.826	2.710	1,040	810

* Thicknesses listed are minimum values. Tolerance is generally − 0 + 10%.
† These representative values are not specified in ASTM D1785-85.
NOTES:
1. Materials are PVC 1120, 1220, and 4116. A fiber stress of 6,400 lb/in² was used in bursting pressure calculations.
2. Materials are PVC 2112, 2116, and 2120. A fiber stress of 5,000 lb/in² was used in bursting pressure calculations.
SOURCE: Abstracted from ASTM Specification D1785-85.

Table 8.7.32 Dimensions of Plastic Pipe for Gas Service (AGA Requirements)

Nominal size, in	Nominal OD, in	Nominal ID, in	Sleeved weight, lb	Max working pressure at 73°F, lb/in²
½	0.625	0.500	0.054	250
¾	0.875	0.750	0.079	175
1	1.125	1.000	0.103	125
1¼	1.375	1.250	0.127	110
1½	1.625	1.500	0.152	90
1¾	1.875	1.750	0.177	80
2	2.125	2.000	0.200	75
2½	2.660	2.500	0.320	75
3	3.190	3.000	0.457	75
4	4.250	4.000	0.800	75

made with wood or metal collars, or with inserted joints. **Continuous-stave pipe** is manufactured in units consisting of staves, bands, and shoes, shipped in knocked-down form, and constructed in the trench. In building this type of pipe, the staves are laid so as to break joints and the completed pipe is without joints. Continuous-stave pipe is banded with individual bands, ranging in size from ⅜ to 1 in (0.95 to 2.54 cm), depending upon the size of the pipe. A factor of safety of 4 is maintained in the band, based on an ultimate strength of 60,000 lb/in² (414 MPa) of cross section. The maximum pressure to which a continuous-stave pipe may be subjected depends upon the size of the pipe. The head for small pipes may run as high as 400 ft (121.9 m) while in the largest sizes the head would be less than 200 ft (61 m).

Machine-banded pipe is made for pressures of 50 to 400 ft (15.2 to 121.9 m). The staves are made from redwood or Douglas-fir lumber, dried and carefully selected. The inside and outside of the staves are dressed to conform to the circumferential lines, and the edges of the staves dressed to conform to the radial lines.

Wooden pipe is largely built in western sections of the United States, close to the natural lumber market. The sizes of machine-banded pipe range from 2 to 24 in (5.08 to 61 cm), and of the continuous-stave pipe from 6- to 20-ft (15.2-cm to 6.1-m) inside diameter.

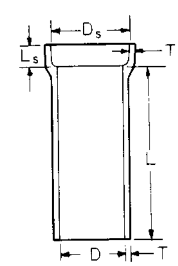


Fig. 8.7.6 Joint for vitrified pipe.

Pipe made from **plywood** is molded in lengths up to 11 ft (3.35 m) in diameters 3 in (7.62 cm) and up, and in wall thicknesses to specifications. Tubes made from fiber, by a molding process, are obtainable in a variety of sizes and lengths.

Concrete pipe is an important factor in sewer, conduit, railroad, culvert, and water-pipe construction. The pipe, as usually made, is constructed of concrete reinforced longitudinally with bars and transversely with wire mesh or steel bands. It is made in sections of definite length, with the longitudinal reinforcement so disposed as to provide for the interlocking of one section with another, and so formed that when these are locked together and cemented they form a continuous line of pipe free from leakage or seepage. Various forms of joints are used, all capable of taking care of expansion. Figure 8.7.8 shows one type of construction. Concrete pipe is manufactured in a great variety of diameters, thicknesses, and lengths to suit almost any requirement arising in practice. (See also Sec. 6.)

Table 8.7.33 Corrosion-Resistance Data, Polyethylene Pipe

Reagent	Performance at: 75°F (24°C)	Performance at: 120°F (49°C)	Reagent	Performance at: 75°F (24°C)	Performance at: 120°F (49°C)	Reagent	Performance at: 75° (24°C)	Performance at: 120°F (49°C)
Acetic acid, glacial*	F	NG	Ferrous sulfate, 15 percent aq.	E	E	Potassium chloride, saturated	E	E
Acetic acid, 10 percent*	E	E	Fluorine	E	NG	Potassium dichromate	E	E
Acetone	NG	NG	Fluosilicic acid, concentrated	E	F	Potassium hydroxide	E	E
Ammonia, dry gas	E	E	Formaldehyde, 40 percent	E	E	Potassium nitrate	E	E
Ammonium hydroxide, 10 percent	E	E	Formic acid, 50 percent	E	E	Potassium permanganate	E	E
Ammonium hydroxide, 28 percent	E	E	Furfuryl alcohol	NG	NG	Silicic acid	E	E
Amyl acetate	NG	NG	Gasoline	NG	NG	Silver nitrate	E	E
Aniline	E	F	Hydrobromic acid	E	E	Sodium benzoate	E	E
Benzene	NG	NG	Hydrochloric acid, 10 percent	E	E	Sodium bisulfite	E	E
Bromine	NG	NG	Hydrochloric acid, 37 percent	E	E	Sodium carbonate, concentrated	E	E
Butyraldehyde	E	G	Hydrofluoric acid, 48 percent	E	E	Sodium chloride, saturated solution	E	E
Calcium chloride, saturated	E	E	Hydrofluoric acid, 75 percent	E	F	Sodium hydroxide, 10 percent	E	E
Calcium hydroxide	E	E	Hydrogen peroxide, 30 percent	E	G	Sodium hydroxide, 50 percent	E	E
Calcium hypochlorite	E	E	Hydrogen peroxide, 90 percent	G	NG	Sodium sulfate	E	E
Carbon disulfide	NG	NG	Lactic acid, 90 percent	E	E	Stannic chloride, saturated	E	E
Carbon tetrachloride	NG	NG	Linseed oil	E	E	Stearic acid, 100 percent	E	E
Carbonic acid	E	E	Lubricating oil	NG	NG	Sulfuric acid, 10 percent	E	E
Chlorine, dry gas	F	NG	Magnesium chloride	E	E	Sulfuric acid, 30 percent	E	E
Chlorine, liquid	NG	NG	Magnesium sulfate	E	E	Sulfuric acid, 60 percent	E	F
Chlorosulfonic acid	NG	NG	Methyl bromide	NG		Sulfuric acid, 98 percent	F	NG
Citric acid, saturated	E	E	Methyl isobutyl ketone	F	NG	Tannic acid	E	E
Copper sulfate	E	E	Nitric acid, 10 percent	E	E	Toluene	NG	NG
Cyclohexanone	NG	NG	Nitric acid, 30–50 percent	E	E	Trichlorobenzene	F	NF
Diethylene glycol	E	E	Nitric acid, 70 percent	E	E	Trichloroethylene	NG	NG
Dioxane	E	G	Oleic acid	F	NG	Vinegar	E	E
Ethyl acetate	F	NG	Phosphoric acid, 30 percent	E	E	Xylene	NG	NG
Ethyl alcohol, 35 percent	NG	NG	Phosphoric acid, 90 percent	E	NG	Zinc chloride	E	E
Ethyl butyrate	F	NG	Photographic developer	E	E	Zinc sulfate	E	E
Ethylene dichloride	NG	NG	Potassium borate	E	E			
Ferric chloride	E	E	Potassium carbonate	E	E			

Corrosion-resistance data given in this table based on laboratory tests conducted by the manufacturers of the materials covered, and are indicative only of the conditions under which the tests were made. This information may be considered as a basis for recommendation, but not as a guarantee. Materials should be tested under actual service to determine suitability for a particular purpose.
E = excellent, G = good, F = fair, NG = not good.
* Polyethylene is permeable to acetic acid.

Asbestos-cement pipe, known by the trade name **Transite** pipe in this country, was developed initially in Europe. It was widely used in the United States for many years in a large variety of services. It was made of a mixture of portland cement and asbestos fiber, was highly resistant to corrosion, and provided outstanding service in mine drainage systems, waterworks systems, gas lines, and sewerage systems. It was manufactured in diameters from 3 to 36 in (7.6 to 91 cm), in stock lengths of 13 ft (4 m), and in pressure classes of 50, 100, 150, and 200 lb/in² (349, 689, 1,034, and 1,379 kPa). While it is no longer manufactured, it is still in place in some of the above-cited applications. Repairs to existing Transite piping generally devolves into replacing it with pipe made of a material compatible with its service, often with fiberglass-reinforced pipe, plastic pipe, or some form of metal pipe.

Table 8.7.34 Standard-Strength Vitrified Clay Pipe
(Dimensions refer to Fig. 8.7.6)

Size D, in	Laying length L: Nominal, ft	Laying length L: Limit of minus variation,* in/ft length	Max difference in length of two opposite sides, in	OD of barrel, in: Min	OD of barrel, in: Max	ID of socket ½ in above base D_s, in: Min	ID of socket ½ in above base D_s, in: Max	Depth of socket L_s, in: Nominal	Depth of socket L_s, in: Min	Thickness of barrel T, in: Nominal	Thickness of barrel T, in: Min	Thickness of socket at ½ in from outer end T_s, in: Nominal	Thickness of socket at ½ in from outer end T_s, in: Min
4	2, 2½, 3	¼	5⁄16	4⅞	5⅛	5¾	6⅛	1¾	1½	½	7⁄16	7⁄16	⅜
6	2, 2½, 3	¼	⅜	7 1⁄16	7 7⁄16	8 3⁄16	8⅝	2½	2	⅝	9⁄16	½	7⁄16
8	2, 2½, 3	¼	7⁄16	9¼	9¾	10½	11	2½	2¼	¾	11⁄16	9⁄16	½
10	2, 2½, 3	¼	7⁄16	11½	12	12¾	13¼	2⅝	2⅜	⅞	13⁄16	⅝	9⁄16
12	2, 2½, 3	¼	7⁄16	13¾	14 5⁄16	15⅛	15¾	2¾	2½	1	15⁄16	¾	11⁄16
15	3, 4	¼	½	17 3⁄16	17 13⁄16	18⅝	19¼	2⅞	2⅝	1¼	1⅛	15⁄16	⅞
18	3, 4	¼	½	20⅝	21 7⁄16	22¼	23	3	2¾	1½	1⅜	1⅛	1 1⁄16
21	3, 4	¼	9⁄16	24⅛	25	25⅞	26¾	3¼	3	1¾	1⅝	1 5⁄16	1 3⁄16
24	3, 4	⅜	9⁄16	27½	28½	29⅜	30⅜	3⅜	3⅛	2	1⅞	1½	1⅜
27	3, 4	⅜	9⁄16	31	32⅛	33	34⅛	3½	3¼	2¼	2⅛	1 11⁄16	1 9⁄16
30	3, 4	⅜	⅝	34⅜	35⅝	36½	37¾	3⅝	3⅜	2½	2⅜	1⅞	1¾
33	3, 4	⅜	⅝	37⅝	38 15⁄16	39⅞	41¼	3¾	3½	2⅝	2½	2	1 13⁄16
36	3, 4	⅜	11⁄16	40¾	42¼	43¼	44¾	4	3¾	2¾	2⅝	2 1⁄16	1⅞

When ordering standard-strength vitrified-clay pipe, give the size of pipe (ID) and the laying strength wanted, and refer to ASTM Specification C-700. Standard lengths of pipe shown meet normal practice in various sections of the country. Manufacturers' stocks include those lengths conforming to local practice.
* There is no limit for plus variation.

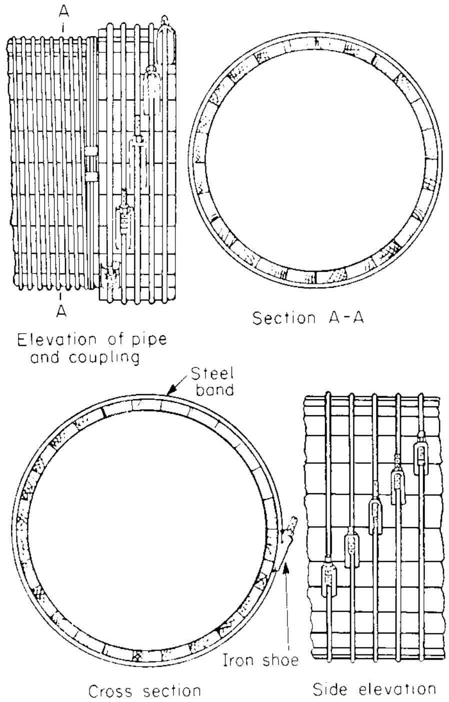


Fig. 8.7.7 Wood-stave pipes.

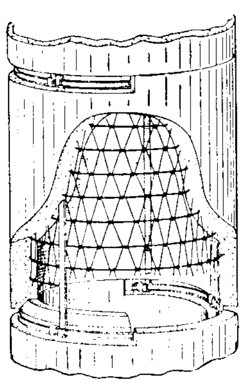


Fig. 8.7.8 Reinforced-concrete pipe.

FITTINGS FOR STEEL PIPE

American National Standard Cast-Iron Pipe Flanges and Flanged Fittings for Maximum Working Saturated Steam Pressure of 25, 125, and 250 lb/in^2 (172, 862, and 1,724 kPa)

INTRODUCTORY NOTES

Sizes The sizes of the fittings in the following tables are nominal pipe sizes. In the class 25 standard, the nominal pipe size is the same as the port diameter of the fittings, for *all* sizes. In the class 125 and class 250 standards the nominal pipe size is the same as the port diameter of fittings for pipe having inside diameters of 12 in (30.5 cm) and smaller. For pipe 14 in (35.6 cm) and larger, the corresponding outside diameter of the pipe is given, and consequently the fittings will have a smaller port diameter.

Pressure Rating In the class 25 standard the sizes 36 in (91.4 cm) and smaller may also be used for maximum nonshock working hydraulic pressures of 43 lb/in^2 gage (296 kPa) or a maximum gas pressure of 25 lb/in^2 gage (172 kPa), at or near the ordinary range of air temperatures. In the class 125 standard, the sizes 12 in (30.5 cm) and smaller may also be used for maximum nonshock working hydraulic pressure of 175 lb/in^2 gage (1,207 kPa) at or near the ordinary range of air temperatures. In the class 250 standard, the sizes 12 in and smaller may be used for maximum nonshock working hydraulic pressures of 400 lb/in^2 gage (2,758 kPa) at or near the ordinary range of air temperatures.

Facing All class 25 and class 125 cast-iron flanges and flanged fittings are plain faced, i.e., without projection or raised face. All class 250 cast-iron flanges and flanged fittings have a raised face 1⁄16 (0.16 cm) high, of the diameters given in Table 8.7.35. The raised face is included in the minimum flange thickness and center-to-face dimensions.

An **inspection limit** of ± 1⁄32 in (0.08 cm) is allowed on all center-to-contact-surface dimensions for sizes up to and including 10 in (25.4 cm) and ± 1⁄8 in on sizes larger than 10 in.

Dimensions In the class 25 standard, the flange diameters, bolt circles, and number of bolts are the same as in the class 125 ANSI Standard B16.1-1975, with a reduction in the thickness of flanges and bolt diameters, thereby maintaining interchangeability between the two standards.

The center-to-face and face-to-face dimensions of class 25 standard fittings are the same as for the class 125 standard.

Bolting Drilling templates are in multiples of four, so that fittings may be made to face in any quarter. **Bolt-holes** straddle the centerline. For bolts smaller than 1¾ in (4.45 cm) the bolt-holes are drilled 1⁄8 in larger in diameter than the nominal size of the bolt. Holes for bolts 1¾ in and larger are drilled ¼ in (0.64 cm) larger than nominal diameter of bolts. **Bolts** of steel are with standard ''rough square heads'' and the nuts are of steel with standard ''rough hexagonal'' dimensions; all as given in the American Standard on Wrench Head Bolts and Nuts and Wrench Openings of the National Screw Thread Commission. For bolts, 1¾-in (4.45-cm) diam and larger, bolt studs with a nut on each end are recommended.

Hexagonal nuts for pipe sizes 1 to 48 in (2.54 to 122 cm) in the class 125 standard and 1 to 16 in (40.6 cm) in the class 250 standard can be conveniently pulled up with open wrenches of minimum design of heads. Hexagonal nuts for pipe sizes 48 to 96 in (244 cm) in the class 125 standard and 18 to 48 in (45.7 to 122 cm) in the class 250 standard can be conveniently pulled up with box wrenches.

Spot Facing The bolt-holes of classes 25, 125, and 250 cast-iron flanges and flanged fittings are not spot-faced for ordinary service. When required, the flanges and fittings in sizes 30 in (76.2 cm) and larger may be spot-faced or back-faced to the minimum thickness of flange with a plus tolerance of 1⁄8 in (0.32 cm).

Reducing Fittings Reducing elbows and side-outlet elbows carry same dimensions center to face as straight-size elbows corresponding to the size of the larger opening.

Tees, side-outlet tees, crosses, and laterals sizes 16 in (40.6 cm) and smaller, reducing on the outlet or branch, have the same dimension center to face and face to face as straight-size fittings corresponding to the size of the larger opening. Sizes 18 in (45.7 cm) and larger, reducing on the outlet or branch, are made in two lengths depending on the size of the outlet as given in the tables of dimensions.

Tees, crosses, and laterals, reducing on the run only, have the same dimensions center to face and face to face as straight-size fittings corresponding to the size of the larger opening.

Reducers and eccentric reducers for all reduction have the same face-to-face dimensions for the larger opening.

Special double-branch elbows whether straight or reducing have the same dimension center to face as straight-size elbows corresponding to the size of the larger opening.

Side-outlet elbows and side-outlet tees have all openings on intersecting centerlines.

Table 8.7.35 Templates for Drilling Cast-Iron Pipe Flanges, Flanged Valves, and Fittings*

Nominal pipe size, in	Diameter of flange, in	Thickness of flange (minimum), in	Diameter of raised face, in	Diameter of bolt circle, in	Number of bolts	Diameter of bolts, in	Diameter of drilled bolt-holes, in	Length of bolts, in	Length of bolt stud with two nuts, in	Total effective area bolt metal, in^2	Stress in bolt metal, lb/in^2†	Size of ring gasket, in
Class 25 standard (922 N/m^2)												
4	9	¾		7½	8	⅝	¾	2¼		1.616	570	4 × 6⅞
5	10	¾		8½	8	⅝	¾	2¼		1.616	750	5 × 7⅞
6	11	¾		9½	8	⅝	¾	2¼		1.616	930	6 × 8¾
8	13½	¾		11¾	8	⅝	¾	2¼		1.616	1470	8 × 11
10	16	⅞		14¼	12	⅝	¾	2½		2.424	1440	10 × 13⅜
12	19	1		17	12	⅝	¾	2¾		2.424	2195	12 × 16⅛
14	21	1⅛		18¾	12	¾	⅞	3¼		3.620	1750	14 × 18
16	23½	1⅛		21¼	16	¾	⅞	3¼		4.830	1710	16 × 20½
18	25	1¼		22¾	16	¾	⅞	3½		4.830	1965	18 × 22
20	27½	1¼		25	20	¾	⅞	3½		6.040	1920	20 × 24½
24	32	1⅜		29½	20	¾	⅞	3¾		6.040	2690	24 × 28¾
30	38¾	1½		36	28	⅞	1	4¼		11.760	2030	30 × 35⅛
36	46	1⅝		42¾	32	⅞	1	5		13.440	2610	36 × 41⅞
42	53	1¾		49½	36	1	1⅛	5¼		19.800	2315	42 × 48½
48	59½	2		56	44	1	1⅛	5½		24.200	2475	48 × 55
54	66¼	2¼		62¾	44	1	1⅛	5¾		24.200	3195	54 × 61¾
60	73	2¼		69¼	52	1⅛	1¼	6		36.020	2515	60 × 68⅛
72	86½	2½		82½	60	1⅛	1¼	6¼		41.570	3120	72 × 81⅜
84	99¾	2¾		95½	64	1¼	1⅜	7¼		57.140	3005	84 × 94¼
96	113¼	3		108½	68	1¼	1⅜	7¾		60.570	3705	96 × 107¼
Class 125 standard (4,612 N/m^2)												
1	4¼	7/16		3⅛	4	½	⅝	1¾				1 × 2⅝
1¼	4⅝	½		3½	4	½	⅝	2				1¼ × 3
1½	5	9/16		3⅞	4	½	⅝	2				1½ × 3⅜
2	6	⅝		4¾	4	⅝	¾	2¼				2 × 4⅛
2½	7	11/16		5½	4	⅝	¾	2½				2½ × 4⅞
3	7½	¾		6	4	⅝	¾	2½				3 × 5⅜
3½	8½	13/16		7	8	⅝	¾	2¾				3½ × 6⅜
4	9	15/16		7½	8	⅝	¾	3				4 × 6⅞
5	10	15/16		8½	8	¾	⅞	3				5 × 7¾
6	11	1		9½	8	¾	⅞	3¼				6 × 8¾
8	13½	1⅛		11¾	8	¾	⅞	3½				8 × 11
10	16	1 3/16		14¼	12	⅞	1	3¼				10 × 13⅜
12	19	1¼		17	12	⅞	1	3¾				12 × 16⅛
14 OD	21	1⅜		18¾	12	1	1⅛	4¼				14 × 17¾
16 OD	23½	1 7/16		21¼	16	1	1⅛	4½				16 × 20¼
18 OD	25	1 9/16		22¾	16	1⅛	1¼	4¾				18 × 21⅝
20 OD	27½	1 11/16		25	20	1⅛	1¼	5				20 × 23⅞
24 OD	32	1⅞		29½	20	1¼	1⅜	5½				24 × 28¼
30 OD	38¾	2⅛		36	28	1¼	1⅜	6¼				30 × 34⅝
36 OD	46	2⅜		42¾	32	1½	1⅝	7				36 × 41¼
42 OD	53	2⅝		49½	36	1½	1⅝	7½				42 × 48
48 OD	59½	2¾		56	44	1½	1⅝	7¾				48 × 54½
54 OD	66¼	3		62¾	44	1¾	2	8½	10½			54 × 61
60 OD	73	3⅛		69¼	52	1¾	2	8¾	11			60 × 67½
72 OD	86½	3½		82½	60	1¾	2	9½	12			72 × 80¾
84 OD	99¾	3⅞		95½	64	2	2¼	10½	13			84 × 93½
96 OD	113¼	4¼		108½	68	2¼	2½	11½	14½			96 × 106¼
Class 250 standard (9,224 N/m^2)												
1	4⅞	11/16	2 11/16	3½	4	⅝	¾	2¼				1 × 2⅞
1¼	5¼	¾	3 1/16	3⅞	4	⅝	¾	2½				1¼ × 3¼
1½	6⅛	13/16	3 9/16	4½	4	¾	⅞	2½				1½ × 3¾
2	6½	⅞	4 3/16	5	8	⅝	¾	2½				2 × 4⅜
2½	7½	1	4 15/16	5⅞	8	¾	⅞	3				2½ × 5⅛
3	8¼	1⅛	5 11/16	6⅝	8	¾	⅞	3¼				3 × 5⅞
3½	9	1 3/16	6 5/16	7¼	8	¾	⅞	3¼				3½ × 6½
4	10	1¼	6 15/16	7⅞	8	¾	⅞	3½				4 × 7⅛
5	11	1⅜	8 5/16	9¼	8	¾	⅞	3¾				5 × 8½
6	12½	1 7/16	9 11/16	10⅝	12	¾	⅞	3¾				6 × 9⅞
8	15	1⅝	11 15/16	13	12	⅞	1	4¼				8 × 12⅛
10	17½	1⅞	14 1/16	15¼	16	1	1⅛	5				10 × 14¼
12	20½	2	16 7/16	17¾	16	1⅛	1¼	5½				12 × 16⅝
14 OD	23	2⅛	18 15/16	20¼	20	1⅛	1¼	5¾				13¼ × 19⅛
16 OD	25½	2¼	21 1/16	22½	20	1¼	1⅜	6				15¼ × 21¼
18 OD	28	2⅜	23 5/16	24¾	24	1¼	1⅜	6¼				17 × 23½
20 OD	30½	2½	25 9/16	27	24	1¼	1⅜	6½				19 × 25¾
24 OD	36	2¾	30 5/16	32	24	1½	1⅝	7½	9½			23 × 30½
30 OD	43	3	37 3/16	39¼	28	1¾	2	8¼	10½			29 × 37½
36 OD	50	3⅜	43 11/16	46	32	2	2¼	9¼	11½			34½ × 44
42 OD	57	3 11/16	50 7/16	52¾	36	2	2¼	9¾	12			40¼ × 50¾
48 OD	65	4	58 7/16	60¾	40	2	2¼	10½	13			46 × 58¾

* To obtain linear dimensions in cm, multiply tabular values in in by 2.54. To obtain areas in cm^2, multiply tabular values in in^2 by 6.45. To obtain stress in N/m^2, multiply values in lb/in^2 by 36.895.

† The stress shown is that of internal pressure assumed to act only on a circular area equal in diameter to the outside diameter of the ring gasket covering the flange to the inside of the bolts for the 25-lb standard.

SOURCE: ANSI B16.1-1975.

Elbows Special degree elbows ranging from 1 to 45° have the same center-to-face dimensions given for 45° elbows, and those over 45° and up to 90° have the same center-to-face dimensions given for 90° elbows. The angle designation of an elbow is its deflection from straight-line flow and is the angle between its flange faces.

Threaded Companion Flanges Threaded companion flanges in the class 25 standard should not be thinner than those in the class 125 standard on sizes 24 in (61 cm) and smaller. Other types of flanges may have thicknesses as given in Table 8.7.35.

Laterals Laterals (Y branches) both straight and reducing sizes 8 in and larger are reinforced to compensate for the inherent weakness in the casting design.

The American National Standard B16.1-1975 covers also dimensions (not included in the tables) of base elbows and base tees and anchorage bases for straight tees and reducing tees.

American National Standard cast-iron pipe flanges and flanged fittings are available for maximum nonshock working hydraulic pressure of 800 lb/in^2 gage (5,516 kPa) at ordinary air temperatures.

Assembly of Flanged Joints The optimum degree of tightening occurs when a stress of 30,000 lb/in^2 (207 MPa) is uniformly reached in each flange stud or bolt. For a modulus of elasticity of 30,000,000 lb/in^2 (207 GPa) a stress of 30,000,000 lb/in^2 occurs when the elongation, determined with a dial indicator or micrometer, is 0.001 in/in of stud length measured between centers of nuts. Uniform tension in flange bolts may also be obtained by use of a torque wrench; bearing surfaces of nuts must have a good machine finish, and threads must be properly lubricated for reliable results with a torque wrench. The following torque values have been found to give 30,000 lb/in^2 stress in studs:

Study diam, in	*Threads per inch*	*Torque, lb · ft**
⅝	11	89
¾	10	107
⅞	9	162
1	8	244
1⅛	8	322
1¼	8	410
1⅜	8	510
1½	8	615

* Multiply by 0.149 to obtain torque in kilogram-metres.

American National Standard Steel Pipe Flanges and Flanged Fittings

INTRODUCTORY NOTES

Pressure Ratings and Tests These standards are known as ''American Class 150, 300, 400, 600, 900, 1,500, and 2,500 Steel Flange Standards'' (ANSI B16.5-1981), said pressure designation being the recommended rating at the temperatures given in Table 8.7.35. This table shows recommended ratings for various temperatures. For other tables, refer to ANSI B16.5-1981.

Sizes The size of the fittings and companion flanges in the tables is identified by the corresponding nominal pipe size. For pipe NPS 14 (35.6 cm) and larger, the corresponding outside diameter of the pipe is given.

Materials The flanged fittings and flanges should be either steel castings or steel forgings of the grade complying with the ASTM specifications recommended under these standards for the various pressure-

Table 8.7.36 Pressure-Temperature Ratings for Carbon-Steel Flanges and Flanged Fittings, Working Pressure in lb/in^2 Gage

	Class 150			Class 300			Class 1500		
	Material group*			Material group*			Material group*		
	1.1	1.2	1.4	1.1	1.2	1.4	1.1	1.2	1.4
	Carbon steel			Carbon steel			Carbon steel		
Temperature, °F	Norm.	High	Low	Norm.	High	Low	Norm.	High	Low
− 20–100	285	290	235	740	750	620	3,705	3,750	3,085
200	260	260	215	675	750	560	3,375	3,750	2,810
300	230	230	210	655	730	550	3,280	3,640	2,735
400				635	705	530	3,170	3,530	2,645
500				600	665	500	2,995	3,325	2,490
600				550	605	455	2,735	3,025	2,285
650				535	590	450	2,685	2,940	2,245
700				535	570	450	2,665	2,840	2,245
750				505	505	445	2,520	2,520	2,210
800				410	410	370	2,060	2,060	1,850
850					270			1,340	
900					170			860	
950					105			515	
1,000					50			260	

Material group	Materials (spec. grade)	See notes
1.1	A105, A181-II, A216-WCB, A515-70	†, ¶
	A516-70	†, §
	A350-LF2, A537-C1.1	‡
1.2	A203-B, A203-E, A216-WCC	†, ¶
	A350-LF3, A352-LC2, A352-LC3	‡
1.4	A181-I, A515-60	†, ¶
	A516-60	†, §
	A350-LF1	‡

* Ratings shown apply to other material groups where columns dividing lines are omitted.
† Permissible but not recommended for prolonged use above about 800°F.
‡ Not to be used over 650°F.
§ Not to be used over 850°F.
¶ Not to be used over 1,000°F.
SOURCE: Abstracted from ASME B16.5-81 with permission.

temperature ratings for which these standards are designed. A few of these characteristics from ANSI B16.5-1981 are given in Table 8.7.36.

Bolting material including nuts and washers are based on a high-grade product equal to that given in ASTM Standard Specifications for Alloy-Steel Bolting Material for High Temperature Service (Table 8.7.37) and with physical and chemical requirements in accordance with the tables given under ANSI B16.5-1981. **Commercial steel bolts should not be used at steam pressures over 250** lb/in^2 (1,724 kPa) and temperatures over 450°F (232°C). Nuts should be of carbon or alloy steel. Washers when used under nuts should be of forged or rolled carbon steel.

Bolting Drilling templates are in multiples of four, so that fittings may be made to face in any quarter. Bolt-holes straddle the centerlines. Bolt-holes are drilled ⅛ in (0.32 cm) larger in diameter than the nominal size of bolt. Bolts or bolt studs threaded at both ends may be used and should be equipped with cold-punched or cold-pressed semifinished nuts of American National Standard rough dimensions, chamfered and trimmed.

All bolts and bolt studs having diameters 1 in and smaller, and the corresponding nuts are threaded with the American National Standard screw thread, coarse thread series, medium fit, class 3, while those bolts and bolt studs whose diameters are 1⅛ in (2.86 cm) and larger have special threads of the American National form whose pitch is ⅛ in (8 threads per inch). It is recommended that these special threads be allowed a pitch-diameter tolerance of $-$ 0.006 in and a lead tolerance of $\pm$ 0.002 in.

Bolt studs with a nut at each end are recommended for high-temperature service.

The allowable working fiber stress, considering internal allowable working pressure only, in bolting material for valve bonnet flanges, cleanout flanges, etc., is not to exceed 9,000 lb/in^2 (62 MPa) assuming the pressure to act upon an area circumscribed by the periphery of the outside of the contact surface.

Metal Thickness Minimum metal thicknesses specified in the tables are based on an allowable fiber stress of 7,000 lb/in^2 (48 MPa), using the modified Barlow formula of the ASME Boiler Construction Code for cylindrical sections and adding 50 percent to the thickness thus determined to compensate for the shape of the fittings. The minimum commercial casting thickness is considered to be ¼ in; therefore, the standards do not show thicknesses less than this. The minimum thickness in these standards means the minimum thickness in any part of the finished casting.

The modified Barlow formula is as follows: For pipes having nominal diameters of ¼ to 5 in, $P + 125 = 2S(t - 0.065)/D$. For pipes of nominal diameters over 5 in, $P = 2S(t - 0.1)/D$, where P is the working pressure, lb/in^2, t is the thickness of wall of pipe, in; D is the actual outside diameter of pipe, in; and S is 7,000 lb/in^2 (48 MPa).

Ring Joints The dimensions used for ring and groove joint facings were developed by a committee of the API. The corresponding dimensions and ring numbers incorporated in the ANSI Standard are identical with those given in API Standard 5G. The dimension for the depth of groove is added to the basic flange thickness, which makes it necessary to include separate tables of dimensions for fittings having the ring joint facing.

Fitting Dimensions An inspection limit of $\pm$ 1⁄32 in is allowed on all center-to-contact surface dimensions for sizes up to and including 10 in, and $\pm$ 1⁄16 in on sizes larger than NPS 10. An inspection limit of $\pm$ 1⁄16 in is allowed on all contact-surface to contact-surface dimensions for sizes up to and including NPS 10, and $\pm$ ⅛ in, on sizes larger than NPS 10.

When elbows having longer radii than specified in the standards are required, the use of pipe bends is recommended.

Laterals The 45° laterals of the larger sizes may require additional reinforcement to compensate for the inherent weakness in this shape of casting.

Valve Dimensions Face-to-face and end-to-end dimensions of ferrous valves for the various pressures are in accordance with the requirements of ANSI B16.10-1973 for ferrous flanged valves.

Reducing Fittings Reducing fittings have the same center-to-flange edge dimensions as those of straight-size fittings of the largest opening.

Side-Outlet Fittings All side-outlet fittings have all openings on the intersecting centerlines.

Welding Neck Flanges The materials, facings, spot facings, etc., conform to the requirements given for other flanges, with the additional provision that the carbon content of the steel shall not exceed 0.35 percent.

Templates for drilling and center to contact-surface dimensions of the American Standard class 150 steel flanges and flanged fittings are the same as for the American Standard class 125 cast-iron flanged fitting standard.

Templates for drilling and center to contact-surface dimensions of the ANSI class 300 steel flanges and flanged fittings are the same as for the ANSI class 600 steel flanged fitting standard for sizes NPS ½ to 1½ (Table 8.7.38); and the same as for the ANSI class 250 cast-iron flanged fitting standard for sizes NPS 2 to NPS 24.

Flanged Pipe Joints

The usual form of pipe joint is that made up by bolting together flanges cast or forged integral with the pipe or fitting, threaded flanges, loose flanges on pipes with lapped ends, and flanges arranged for welding. These forms are illustrated in Tables 8.7.39 and 8.7.40 and in Fig. 8.7.9. The threaded joint is satisfactory for low and medium steam pressures. The lapped joint is permitted in the same sizes and service ratings as for joints with integral flanges. It is extensively used in high-class work. With the ring joint a higher pressure can be maintained with the same total bolt stress than is possible with the flat gasket type of joint. The welded joint eliminates possibility of leakage between flange and pipe. It is very successful in lines subject to high temperatures and pressures and heavy expansion strains. The welding-neck flange is available in the various pipe sizes. Specific requirements covering the application of all the types of joints in common use are outlined in the Code for Power Piping (ASME B31.1 and B31.3).

Facing of Flanges Various styles of finish are used on the faces of flanges, for the purpose of the retention of the gasket used to make a tight joint. Those in general use are as follows (see Table 8.7.39): plain straight face, plain face corrugated or scored, male and female, tongue and groove, and raised face.

The **plain straight face** has the entire face of the flange faced straight across and may be used with either a full face or ring gasket. The **plain face, serrated or V-grooved,** is a plain face upon which concentric grooves have been cut with either a round-nose or V-shaped tool. This finish is sometimes of advantage when the service demands an exceptionally thick, loosely woven fibrous or soft metallic gasket, because the roughening of the faces of the flanges tends to keep the gasket from blowing out. The **male-and-female** facing consists of a recess in one flange and a corresponding raised face or projection on the other mating flange extending from the inside of the pipe nearly to the inside of the bolt holes. In the **tongue-and-groove** facing, the tongue or raised face and the groove or recess are narrow rings located between the bolt holes and the port. The male-and-female and the tongue-and-groove facing have been extensively used, particularly on hydraulic lines. To a more limited extent they have been used also on high-pressure steam lines. Both these types, however, have in common several objectionable features from the standpoint of manufacture, erection, and maintenance. These objections are removed by the use of the **raised-face** facing, which consists of a high narrow raised ring on each of the mating flanges, whose inside diameters is the same as that of the pipe or port. It is particularly recommended for high-pressure steam and hydraulic lines. **Gaskets** used in this type of joint are either soft fibrous material or soft metal and extend from the inside of the pipe to the bolt holes. Only the small portion in contact with the narrow raised face is subjected to the compressive effect of the bolts.

The following advantages are claimed for the raised-face type of facing: all mating of flanges has been eliminated; any valve or fitting may be removed from the line without springing the line apart; the gasket is automatically centered by its outer edge coming in contact with the bolts; the outside edges of the flanges are far enough apart to make it possible to determine whether the joint has been properly made.

Table 8.7.37 Mechanical Properties of Alloy Steel Bolting Material for High-Temperature Service

Ferritic steels

Grade	Diameter, in (mm)	Minimum tempering temperature,* °F (°C)	Tensile strength, min ksi (MPa)	Yield strength, min 0.2% offset, ksi (MPa)	Elongation in 2 in (50.8 mm), min. %	Reduction of area, min %	Hardness, max
B5, 4 to 6% chromium	Up to 4 (101.6), incl.	1,100 (593)	100 (690)	80 (550)	16	50	
B6, 13% chromium	Up to 4 (101.6), incl.	1,100 (593)	110 (760)	85 (585)	15	50	
B6X, 13% chromium	Up to 4 (101.6), incl.	1,100 (593)	90 (620)	70 (485)	16	50	26 HRC
B7, chromium-molybdenum	2½ (63.5) and under	1,100 (593)	125 (860)	105 (720)	16	50	
	Over 2½ to 4 (63.5 to 101.6)	1,100 (593)	115 (790)	95 (655)	16	50	
	Over 4 to 7 (101.6 to 117.8)	1,100 (593)	100 (690)	75 (515)	18	50	
B7M,† chromium-molybdenum	2½ (63.5) and under	1,150 (620)	100 (690)	80 (550)	18	50	235 HB or 99 HRB
B16, chromium-molybdenum-vanadium	2½ (63.5) and under	1,200 (650)	125 (860)	105 (720)	18	50	
	Over 2½ to 4 (63.5 to 101.6)	1,200 (650)	110 (760)	95 (655)	17	45	
	Over 4 to 7 (101.6 to 177.8)	1,200 (650)	100 (690)	85 (585)	16	45	

Austenitic steels

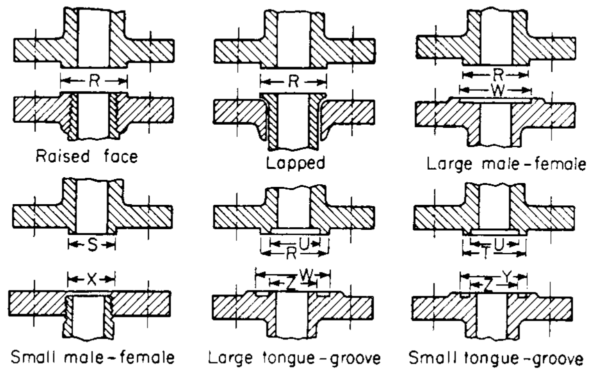
Class and grade, diameter, in (mm)	Heat treatment‡	Tensile strength, min ksi (MPa)	Yield strength, min 0.2% offset, ksi (MPa)	Elongation in 2 in (50.8 mm), min %	Reduction of area, min %	Hardness, max
Class 1: B8, B8C, B8M, B8P, B8T, B8LN, B8MLN, all diameters	Carbide solution treated	75 (515)	30 (205)	30	50	223 HB or 96 HRB
Class 1A: B8A, B8CA, B8MA, B8PA, B8TA, B8LNA, B8MLNA, B8NA, B8MNA, all diameters	Carbide solution treated in finished condition	75 (515)	30 (205)	30	50	192 HB or 90 HRB
Class 1B: B8N, B8MN, all diameters	Carbide solution treated	80 (550)	35 (240)	30	40	223 HB or 96 HRB
Class 1C: B8R, all diameters	Carbide solution treated	100 (690)	55 (380)	35	55	271 HB or 28 HRC
B8RA, all diameters	Carbide solution treated in finished condition	100 (690)	55 (380)	35	55	271 HB or 28 HRC
B8S, all diameters	Carbide solution treated	95 (655)	50 (345)	35	55	271 HB or 28 HRC
B8SA, all diameters	Carbide solution treated in finished condition	95 (655)	50 (345)	35	55	271 HB or 28 HRC
Class 2: B8, B8C, B8P, B8T, B8N, ¾ (19.05) and under	Carbide solution treated and strain-hardened	125 (860)	100 (690)	12	35	321 HB or 35 HRC
Over ¾ to 1 (19.05 to 25.4) incl.		115 (790)	80 (550)	15	35	321 HB or 35 HRC
Over 1 to 1¼ (25.4 to 31.6) incl.		105 (720)	65 (450)	20	35	321 HB or 35 HRC
Over 1¼ to 1½ (31.6 to 37.9) incl.		100 (690)	50 (345)	28	45	321 HB or 35 HRC
Class 2: B8M, B8MN, ¾ (19.05) and under	Carbide solution treated and strain-hardened	110 (760)	95 (655)	15	45	321 HB or 35 HRC
Over ¾ to 1 (19.05 to 25.4) incl.		100 (690)	80 (550)	20	45	321 HB or 35 HRC
Over 1 to 1¼ (25.4 to 31.6) incl.		95 (655)	65 (450)	25	45	321 HB or 35 HRC
Over 1¼ to 1½ (31.6 to 37.9) incl.		90 (620)	50 (345)	30	45	321 HB or 35 HRC

* For sizes ¾ in. in diameter and smaller, a maximum hardness of 241 HB (100 HRB) is permitted.
† To meet the tensile requirements, the Brinell hardness shall be over 201 HB (94 HRB).
‡ Class 1 is solution treated. Class 1A is solution treated in the finished condition for corrosion resistance: heat treatment is critical due to physical property requirement. Class 2 is solution-treated and strain-hardened. Austenitic steels in the strain-hardened condition may not show uniform properties throughout the section particularly in sizes over ¾ in (19.05 mm) in diameter.

Table 8.7.38 Templates for Drilling, American National Standard Steel Pipe Flanges and Flanged Fittings (ANSI B16.5-1981)
(All dimensions in inches)

	Class 400 standard					Class 600 standard					Class 900 standard					Class 1,500 standard				
Nominal pipe size	Outside diam of flange	Thickness of flange, minimum	Diam of bolt circle	Number of bolts	Size of bolts	Outside diam of flange	Thickness of flange, minimum	Diam of bolt circle	Number of bolts	Size of bolts	Outside diam of flange	Thickness of flange, minimum	Diam of bolt circle	Number of bolts	Size of bolts	Outside diam of flange	Thickness of flange, minimum	Diam of bolt circle	Number of bolts	Size of bolts
½	For sizes below 4 in use					3¾	9⁄16	2⅝	4	½	For sizes below 3 in use					4¾	⅞	3¼	4	¾
¾	dimensions of 600-lb					4⅝	⅝	3¼	4	⅝	dimensions of 1,500-lb					5⅛	1	3½	4	¾
1	fittings					4⅞	11⁄16	3½	4	⅝	fittings					5⅞	1⅛	4	4	1⅞
1¼	—	—	—	—	—	5¼	13⁄16	3⅞	4	⅝	—	—	—	—	—	6¼	1⅛	4⅜	4	⅞
1½	—	—	—	—	—	6⅛	⅞	4½	4	¾	—	—	—	—	—	7	1¼	4⅞	4	1
2	—	—	—	—	—	6½	1	5	8	⅝	—	—	—	—	—	8½	1½	6½	8	⅞
2½	—	—	—	—	—	7½	1⅛	5⅞	8	¾	—	—	—	—	—	9⅝	1⅝	7½	8	1
3	—	—	—	—	—	8¼	1¼	6⅝	8	¾	9½	1½	7½	8	⅞	10½	1⅞	8	8	1⅛
3½	—	—	—	—	—	9	1⅜	7¼	8	⅞										
4	10	1⅜	7⅞	8	⅞	10¾	1½	8½	8	⅞	11½	1¾	9¼	8	1⅛	12¼	2⅛	9½	8	1¼
5	11	1½	9¼	8	⅞	13	1¾	10½	8	1	13¾	2	11	8	1¼	14¾	2⅞	11½	8	1½
6	12½	1⅝	10⅝	12	⅞	14	1⅞	11½	12	1	15	2 3⁄16	12½	12	1⅛	15½	3¼	12½	12	1⅜
8	15	1⅞	13	12	1	16½	2 3⁄16	13¾	12	1⅛	18½	2½	15½	12	1⅜	19	3⅝	15½	12	1⅝
10	17½	2⅛	15¼	16	1⅛	20	2½	17	16	1¼	21½	2¾	18½	16	1⅜	23	4¼	19	12	1⅞
12	20½	2¼	17¾	16	1¼	22	2⅝	19¼	20	1¼	24	3⅛	21	20	1⅜	26½	4⅞	22½	16	2
14 OD	23	2⅜	20¼	20	1¼	23¾	2¾	20¾	20	1⅜	25¼	3⅜	22	20	1½	29½	5¼	25	16	2¼
16 OD	25½	2½	22½	20	1⅜	27	3	23¾	20	1½	27¾	3½	24¼	20	1⅝	32½	5¾	27¾	16	2½
18 OD	28	2⅝	24¾	24	1⅜	29¼	3¼	25¾	20	1⅝	31	4	27	20	1⅞	36	6⅜	30½	16	2¾
20 OD	30½	2¾	27	24	1½	32	3½	28½	24	1⅝	33¾	4¼	29½	20	2	38¾	7	32¾	16	3
24 OD	36	3	32	24	1¾	37	4	33	24	1⅞	41	5½	35½	20	2½	46	8	39	16	3½

Table 8.7.39 Facing Dimensions for the American Class 150, 300, 400, 600, 900, 1,500, and 2,500 Steel Flanges (ANSI B16.5-1981)



Nominal pipe size, in	Outside diameter, in: Raised face, lapped, large male, and large tongue, *R*	Outside diameter, in: Small male, *S*	Outside diameter, in: Small tongue, *T*	ID of large and small tongue, *U*	Outside diameter, in: Large female and large groove, *W*	Outside diameter, in: Small female, *X*	Outside diameter, in: Small groove, *Y*	ID of large and small groove, *Z*	Height, in: Raised face, 150 and 300 lb std*	Height, in: Raised face, large and small male and tongue 400-, 600-, 900-, 1,500- and 2,500-lb studs	Depth of groove or female companion flanges
½	1⅜	23⁄32	1⅜	1	1 7⁄16	25⁄32	1 7⁄16	15⁄16	1⁄16	¼	3⁄16
¾	1 11⁄16	15⁄16	1 11⁄16	1 5⁄16	1¾	1	1¾	1¼	1⁄16	¼	3⁄16
1	2	1 3⁄16	1⅞	1½	2 1⁄16	1¼	1 15⁄16	1 7⁄16	1⁄16	¼	3⁄16
1¼	2½	1½	2¼	1⅞	2 9⁄16	1 9⁄16	2 5⁄16	1 13⁄16	1⁄16	¼	3⁄16
1½	2⅞	1¾	2½	2⅛	2 15⁄16	1 13⁄16	2 9⁄16	2 1⁄16	1⁄16	¼	3⁄16
2	3⅝	2¼	3¼	2⅞	3 11⁄16	2 5⁄16	3 5⁄16	2 13⁄16	1⁄16	¼	3⁄12
2½	4⅛	2 11⁄16	3¾	3⅜	4 3⁄16	2¾	3 13⁄12	3 5⁄16	1⁄16	¼	3⁄16
3	5	3 5⁄16	4⅝	4¼	5 1⁄16	3⅜	4 11⁄16	4 3⁄16	1⁄16	¼	3⁄16
3½	5½	3 13⁄16	5⅛	4¾	5 9⁄16	3⅞	5 3⁄16	4 11⁄16	1⁄16	¼	3⁄16
4	6 3⁄16	4 5⁄16	5 11⁄16	5 3⁄16	6¼	4⅜	5¾	5⅛	1⁄16	¼	3⁄16
5	7 5⁄16	5⅜	6 13⁄16	6 5⁄16	7⅜	5 7⁄16	6⅞	6¼	1⁄16	¼	3⁄16
6	8½	6⅜	8	7½	8 9⁄16	6 7⁄16	8 1⁄16	7 7⁄16	1⁄16	¼	3⁄16
8	10⅝	8⅜	10	9⅜	10 11⁄16	8 7⁄16	10 1⁄16	9 5⁄16	1⁄16	¼	3⁄16
10	12¾	10½	12	11¼	12 13⁄16	10 9⁄16	12 1⁄16	11 3⁄16	1⁄16	¼	3⁄16
12	15	12½	14¼	13½	15 1⁄16	12 9⁄16	14 5⁄16	13 7⁄16	1⁄16	¼	3⁄16
14 OD	16¼	13¾	15½	14¾	16 5⁄16	13 13⁄16	15 9⁄16	14 11⁄16	1⁄16	¼	3⁄16
16 OD	18½	15¾	17⅝	16¾	18 9⁄16	15 13⁄16	17 11⁄16	16 11⁄16	1⁄16	¼	3⁄16
18 OD	21	17¾	20⅛	19¼	21 1⁄16	17 13⁄16	20 3⁄16	19 3⁄16	1⁄16	¼	3⁄16
20 OD	23	19¾	22	21	23 1⁄16	19 13⁄16	22 1⁄16	20 15⁄16	1⁄16	¼	3⁄16
24 OD	27¼	23¾	26¼	25¼	27 5⁄16	23 13⁄16	26 5⁄16	25 3⁄16	1⁄16	¼	3⁄16

* Included in the minimum flange thickness dimensions. A 1⁄16-in raised face is also permitted on the class 400, 600, 900, 1,500, and 2,500 flange standards, but it must be added to the minimum flange thicknesses. Regular facing for class 400, 600, 900, 1,500, and 2,500 flange standards is a ¼-in raised face not included in minimum flange thickness dimensions.

A tolerance of 1⁄64 in is allowed on the inside and outside diameters of all facings.

Gaskets for male-female and tongue-groove joints should cover the bottom of the recess with minimum clearance taking into account the tolerances stated above.

Unions may be classified as **screw** and **flange**. Typical designs are shown in Fig. 8.7.10, where at the top left is represented a female screw union of the gasket type, at the top right a female screw union having a brass to iron seat that is noncorrosive and a ground joint that eliminates the need for a gasket, and at the bottom a flange union of the gasket type. As in the case of other pipe fittings, unions and union fittings are available in the various pipe sizes and in materials and designs suitable for any service conditions. Very large flange unions can be made by bolting together two screwed companion flanges.

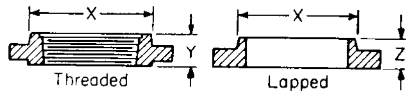
Threaded Fittings

Threaded fittings are made of cast iron, malleable iron, cast steel, forged steel, or brass. Plain standard fittings are generally used for low-pressure gas and water, as in house plumbing and railing work, while the beaded fitting is the standard steam, air, gas, or oil fitting. Screwed fittings are supplied with a large factor of safety. The questions of strength involve much more than the pressure from within the pipe which induces a comparatively low stress in the material. The greater strains come from expansion, contraction, weight of piping, settling, water hammer, etc. Dimensions of cast-iron and malleable-iron screwed fittings of the American National Standard are given in Tables 8.7.41 and 8.7.42.

The dimensions of ferrous plugs, bushings, locknuts, and caps with pipe threads are covered by ANSI B16.14-1983. The dimensions of pipe plugs from this standard are given in Table 8.7.43.

The normal **amount of thread engagement** necessary to make a tight

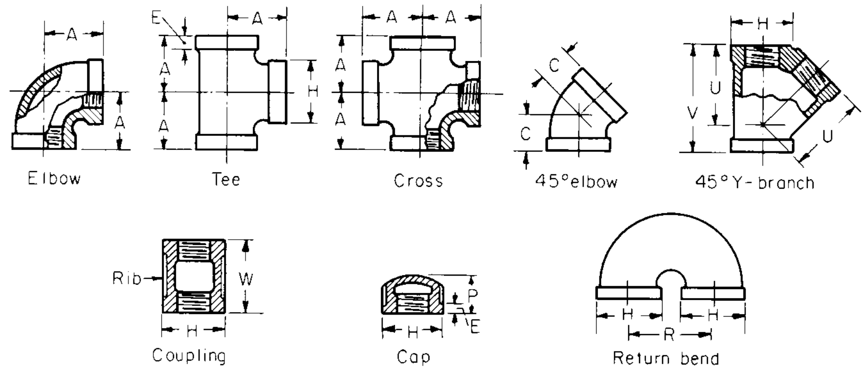
Table 8.7.40 Dimensions of American National Standard Companion Flanges (ANSI B16.5-1981)*



Nominal pipe size	Class 150			Class 300			Class 400			Class 600			Class 900			Class 1,500		
	X	*Y*	*Z*	*X*	*Y*	*Z*	*X*	*Y*	*Z*	*X*	*Y*	*Z*	*X*	*Y*	*Z*	*X*	*Y*	*Z*
½	1 3/16	⅝	⅝	1½	⅞	⅞		For sizes below 4 in,		1½	⅞	⅞		For sizes below 3 in,		1½	1¼	1¼
¾	1½	⅝	⅝	1⅞	1	1		use dimensions of		1⅞	1	1		use dimensions of		1¾	1⅜	1⅜
1	1 15/16	11/16	11/16	2⅛	1 1/16	1 1/16		600-lb flanges		2⅛	1 1/16	1 1/16		1,500-lb flanges		2 1/16	1⅝	1⅝
1¼	2 5/16	13/16	13/16	2½	1 1/16	1 1/16	—	—	—	2½	1⅛	1⅛	—	—	—	2½	1⅝	1⅝
1½	2 9/16	⅞	⅞	2¾	1 3/16	1 3/16	—	—	—	2¾	1¼	1¼	—	—	—	2¾	1¾	1¾
2	3 1/16	1	1	3 5/16	1 5/16	1 5/16	—	—	—	3 5/16	1 7/16	1 7/16	—	—	—	4⅛	2¼	2¼
2½	3 9/16	1⅛	1⅛	3 15/16	1½	1½	—	—	—	3 15/16	1⅝	1⅝	—	—	—	4⅞	2½	2½
3	4¼	1 3/16	1 3/16	4⅝	1 11/16	1 11/16	—	—	—	4⅝	1 13/16	1 13/16	5	2⅛	2⅛	5¼	2⅞	2⅞
3½	4 13/16	1¼	1¼	5¼	1¾	1¾	—	—	—	5¼	1 15/16	1 15/16						
4	5 5/16	1 5/16	1 5/16	5¾	1⅞	1⅞	5¾	2	2	6	2⅛	2⅛	6¼	2¾	2¾	6⅜	3 9/16	3 9/16
5	6 7/16	1 7/16	1 7/16	7	2	2	7	2⅛	2⅛	7 7/16	2⅜	2⅜	7½	3⅛	3⅛	7¾	4⅛	4⅛
6	7 9/16	1 9/16	1 9/16	8⅛	2 1/16	2 1/16	8⅛	2¼	2¼	8¾	2⅝	2⅝	9¼	3⅜	3⅜	9	4 11/16	4 11/16
8	9 11/16	1¾	1¾	10¼	2 7/16	2 7/16	10¼	2 11/16	2 11/16	10¾	3	3	11¾	4	4½	11½	5⅝	5⅝
10	12	1 15/16	1 15/16	12⅝	2⅝	3¾	12⅝	2⅞	4	13½	3⅜	4⅜	14½	4¼	5	14½	6¼	7
12	14⅜	2 3/16	2 3/16	14¾	2⅞	4	14¾	3⅛	4¼	15¾	3⅝	4⅝	16½	4⅝	5⅝	17¾	7⅛	8⅝
14 OD	15¾	2¼	3⅛	16¾	3	4⅜	16¾	3 5/16	4⅝	17	3 11/16	5	17¾	5⅛	6⅛	19½	—	9½
16 OD	18	2½	3 7/16	19	3¼	4¾	19	3 11/16	5	19½	4 3/16	5½	20	5¼	6½	21¾	—	10¼
18 OD	19⅞	2 11/16	3 13/16	21	3½	5⅛	21	3⅞	5⅜	21½	4⅝	6	22¼	6	7½	23½	—	10⅞
20 OD	22	2⅞	4 1/16	23⅛	3¾	5½	23⅛	4	5¾	24	5	6¼	24½	6½	8¼	25¼	—	11½
24 OD	26⅛	3¼	4⅜	27⅝	4 3/16	6	27⅝	4½	6¼	28¼	5½	7¼	29½	8	10½	30	—	13

* Other dimensions are given in Tables 8.7.38 and 8.7.39. Finished bore on lapped flange to be such as method of attachment of pipe requires.

Table 8.7.41 Dimensions of ANSI Class 150 Standard Malleable-Iron Threaded Fittings*
(All dimensions in inches)



Size	*A*	*H*	*E*	*C*	*V*	*U*	*W*	*P*	*R* Close	*R* Medium	*R* Open
⅛	0.69	0.693	0.200				0.96				
¼	0.81	0.844	0.215	0.73			1.06				
⅜	0.95	1.015	0.230	0.80	1.93	1.43	1.16				
½	1.12	1.197	0.249	0.88	2.32	1.71	1.34	0.87	1.000	1.25	1.50
¾	1.31	1.458	0.273	0.98	2.77	2.05	1.52	0.97	1.250	1.50	2.00
1	1.50	1.771	0.302	1.12	3.28	2.43	1.67	1.16	1.500	1.875	2.50
1¼	1.75	2.153	0.341	1.29	3.94	2.92	1.93	1.28	1.750	2.25	3.00
1½	1.94	2.427	0.368	1.43	4.38	3.28	2.15	1.33	2.188	2.50	3.50
2	2.25	2.963	0.422	1.68	5.17	3.93	2.53	1.45	2.625	3.00	4.00
2½	2.70	3.589	0.478	1.95	6.25	4.73	2.88	1.70			4.50
3	3.08	4.285	0.548	2.17	7.26	5.55	3.18	1.80			5.00
3½	3.42	4.843	0.604	2.39			3.43	1.90			
4	3.79	5.401	0.661	2.61	8.98	6.97	3.69	2.08			
5	4.50	6.583	0.780	3.05				2.32			
6	5.13	7.767	0.900	3.46				2.55			

* The complete standard (ANSI B16.3-1977) covers also reducing couplings, elbows, tees, crosses, and service or street elbows and tees.
SOURCE: ANSI B16.3-1977.

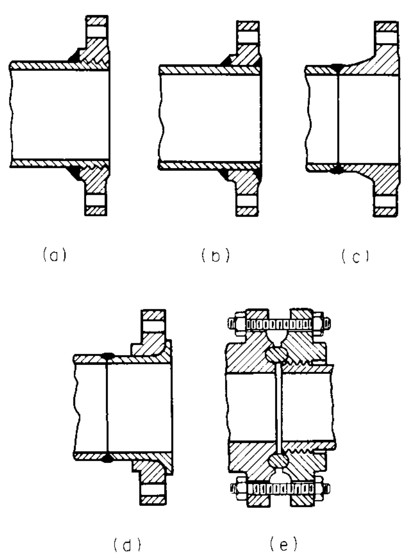


Fig. 8.7.9 Welded flange joints and ring joint. (*a*) Forged steel, screwed flange, back-welded and refaced; (*b*) forged steel, slip-on welding flange, welded front and back, refaced; (*c*) forged steel, welding neck flange, butt-welded to pipe; (*d*) lap-welding nipple, butt-welded to pipe; (*e*) ring joint.

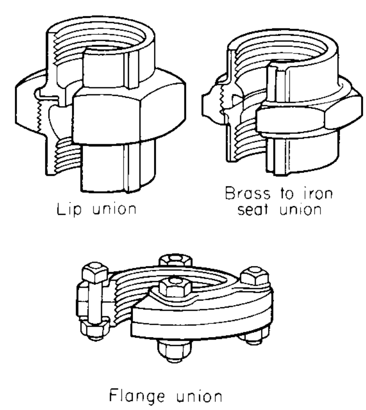


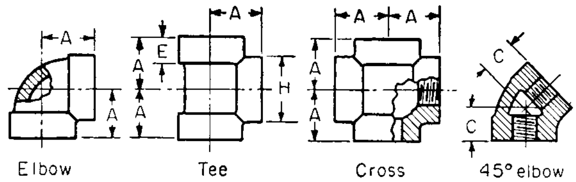
Fig. 8.7.10 Types of pipe unions.

joint for ANSI Standard pipe thread joints as recommended by Crane Co. is as follows:

Size of pipe, in	⅛	¼	⅜	½	¾	1	1¼	1½	2
Length of thread, in	¼	⅜	⅜	½	9⁄16	11⁄16	11⁄16	11⁄16	¾

Size of pipe, in	2½	3	3½	4	5	6	8	10	12
Length of thread, in	15⁄16	1	1 1⁄16	1⅛	1¼	1 5⁄16	1 7⁄16	1⅝	1¾

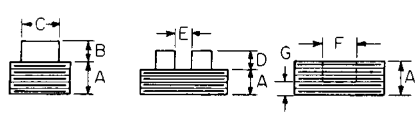
Table 8.7.42 Dimensions of Class 125 and Class 250 Standard Cast-Iron Threaded Fittings*
(All dimensions in inches)



	Class 125				Class 250			
Size	*A*	*H*	*E*	*C*	*A*	*H*	*E*	*C*
¼	0.81	0.93	0.38	0.73	0.94	1.17	0.49	0.81
⅜	0.95	1.12	0.44	0.80	1.06	1.36	0.55	0.88
½	1.12	1.34	0.50	0.88	1.25	1.59	0.60	1.00
¾	1.13	1.63	0.56	0.98	1.44	1.88	0.68	1.13
1	1.50	1.95	0.62	1.12	1.63	2.24	0.76	1.31
1¼	1.75	2.39	0.69	1.29	1.94	2.73	0.88	1.50
1½	1.94	2.68	0.75	1.43	2.13	3.07	0.97	1.69
2	2.25	3.28	0.84	1.68	2.50	3.74	1.12	2.00
2½	2.70	3.86	0.94	1.95	2.94	4.60	1.30	2.25
3	3.08	4.62	1.00	2.17	3.38	5.36	1.40	2.50
3½	3.42	5.20	1.06	2.39	3.75	5.98	1.49	2.63
4	3.79	5.79	1.12	2.61	4.13	6.61	1.57	2.81
5	4.50	7.05	1.18	3.05	4.88	7.92	1.74	3.19
6	5.13	8.28	1.28	3.46	5.63	9.24	1.91	3.50
8	6.56	10.63	1.47	4.28	7.00	11.73	2.24	4.31
10	8.08	13.12	1.68	5.16	8.63	14.37	2.58	5.19
12	9.50	15.47	1.88	5.97	10.00	16.84	2.91	6.00

* This applies to elbows and tees only.
The class 125 standard covers also reducing elbows and tees. The class 250 standard covers only the straight sizes.
SOURCE: ANSI B16.4-1971.

Table 8.7.43 Dimensions of Class 125, 150, and 250 Pipe Plugs*
(All dimensions in inches)



Nominal pipe size	Square-head pattern			Slotted pattern			Countersunk pattern† (square sockets)		
	A	*B*	*C*	*A*	*D*	*E*	*A*	*F*	*G*
⅛	0.37	0.24	9⁄32						
¼	0.44	0.28	⅜						
⅜	0.48	0.31	7⁄16						
½	0.56	0.38	9⁄16				0.56	⅜	0.16
¾	0.63	0.44	⅝				0.63	½	0.18
1	0.75	0.50	13⁄16				0.75	½	0.20
1¼	0.80	0.56	15⁄16				0.80	¾	0.22
1½	0.83	0.62	1⅛				0.83	¾	0.24
2	0.88	0.68	1 5⁄16				0.88	⅞	0.26
2½	1.07	0.74	1½				1.07	1⅛	0.29
3	1.13	0.80	1 11⁄16				1.13	1⅜	0.31
3½	1.18	0.86	1⅞				1.18	1½	0.34
4				1.22	1.00	0.88	1.22	2	0.37
5				1.31	1.00	0.88	1.31	2¼	0.46
6				1.40	1.25	1.25	1.40	2½	0.52
8				1.57	1.38	1.50			

* The material of (ANSI B16.14-1983) is to be cast iron, malleable iron, or steel, for use in connection with fittings covered by the American National Standard class 125 cast-iron threaded fittings (ANSI B16.4) and the American National Standard class 150 malleable-iron screwed fittings (ANSI B16.3).
† Hexagon sockets (sizes ⅛ to 1 in) have dimensions to fit regular wrenches used with hexagon socket setscrews.
SOURCE: ANSI B16.14-1983.

The Manufacturers' Standardization Society of Valve and Fitting Industry (MSS) has standardized malleable-iron and brass threaded fittings for several pressures.

Cast-bronze threaded fittings are made in both the class 125 and 250 standards. They are used for any water pipe where bad water makes steel pipe undesirable. Bronze fittings may be had in iron pipe sizes. Forged-steel threaded fittings are made for cold water or oil-working pressures up to 6,000 lb/in^2 (41.4 MPa) hydrostatic. The ANSI has approved standard B16.26-1983 for cast copper alloy fittings for flared copper tubes for maximum cold-water service pressure of 175 lb/in^2 (1,207 kPa).

Railing Fittings Fittings of special construction and of tighter weight than standard steam, gas, and water pipe fittings are widely used for hand railings around areaways, on stairs, for office enclosures with gates, and for permanent ladders. Railing fittings are made in various styles, generally globe-shaped in body, with ends reduced to take thread and recessed to cover all threads. They are furnished in malleable iron, black and galvanized, and in brass.

Special railing-fitting joints are available, such as the slip-and-screwed joint, where the post connection is screwed and the rim of the fitting is so made that the rail will slip into the fitting and allow for an angular variation of several degrees, being fastened by pins which are riveted over and filed smooth. The flush-joint stair-rail fitting is another special style of fitting which provides a hand rail with even surfaces at the joints.

Drainage fittings, as shown in the figures accompanying Table 8.7.44, have no pockets for the lodgment of solids, and the length of the thread chamber is such that when the pipe is threaded to the American National Standard dimensions, the end of the pipe will practically touch the shoulder when screwed in. They are especially adapted to plumbing work and vacuum-cleaning pipe installations. Dimensions in Table 8.7.44 conform to ANSI Standard B16.12-1983, Cast-Iron Threaded Drainage Fittings.

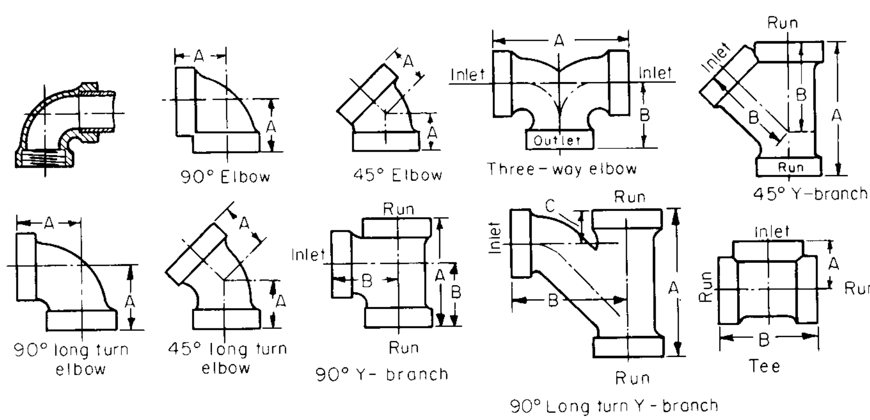
The development of standards for **cast-iron long-turn sprinkler fittings** was begun by the National Fire Protection Assoc. in 1914 with a study of the peculiar needs of fittings intended for fire-protection purposes. These fittings (screwed and flanged) are rated at 175 and 250 lb/in^2 (1,207 and 1,724 kPa).

American National Standard Air Gaps and Backflow Preventers in Plumbing Systems, ANSI A40.4-1942 and A40.6-1943, was prepared to establish minimum requirements for plumbing, including water-supply distributing systems, drainage and venting systems, fixtures, apparatus, and devices, and the standardization of plumbing equipment in general.

Ammonia valves and fittings must provide a high margin of safety against accidents. Flanged valves and fittings have tongue-and-groove faces to assure tightness at the joints and against blowing out gaskets. Gaskets are compressed asbestos sheet. Threaded valves and fittings have long threads and are recessed so that the joints may be soldered. These valves and fittings are made of malleable iron, ductile iron, ferrosteel, or forged steel; depending on the size and style. Valves are all iron, with steel stems, and have special lead disk faces or steel disks. Copper or brass must not be used in their construction. Flanged valves are generally interchangeable with flanged fittings. All valves and fittings for ammonia are tested to 300 lb/in^2 (2,069 kPa) air pressure under water. For dimensions of valves, fittings, and specialties for ammonia, refer to manufacturers' catalogs.

Soldered-Joint Fittings The American standard for these fittings

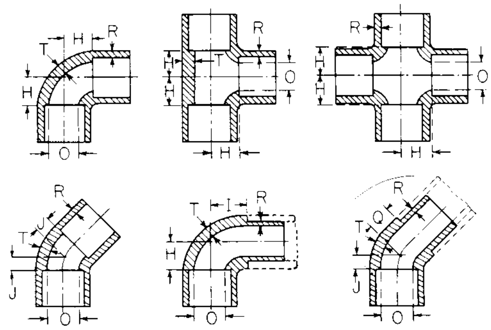
Table 8.7.44 Dimensions of American National Standard Cast-Iron Threaded Drainage Fittings
(All dimensions in inches)



Size, in	90° elbows*	45° elbows*	90° long-turn elbows	45° long-turn elbows	Three-way elbows†		Tees*		90° Y branches		90° long-turn *Y* branches			45° Y branches*	
	A	*A*	*A*	*A*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*C*	*A*	*B*
1¼	1¾	1⁵⁄₁₆	2¼	1¾	4½	2¼	1¾	3½	3¾	2¼	4¾	3⅝	1⅛	5	3¼
1½	1¹⁵⁄₁₆	1⁷⁄₁₆	2½	1⅞	5	2½	1¹⁵⁄₁₆	3⅞	4¼	2½	5⅜	4⅛	1¼	5½	3⅝
2	2¼	1¹¹⁄₁₆	3¹⁄₁₆	2¼	6⅛	3¹⁄₁₆	2¼	4½	5³⁄₁₆	3¹⁄₁₆	7	5¼	1¾	6½	4⅜
2½	2¹¹⁄₁₆	1¹⁵⁄₁₆	3¹¹⁄₁₆	2⅝	7⅝	3¹¹⁄₁₆	2¹¹⁄₁₆	5⅜	6⁵⁄₁₆	3¹¹⁄₁₆	8¼	6¼	2	7⅞	5⅜
3	3¹⁄₁₆	2³⁄₁₆	4¼	2¹⁵⁄₁₆	8½	4¼	3¹⁄₁₆	6⅛	7¼	4¼	9⅞	7½	2⅜	9	6³⁄₁₆
4	3¹³⁄₁₆	2⅝	5³⁄₁₆	3½	10⅜	5³⁄₁₆	3¹³⁄₁₆	7⅝	8¾	5³⁄₁₆	13	9⅞	3⅛	10⅞	7¹¹⁄₁₆
5	4½	3¹⁄₁₆	6⅛	4⅛	12¼	6⅛	4½	9	10⁵⁄₁₆	6⅛	15¾	12¼	3½	12¹⁵⁄₁₆	9¾
6	5⅛	3⁷⁄₁₆	7⅛	4⅞	14¼	7⅛	5⅛	10¼	11¹⁵⁄₁₆	7⅛	18¾	14⅝	4⅛	14⅞	10¾

* Same as adopted for Class 125 Cast-iron Threaded Fittings, ANSI B16.4-1983.
† Three-way elbows have same dimensions as 90° long-radius elbows.
Double Y branches have the same dimensions as single Y branches.
Other fittings which are available are as follows: 5⅝, 11¼, and 60° elbows; basin tees and crosses; double 90° Y branches; double 90° long-turn Y branches; 45° double Y branches; S traps; half S traps; offsets, couplings, increasers, and reducing sizes.
SOURCE: ANSI B16.2-1983.

Table 8.7.45 Soldered-Joint Fittings—Dimensions of Elbows, Tees, and Crosses
(All dimensions in inches)



Nominal size	Cast brass†							Wrought metal
	*H**	*I*	*J*	*Q*	*O*‡	*T*	*R*	(*T* and *R*)§,¶
¼	¼	⅜	3⁄16	¼	0.31	0.08	0.048	0.030
⅜	5⁄16	7⁄16	3⁄16	5⁄16	0.43	0.08	0.048	0.035
½	7⁄16	9⁄16	3⁄16	5⁄16	0.54	0.09	0.054	0.040
¾	9⁄16	11⁄16	¼	⅜	0.78	0.10	0.060	0.045
1	¾	⅞	5⁄16	7⁄16	1.02	0.11	0.066	0.050
1¼	⅞	1	7⁄16	9⁄16	1.26	0.12	0.072	0.055
1½	1	1⅛	½	⅝	1.50	0.13	0.078	0.060
2	1¼	1⅜	9⁄16	¾	1.98	0.15	0.090	0.070
2½	1½	1⅝	⅝	⅞	2.46	0.17	0.102	0.080
3	1¾	1⅞	¾	1	2.94	0.19	0.114	0.090
3½	2	2⅛	⅞	1⅛	3.42	0.20	0.120	0.100
4	2¼	2⅜	15⁄16	1¼	3.90	0.22	0.132	0.110
5	3⅛		17⁄16		4.87	0.28	0.168	0.125
6	3⅝		1⅝		5.84	0.34	0.204	0.140

Wrought fittings as well as cast fittings, must be provided with a shoulder or stop at the bottom end of socket.
* Dimensions for reducing elbows, reducing crosses, reducing tees, couplings, caps, bushings, adapters, and fittings with pipe thread on one end are also included in this standard.
† These dimensions may be used for wrought metal fittings as well as for cast brass fittings at manufacturer's option.
‡ This dimension is the same as the inside diameter class L tubing (ASTM B88-1983).
§ This dimension has the same thickness as class L tubing.
¶ These dimensions are minimum, but in every case the thickness of wrought fittings should be at least as heavy as the tubing with which it is to be used.
SOURCE: ANSI B16.18-1984.

(ANSI B16.18-1984) covers certain dimensions of soldered-joint wrought metal and cast brass fittings for copper water tubing including (1) detailed dimensions of the bore, (2) minimum specifications for materials, (3) minimum inside diameter of the fitting, (4) metal thickness for both wrought metal and cast brass fittings, and (5) general dimensions for cast brass fittings including center-to-shoulder dimensions for both straight and reducing cast fittings. Pressure and temperature ratings are also given. Sizes of the fittings are identified by the nominal tubing size as covered by the Specifications for Copper Water Tube (ASTM B88-1983). Dimensions of some of the fittings from this standard are given in Table 8.7.45.

Valves

The face-to-face dimensions of ferrous flanged and welding end valves are given in ANSI B16.10-1973. The types covered are:

Wedge-Gate Valves Cast iron, for 125-, 175-, and 250-lb/in² (862, 1,207, and 1,724 kPa) steam service pressure and 800-lb/in² (5,516 kPa) hydraulic pressure, and steel, for 150-, 300-, 400-, 600-, 900-, and 1,500-lb/in² (1,034-, 2,068-, 2,758-, 4,137-, 6,206-, and 10,343-kPa) steam service pressures (see Fig. 8.7.11).

Double-Disk Gate Valves Cast iron, for 125- and 250-lb/in² (862- and 1,724-kPa) steam service pressure and 800-lb/in² (5,516-kPa) hydraulic pressure.

Globe and Angle Valves Cast iron, for 125- and 250-lb/in² (862- and 1,724-kPa) steam service pressure, and steel, for 150-, 300-, 400-, 600-, 900-, 1,500-, and 2,500-lb/in² (862-, 2,068-, 2,758-, 4,137-, 6,206-, 10,343-, and 17,238-kPa) steam service pressures (see Fig. 8.7.12).

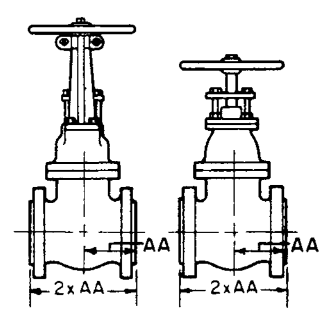


Fig. 8.7.11 Wedge gate valves.

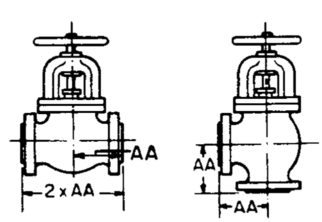


Fig. 8.7.12 Globe valve and angle valve.

Swing-Check Valves Cast iron, for 125- and 250-lb/in^2 (862- and 1,724-kPa) steam service pressure and 800-lb/in^2 (5,516-kPa) hydraulic pressure, and steel, for 150-, 300-, 400-, and 600-lb/in^2 (1,034-, 2,068-, 2,758-, and 4,137-kPa) steam service pressures.

Except for ring-joint facings to the face-to-face dimension for flanged valves is the distance between the faces of the connecting end flanges upon which the gaskets are actually compressed, i.e., the ''contact surfaces.''

All flanges for class 125 cast-iron valves are plain-faced. The facings of the class 250 cast-iron, and the class 150 and 300 steel valves have a 1⁄16-in raised face which is included in the contact-surface to contact-surface dimensions. The contact-surface to contact-surface dimensions of steel valves for class 400 and higher pressures and for cast-iron valves for class 800 hydraulic pressure include a ¼-in raised face.

The end-to-end dimensions for **welding-end** valves for sizes NPS 1 to 8 are the same as the contact-surface to contact-surface dimensions given in the tables for steel valves. For details of welding bevel see ANSI B16.10-1973 and Fig. 8.7.15.

A plus or minus **tolerance** of 1⁄16 in is allowed on all face-to-face dimensions of valves NPS 10 and smaller, and a tolerance of ⅛ on sizes NPS 12 and larger.

Cocks The ordinary plug cock operated by a handle or wrench is a form of valve in comparatively small sizes suitable for ordinary service only. The ASME Code for Pressure Piping requires that where cocks are used for high-temperature service they shall be so designed as to prevent galling, either by making the plugs of different material from the body of the cock or by treating the plugs to ensure different physical properties. By means of special design features that eliminate the tendency to leak and stick, the plug-cock type of valve has become available in large sizes and for severe service conditions. Sizes are listed as high as 30 in and are gear-operated in the larger sizes. For further details, refer to manufacturers' catalogs.

Expansion and Flexibility

Piping systems must be designed so that they (1) will not fail because of excessive stresses, (2) will not produce excessive thrusts or moments at connected equipment, or (3) will not leak at joints because of expansion of the pipe. Flexibility is provided by changes of direction in the piping through the use of bends or loops, or provision may be made to absorb thermal strains by use of expansion joints. All, or portions, of the pipe may be corrugated to improve flexibility; in many systems, however, sufficient change is provided by the geometry of the layout to make unnecessary the use of either expansion joints or corrugated sections of piping. Proper cold springing is beneficial in assisting the piping system to attain its most favorable condition. Because of plastic flow of the piping material, hot stresses tend to decrease with time while cold stresses tend to increase with time; their sum, called the stress range, remains substantially constant. For this reason no credit is warranted with regard to stresses; for calculation of forces and moments, the effect of cold spring is recognized by use of a cold-spring factor varying from 0 to 1 for cold spring varying from 0 to 100 percent.

The allowable stress range S_A is calculated by

$$S_A = f(1.25S_c + 0.25S_h)$$

where S_c and S_h are the S values for the minimum cold and maximum hot conditions, respectively, as given in Table 8.7.15. The stress-reduction factor f is a function of the number of hot-to-cold-to-hot (full) temperature cycles anticipated over the life of the plant, as follows:

Total no. of full temp cycles over expected life	*Stress-reduction factor*
7,000 and less	1.0
14,000 and less	0.9
22,000 and less	0.8
45,000 and less	0.7
100,000 and less	0.6
Over 100,000	0.5

The bending and torsional stresses calculated (see paragraph 119.6.4 of ANSI B31.1.0-1983) are used to determine the maximum computed expansion stress $S_E = \sqrt{S_b^2 + 4S_t^2}$, where S_b and S_t are bending and torsional stresses, respectively. S_E must not exceed the allowable stress range S_A.

In recent years, many principal high-temperature steam lines have either been analyzed, tested in a model-testing machine, or both. No rigid rule is stipulated for the requirement of analysis or model test; however, the Code for Pressure Piping suggests that when the following criterion is not satisfied, need for an analysis is indicated: $DY/(L - U)^2 \leq 0.03$, where D is the nominal pipe size, in; Y is the resultant of movements to be absorbed by pipeline, in; U is the length of straight line joining the anchor points, ft; and L is the length of the developed line axis, ft.

Expansion Joints for Steam Pipelines In many instances it may be economical to care for thermal expansion by use of expansion joints. For low-pressure steam lines, the use of packed expansion joints may be feasible; experience has indicated that packed joints are difficult to maintain when used on high-pressure lines. Figure 8.7.13 shows a type of joint that has been successfully used for high-pressure, high-temperature service. The bellows is designed to take either axial, lateral, or combined axial and lateral deflections. The internal sleeve guides movement of the joint and also protects the flexible bellows from direct contact with the fluid being handled. Face-to-face dimensions, as well as permissible axial and lateral deflections, are indicated in Table 8.7.46.

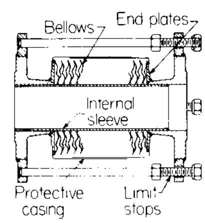


Fig. 8.7.13 Expansion joint for steam line. *(Croll-Reynolds, Inc.)*

Where large lateral deflections are to be absorbed, two expansion joints separated by a length of pipe as shown in Fig. 8.7.14 may be used. With such an arrangement, the lateral deflection permissible with one joint only may be increased many times. Tie rods, as shown, should always be installed to protect the joint against overtravel and externally to guide movement of the joint.

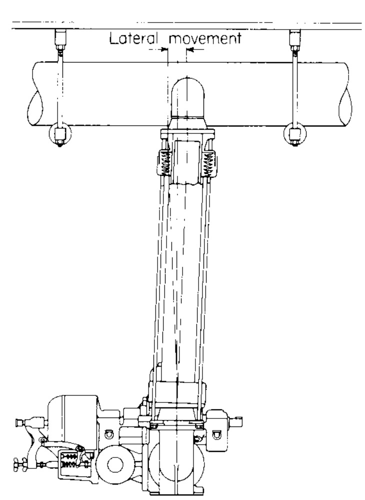


Fig. 8.7.14 Arrangement of expansion joints for large lateral deflection. *(Croll-Reynolds, Inc.)*

Table 8.7.46 Dimensions of Expansion Joints*

Pipe size	Pressure series, lb/in² gage	Face-to-face dimensions,† axial movements of 1 in	2 in	3 in	Lateral movements‡ equivalent to axial movements of 1 in	2 in	3 in
4	150	8½	11	15½	1/16	1/4	9/16
	300	12	17	24	1/8	1/2	1
	600	17½	26½		1/4	1	
	900	31½			11/16		
6	150	9½	12	16½	1/16	3/16	3/8
	300	13	18	25	3/32	13/32	15/16
	600	18½	27½		7/32	27/32	
	900	33½			1/2		
8	150	10½	13	17½	1/32	3/16	3/8
	300	14	19	26	3/32	3/8	27/32
	600	20	29		3/16	3/4	
	900	35½			15/32		
10	150	10½	13	17½	1/32	1/8	5/16
	300	14	19	26	1/16	5/16	23/32
	600	21½	30½		5/32	5/8	
	900	37			3/8		
12	150	22½	14	18½	1/32	1/8	1/4
	300	15	20	27	1/16	1/4	19/32
	600	21½	30½		1/8	1/2	
	900	38½			5/16		
14	100	12½	15½		1/32	1/8	
	150	15	20		1/16	7/32	
	300	20½			3/32		
16	100	12½	15½		1/32	3/32	
	150	15	20		1/32	3/16	
	300	20½			3/32		
18	100	13½	16½		1/32	3/32	
	150	16	21		1/32	3/16	
	300	21½			3/32		
20	100	14½	16½			3/32	
	150	16	21		1/32	3/16	
	300	21½			3/32		
24	100	14½	17½			1/16	
	150	17	22		1/32	5/32	
	300	22½			1/16		
30	100	9½	12½			1/16	
	150	12	17		1/32	1/8	
	300	19½			1/16		

* Croll-Reynolds, Inc.
† For welding ends, add 4 in to face-to-face dimension shown.
‡ Consult manufacturer for permissible combined axial and lateral deflection.

Table 8.7.47 Thermal Expansion Data

Material	Coefficient	Temp range: 70°F (21°C) to: 70 (21)	200 (93)	300 (149)	400 (205)	500 (260)	600 (316)	700 (371)	800 (427)	900 (482)	1,000 (538)	1,100 (593)	1,200 (649)	1,300 (705)	1,400 (760)
Carbon steel: carbon-moly steel low-chrome steels (through 3% Cr)	*A*		6.38	6.60	6.82	7.02	7.23	7.44	7.65	7.84	7.97	8.12	8.19	8.28	8.36
	B	0	0.99	1.82	2.70	3.62	4.60	5.63	6.70	7.81	8.89	10.04	11.10	12.22	13.34
Intermediate alloy steels: 5 Cr Mo-9 Cr Mo	*A*		6.04	6.19	6.34	6.50	6.66	6.80	6.96	7.10	7.22	7.32	7.41	7.49	7.55
	B	0	0.94	1.71	2.50	3.35	4.24	5.14	6.10	7.07	8.06	9.05	10.00	11.06	12.05
Austenitic stainless steels	*A*		9.34	9.47	9.59	9.70	9.82	9.92	10.05	10.16	10.29	10.39	10.48	10.54	10.60
	B	0	1.46	2.61	3.80	5.01	6.24	7.50	8.80	10.12	11.48	12.84	14.20	15.56	16.92
Straight chromium stainless steels: 12 Cr, 17 Cr, and 27 Cr	*A*		5.50	5.66	5.81	5.96	6.13	6.26	6.39	6.52	6.63	6.72	6.78	6.85	6.90
	B	0	0.86	1.56	2.30	3.08	3.90	4.73	5.60	6.49	7.40	8.31	9.20	10.11	11.01
25 Cr-20 Ni	*A*		7.76	7.92	8.08	8.22	8.38	8.52	8.68	8.81	8.02	9.00	9.08	9.12	9.18
	B	0	1.21	2.18	3.20	4.24	5.33	6.44	7.60	8.78	9.95	11.12	12.31	13.46	14.65
Monel 67: Ni-30 Cu	*A*		7.84	8.02	8.20	8.40	8.58	8.78	8.96	9.16	9.34	9.52	9.70	9.88	10.04
	B	0	1.22	2.21	3.25	4.33	5.46	6.64	7.85	9.12	10.42	11.77	13.15	14.58	16.02
Monel 66: Ni-29 CuAl	*A*		7.48	7.68	7.90	8.09	8.30	8.50	8.70	8.90	9.10	9.30	9.50	9.70	9.89
	B	0	1.17	2.12	3.13	4.17	5.28	6.43	7.62	8.86	10.16	11.50	13.00	14.32	15.78
Aluminum	*A*		12.95	13.28	13.60	13.90	14.20								
	B	0	2.00	3.66	5.39	7.17	9.03								
Gray cast iron	*A*		5.75	5.93	6.10	6.28	6.47	6.65	6.83	7.00	7.19				
	B	0	0.90	1.64	2.42	3.24	4.11	5.03	5.98	6.97	8.02				
Bronze	*A*		10.03	10.12	10.23	10.32	10.44	10.52	10.62	10.72	10.80	10.90	11.00		
	B	0	1.56	2.79	4.05	5.33	6.64	7.95	9.30	10.68	12.05	13.47	14.92		
Brass	*A*		9.76	10.00	10.23	10.47	10.69	10.92	11.16	11.40	11.63	11.85	12.09		
	B	0	1.52	2.76	4.05	5.40	6.80	8.26	9.78	11.35	12.98	14.65	16.39		
Wrought iron	*A*		7.32	7.48	7.61	7.73	7.88	8.01	8.13	8.29	8.39				
	B	0	1.14	2.06	3.01	3.99	5.01	6.06	7.12	8.26	9.36				
Copper-nickel (70/30)	*A*		8.54	8.71	8.90										
	B	0	1.33	2.40	3.52										

A = mean coefficient of thermal expansion × 10^6, in/(in · °F) in going from 70°F (21°C) to indicated temperature.
B = linear thermal expansion, in/100 ft in going from 70°F (21°C) to indicated temperature.
Multiply values of A shown by 1.8 to obtain coefficient of expansions in cm/(cm · °C).
Multiply values of B shown by 8.33 to obtain linear expansion in cm per 100 m.
SOURCE: ANSI B31.1-1983.

Table 8.7.47, extracted from the Code for Pressure Piping, lists thermal-expansion data for both ferrous and nonferrous piping. For expansion at temperatures intermediate between those shown, straight-line interpolation is permitted.

The **rubber expansion joint** has become an established part of pipeline equipment. Its special field of application is on low-pressure and vacuum lines in condenser applications, etc., and it is recommended for pressures up to 25 lb/in² (172 kPa) gage where the maximum temperature does not exceed 250°F. Standard joints for pressure installations are reinforced to withstand working pressures up to 125 lb/in² (862 kPa) gage and temperatures up to 200°F. Joints are available in all standard pipe sizes.

Welding in Power-Plant Piping

(For dimensions of welding fittings see Tables 8.7.48 to 8.7.51; for welding techniques see also Sec. 13.3.)

The majority of main-cycle and service steel piping in modern steam power plants is of welded construction. Steel pipe of NPS 2 and smaller is generally socket-welded; larger-size piping is usually butt-welded. Frequently, depending on location and scheduling, piping larger than NPS 2 is prefabricated; smaller piping is shipped to the construction site in random lengths and is fabricated concurrently with installation. Small-sized chromium-molybdenum piping requiring bending is frequently also shop-fabricated so as to avoid high field preheat, welding, and stress-relieving costs. It is desirable to schedule shipment of

Table 8.7.48 Dimensions of Long-Radius 90° Butt-Welding Elbows
(Standard weight—ANSI B16.9-1978, ASTM A234)
(All dimensions in inches)

Nominal pipe size	OD	ID	Wall thickness	Center to face	Pipe schedule numbers	Approx wt, lb
2½	2.875	2.469	0.203	3¾	40	2.92
3	3.500	3.068	0.216	4½	40	4.58
3½	4.000	3.548	0.226	5¼	40	6.43
4	4.500	4.026	0.237	6	40	8.70
5	5.563	5.047	0.258	7½	40	14.7
6	6.625	6.065	0.280	9	40	22.9
8	8.625	7.981	0.322	12	40	46.0
10	10.750	10.020	0.365	15	40	81.5
12	12.750	12.000	0.375	18	ST*	119
14	14.000	13.250	0.375	21	30	154
16	16.000	15.250	0.375	24	30	201
18	18.000	17.250	0.375	27	ST*	256
20	20.000	19.250	0.375	30	20	317
22	22.000	21.250	0.375	33	ST*	385
24	24.000	23.250	0.375	36	20	458
26	26.000	25.250	0.375	39	ST*	539
30	30.000	29.250	0.375	45	ST*	720
34	34.000	33.250	0.375	51	ST*	926
36	36.000	35.250	0.375	54	ST*	1,040
42	42.000	41.250	0.375	63	ST*	1,420

* Standard weight.

Table 8.7.49 Dimensions of Straight Butt-Welding Tees
(Standard weight—ANSI B16.9-1978, ASTM A234)
(Dimensions in inches)

Nominal pipe size	OD	ID	Wall thickness	Center to end	Pipe schedule numbers	Approx wt, lb
2½	2.875	2.469	0.203	3	40	5.21
3	3.500	3.068	0.216	3⅜	40	7.44
3½	4.000	3.548	0.226	3¾	40	9.85
4	4.500	4.026	0.237	4⅛	40	12.6
5	5.563	5.047	0.258	4⅞	40	19.8
6	6.625	6.065	0.280	5⅝	40	29.3
8	8.625	7.981	0.322	7	40	53.7
10	10.750	10.020	0.365	8½	40	91.2
12	12.750	12.000	0.375	10	ST*	132
14	14.000	13.250	0.375	11	30	172
16	16.000	15.250	0.375	12	30	219
18	18.000	17.250	0.375	13½	ST*	282
20	20.000	19.250	0.375	15	20	354
22	22.000	21.250	0.375	16½	ST*	437
24	24.000	23.250	0.375	17	20	493
26	26.000	25.250	0.375	19½	ST*	634
30	30.000	29.250	0.375	22	ST*	855
34	34.000	33.250	0.375	25	ST*	1,136
36	36.000	35.250	0.375	26½	ST*	1,294

* Standard weight.

Table 8.7.50 Dimensions of Long-Radius 45° Butt-Welding Elbows
(Standard weight—ANSI B16.9-1978, ASTM A234)
(Dimensions in inches)

Nominal pipe size	OD	ID	Wall thickness	Center to face	Radius	Pipe schedule numbers	Approx wt, lb
2½	2.875	2.469	0.203	1¾	3¾	40	1.64
3	3.500	3.068	0.216	2	4½	40	2.43
3½	4.000	3.548	0.226	2¼	5¼	40	3.29
4	4.500	4.026	0.237	2½	6	40	4.31
5	5.563	5.047	0.258	3⅛	7½	40	7.30
6	6.625	6.065	0.280	3¾	9	40	11.3
8	8.625	7.981	0.322	5	12	40	22.8
10	10.750	10.020	0.365	6¼	15	40	40.4
12	12.750	12.000	0.375	7½	18	ST*	59.5
14	14.000	13.250	0.375	8¾	21	30	76.5
16	16.000	15.250	0.375	10	24	30	100
18	18.000	17.250	0.375	11¼	27	ST*	128
20	20.000	19.250	0.375	12½	30	20	158
22	22.000	21.250	0.375	13½	33	ST*	192
24	24.000	23.250	0.375	15	36	20	229
26	26.000	25.250	0.375	16	39	ST*	269
30	30.000	29.250	0.375	18½	45	ST*	358
34	34.000	33.250	0.375	21	51	ST*	463
36	36.000	35.250	0.375	22¼	54	ST*	518
42	42.000	41.250	0.375	26	63	ST*	707

* Standard weight.

hangers so that they will be available at the job site upon arrival of the prefabricated piping; this avoids the expense of providing, installing, and later removing temporary hangers and supports. Aside from the economy of welded construction, it is a virtual necessity in high-pressure, high-temperature work because of danger of leakage if joints are flanged.

Shop welds are frequently made by automatic or semiautomatic submerged-arc or inert-gas shielded-arc processes; field welds are gener-

Table 8.7.51 Dimensions of Concentric and Eccentric Butt-Welding Reducers
(Standard weight—ANSI B16.9-1978, ASTM A234)
(Dimensions in inches)

Nominal pipe size	Length	Approx wt, lb	Nominal pipe size	Length	Approx wt, lb	Nominal pipe size	Length	Approx wt, lb	
2½ × 1	3½	1.30	8 × 6	6	13.4	22 × 20	20	157	
2½ × 1¼	3½	1.47	8 × 6	6	13.9	24 × 16	20	160	
2½ × 1½	3½	1.51	10 × 4	7	21.1	24 × 18	20	163	
2½ × 2	3½	1.60	10 × 5	7	21.8	24 × 20	20	167	
3 × 1¼	3½	1.70	10 × 6	7	22.3	26 × 18	24	200	
3 × 1½	3½	1.89	10 × 8	7	23.2	26 × 20	24	200	
3 × 2	3½	2.00	12 × 5	8	30.5	26 × 22	24	200	
3 × 2½	3½	2.16	12 × 6	8	31.1	26 × 24	24	200	
3½ × 1¼	4	2.35	12 × 8	8	32.1	30 × 20	24	220	
3½ × 1½	4	2.52	12 × 10	8	33.4	30 × 24	24	220	
3½ × 2	4	2.71	14 × 6	13	55.8	30 × 26	24	220	
3½ × 2½	4	2.96	14 × 8	13	57.2	30 × 28	24	220	
3½ × 3	4	3.05	14 × 10	13	60.4				
4 × 1½	4	2.73	14 × 12	13	63.4			Conc.	Ecc.
4 × 2	4	3.17	16 × 8	14	70.2	34 × 24	24	270	229
4 × 2½	4	3.34	16 × 10	14	72.9	34 × 26	24	270	237
4 × 3	4	3.50	16 × 12	14	75.6	34 × 30	24	270	253
4 × 3½	4	3.61	16 × 14	14	77.5	34 × 32	24	270	261
5 × 2	5	5.05	18 × 10	15	86.9	36 × 24	24	340	237
5 × 2½	5	5.52	18 × 12	15	89.2	36 × 26	24	340	245
5 × 3	5	5.73	18 × 14	15	90.9	36 × 30	24	340	261
5 × 3½	5	5.86	18 × 16	15	94.0	36 × 32	24	340	269
5 × 4	5	5.99	20 × 12	20	134	36 × 34	24	340	277
6 × 2½	5½	7.61	20 × 14	20	135	42 × 24	24	260	
6 × 3	5½	8.00	20 × 16	20	138	42 × 26	24	270	
6 × 3½	5½	8.14	20 × 18	20	142	42 × 30	24	285	
6 × 4	5½	8.19	22 × 14	20	148	42 × 32	24	295	
6 × 5	5½	8.65	22 × 16	20	151	42 × 34	24	300	
8 × 3½	6	12.8	22 × 18	20	154	42 × 36	24	310	
8 × 4	6	13.1							

ally of the manual type and may be done by the shielded metal-arc and/or inert-gas metal-arc processes. Welding in power piping systems, whether in the shop or at the job site, must be done by welders who have qualified under provisions of the Code for Pressure Piping or the ASME Boiler and Pressure Vessel Code.

End Preparation for Butt Welds Figure 8.7.15 shows the end preparation recommended (not required) for piping whose wall thickness is ¾ in or less, and Fig. 8.7.16 shows that required for piping with wall thickness above ¾ in. During the welding process, to avoid entrance of welding material into the pipe, backing rings may be used as shown in Fig. 8.7.17*a, b,* and *c*.* Note that thick-walled pipes (over ¾ in) are taper-bored on the inside in order that they may receive a tapered, machined backing ring.

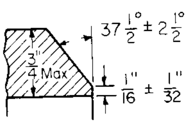


Fig. 8.7.15 Recommended end preparation for pipe wall thickness of ¾ in or less.

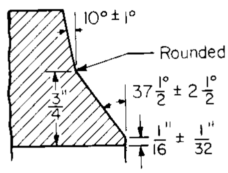


Fig. 8.7.16 Recommended end preparation for pipe wall thickness greater than ¾ in.

Preheating Prior to start of welding, many materials require preheat to a specified temperature: preheat may be done by electrical-resistance or induction heating or by ring-type gas burners placed concentrically with the pipe. The preheat temperature is measured by indicating crayons or by thermocouple pyrometers and must be maintained during the welding operation. Table 1, Appendix D, of the Code for Pressure Piping lists materials used in piping systems and the appropriate temperatures for preheat. In general, the following is indicative of the intent only; for specific instances, the Code must be consulted.

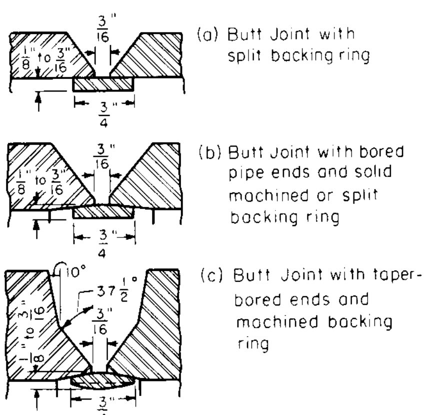


Fig. 8.7.17 Recommended backing ring types. (*a*) Butt joint with split backing ring; (*b*) butt joint with bored pipe ends and solid machined or split backing ring; (*c*) butt joint with taper-bore ends and machined backing ring.

Carbon steel and **wrought iron** should be preheated to a ''hand-hot'' condition if the ambient temperature at time of field installation is 32°F (0°C) or less: carbon steels which have minimum tensile properties of 70,000 lb/in^2 (483 MPa) or higher should be preheated to 250°F (121°C); under other conditions, preheat is not mandatory, but some purchasers insist that the contractor preheat heavy-walled piping such as boiler feed.

* Consumable inserts are also available. They are recommended for installation in piping systems which require a smooth, unobstructed interior surface.

Low-alloy steels with a chromium content not exceeding ¾ percent and low-alloy steels with a total alloy content not exceeding 2 percent are required to be preheated to a minimum temperature of 300°F (149°C).

Alloy steels with a chromium content between ¾ and 2 percent and low-alloy steels with a total alloy content not exceeding 2¾ percent require preheating to 375°F (191°C) minimum. Those with a total alloy content greater than 2¾ percent but not exceeding 10 percent require preheating to a temperature of 450°F (232°C) minimum.

High-alloy steels containing the martensitic phase require preheating to 450°F (232°C) minimum; preheating is a matter of agreement between the purchaser and contractor in the case of welding high-alloy ferritic steels (ASTM A240 and A268). The possible advantages of preheat have not been established in the case of welding high-alloy austenitic steels, and for this reason the Code for Pressure Piping states that preheat is optional for these materials.

Welding procedure varies with material and welding process. In general, the pipe ends must be cleaned of oil or grease, and excessive amounts of scale or rust should be removed. The size and type of welding rod must be stated; the number of layers or passes is determined by the thickness of the pieces being joined. All slag or flux remaining on any bead of welding must be removed before laying down the next successive bead; any cracks or blowholes that appear on the surface of any bead must be chipped or ground away before the next bead of weld material is deposited. Throughout the welding process, it is essential that the minimum specified preheat temperature be maintained.

Stress Relieving Welded joints in all carbon-steel material whose thickness is ¾ in (1.91 cm) or greater must be stress-relieved at a temperature of 1,100°F (593°C) or over for a period of time proportioned on the basis of at least 1 h/in of pipe-wall thickness (but in no case less than ½ h) and then allowed to cool slowly (generally under a blanket) and uniformly. No stress relief is required for joints in carbon-steel piping whose wall thickness is less than ¾ in.

Welded joints in alloy steels with a wall thickness of ½ in (1.27 cm) or greater, having a chromium content not exceeding ¾ percent, and low-alloy steels with a total alloy content not exceeding 2 percent require stress-relieving at a temperature of 1,200°F (649°C) or over for a period of time proportioned on the basis of at least 1 h/in (0.4 h/cm) of wall thickness, but in no case less than ½ h.

Welded joints in alloy steels having a chromium content exceeding ¾ percent, or a total alloy content exceeding 2 percent, except high-alloy ferritic (ASTM A240, A268) and austenitic steels, regardless of wall thickness, require stress relief at a temperature of 1,200°F or over for a period of time proportioned on the basis of at least 1 h/in of wall thickness, but in no case less than ½ h. Stress relief of high-alloy ferritic steel (A240, A268) and austenitic steels is not required but may be performed as agreed upon by purchaser and contractor. In welds between austenitic and ferritic materials, stress relieving is optional and, if used, shall be a matter of agreement between the purchaser and contractor. Because of the difference between the coefficients of thermal expansion of the two dissimilar materials, careful consideration should be given to the selection of a heat treatment, if any, that will be beneficial to the welded joint.

Graphitization is precipitation of carbon at the grain boundaries in the heat-affected zone during the welding process. Such a phenomenon occurs when some metals operate at high temperatures for extended periods. It has been observed particularly in carbon-molybdenum steels that operate at 900°F (482°C) or higher.

Graphitization does not generally occur in carbon-molybdenum steels with over 1 percent molybdenum. It also has generally not occurred in the chromium-molybdenum low-alloy steels operated at temperatures between 900°F (482°C) and 1,050°F (566°C). Where graphitization has occurred, the two most commonly used methods for rehabilitation of the pipe are (1) gouging out the heat-affected zone of the weld deposit and rewelding the area with electrodes depositing carbon-molybdenum weld metal, followed by a stabilization heat treatment at 1,300°F (704°C) for 4 h, or (2) solution annealing the weld joints at 1,800°F (982°C), followed by a stabilization heat treatment.

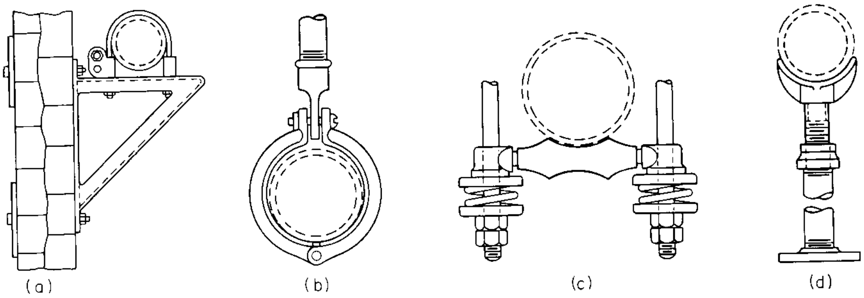


Fig. 8.7.18 Methods of supporting pipes.

Pipe Supports

The Code for Pressure Piping includes many types of supports and gives directions for their application. A proper pipe support must have a strong rigid base properly supported, and an adjustable roll construction which will maintain the alignment in any direction. It is important to avoid friction caused by the movement of the pipe in the support and to have all parts of sufficient strength to maintain alignment at all times. Wire hangers, band iron hangers, wooden hangers, hangers made from small pipe, and hangers having one vertical pipe support do not maintain alignment.

The direction of expansion in a pipe run can be predetermined by anchoring one end, both ends, or the middle. **Anchors** must be firmly fastened to a rigid and heavy part of the power-plant structure, and must also be securely fastened to the pipe; otherwise the equipment for absorbing expansion is useless, and severe stresses may be thrown on parts of the piping system. Some methods of support are shown in Figs. 8.7.18 and 8.7.19. Welded steel brackets (Fig. 8.7.18*a*) are available in light, medium, and heavy weights. Many types of supports can be mounted on these brackets, such as the anchor chair shown on the bracket at (*a*), pipe roller supports of the type at (*c*), pipe roll stands of various types such as shown in Fig. 8.7.19, pipe seats, etc. Figure 8.7.18*b* illustrates one of the many types of adjustable ring hangers in use. The split ring hanger can be applied after the pipeline is in place. At (*c*) in Fig. 8.7.18 is shown a spring cushion pipe roll hanger recommended for service where constant support* is required and compensation must be made for movement of the piping. The springs provide an efficient means of absorbing the vibration. Figure 8.7.18*d* shows one of the many types of pipe saddle supports available. Figure 8.7.19 shows a cast-iron pipe roll stand designed for cases where vertical adjustment is not necessary but where provision must be made for expansion and contraction of the pipeline. Several designs of such stands with provision for vertical adjustment and of the same general dimensions are also available. One type of cast-iron roll and plate, illustrated in Fig. 8.7.19, provides for expansion and contraction where vertical adjustment is not necessary. If necessary, the baseplate can be raised or lowered by use of shims. Detailed information and dimensions of a great variety of pipe supports can be found in manufacturers' catalogs.

In supporting a high-temperature piping system, it is necessary to provide for expansion and contraction due to cyclic changes. It is often possible to find a point of zero movement along the run of a long line and to support a considerable portion of the total load by a rigid hanger or support of the type shown in Figs. 8.7.18 and 8.7.19. However, for other portions of the run, some form of spring support is often indicated. For relatively light lines, which are not subjected to excessive movements from hot to cold positions, a variable spring hanger will frequently suffice; for heavy lines, or those in which expansion movements are great, it is advisable to use constant support of counterweighted hangers so that transfer of weight to other hangers or equipment connections is prevented. Parts (*a*) and (*b*) of Fig. 8.7.20 indicate, respectively, a horizontal and vertical run of piping supported by a **constant-support hanger**. Figure 8.7.20*c* and 8.7.21*a* indicate horizontal runs supported by **variable-spring hangers**. Figure 8.7.21*b* shows a riser supported by a variable spring beneath a base elbow. Figure 8.7.21*c* indicates a **sway brace** that is used to control vibration and undesirable movement in a piping system.

The principal supports utilized for the support of critical piping in-

* The support afforded by the hanger of Fig. 8.7.18*c* is constant only in the sense that some degree of support is always present. It might be more appropriately termed a variable-support device.

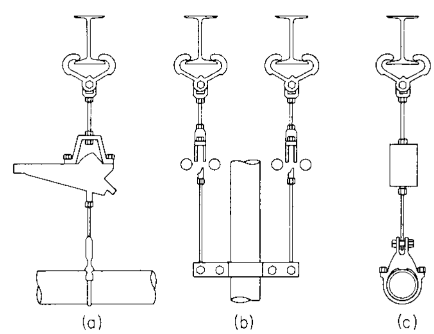


Fig. 8.7.20 Constant support and variable-spring hangers.

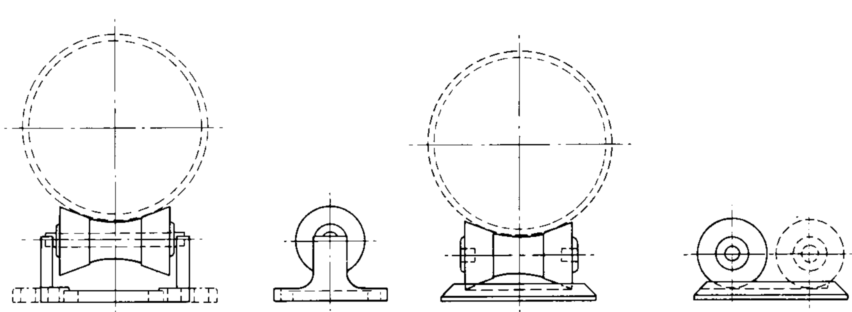


Fig. 8.7.19 Pipe supports on cast-iron rolls.

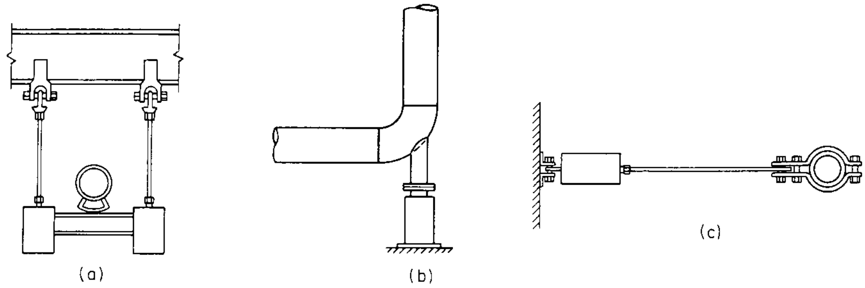


Fig. 8.7.21 Spring hangers and sway brace.

volve constant-support hangers, variable-spring hangers, rigid hangers, and restraints.

Constant-Support Hangers This type of hanger provides a constant supporting force for the piping system throughout its full range of vertical pipe movement. This is accomplished through use of a spring coil working in conjunction with a lever in such a way that the spring force times its distance to the lever pivot is always equal to the pipe load times its distance to the lever pivot. This type of support is ''thermally invisible,'' as the supporting force equals the pipe weight throughout its entire expansion or contraction cycles.

These hangers are used on systems or at locations where stresses are considered critical. Pipe weight reactions or transfer of loads are not imposed in the system or connections with this type of device.

As the load is considered constant with the unit in travel, readings from inspections are based on travel. The readings are taken from the position of an indicator and its relation to the numbers on the travel scale. The scale is divided into 10 divisions. H or high on the scale is equal to (0.0), M or midway on the scale is equal to (5.0), and L or low on the scale is equal to 10.

The design settings were obtained from the position of the factory-installed buttons that are placed adjacent to the travel scale.

''Perfect'' readings would be if the indicator were to line up with the white button (cold) and the red (hot); however, this is rarely the case. Generally, readings are considered acceptable and not noteworthy as long as they reflect movement consistent with design in both direction and length. A general rule is that when the hot setting is higher than the cold setting, then movement is down from cold to hot. If the cold setting is higher, movement is up.

The following terms are normally applied to these devices:

Actual travel: Anticipated movement of the pipe from design. The hot and cold position stickers are a function of this movement.

Total travel: The maximum movement a support can accept without danger of topping or bottoming out. The scale from H (0.0) to L (10.0) is a function of this.

Topped out: The indicator is above the high point and in contact with the end of the slot. This condition means the support is unloaded or is in the process of unloading.

Bottomed out: The indicator is below the (10.0) point and in contact with the end of the travel slot; the support is overloaded.

Variable-Spring Hangers These devices are installed at locations where stresses are not considered to be critical or where movement and economics permit their use.

The inherent characteristics of a variable are such that the supporting force varies with the spring's deflection. Movement of the pipe causes the spring to extend or compress. This results in a change or variance in the supporting force. Since the weight of the pipe is the same in either condition, hot or cold, the variation results in pipe weight transfer to equipment and adjacent hangers and consequently additional stresses in the piping system. The effects of this variation are usually considered during the original design.

In addition, as it is desirable to support the actual weight of the pipe when the system is hot, when the stresses tend to become most critical, the hot load is the dead weight of the pipe. The cold load is actually under- or oversupporting the pipe, depending on the movement from cold to hot.

The general rule for determining movement is similar to that of constant supports. If the hot load is higher than the cold, then pipe movement is down from cold to hot. If the cold load is higher, then movement is up.

Unlike constant supports, the readings from variables are measured in pounds. The readings are taken by noting the position of the indicator relative to a load scale that is adjacent to the travel slot.

The **distance between supports** will vary with the kind of piping and the number of valves and fittings. Supports should be provided near changes in direction, branch lines, and particularly near valves. The weight of piping must not be carried through valve bodies. In establishing the location of pipe supports, the designer should be guided by two requirements: (1) the horizontal span must not be so long that sag in the pipe will impose an excessive stress in the pipe wall and (2) the pipeline must be pitched downward so that the outlet of each span is lower than maximum sag in the span. Otherwise entrapped water can result in severe water hammer and pipe swings, particularly during plant start-up of steam piping.

Fabrication and installation practices are provided in MSS Standard Practice SP-89. Table 8.7.52 lists spacing for standard-weight pipe supports.

Pipe Insulation

(see Secs. 4 and 6 for heat-transmission data.)

The value of a steam-pipe covering is measured by its ability to reduce heat losses. This might range from 50 percent for small, low-temperature lines to 90 percent for large, high-temperature lines. Many **pipe-insulating materials** are available: 85 percent magnesia, foam glass, calcium silicate, and various forms of diatomaceous earths. Some of these materials are suited for relatively low temperatures only, others are best suited for high temperatures, and still others are suitable over a considerable temperature range.

Pipe insulation is applied in molded sections 3 ft long. For high-temperature work, the insulation is applied in at least two layers with the

Table 8.7.52 Maximum Spacing of Pipe Supports at 750°F (399°C)*

Nominal pipe size, in	1	1½	2	3	4	6	8	10	12	14	16	18	20	24
Maximum span, ft	7	9	10	12	14	17	19	22	23	25	27	28	30	32
Maximum span, m	2.13	2.74	3.05	3.66	4.27	5.18	5.79	6.71	7.01	7.62	8.23	8.53	9.14	9.75

* This tabulation assumes that concentrated loads, such as valves and flanges, are separately supported. Spacing is based on a combined bending and shear stress of 1,500 lb/in^2 when pipe is filled with water; under this condition, sag in pipeline between supports will be approximately 0.1 in.

joints staggered so as to prevent a direct channel for heat loss. Because of its maximum-temperature limitation of about 600°F (316°C), 85 percent magnesia is used as the second layer with a high-temperature-resistant material placed in direct contact with the pipe. The molded insulation is fastened securely in place with copper or galvanized wire and is then given a surface finish; indoor pipes are first sheathed with resin paper and covered with canvas, either pasted or sewed; outdoor pipes may be weather-protected by a coating of asphaltic-type waterproofing compound, they may be sheathed and canvased and then given a weatherproof surface, so they may be encased in metallic (steel or aluminum) jackets.

The heat loss from an insulated pipe appears in three phases: heat passes by *conduction* through the metallic pipe walls and through the insulating material; it then is dissipated from the outdoor surface of the insulation by *convection* and by *radiation.* Extremely accurate calculations must also take into account the temperature drop by convection through the film on the inside surface of the pipe. The task of accurately calculating heat losses is somewhat tedious, since the convection and radiation losses are related to the surface temperature (outside of insulation), which is unknown until conduction losses are balanced against surface losses.

For combined convection and radiation coefficients for bare pipes, and all necessary formulas to permit trial-and-error calculations, see Sec. 4. Insulation manufacturers publish data which give heat losses for wide ranges of pipe size and temperature.

Identification of Piping

The American National Standards Institute has approved a **Scheme for the Identification of Piping Systems** (ANSI A13.1-1981). This scheme is limited to the identification of piping systems in industrial plants, not including pipes buried in the ground, and electric conduits. Fittings, valves, and pipe coverings are included, but not supports, brackets, or other accessories.

Classification by Color All piping systems are classified by the nature of the material carried. Each piping system is placed, by the nature of its contents, in the following classifications:

Class	*Color*
F—Fire-protection equipment	Red
D—Dangerous materials	Yellow (or orange)
S—Safe materials	Green (or the achromatic colors, white, black, gray, or aluminum)
P—Protective materials	Bright blue
V—Extra valuable materials	Deep purple

Method of Identification At conspicuous places throughout a piping system, color bands should be painted on the pipes to designate to which one of the five main classes it belongs. If desired, the entire length of the piping system may be painted the main classification color.

Further, the actual contents of a piping system may be indicated by, preferably, a stenciled legend of standard size giving the name of the contents in full or abbreviated form. These legends should be placed on the color bands. The identification scheme may be extended by the use of colored stripes placed at the edges of the colored bands.

The bands, legends, and stripes should be placed at intervals throughout the piping system, preferably adjacent to valves and fittings to ensure ready recognition during operation, repairs, and at times of emergency.

A recommended classification, under this color scheme of materials carried in pipes, includes, as dangerous, combustible gases and oils, hot water and steam above atmospheric pressure; as safe, compressed air, cold water, and steam under vacuum.

Pressure Hose

Hose with durable rubber lining may be obtained to withstand any needed pressure. If the rubber compound is properly made, the life of a hose will be 7 to 10 years, while a cheaper hose, lined with inferior material, will probably not last more than 3 or 4 years. See also Secs. 3 and 12.

American National Fire-Hose Coupling Screw Thread (ANSI B1.20-7-1966) This standard is intended to cover the threaded part of fire-hose couplings, hydrant outlets, standpipe connections, and at other special fittings on fire lines, where fittings of the nominal diameters given in Table 8.7.53 are used. It also includes the limiting dimensions of the field inspection gages. The American National Standard form of thread must be used.

Table 8.7.53 Dimensions of Standard Fire-Hose Couplings
(All dimensions in inches. Letters refer to Fig. 8.7.22)

Inside diam, *C*	Diam of thread, *D*	No. of threads per inch	*L*	*I*	*H*	*J*	*T*
2½	3¹⁄₁₆	7½	1	¼	¹⁵⁄₁₆	³⁄₁₆	¹¹⁄₁₆
3	3⅝	6	1⅛	⁵⁄₁₆	1¹⁄₁₆	¼	¹³⁄₁₆
3½	4¼	6	1⅛	⁵⁄₁₆	1¹⁄₁₆	¼	¹³⁄₁₆
4½	5¾	4	1¼	⁷⁄₁₆	1³⁄₁₆	⅜	¹⁵⁄₁₆

SOURCE: ANSI B1.20.7-1966.

American National Standard Hose-Coupling Screw Threads (ANSI B1.20.7-1966) These standards apply to the threaded parts of hose couplings, valves, nozzles, and all other fittings used in direct connection with hose intended for fire protection or for domestic, industrial, or general service in nominal sizes given in Table 8.7.54. The American

Table 8.7.54 Dimensions of Standard Hose Couplings
(All dimensions in inches. Letters refer to Fig. 8.7.22)

Service and nominal size	Inside diam, *C*	Diam of thread, *D*	No. of threads per inch	*L*	*I*	*H*	*T*
Garden:							
½, ⅝, ¾	²⁵⁄₃₂	1¹⁄₁₆	11½	⁹⁄₁₆	⅛	¹⁷⁄₃₂	⅜
Chemical:							
¾, 1	1¹⁄₃₂	1⅜	8	⅝	⁵⁄₃₂	¹⁹⁄₃₂	¹⁵⁄₃₂
Fire:							
1½	1¹⁷⁄₃₂	2	9	⅝	⁵⁄₃₂	¹⁹⁄₃₂	¹⁵⁄₃₂
Other connections:							
½	¹⁷⁄₃₂	¹³⁄₁₆	14	½	⅛	¹⁵⁄₃₂	⁵⁄₃₂
¾	²⁵⁄₃₂	1¹⁄₃₂	14	⁹⁄₁₆	⅛	¹⁷⁄₃₂	⅜
1	1¹⁄₃₂	1⁹⁄₃₂	11½	⁹⁄₁₆	⁵⁄₃₂	¹⁷⁄₃₂	⅜
1¼	1⁹⁄₃₂	1⅝	11½	⅝	⁵⁄₃₂	¹⁹⁄₃₂	¹⁵⁄₃₂
1½	1¹⁷⁄₃₂	1⅞	11½	⅝	⁵⁄₃₂	¹⁹⁄₃₂	¹⁵⁄₃₂
2	2¹⁄₃₂	2¹¹⁄₃₂	11½	¾	³⁄₁₆	²³⁄₃₂	¹⁹⁄₃₂

SOURCE: ANSI B1.20.7-1966.

National Standard thread form is used. This coupling is similar in design to the fire-hose couplings illustrated in Fig. 8.7.22.

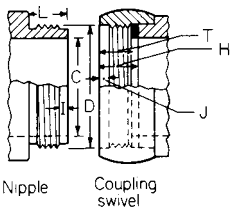


Fig. 8.7.22 Typical form of standard coupling.

Flexible metal hose and tubing are available for a wide range of conditions of temperature, pressure, vibration, and corrosion, and are made in two basic constructions, corrugated or interlocked, and in either bronze or steel.

The corrugated type (Fig. 8.7.23) may have either annular or helical corrugated formations, usually covered with metal braid, and is adapted to high-pressure high-temperature leak-proof service. Some typical applications include diesel-engine exhaust hose, reciprocating flexible connections, loading and unloading hose, saturated and superheated steam lines, lubricating lines, gas and oil lines, vibration connections, etc.

The interlocked type is made in several ways; the fully interlocked type is illustrated in Fig. 8.7.24. Typical applications include wiring conduit, cable armor, decorative wiring covering, dust-collective tubing, grease and oil connections, flexible spouts, and moderate-pressure oil lines.

Standard couplings and fittings can be attached to flexible metal hose or tubing by various methods such as brazing or welding. Each type of hose construction has limits of service use and proved application usages. Information and recommendations as to the type and size to use under any given conditions should be obtained from the manufacturers.

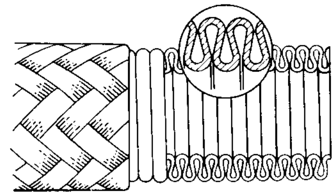


Fig. 8.7.23 Flexible metal hose.

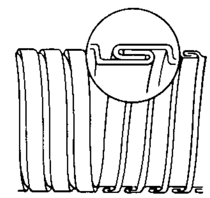


Fig. 8.7.24 Interlocked flexible metal hose.

8.8 PREFERRED NUMBERS

by C. H. Berry

REFERENCES: Hirshfeld and Berry, Size Standardization by Preferred Numbers, *Mech. Eng.,* Dec. 1922. Schlink, A New Tool for Standardizers, *Am. Mach.,* July 12, 1923. ANSI Standard Z17.1.

Many manufactured articles are made in several sizes which may be designated by some dimension, speed, capacity, or other feature. Each such series of products may be paralleled by a series of numbers.

It is generally agreed that such number series should be **geometric progressions;** i.e., each term should be a fixed percentage larger than the preceding. A geometric series provides small steps for small numbers, large steps for large numbers, and this best meets most requirements. The small steps in the diameter of the numbered twist drills would be absurd in drills of 1 in diameter and larger.

In the case of sized objects that are used principally as raw material, e.g., steel rod, an arithmetic progression may be preferable because it tends to reduce the cost of machining. It is desirable to be able to buy raw material a fixed amount (rather than a fixed percentage) larger than the finished article.

Preferred numbers is the name given to various series proposed for general use. These are either geometric progressions or approximations thereto. A geometric series is defined by one term and the ratio of each term to the preceding one. On the choice of these elements for a preferred number series, there is as yet no general agreement. The same value would hardly be satisfactory for all cases. The idea of preferred numbers is to provide a master series from which terms can be chosen to suit any needs. This would ultimately lead to a comprehensive plan in all fields of manufacture, so that, for example, the sizes of shafting would be in accord with the sizes of bearings, and indeed with all manner of cylindrical machine elements.

An advantage of a geometric series is that if linear dimensions are chosen in the series, areas, volumes, and other functions of powers of dimensions are also members of the same series.

In one of the most carefully considered systems of preferred numbers the base term is 1, and the ratio is $\sqrt[80]{10}$. In this series, the 81st term is 10, and accordingly the series from 10 to 100 or from 0.01 to 0.1, or, in general, from 10^n to 10^{n+1} is identical with the series from 1 to 10 with the decimal point shifted. This series will rarely be used in full; some will choose alternate terms, some every fourth, fifth, tenth, or twentieth

Table 8.8.1 Basic Series of Preferred Numbers: R 80 Series

1.00	1.80	3.15	5.60
1.03	1.85	3.25	5.80
1.06	1.90	3.35	6.00
1.09	1.95	3.45	6.15
1.12	2.00	3.55	6.30
1.15	2.06	3.65	6.50
1.18	2.12	3.75	6.70
1.22	2.18	3.87	6.90
1.25	2.24	4.00	7.10
1.28	2.30	4.12	7.30
1.32	2.36	4.25	7.50
1.36	2.43	4.37	7.75
1.40	2.50	4.50	8.00
1.45	2.58	4.62	8.25
1.50	2.65	4.75	8.50
1.55	2.72	4.87	8.75
1.60	2.80	5.00	9.00
1.65	2.90	5.15	9.25
1.70	3.00	5.30	9.50
1.75	3.07	5.45	9.75

SOURCE: American National Standard Preferred Numbers Z17.1, reproduced with permission of ANSI.

term. The index of the root, 80, has as factors, 2^4 and 5, so that the series readily yields subseries having as ratios the roots of 10 with indices 2, 4, 8, 16, 5, 10, 20, 40, thus giving a wide range of choice. See Table 8.8.1.

The strict logic of this series has been somewhat impaired by the adoption of rounded values that are slightly different in the 1-to-10 and 10-to-100 intervals. For the United States, ANSI has adopted a Table of Preferred Numbers (ANSI Z17.1) which differs slightly from the system described in the preceding paragraph.

Another type of series is the **semigeometric series** consisting of a basic geometric series with 1 as the base term and a ratio of 2, giving a series . . . ⅛, ¼, ½, 1, 2, 4, ... Between consecutive terms are inserted arithmetic series of 2, 4, 8, or 16 terms, in general using different numbers of terms in different intervals.

A similar procedure is used to establish the numbers of teeth in a prescribed number of gears that are intended for use in a gear train to provide a stepped gradation of rotational speeds within an upper and lower bound.