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PLAIN BEARINGS

On the following pages are given data and procedures for designing full-film or hydrodynamically lubricated bearings of the journal and thrust types. However, before proceeding to these design methods, it is thought useful to first review those bearing aspects concerning the types of bearings available; lubricants and lubrication methods; hardness and surface finish; machining methods; seals; retainers; and typical length-to-diameter ratios for various applications.

The following paragraphs preceding the design sections provide guidance in these matters and suggest modifications in allowable loads when other than full-film operating conditions exist in a bearing.

Classes of Plain Bearings.—Bearings that provide sliding contact between mating surfaces fall into three general classes: *radial bearings* that support rotating shafts or journals; *thrust bearings* that support axial loads on rotating members; and *guide* or *slipper bearings* that guide moving parts in a straight line. Radial sliding bearings, more commonly called sleeve bearings, may be of several types, the most usual being the plain full journal bearing, which has 360-degree contact with its mating journal, and the partial journal bearing, which has less than 180-degree contact. This latter type is used when the load direction is constant and has the advantages of simplicity, ease of lubrication, and reduced frictional loss.

The relative motions between the parts of plain bearings may take place: 1) As pure sliding without the benefit of a liquid or gaseous lubricating medium between the moving surfaces such as with the dry operation of nylon or Teflon; 2) with hydrodynamic lubrication in which a wedge or film buildup of lubricating medium is produced, with either whole or partial separation of the bearing surfaces; 3) with hydrostatic lubrication in which a lubricating medium is introduced under pressure between the mating surfaces causing a force opposite to the applied load and a lifting or separation of these surfaces; and 4) with a hybrid form or combination of hydrodynamic and hydrostatic lubrication.

Listed below are some of the advantages and disadvantages of sliding contact (plain) bearings as compared with rolling contact (antifriction) bearings.

Advantages: 1) Require less space; 2) are quieter in operation; 3) are lower in cost, particularly in high-volume production; 4) have greater rigidity; and 5) their life is generally not limited by fatigue.

Disadvantages: 1) Have higher frictional properties resulting in higher power consumption; 2) are more susceptible to damage from foreign material in lubrication system; 3) have more stringent lubrication requirements; and 4) are more susceptible to damage from interrupted lubrication supply.

Types of Journal Bearings.—Many types of journal bearing configurations have been developed; some of these are shown in Fig. 1.

Circumferential-groove bearings, Fig. 1(a), have an oil groove extending circumferentially around the bearing. The oil is maintained under pressure in the groove. The groove divides the bearing into two shorter bearings that tend to run at a slightly greater eccentricity. However, the advantage in terms of stability is slight, and this design is most commonly used in reciprocating-load main and connecting-rod bearings because of the uniformity of oil distribution.

Short cylindrical bearings are a better solution than the circumferential-groove bearing for high-speed, low-load service. Often the bearing can be shortened enough to increase the unit loading to a substantial value, causing the shaft to ride at a position of substantial eccentricity in the bearing. Experience has shown that instability rarely results when the shaft eccentricity is greater than 0.6. Very short bearings are not often used for this type of application, because they do not provide a high temporary rotating-load capacity in the event some unbalance should be created in the rotor during service.

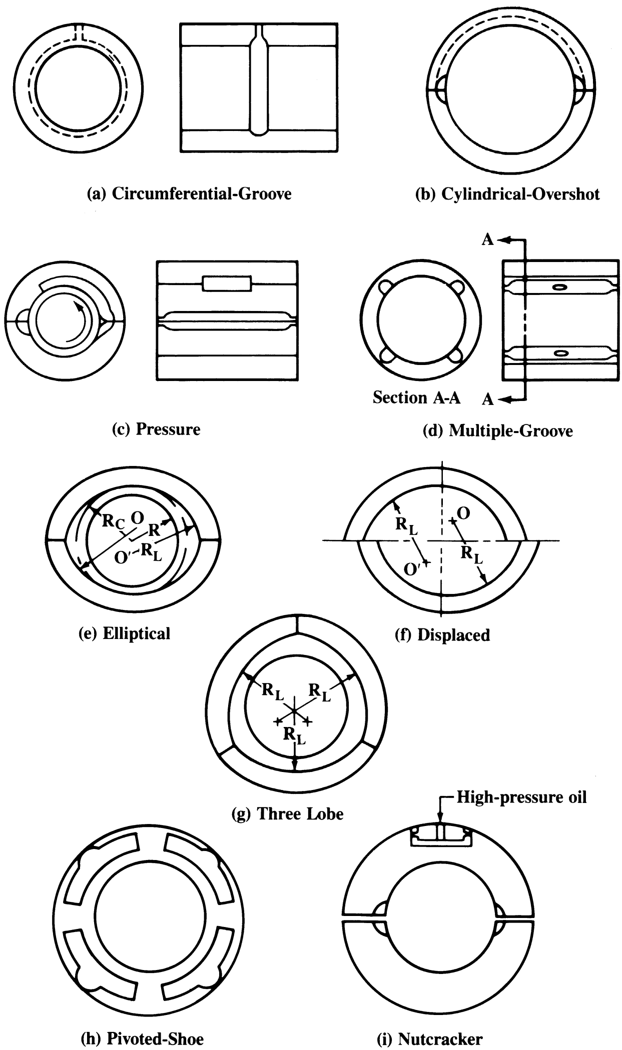

Fig. 1. Typical shapes of several types of pressure-fed bearings.

Cylindrical-overshot bearings,: Fig. 1(b), are used where surface speeds of 10,000 fpm or more exist, and where additional oil flow is desired to maintain a reasonable bearing temperature. This bearing has a wide circumferential groove extending from one axial oil groove to the other over the upper half of the bearing. Oil is usually admitted to the trailing-edge oil groove. An inlet orifice is used to control the oil flow. Cooler operation results from the elimination of shearing action over a large section of the upper half of the bearing and, to a great extent, from the additional flow of cool oil over the top half of the bearing.

Pressure bearings,: Fig. 1(c), employ a groove over the top half of the bearing. The groove terminates at a sharp dam about 45 degrees beyond the vertical in the direction of shaft rotation. Oil is pumped into this groove by shear action from the rotation of the shaft and is then stopped by the dam. In high-speed operation, this situation creates a high oil pressure over the upper half of the bearing. The pressure created in the oil groove and surrounding upper half of the bearing increases the load on the lower half of the bearing. This self-generated load increases the shaft eccentricity. If the eccentricity is increased to 0.6 or greater, stable operation under high-speed, low-load conditions can result. The central oil groove can be extended around the lower half of the bearing, further increasing the effective loading. This design has one primary disadvantage: Dirt in the oil will tend to abrade the sharp edge of the dam and impair ability to create high pressures.

Multiple-groove bearings,: Fig. 1(d), are sometimes used to provide increased oil flow. The interruptions in the oil film also appear to give this bearing some merit as a stable design.

Elliptical bearings,: Fig. 1(e), are not truly elliptical, but are formed from two sections of a cylinder. This two-piece bearing has a large clearance in the direction of the split and a smaller clearance in the load direction at right angles to the split. At light loads, the shaft runs eccentric to both halves of the bearing, and hence, the elliptical bearing has a higher oil flow than the corresponding cylindrical bearing. Thus, the elliptical bearing will run cooler and will be more stable than a cylindrical bearing.

Elliptical-overshot bearings: (not shown) are elliptical bearings in which the upper half is relieved by a wide oil groove connecting the axial oil grooves. They are analogous to cylindrical-overshot bearings.

Displaced elliptical bearings,: Fig. 1(f), shift the centers of the two bearing arcs in both a horizontal and a vertical direction. This design has greater stiffness than a cylindrical bearing, in both horizontal and vertical directions, with substantially higher oil flow. It has not been extensively used, but offers the prospect of high stability and cool operation.

Three-lobe bearings,: Fig. 1(g), are made up in cross section of three circular arcs. They are most effective as antioil whip bearings when the centers of curvature of each of the three lobes lie well outside the clearance circle that the shaft center can describe within the bearing. Three axial oil-feed grooves are used. It is a more difficult design to manufacture, because it is almost necessary to make it in three parts instead of two. The bore is machined with shims between each of the three parts. The shims are removed after machining is completed.

Pivoted-shoe bearings,: Fig. 1(h), are one of the most stable bearings. The bearing surface is divided into three or more segments, each of which is pivoted at the center. In operation, each shoe tilts to form a wedge-shaped oil film, thus creating a force tending to push the shaft toward the center of the bearing. For single-direction rotation, the shoes are sometimes pivoted near one end and forced toward the shaft by springs.

Nutcracker bearings,: Fig. 1(i), consist of two cylindrical half-bearings. The upper half-bearing is free to move in a vertical direction and is forced toward the shaft by a hydraulic cylinder. External oil pressure may be used to create load on the upper half of the bearing through the hydraulic cylinder. Or the high-pressure oil may be obtained from the lower half of the bearing by tapping a hole into the high-pressure oil film, thus creating a self-loading bearing. Either type can increase bearing eccentricity to the point where stable operation can be achieved.

Hydrostatic Bearings.—Hydrostatic bearings are used when operating conditions require full film lubrication that cannot be developed hydrodynamically. The hydrostatically lubricated bearing, either thrust or radial, is supplied with lubricant under pressure from an external source. Some advantages of the hydrostatic bearing over bearings of other types are: low friction; high load capacity; high reliability; high stiffness; and long life.

Hydrostatic bearings are used successfully in many applications including machine tools, rolling mills, and other heavily loaded slow-moving machinery. However, specialized techniques, including a thorough understanding of hydraulic components external to the bearing package is required. The designer is cautioned against use of this type of bearing without a full knowledge of all aspects of the problem. Determination of the operating performance of hydrostatic bearings is a specialized area of the lubrication field and is described in specialized reference books.

Design.—The design of a sliding bearing is generally accomplished in one of two ways:

1) a bearing operating under similar conditions is used as a model or basis from which the new bearing is designed; and 2) in the absence of any previous experience with similar bearings in similar environments, certain assumptions concerning operating conditions and requirements are made and a tentative design prepared based on general design parameters or rules of thumb. Detailed lubrication analysis is then performed to establish design and operating details and requirements.

Modes of Bearing Operation.—The load-carrying ability of a sliding bearing depends upon the kind of fluid film that is formed between its moving surfaces. The formation of this film is dependent, in part, on the design of the bearing and, in part, on the speed of rotation. The bearing has three modes or regions of operation designated as *full-film, mixed-film,* and *boundary* lubrication with effects on bearing friction, as shown in Fig. 2.

In terms of physical bearing operation these three modes may be further described as follows:

1) Full-film, or hydrodynamic, lubrication produces a complete physical separation of the sliding surfaces resulting in low friction and long wear-free service life.

To promote full-film lubrication in hydrodynamic operation, the following parameters should be satisfied: 1) Lubricant selected has the correct viscosity for the proposed operation; 2) proper lubricant flow rates are maintained; 3) proper design methods and considerations have been utilized; and 4) surface velocity in excess of 25 feet per minute is maintained.

When full-film lubrication is achieved, a coefficient of friction between 0.001 and 0.005 can be expected.

2) Mixed-film lubrication is a mode of operation between the full-film and boundary modes. With this mode, there is a partial separation of the sliding surfaces by the lubricant film; however, as in boundary lubrication, limitations on surface speed and wear will result. With this type of lubrication, a surface velocity in excess of 10 feet per minute is required with resulting coefficients of friction of 0.02 to 0.08.

3) Boundary lubrication takes place when the sliding surfaces are rubbing together with only an extremely thin film of lubricant present. This type of operation is acceptable only in applications with oscillating or slow rotary motion. In complete boundary lubrication, the oscillatory or rotary motion is usually less than 10 feet per minute with resulting coefficients of friction of 0.08 to 0.14. These bearings are usually grease lubricated or periodically oil lubricated.

In starting up and accelerating to its operating point, a journal bearing passes through all three modes of operation. At rest, the journal and bearing are in contact, and thus when starting, the operation is in the boundary lubrication region. As the shaft begins to rotate more rapidly and the hydrodynamic film starts to build up, bearing operation enters the region of mixed-film lubrication. When design speeds and loads are reached, the hydrodynamic action in a properly designed bearing will promote full-film lubrication.

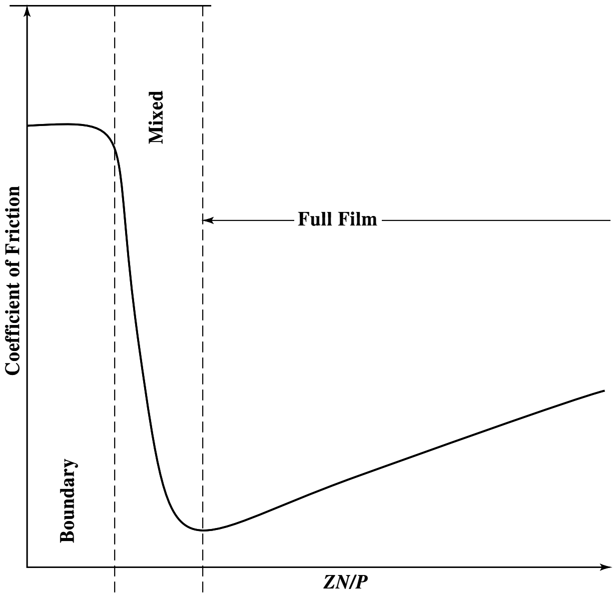

Fig. 2. Three modes of bearing operation.

Methods of Retaining Bearings.— Several methods are available to ensure that a bearing remains in place within a housing. Which method to use depends upon the particular application but requires first that the unit lends itself to convenient assembly and disassembly; additionally, the bearing wall should be of uniform thickness to avoid introduction of weak points in the construction that may lead to elastic or thermal distortion.

Press or Shrink Fit: One common and satisfactory technique for retaining the bearing is to press or shrink the bearing in the housing with an interference fit. This method permits the use of bearings having uniform wall thickness over the entire bearing length.

Standard bushings with finished inside and outside diameters are available in sizes up to approximately 5 inches inside diameter. Stock bushings are commonly provided 0.002 to 0.003 inch over nominal on outside diameter sizes of 3 inches or less. For diameters greater than 3 inches, outside diameters are 0.003 to 0.005 inch over nominal. Because these tolerances are built into standard bushings, the amount of press fit is controlled by the housing-bore size.

As a result of a press or shrink fit, the bore of the bearing material "closes in" by some amount. In general, this diameter decrease is approximately 70 to 100 per cent of the amount of the interference fit. Any attempt to accurately predict the amount of reduction, in an effort to avoid final clearance machining, should be avoided.

Shrink fits may be accomplished by chilling the bearing in a mixture of dry ice and alcohol, or in liquid air. These methods are easier than heating the housing and are preferred. Dry ice in alcohol has a temperature of −110 degrees F and liquid air boils at −310 degrees F.

When a bearing is pressed into the housing, the driving force should be uniformly applied to the end of the bearing to avoid upsetting or peening of the bearing. Of equal importance, the mating surfaces must be clean, smoothly finished, and free of machining imperfections.

Keying Methods: A variety of methods can be used to fix the position of the bearing with respect to its housing by "keying" the two together.

Methods of Bearing Retention

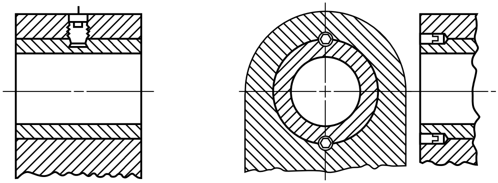

Fig. 3a. Set Screws

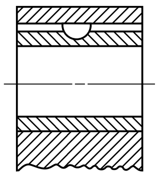

Fig. 3b. Woodruff Key

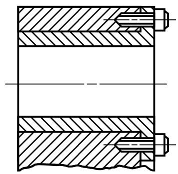

Fig. 3c. Bolts through Flange

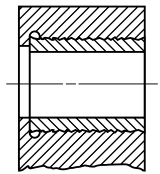

Fig. 3d. Bearing Screwed into Housing

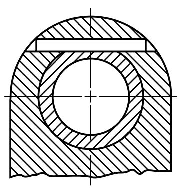

Fig. 3e. Dowel Pin

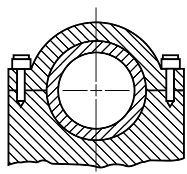

Fig. 3f. Housing Cap

Possible keying methods are shown in Figs. 3a through 3f including: D) set screws; E) Woodruff keys; F) bolted bearing flanges; G) threaded bearings; H) dowel pins; and I) housing caps.

Factors to be considered when selecting one of these methods are as follows:

1) Maintaining uniform wall thickness of the bearing material, if possible, especially in the load-carrying region of the bearing.

2) Providing as much contact area as possible between bearing and housing. Mating surfaces should be clean, smooth, and free from imperfections to facilitate heat transfer.

3) Preventing any local deformation of the bearing that might result from the keying method. Machining after keying is recommended.

4) Considering the possibility of bearing distortion resulting from the effect of temperature changes on the particular keying method.

Methods of Sealing.—In applications where lubricants or process fluids are utilized in operation, provision must be made normally to prevent leakage to other areas. This provision is made by the use of static and dynamic type sealing devices. In general, three terms are used to describe the devices used for sealing:

Seal: A means of preventing migration of fluids, gases, or particles across a joint or opening in a container.

Packing: A dynamic seal, used where some form of relative motion occurs between rigid members of an assembly.

Gaskets: A static seal, used where there is no relative motion between joined parts.

Two major functions must be achieved by all sealing applications: prevent escape of fluid; and prevent migration of foreign matter from the outside.

The first determination in selecting the proper seal is whether the application is static or dynamic. To meet the requirements of a static application there must be no relative motion between the joining parts or between the seal and the mating part. If there is any relative motion, the application must be considered dynamic, and the seal selected accordingly.

Dynamic sealing requires control of fluids leaking between parts with relative motion. Two primary methods are used to this end: positive contact or rubbing seals; and controlled clearance noncontact seals.

Positive Contact or Rubbing Seals: These seals are used where positive containment of liquids or gases is required, or where the seal area is continuously flooded. If properly selected and applied, contact seals can provide zero leakage for most fluids. However, because they are sensitive to temperature, pressure, and speed, improper application can result in early failure. These seals are applicable to rotating and reciprocating shafts. In many assemblies, positive-contact seals are available as off-the-shelf items. In other instances, they are custom-designed to the special demands of a particular application. Custom design is offered by many seal manufacturers and, for extreme cases, probably offers the best solution to the sealing problem.

Controlled Clearance Noncontact Seals: Representative of the controlled-clearance seals, which includes all seals in which there is no rubbing contact between the rotating and stationary members, are throttling bushings and labyrinths. Both types operate by fluid-throttling action in narrow annular or radial passages.

Clearance seals are frictionless and very insensitive to temperature and speed. They are chiefly effective as devices for limiting leakage rather than stopping it completely. Although they are employed as primary seals in many applications, the clearance seal also finds use as auxiliary protection in contact-seal applications. These seals are usually designed into the equipment by the designer himself, and they can take on many different forms.

Advantages of this seal are that friction is kept to an absolute minimum and there is no wear or distortion during the life of the equipment. However, there are two significant disadvantages: The seal has limited use when leakage rates are critical; and it becomes quite costly as the configuration becomes elaborate.

Static Seals: Static seals such as gaskets. “O” rings, and molded packings cover very broad ranges of both design and materials.

Some of the typical types are as follows: 1) Molded packings:A. lip type, B. squeeze-molded; 2) simple compression packings; 3) diaphragm seals; 4) nonmetallic gaskets; 5) “O” rings;; and 6) metallic gaskets and “O” rings..

Data on “O” rings are found starting on page 2482.

Detailed design information for specific products should be obtained directly from manufacturers.

Hardness and Surface Finish.—Even in well-lubricated full-film sleeve bearings, momentary contact between journal and bearing may occur under such conditions as starting, stopping, or overloading. In mixed-film and boundary-film lubricated sleeve bearings, continuous metal-to-metal contact occurs. Hence, to allow for any necessary wearing-in, the journal is usually made harder than the bearing material. This arrangement allows the effects of scoring or wearing to take place on the bearing, which is more easily replaced, rather than on the more expensive shaft. As a general rule, recommended Brinell (Bhn) hardness of the journal is at least 100 points harder than the bearing material.

The softer cast bronzes used for bearings are those with high lead content and very little tin. Such bronzes give adequate service in boundary-and mixed-film applications where full advantage is taken of their excellent "bearing" characteristics.

High-tin, low-lead content cast bronzes are the harder bronzes and these have high ulimate load-carrying capacity: higher journal hardnesses are required with these bearing bronzes. Aluminum bronze, for example, requires a journal hardness in the range of 550 to 600 Bhn.

In general, harder bearing materials require better alignment and more reliable lubrication to minimize local heat generation if and when the journal touches the shaft. Also, abrasives that find their way into the bearing are a problem for the harder bearing materials and greater care should be taken to exclude them.

Surface Finish: Whether bearing operation is complete boundary, mixed film, or fluid film, surface finishes of the journal and bearing must receive careful attention. In applications where operation is hydrodynamic or full-film, peak surface variations should be less than the expected minimum film thickness; otherwise, peaks on the journal surface will contact peaks on the bearing surface, with resulting high friction and temperature rise. Ranges of surface roughness obtained by various finishing methods are: boring, broaching, and reaming, 32 to 64 microinches, rms; grinding, 16 to 64 microinches, rms; and fine grinding, 4 to 16 microinches, rms.

In general, the better surface finishes are required for full-film bearings operating at high eccentricity ratios because full-film lubrication must be maintained with small clearances, and metal-to-metal contact must be avoided. Also, the harder the material, the better the surface finish required. For boundary- and mixed-film applications, surface finish requirements may be somewhat relaxed because bearing wear-in will in time smooth the surfaces.

Fig. 4 is a general guide to the ranges required for bearing and journal surface finishes. Selecting a particular surface finish in each range can be simplified by observing the general rule that smoother finishes are required for the harder materials, for high loads, and for high speeds.

Machining.—The methods most commonly used in finishing journal bearing bores are boring, broaching, reaming, and burnishing.

Broaching is a rapid finishing method providing good size and alignment control when adequate piloting is possible. Soft babbitt materials are particularly compatible with the broaching method. A third finishing method, reaming, facilitates good size and alignment control when piloting is utilized. Reaming can be accomplished both manually or by machine, the machine method being preferred. Burnishing is a fast sizing operation that gives good alignment control, but does not give as good size control as the cutting methods. It is not recommended for soft materials such as babbitt. Burnishing has an ironing effect that gives added seating of the bushing outside diameter in the housing bore; consequently, it is often used for this purpose, especially on a $\frac{1}{32}$-inch wall bushing, even if a further sizing operation is to be used subsequently.

Boring of journal bearings provides the best concentricity, alignment, and size control and is the finishing method of choice when close tolerances and clearances are desirable.

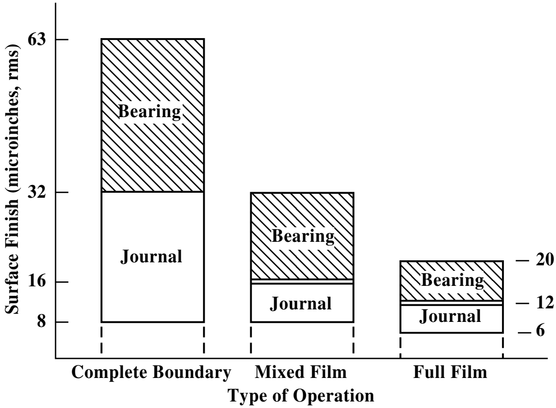

Fig. 4. Recommended ranges of surface finish for the three types of sleeve bearing operations.

Methods of Lubrication.—There are numerous ways to supply lubricant to bearings. The more common of these are described in the following.

Pressure lubrication,: in which an abundance of oil is fed to the bearing from a central groove, single or multiple holes, or axial grooves, is effective and efficient. The moving oil assists in flushing dirt from the bearing and helps keep the bearing cool. In fact, it removes heat faster than other lubricating methods and, therefore, permits thinner oil films and unimpaired load capacities. The oil-supply pressure needed for bushings carrying the basic load is directly proportional to the shaft speed, but for most installations, 50 psi will be adequate.

Splash fed: applies to a variety of intermittently lubricated bushings. It includes everything from bearings spattered with oil from the action of other moving parts to bearings regularly dipped in oil. Like oil bath lubrication, splash feeding is practical when the housing can be made oiltight and when the moving parts do not churn the oil. The fluctuating nature of the load and the intermittent oil supply in splash fed applications requires the designer to use experience and judgment when determining the probable load capacity of bearings lubricated in this way.

Oil bath lubrication,: in which the bushing is submerged in oil, is the most reliable of all methods except pressure lubrication. It is practical if the housing can be made oil tight, and if the shaft speed is not so great as to cause excessive churning of the oil.

Oil ring lubrication,: in which oil is supplied to the bearing by a ring in contact with the shaft, will, within reasonable limits, bring enough oil to the bearing to maintain hydrodynamic lubrication. If the shaft speed is too low, little oil will follow the ring to the bearing; and, if the speed is too high, the ring speed will not keep pace with the shaft. Also, a ring revolving at high speed will lose oil by centrifugal force. For best results, the peripheral speed of the shaft should be between 200 and 2000 feet per minute. Safe load to achieve hydrodynamic lubrication should be one-half of that for pressure fed bearings. Unless the load is light, hydrodynamic lubrication is doubtful. The safe load, then, to achieve hydrodynamic lubrication, should be one-quarter of that of pressure fed bearings.

Table 1. Oil Viscosity Unit Conversion

Convert from	Convert to: Poise (P)	Centipoise (Z)	Reyn (μ)	Stoke (S)	Centistoke (v)
	Multiplying Factors				
Poise (P) $\frac{\text{dyne-s}}{\text{cm}^2}$ or $\frac{\text{gram mass}}{\text{cm-s}}$	1	100	1.45×10^{-5}	$\frac{1}{\rho}$	$\frac{100}{\rho}$
Centipoise (Z) $\frac{\text{dyne-s}}{100\ \text{cm}^2}$ or $\frac{\text{gram mass}}{100\text{cm-s}}$	0.01	1	1.45×10^{-7}	$\frac{0.01}{\rho}$	$\frac{1}{\rho}$
Reyn (μ) $\frac{\text{lb force-s}}{\text{in}^2}$	6.9×10^4	6.9×10^4	1	$\frac{6.9 \times 10^4}{\rho}$	$\frac{6.9 \times 10^6}{\rho}$
Stoke (S) $\frac{\text{cm}^2}{\text{s}}$	ρ	$100\ \rho$	$1.45 \times 10^{-5}\ \rho$	1	100
Centistoke (v) $\frac{\text{cm}^2}{100\ \text{s}}$	$0.01\ \rho$	ρ	$1.45 \times 10^{-7}\ \rho$	0.01	1

ρ = Specific gravity of the oil.

To convert from a value in the "Convert from" column to a value in a "Convert to" column, multiply the "Convert from" column by the figure in the intersecting block, e.g. to change from Centipoise to Reyn, multiply Centipoise value by 1.45×10^{-7}.

Wick or waste pack lubrication: delivers oil to a bushing by the capillary action of a wick or waste pack; the amount delivered is proportional to the size of the wick or pack.

Lubricants: The value of an oil as a lubricant depends mainly on its film-forming capacity, that is, its capability to maintain a film of oil between the bearing surfaces. The film-forming capacity depends to a large extent on the viscosity of the oil, but this should not be understood to mean that oil of the highest viscosity is always the most suitable lubricant. For practical reasons, an oil of the lowest viscosity that will retain an unbroken oil film between the bearing surfaces is the most suitable for purposes of lubrication. A higher viscosity than that necessary to maintain the oil film results in a waste of power due to the expenditure of energy necessary to overcome the internal friction of the oil itself.

Fig. 5 provides representative values of viscosity in centipoises for SAE mineral oils. Table 1 is provided as a means of converting viscosities of other units to centipoises.

Grease: packed in a cavity surrounding the bushing is less adequate than an oil system, but it has the advantage of being more or less permanent. Although hydrodynamic lubrication is possible under certain very favorable circumstances, boundary lubrication is the usual state.

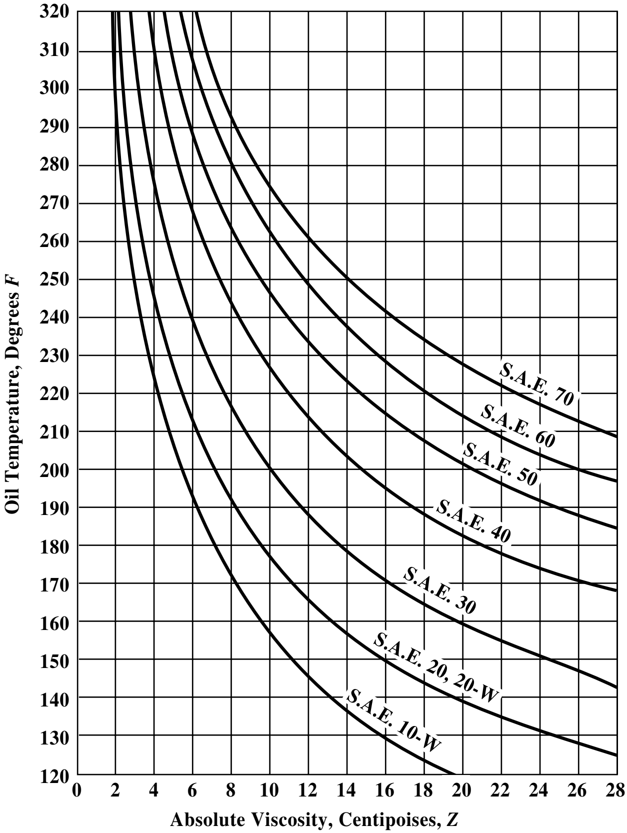

Fig. 5. Viscosity vs. temperature—SAE oils.

Lubricant Selection.—In selecting lubricants for journal bearing operation, several factors must be considered: 1) Type of operation (full, mixed, or boundary film) anticipated;

2) Surface speed; and 3) Bearing loading.

Fig. 6 combines these factors and facilitates general selection of the proper lubricant viscosity range.

As an example of using these curves, consider a lightly loaded bearing operating at 2000 rpm. At the bottom of the figure, locate 2000 rpm and move vertically to intersect the light-load full-film lubrication curve, which indicates an SAE 5 oil.

As a general rule-of-thumb, heavier oils are recommended for high loads and lighter oils for high speeds.

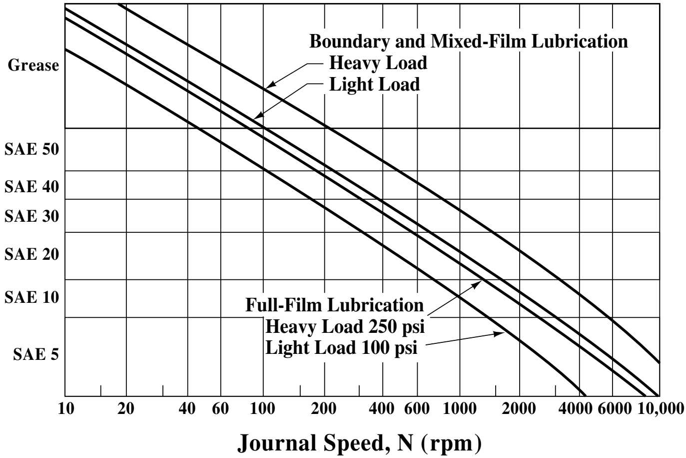

Fig. 6. Lubricant Selection Guide

In addition, other than using conventional lubrication oils, journal bearings may be lubricated with greases or solid lubricants. Some of the reasons for use of these lubricants are to:

1) Lengthen the period between relubrication:

2) Avoid contaminating surrounding equipment or material with "leaking" lubricating oil;

3) Provide effective lubrication under extreme temperature ranges;

4) Provide effective lubrication in the presence of contaminating atmospheres; and

5) Prevent intimate metal-to-metal contact under conditions of high unit pressure which might destroy boundary lubricating films.

Greases: Where full-film lubrication is not possible or is impractical for slow-speed fairly high-load applications, greases are widely used as bearing lubricants. Although full-film lubrication with grease is possible, it is not normally considered since an elaborate pumping system is required to continuously supply a prescribed amount of grease to the bearing. Bearings supplied with grease are usually lubricated periodically. Grease lubrication, therefore, implies that the bearing will operate under conditions of complete boundary lubrication and should be designed accordingly.

Lubricating greases are essentially a combination of a mineral lubricating oil and a thickening agent, which is usually a metallic soap. When suitably mixed, they make excellent bearing lubricants. There are many different types of greases which, in general, may be classified according to the soap base used. Information on commonly used greases is shown in Table 2.

Synthetic greases are composed of normal types of soaps but use synthetic hydrocarbons instead of normal mineral oils. They are available in a range of consistencies in both water-soluble and insoluble types. Synthetic greases can accommodate a wide range of variation in operating temperature; however, recommendations on special-purpose greases should be obtained from the lubricant manufacturer.

Application of grease is accomplished by one of several techniques depending upon grease consistency. These classifications are shown in Table 3 along with typical methods of application. Grooves for grease are generally greater in width, up to 1.5 times, than for oil.

Table 2. Commonly Used Greases and Solid Lubricants

Type	Operating Temperature, Degrees F	Load	Comments
Greases			
Calcium or lime soap	160	Moderate	…
Sodium soap	300	Wide	For wide speed range
Aluminum soap	180	Moderate	…
Lithium soap	300	Moderate	Good low temperature
Barium soap	350	Wide	…
Solid Lubricants			
Graphite	1000	Wide	…
Molybdenum disulfide	−100 to 750	Wide	…

Table 3. NLGI Consistency Numbers

NLGI Consistency No.	Consistency of Grease	Typical Method of Application
0	Semifluid	Brush or gun
1	Very soft	Pin-type cup or gun
2	Soft	Pressure gun or centralized pressure system
3	Light cup grease	Pressure gun or centralized pressure system
4	Medium cup grease	Pressure gun or centralized pressure system
5	Heavy cup grease	Pressure gun or hand
6	Block grease	Hand, cut to fit

NLGI is National Lubricating Grease Institute

Coefficients of friction for grease-lubricated bearings range from 0.08 to 0.16, depending upon consistency of the grease, frequency of lubrication, and type of grease. An average value of 0.12 may be used for design purposes.

Solid Lubricants: The need for effective high-temperature lubricants led to the development of several solid lubricants. Essentially, solid lubricants may be described as low-shear-strength solid materials. Their function within a bronze bearing is to act as an intermediary material between sliding surfaces. Since these solids have very low shear strength, they shear more readily than the bearing material and thereby allow relative motion. So long as solid lubricant remains between the moving surfaces, effective lubrication is provided and friction and wear are reduced to acceptable levels.

Solid lubricants provide the most effective boundary films in terms of reduced friction, wear, and transfer of metal from one sliding component to the other. However, there is a significant deterioration in these desirable properties as the operating temperature of the boundary film approaches the melting point of the solid film. At this temperature the friction may increase by a factor of 5 to 10 and the rate of metal transfer may increase by as much as 1000. What occurs is that the molecules of the lubricant lose their orientation to the surface that exists when the lubricant is solid. As the temperature further increases, additional deterioration sets in with the friction increasing by some additional small amount but the transfer of metal accelerates by an additional factor of 20 or more. The final effect of too high temperature is the same as metal-to-metal contact without benefit of lubricant. These changes, which are due to the physical state of the lubricant, are reversed when cooling takes place.

The effects just described also partially explain why fatty acid lubricants are superior to paraffin base lubricants. The fatty acid lubricants react chemically with the metallic surfaces to form a metallic soap that has a higher melting point than the lubricant itself, the result being that the breakdown temperature of the film, now in the form of a metallic soap is raised so that it acts more like a solid film lubricant than a fluid film lubricant.

Journal or Sleeve Bearings

Although this type of bearing may take many shapes and forms, there are always three basic components: journal or shaft, bushing or bearing, and lubricant. Fig. 7 shows these components with the nomenclature generally used to describe a journal bearing: W = applied load, N = revolution, e = eccentricity of journal center to bearing center, θ = attitude angle, which is the angle between the applied load and the point of minimum film thickness, d = diameter of the shaft, c_d = bearing clearance, $d + c_d$ = diameter of the bearing and h_o = minimum film thickness.

Grooving and Oil Feeding.—Grooving in a journal bearing has two purposes: 1) to establish and maintain an efficient film of lubricant between the bearing moving surfaces and; and 2) to provide adequate bearing cooling.

The obvious and only practical location for introducing lubricant to the bearing is in a region of low pressure. A typical pressure profile of a bearing is shown by Fig. 8. The arrow W shows the applied load. Typical grooving configurations used for journal bearings are shown in Figs. 10a through 10e.

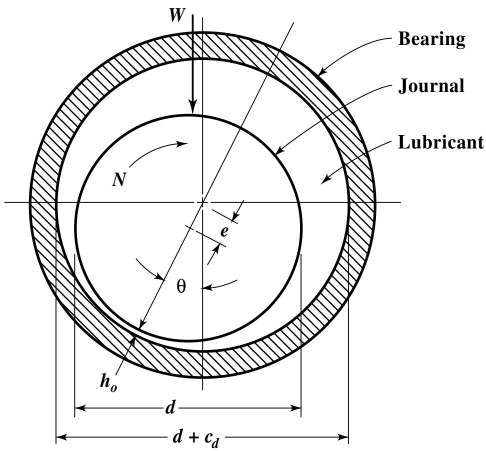

Fig. 7. Basic components of a journal bearing.

Heat Radiating Capacity.—In a self-contained lubrication system for a journal bearing, the heat generated by bearing friction must be removed to prevent continued temperature rise to an unsatisfactory level. The heat-radiating capacity H_R of the bearing in foot-pounds per minute may be calculated from the formula $H_R = Ld\, Ct_R$ in which C is a constant determined by O. Lasche, and t_R is temperature rise in degrees Fahrenheit.

Values for the product Ct_R may be found from the curves in Fig. 9 for various values of bearing temperature rise t_R and for three operating conditions. In this equation, L = total length of the bearing in inches and d = bearing diameter in inches.

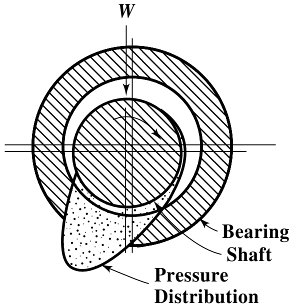

Fig. 8. Typical pressure profile of journal bearing.

Journal Bearing Design Notation.—The symbols used in the following step-by-step procedure for lubrication analysis and design of a plain sleeve or journal bearing are as follows:

c = specific heat of lubricant, Btu/lb/degree F
c_d = diametral clearance, inches
C_n = bearing capacity number
d = journal diameter, inches
e = eccentricity, inches

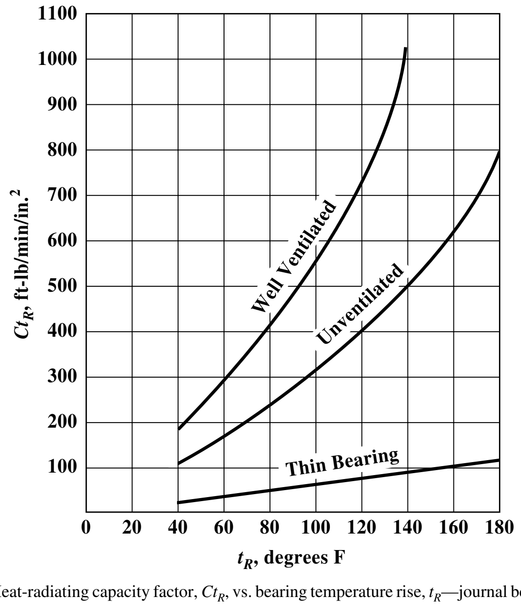

Fig. 9. Heat-radiating capacity factor, Ct_R, vs. bearing temperature rise, t_R—journal bearings.

Types of Journal Bearing Oil Grooving

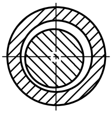

Fig. 10a. Single inlet hole

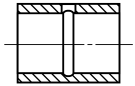

Fig. 10b. Circular groove

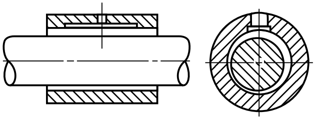

Fig. 10c. Straight axial groove

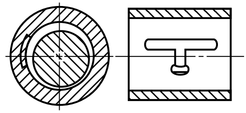

Fig. 10d. Straight axial groove with feeder groove

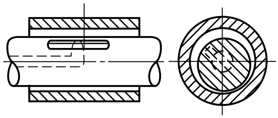

Fig. 10e. Straight axial groove in shaft

h_o = minimum film thickness, inch
K = constants
l = bearing length as defined in Fig. 11, inches
L = actual length of bearing, inches
m = clearance modulus
N = rpm
p_b = unit load, psi
p_s = oil supply pressure, psi
P_f = friction horsepower
P' = bearing pressure parameter
q = flow factor
Q_1 = hydrodynamic flow, gpm
$Q2$ = pressure flow, gpm
Q = total flow, gpm
Q_{new} = new total flow, gpm
Q_R = total flow required, gpm
r = journal radius, inches
Δt = actual temperature rise of oil in bearing, °F
Δt_a = assumed temperature rise of oil in bearing, °F
Δt_{new} = new assumed temperature rise of oil in bearing, °F
t_b = bearing operating temperature, °F
t_{in} = oil inlet temperature, °F
T_f = friction torque, inch-pounds/inch
T' = torque parameter
W = load, pounds
X = factor
Z = viscosity, centipoises
$\in$ = eccentricity ratio — ratio of eccentricity to radial clearance
α = oil density, lbs/inch^3

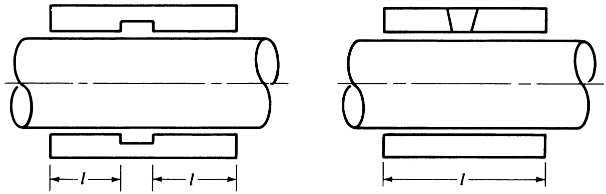

Fig. 11. Length, l, of bearing for circular groove type (left) and single inlet hole type (right).

Journal Bearing Lubrication Analysis.—The following procedure leads to a complete lubrication analysis which forms the basis for the bearing design.

1) *Diameter of bearing d.* This is usually determined by considering strength and/or deflection requirements for the shaft using principles of strength of materials.

2) *Length of bearing L.* This is determined by an assumed l/d ratio in which l may or may not be equal to the overall length, L (See Step 6). Bearing pressure and the possibility of edge loading due to shaft deflection and misalignment are factors to be considered. In general, shaft misalignment resulting from location tolerances and/or shaft deflections should be maintained below 0.0003 inch per inch of length.

3) *Bearing pressure* p_b. The unit load in pound per square inch is calculated from the formula:

$$p_b = \frac{W}{Kld}$$

where $K = 1$ for single oil hole

$K = 2$ for central groove

$W =$ load, pounds

$l =$ bearing length as defined in Fig. 11, inches

$d =$ journal diameter, inches

4) Typical unit loads in service are shown in Table 4. These pressures can be used as a safe guide in selection. However, if space limitations impose a higher limit of loading, the complete lubrication analysis and evaluation of material properties will determine acceptability.

Diametral clearance c_d. This is selected on a trial basis from Fig. 12 which shows suggested diametral clearance ranges for various shaft sizes and for two speed ranges. These are *hot* or *operating* clearances so that thermal expansion of journal and bearing to these temperatures must be taken into consideration in establishing machining dimensions. The optimum operating clearance should be determined on the basis of a complete lubrication analysis (See paragraph following Step 23).

Clearance modulus m. This is calculated from the formula:

$$m = \frac{c_d}{d}$$

Length to diameter ratio l/d. This is usually between 1 and 2; however, with the modern trend toward higher speeds and more compact units, lower ratios down to 0.3 are used. In shorter bearings there is a consequent reduction in load carrying capacity due to excessive end or side leakage of lubricant. In longer bearings there may be a tendency towards edge loading. Length l for a single oil feed hole is taken as the total length of the bearing as shown in Fig. 11. For a central oil groove length, l is taken as one-half the total length.

Typical l/d ratio's use for various types of applications are given in Table 5.

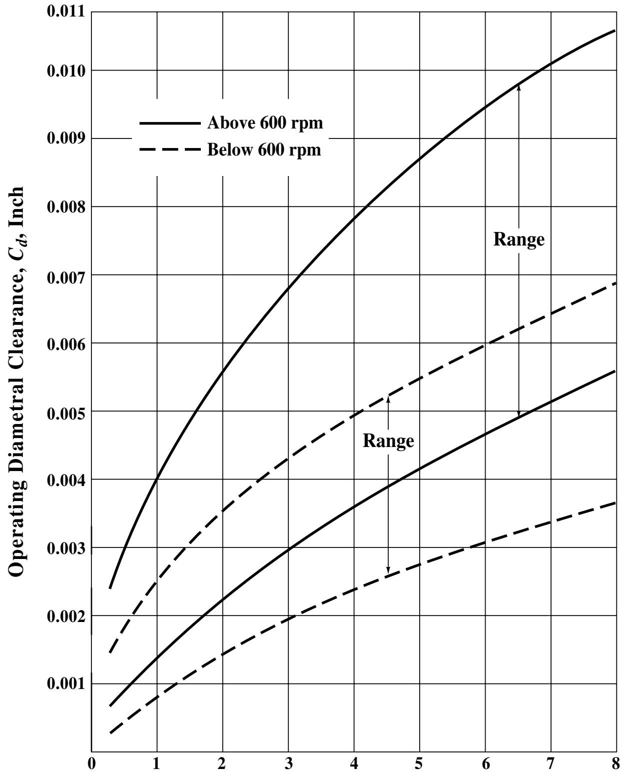

Fig. 12. Operating Diametral Clearance C_d vs. Journal Diameter d.

5) *Assumed operating temperature* t_b. A temperature rise of the lubricant as it passes through the bearing is assumed and the consequent operating temperature in degrees F is calculated from the formula:

$$t_b = t_{in} + \Delta t_a$$

where t_{in} = inlet temperature of oil in °F

Δt_a = assumed temperature rise of oil in bearing in °F

6) An initial assumption of 20°F is usually made.

7) *Viscosity of lubricant Z.* The viscosity in centipoises at the assumed bearing operating temperature is found from the curve in Fig. 5 which shows the viscosity of SAE grade oils versus temperature.

Bearing pressure parameter P′. This value is required to find the eccentricity ratio and is calculated from the formula:

$$P' = \frac{6.9(1000m)^2 p_b}{ZN}$$

where N = rpm

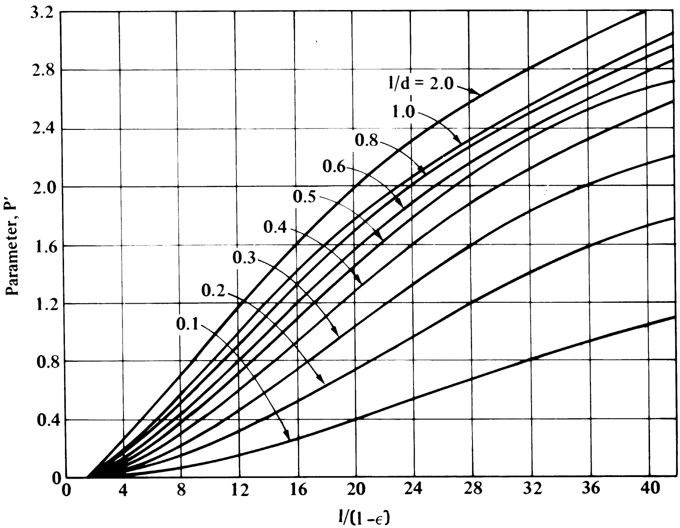

Fig. 13. Bearing parameter. P', vs. eccentricity ratio. $1/(1 - \in)$ — journal bearings.

8) *Eccentricity ratio* $\in$. Using P' and l/d, the value of $1/(1 - \in)$ is determined from Fig. 13 and from this $\in$ can be determined.

9) *Torque parameter T′.* This value is obtained from Fig. 14 or Fig. 15 using $1/(1 - \in)$ and l/d.

Friction torque T. This value is calculated from the formula:

$$T = \frac{T' r^2 ZN}{6900(1000m)}$$

where r = journal radius, inches

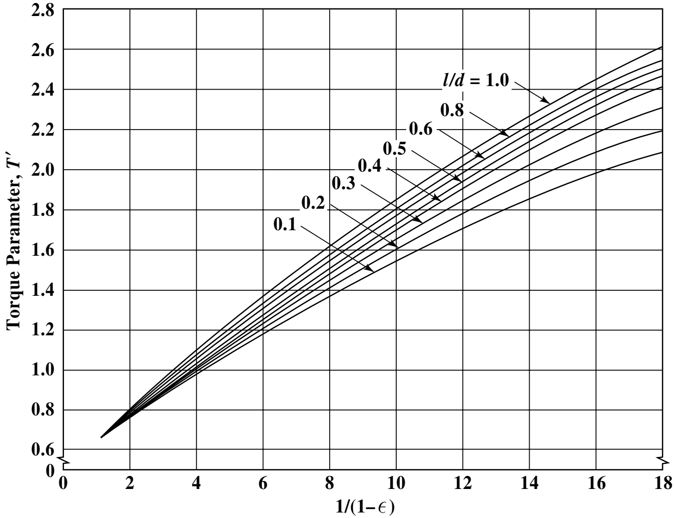

Fig. 14. Torque parameter, T', vs. eccentricity ratio, $1(1 — \in)$ — journal bearings.

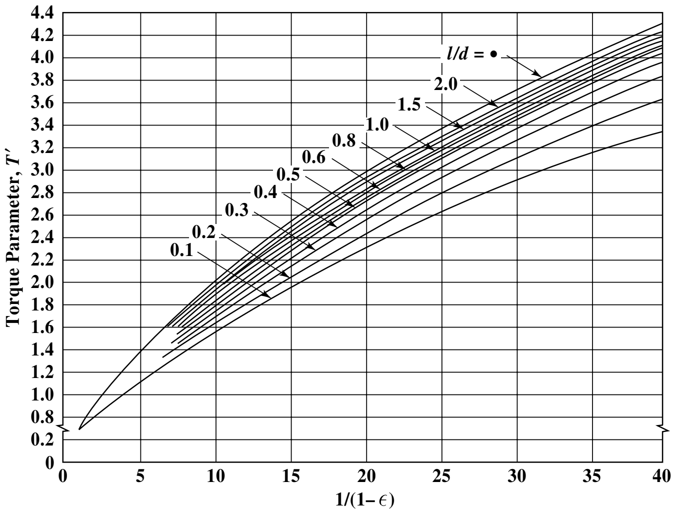

Fig. 15. Torque parameter, T', vs eccentricity ratio, $1/(1 — \in)$ — journal bearings.

10) *Friction horsepower* P_f. This value is calculated from the formula:

$$P_f = \frac{KTNl}{63{,}000}$$

where $K = 1$ for single oil hole, 2 for central groove.

Factor X. This factor is used in the calculation of the lubricant flow and can either be obtained from Table 6 or calculated from the formula:

$$X = 0.1837 / \alpha c$$

where α = oil density in pounds per cubic inch

c = specific heat of lubricant in Btu/lb/°F

Total flow of lubricant required Q_R. This is calculated from the formula:

$$Q_R = \frac{X(P_f)}{\Delta t_a}$$

Bearing capacity number C_n. This value is needed to obtain the flow factor and is calculated from the formula:

$$C_n = \left(\frac{l}{d}\right)^2 / 60P'$$

Flow factor q. This value is obtained from the curve in Fig. 16.

Hydrodynamic flow of lubricant Q_1. This flow in gallons per minute is calculated from the formula:

$$Q_1 = \frac{Nlc_d qd}{294}$$

Pressure flow of lubricant Q_2. This flow in gallons per minute is calculated from the formula:

$$Q_2 = \frac{Kp_s c_d^3 d(1 + 1.5\epsilon^2)}{Zl}$$

where $K = 1.64 \times 10^5$ for single oil hole

$K = 2.35 \times 10^5$ for central groove

p_s = oil supply pressure

Total flow of lubricant Q. This value is obtained by adding the hydrodynamic flow and the pressure flow.

$$Q = Q_1 + Q_2$$

Bearing temperature rise Δt. This temperature rise in degrees F is obtained from the formula:

$$\Delta t = \frac{X(P_f)}{Q}$$

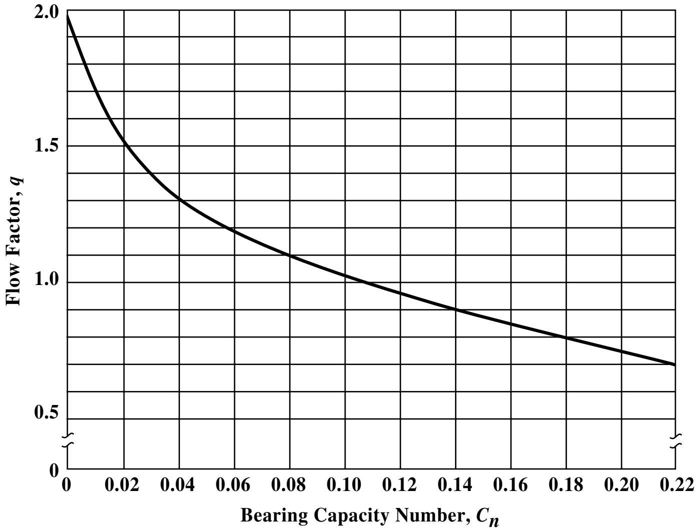

Fig. 16. Flow factor, q, vs. bearing capacity number, C_n—journal bearings.

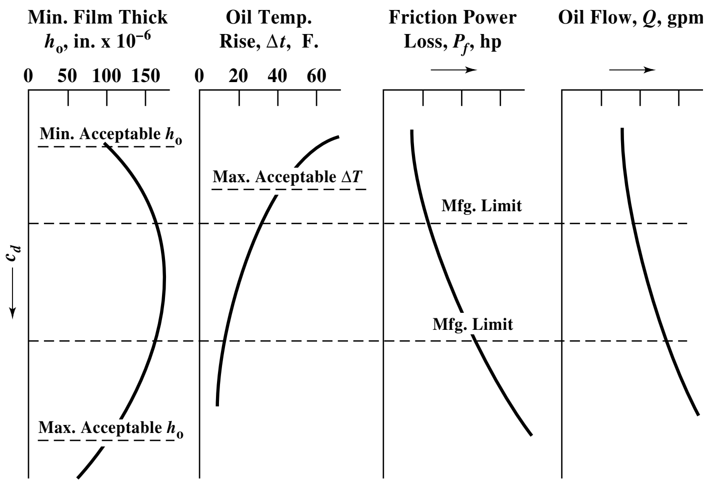

Fig. 17. Example of lubrication analysis curves for journal bearing.

11) *Comparison of actual and assumed temperature rises.* At this point if Δt_a and Δt differ by more than 5 degrees F, Steps 7 through 22 are repeated using a Δt_{new} halfway between the former Δt_a and Δt.

Minimum film thickness h_o. When Step 22 has been satisfied, the minimum film thickness in inches is calculated from the formula: $h_o = \frac{1}{2}C_d(1 — \in)$.

A new diametral clearance c_d is now assumed and Steps 5 through 23 are repeated. When this repetition has been done for a sufficient number of values for c_d, the full lubrication study is plotted as shown in Fig. 17. From this chart a working range of diametral clearance can be determined that optimizes film thickness, differential temperature, friction horsepower and oil flow.

Table 4. Allowable Sleeve Bearing Pressures for Various Classes of Bearings

Types of Bearing or Kind of Service	Pressure, Lbs. per Sq. In.	Types of Bearing or Kind of Service	Pressure, Lbs. per Sq. In.
Electric Motor & Generator		Diesel Engine	
Bearings (General)	100–200	Rod	1000–2000
Turbine & Reduction		Wrist Pins	1800–2000
Gears	100–250	Automotive,	
Heavy Line Shafting	100–150	Main Bearings	500–700
Locomotive Axles	300–350	Rod Bearings	1500–2500
Light Line Shafting	15–35	Centrifugal Pumps	80–100
Diesel Engine, Main	800–1500	Aircraft Rod Bearings	700–3000

These pressures in pounds per square inch of area equal to length times diameter are intended as a general guide only. The allowable unit pressure depends upon operating conditions, especially in regard to lubrication, design of bearings, workmanship, velocity, and nature of load.

Table 5. Representative *l/d* Ratios

Type of Service	*l/d*	Type of Service	*l/d*
Gasoline and diesel engine		Light shafting	2.5 to 3.5
main bearings and crankpins	0.3 to 1.0	Heavy shafting	2.0 to 3.0
Generators and motors	1.2 to 2.5	Steam engine	
Turbogenerators	0.8 to 1.5	Main bearings	1.5 to 2.5
Machine tools	2.0 to 3.0	Crank and wrist pins	1.0 to 1.3

Table 6. *X* Factor vs. Temperature of Mineral Oils

Temperature	*X* Factor
100	12.9
150	12.4
200	12.1
250	11.8
300	11.5

Use of Lubrication Analysis.—Once the lubrication analysis has been completed and plotted as shown in Fig. 17, the following steps lead to the optimum bearing design, taking into consideration both basic operating requirements and requirements peculiar to the application.

1) Examine the curve (Fig. 17)

1) for minimum film thickness and determine the acceptable range of diametral clearance, c_d, based on a) a minimum of 200×10^{-6} inches for small bearings under 1 inch diameter; b) a minimum of 500×10^{-6} inches for bearings from 1 to 4 inches diameter; 1) and

a) a minimum of 750×10^{-6} inches for larger bearings. More conservative designs would increase these requirements.

2) Determine the minimum acceptable c_d based on a maximum Δt of 40°F from the oil temperature rise curve (Fig. 17).

3) If there are no requirements for maintaining low friction horsepower and oil flow, the possible limits of diametral clearance are now defined.

4) The required manufacturing tolerances can now be placed within this band to optimize h_o as shown by Fig. 17.

5) If oil flow and power loss are a consideration, the manufacturing tolerances may then be shifted, within the range permitted by the requirements for h_o and Δt.

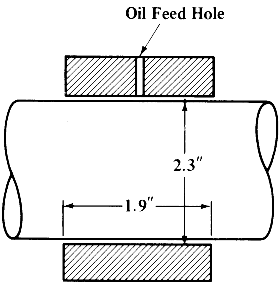

Fig. 18. Full journal bearing example design.

A full journal bearing, Fig. 18, 2.3 inches in diameter and 1.9 inches long is to carry a load of 6000 pounds at 4800 rpm, using SAE 30 oil supplied at 200°F through a single oil hole at 30 psi. Determine the operating characteristics of this bearing as a function of diametral clearance.

1) *Diameter of bearing*. Given as 2.3 inches.

2) *Length of bearing*. Given as 1.9 inches.

3) *Bearing pressure.*

$$p_b = \frac{6000}{1 \times 1.9 \times 2.3} = 1372 \text{ lbs. per sq. in.}$$

4) *Diametral clearance.* Assume c_d is equal to 0.003 inch from Fig. 12 for first calculation.

5) *Clearance modulus.*

$$m = \frac{0.003}{2.3} = 0.0013 \text{ inch}$$

6) *Length-to-diameter ratio.*

$$\frac{l}{d} = \frac{1.9}{2.3} = 0.83$$

7) *Assumed operating temperature.* If the temperature rise Δt_a is assumed to be 20°F,

$$t_b = 200 + 20 = 220°\text{F}$$

8) *Viscosity of lubricant.* From Fig. 5, Z = 7.7 centipoises

9) *Bearing-pressure parameter.*

$$P' = \frac{6.9 \times 1.3^2 \times 1372}{7.7 \times 4800} = 0.43$$

10) *Eccentricity ratio.* From Fig. 13, $\frac{1}{1 - \in} = 6.8$ and $\in = 0.85$

11) *Torque parameter.* From Fig. 14, $T' = 1.46$

12) *Friction torque.*

$$T_f = \frac{1.46 \times 1.15^2 \times 7.7 \times 4800}{6900 \times 1.3} = 7.96 \text{ inch-pounds per inch}$$

13) *Friction horsepower.*

$$P_f = \frac{1 \times 7.96 \times 4800 \times 1.9}{63{,}000} = 1.15 \text{ horsepower}$$

14) *Factor X.* From Table 6, $X = 12$, approximately

15) *Total flow of lubricant required.*

$$Q_R = \frac{12 \times 1.15}{20} = 0.69 \text{ gallon per minute}$$

16) *Bearing-capacity number.*

$$C_n = \frac{0.83^2}{60 \times 0.43} = 0.027$$

17) *Flow factor.* From Fig. 16, $q = 1.43$

18) *Actual hydrodynamic flow of lubricant.*

$$Q_1 = \frac{4800 \times 1.9 \times 0.003 \times 1.43 \times 2.3}{294} = 0.306 \text{ gallon per minute}$$

19) *Actual pressure flow of lubricant.*

$$Q_2 = \frac{1.64 \times 10^5 \times 30 \times 0.003^3 \times 2.3 \times (1 + 1.5 \times 0.85^2)}{7.7 \times 1.9} = 0.044 \text{ gallon per min}$$

20) *Actual total flow of lubricant.*

$$Q = 0.306 + 0.044 = 0.350 \text{ gallon per minute}$$

21) *Actual bearing-temperature rise.*

$$\Delta t = \frac{12 \times 1.15}{0.350} = 39.4°\text{F}$$

22) *Comparison of actual and assumed temperature rises.* Because Δt_a and Δt differ by more than 5°F, a new Δt_a, midway between these two, of 30°F is assumed and Steps 7 through 22 are repeated.

7a. *Assumed operating temperature.*

$$t_b = 200 + 30 = 230°\text{F}$$

8a. *Viscosity of lubricant.* From Fig. 5, $Z = 6.8$ centipoises

9a. *Bearing-pressure parameter.*

$$P' = \frac{6.9 \times 1.3^2 \times 1372}{6.8 \times 4800} = 0.49$$

10a. *Eccentricity ratio.* From Fig. 13,

$$\frac{1}{1 - \in} = 7.4$$

and $\in = 0.86$

11a. *Torque parameter.* From Fig. 14, $T' = 1.53$

12a. *Friction torque.*

$$T_f = \frac{1.53 \times 1.15^2 \times 6.8 \times 4800}{6900 \times 1.3} = 7.36 \text{ inch-pounds per inch}$$

13a. *Friction horsepower.*

$$P_f = \frac{1 \times 7.36 \times 4800 \times 1.9}{63{,}000} = 1.07 \text{ horsepower}$$

14a. *Factor X.* From Table 6, $X = 11.9$ approximately

15a. *Total flow of lubricant required.*

$$Q_R = \frac{11.9 \times 1.07}{30} = 0.42 \text{ gallon per minute}$$

16a. *Bearing-capacity number.*

$$C_n = \frac{0.83^2}{60 \times 0.49} = 0.023$$

17a. *Flow factor.* From Fig. 16, $q = 1.48$

18a. *Actual hydrodynamic flow of lubricant.*

$$Q_1 = \frac{4800 \times 1.9 \times 0.003 \times 1.48 \times 2.3}{294} = 0.317 \text{ gallon per minute}$$

19a. *Pressure flow.*

$$Q_2 = \frac{1.64 \times 10^5 \times 30 \times 0.003^3 \times 2.3 \times (1 + 1.5 \times 0.86^2)}{6.8 \times 1.9} = 0.050 \text{ gallon per minute}$$

20a. *Actual flow of lubricant.*

$$Q_{new} = 0.317 + 0.050 = 0.367 \text{ gallon per minute}$$

21a. *Actual bearing-temperature rise.*

$$\Delta t = \frac{11.9 \times 1.06}{0.367} = 34.4°\text{F}$$

22a. *Comparison of actual and assumed temperature rises.* Now Δt and Δt_a are within 5 degrees F.

23) *Minimum film thickness.*

$$h_o = \frac{0.003}{2}(1 - 0.86) = 0.00021 \text{ inch}$$

This analysis may now be repeated for other values of c_d determined from Fig. 12 and a complete lubrication analysis performed and plotted as shown in Fig. 17. An operating range for c_d can then be determined to optimize minimum clearance, friction horsepower loss, lubricant flow, and temperature rise.

THRUST BEARINGS

As the name implies, thrust bearings are used either to absorb axial shaft loads or to position shafts axially. Brief descriptions of the normal designs for these bearings follow with approximate design methods for each. The generally accepted load ranges for these types of bearings are given in Table 7 and the schematic configurations are shown in Fig. 19.

The parallel or flat plate thrust bearing is probably the most frequently used type. It is the simplest and lowest in cost of those considered; however, it is also the least capable of absorbing load, as can be seen from Table 7. It is most generally used as a positioning device where loads are either light or occasional.

The step bearing, like the parallel plate, is also a relatively simple design. This type of bearing will accept the normal range of thrust loads and lends itself to low-cost, high-volume production. However, this type of bearing becomes sensitive to alignment as its size increases.

The tapered land thrust bearing, as shown in Table 7, is capable of high load capacity. Where the step bearing is generally used for small sizes, the tapered land type can be used in larger sizes. However, it is more costly to manufacture and does require good alignment as size is increased.

The tilting pad or Kingsbury thrust bearing (as it is commonly referred to) is also capable of high thrust capacity. Because of its construction it is more costly, but it has the inherent advantage of being able to absorb significant amounts of misalignment.

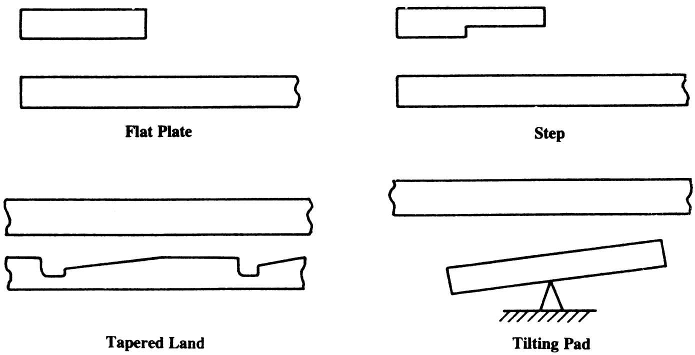

Fig. 19. Types of thrust bearings.

Table 7. Thrust Bearing Loads*

Type	Normal Unit Loads, Lb per Sq. In.	Maximum Unit Loads, Lb per Sq. In.
Parallel surface	<75	<150
Step	200	500
Tapered land	200	500
Tilting pad	200	500

Thrust Bearing Design Notation.—The symbols used in the design procedures that follow for flat plate, step, tapered land, and tilting pad thrust bearings are as follows:

a = radial width of pad, inches

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b = circumferential length of pad at pitch line, inches
b_2 = pad step length
B = circumference of pitch circle, inches
c = specific heat of oil, Btu/gal/°F
D = diameter, inches
e = depth of step, inch
f = coefficient of friction
g = depth of 45° chamfer, inches
h = film thickness, inch
i = number of pads
J = power loss coefficient
K = film thickness factor
K_g = fraction of circumference occupied by the pads; usually, 0.8
l = length of chamfer, inches
M = horsepower per square inch
N = revolutions per minute
O = operating number
p = bearing unit load, psi
p_s = oil-supply pressure, psi
P_f = friction horsepower
Q = total flow, gpm
Q_c = required flow per chamfer, gpm
$Q^o{}_c$ = uncorrected required flow per chamfer, gpm
Q_F = film flow, gpm
s = oil-groove width
Δt = temperature rise, °F
U = velocity, feet per minute
V = effective width-to-length ratio for one pad
W = applied load, pounds
Y_G = oil-flow factor
Y_L = leakage factor
Y_S = shape factor
Z = viscosity, centipoises
α = dimensionless film-thickness factor
δ = taper
ξ = kinetic energy correction factor

Note: In the following, subscript 1 denotes inside diameter and subscript 2 denotes outside diameter. Subscript *i* denotes inlet and subscript *o* denotes outlet.

Flat Plate Thrust Bearing Design.—The following steps define the performance of a flat plate thrust bearing, one section of which is shown in Fig. 20. Although each bearing section is wedge shaped, as shown below right, for the purposes of design calculation, it is considered to be a rectangle with a length *b* equal to the circumferential length along the pitch line of the section being considered, and a width *a* equal to the difference in the external and internal radii.

General Parameters: X) From Table 7, the maximum unit load is between 75 and 100 pounds per square inch; and Y) The outside diameter is usually between 1.5 and 2.5 times the inside diameter.

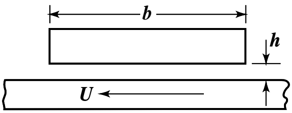
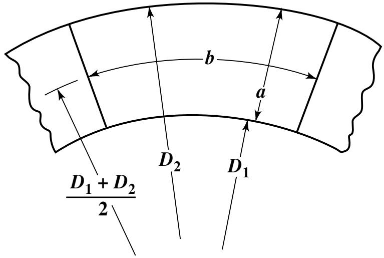


Fig. 20. Basic elements of flat plate thrust bearing.*



Basic elements of flat plate thrust bearing.*

1) *Inside diameter,* D_1. Determined by shaft size and clearance.

2) *Outside diameter,* D_2. Calculated by the formula

$$D_2 = \left(\frac{4W}{\pi K_g p} + D_1^2\right)^{1/2}$$

where W = applied load, pounds
K_g = fraction of circumference occupied by pads; usually, 0.8
p = bearing unit load, psi

3) *Radial pad width,* a. Equal to one-half the difference between the inside and outside diameters.

$$a = \frac{D_2 - D_1}{2}$$

4) *Pitch line circumference,* B. Found from the pitch diameter.

$$B = \pi(D_2 - a)$$

Number of pads, i. Assume an oil groove width, s. If the length of pad is assumed to be optimum, i.e., equal to its width,

$$i_{app} = \frac{B}{a + s}$$

Take i as nearest even number.

5) *Length of pad,* b. If number of pads and oil groove width are known,

$$b = \frac{B - (i \times s)}{i}$$

6) *Actual unit load,* p. Calculated in pounds per square inch based on pad dimensions.

$$p = \frac{W}{iab}$$

7) *Pitch line velocity,* U. Found in feet per minute from

$$U = \frac{BN}{12}$$

where N = rpm

8) *Friction power loss,* P_f. Friction power loss is difficult to calculate for this type of bearing because there is no theoretical method of determining the operating film thickness. However, a good approximation can be made using Fig. 21. From this curve, the value of M, horsepower loss per square inch of bearing surface, can be obtained. The total power loss, P_f, is then calculated from

$$P_f = iabM$$

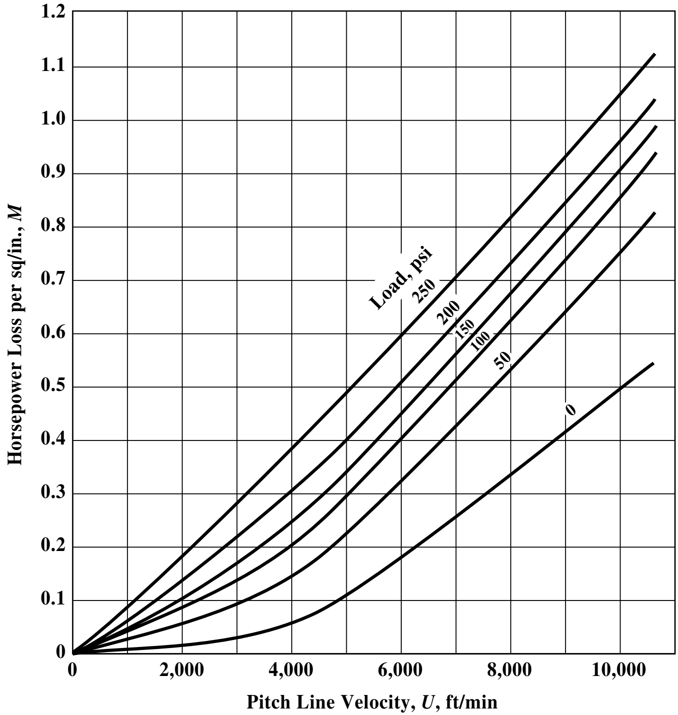


Fig. 21. Friction power loss, M, vs. peripheral speed, U — thrust bearings.*

9) *Oil flow required, Q*. May be estimated in gallons per minute for a given temperature rise from

$$Q = \frac{42.4 P_f}{c \Delta t}$$

where c = specific heat of oil in Btu/gal/°F

Δt = temperature rise of the oil in °F

Note: A Δt of 50°F is an acceptable maximum.

Because there is no theoretical method of predicting the minimum film thickness in this type of bearing, only an approximation, based on experience, of the film flow can be made. For this reason and based on practical experience, it is desirable to have a minimum of one-half of the desired oil flow pass through the chamfer.

10) *Film flow, Q_F*. Calculated in gallons per minute from

$$Q_F = \frac{(1.5)(10^5) i V h^3 p_s}{Z_2}$$

where V = effective width-to-length ratio for one pad, a/b

* See footnote on page 2219.

Z_2 = oil viscosity at outlet temperature
h = film thickness

Note: Because h cannot be calculated, use h = 0.002 inch.

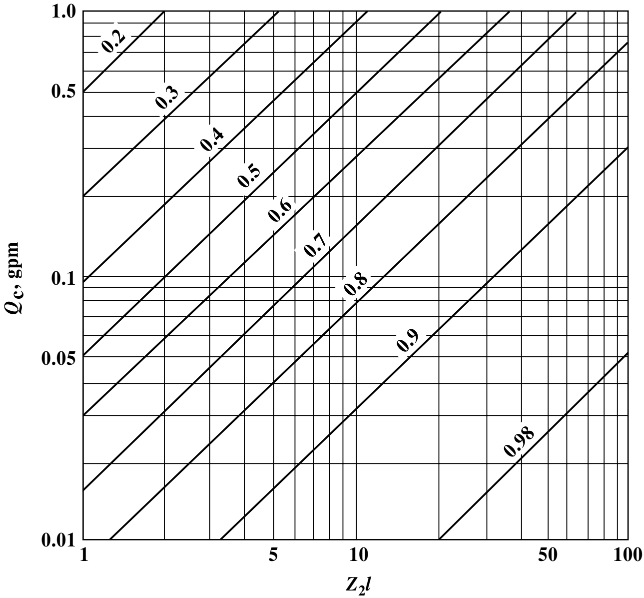


Fig. 22. Kinetic energy correction factor, ξ—thrust bearings.*

11) *Required flow per chamfer, Q_c*. Readily found from the formula

$$Q_c = \frac{Q}{i}$$

12) *Kinetic energy correction factor, ξ*. Found by assuming a chamfer length l and entering Fig. 22 with a value $Z_2 l$ and Q_c.

13) *Uncorrected required flow per chamfer, $Q^0{}_c$*. Found from the formula

$$Q_c^0 = \frac{Q_c}{\xi}$$

14) *Depth of chamfer, g*. Found from the formula

$$g = \sqrt[4]{\frac{Q_c^0 l Z_2}{4.74 \times 10^4 p_s}}$$

Example:—Design a flat plate thrust bearing to carry 900 pounds load at 4000 rpm using an SAE 10 oil with a specific heat of 3.5 Btu/gal/°F at 120°F and 30-psi inlet conditions. The shaft is 2¾ inches in diameter and the temperature rise is not to exceed 40°F. Fig. 23 shows the final design of this bearing.

* See footnote on page 2219.

1) *Inside diameter.* Assumed to be 3 inches to clear shaft.

2) *Outside diameter.* Assuming a unit bearing load of 75 pounds per square inch from Table 7,

$$D_2 = \sqrt{\frac{4 \times 900}{\pi \times 0.8 \times 75} + 3^2} = 5.30 \text{ inches}$$

Use 5½ inches.

3) *Radial pad width.*

$$a = \frac{5.5 - 3}{2} = 1.25 \text{ inches}$$

4) *Pitch-line circumference.*

$$B = \pi \times 4.25 = 13.3 \text{ inches}$$

5) *Number of pads.* Assume an oil groove width of $\frac{3}{16}$ inch. If length of pad is assumed to be equal to width of pad, then

$$i_{\text{app}} = \frac{13.3}{1.25 + 0.1875} = 9+$$

If the number of pads, i, is taken as 10, then

6) *Length of pad.* $b = \dfrac{13.3 - (10 \times 0.1875)}{10} = 1.14 \text{ inches}$

7) *Actual unit load.*

$$p = \frac{900}{10 \times 1.25 \times 1.14} = 63 \text{ psi}$$

8) *Pitch-line velocity.*

$$U = \frac{13.3 \times 4000}{12} = 4{,}430 \text{ ft per min.}$$

9) *Friction power loss.* From Fig. 21, $M = 0.19$

$$P_f = 10 \times 1.25 \times 1.14 \times 0.19 = 2.7 \text{ horsepower}$$

10) *Oil flow required.*

$$Q = \frac{42.4 \times 2.7}{3.5 \times 40} = 0.82 \text{ gallon per minute}$$

(Assuming a temperature rise of 40°F—the maximum allowable according to the given condition—then the assumed operating temperature will be 120°F + 40°F = 160°F and the oil viscosity Z_2 is found from Fig. 5 to be 9.6 centipoises.)

11) *Film flow.*

$$Q_F = \frac{1.5 \times 10^5 \times 10 \times 1 \times 0.002^3 \times 30}{9.6} = 0.038 \text{ gpm}$$

Because 0.038 gpm is a very small part of the required flow of 0.82 gpm, the bulk of the flow must be carried through the chamfers.

12) *Required flow per chamfer.* Assume that all the oil flow is to be carried through the chamfers.

$$Q_c = \frac{0.82}{10} = 0.082 \text{ gpm}$$

13) *Kinetic energy correction factor.* If l, the length of chamfer is made ⅛ inch, then $Z_2 l = 9.6 \times \frac{1}{8} = 1.2$. Entering Fig. 22 with this value and $Q_c = 0.082$,

$$\xi = 0.44$$

14) *Uncorrected required oil flow per chamfer.*

$$Q_c^0 = \frac{0.082}{0.44} = 0.186 \text{ gpm}$$

15) *Depth of chamfer.*

$$g = \sqrt[4]{\frac{0.186 \times 0.125 \times 9.6}{4.74 \times 10^4 \times 30}}$$

$$g = 0.02 \text{ inch}$$

A schematic drawing of this bearing is shown in Fig. 23.

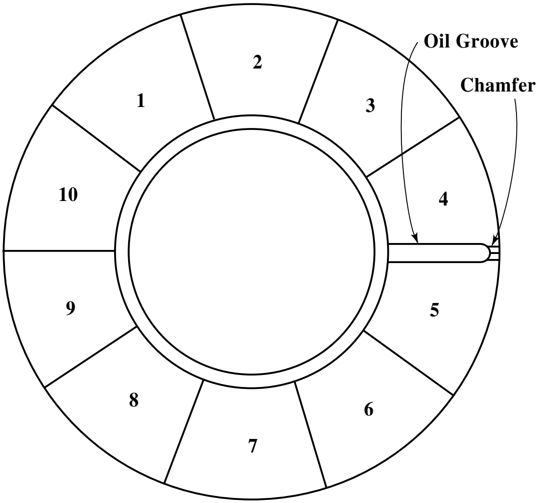


Fig. 23. Flat plate thrust bearing example design.*

Step Thrust Bearing Design.—The following steps define the performance of a step thrust bearing, one section of which is shown in Fig. 24.

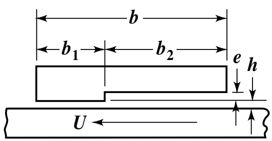


Fig. 24. Basic elements of step thrust bearing.*

Although each bearing section is wedge shaped, as shown at the right in Fig. 24, for the purposes of design calculation it is considered to be a rectangle with a length b equal to the circumferential length along the pitch line of the section being considered, and a width a equal to the difference in the external and internal radii.

General Parameters: For optimum proportions, $a = b$, $b_2 = 1.2b_1$, and $e = 0.7h$.

1) *Internal diameter, D_1.* An internal diameter is assumed that is sufficient to clear the shaft.

* See footnote on page 2219.

2) *External diameter, D_2*. A unit bearing pressure is assumed from Table 7 and the external diameter is then found from the formula

$$D_2 = \sqrt{\frac{4W}{\pi K_g p} + D_1^2}$$

3) *Radial pad width, a*. Equal to the difference between the external and internal radii.

$$a = \frac{D_2 - D_1}{2}$$

4) *Pitch-line circumference, B*. Found from the formula

$$B = \frac{\pi(D_1 + D_2)}{2}$$

5) *Number of pads, i*. Assume an oil groove width, *s* (0.062 inch may be taken as a minimum), and find the approximate number of pads, assuming the pad length is equal to *a*. Note that if a chamfer is found necessary to increase the oil flow (see Step 13), the oil groove width should be greater than the chamfer width.

$$i_{app} = \frac{B}{a + s}$$

Then *i* is taken as the nearest even number.

6) *Length of pad, b*. Readily determined from the number of pads and groove width.

$$b = \frac{B}{i} - s$$

7) *Pitch-line velocity, U*. Found in feet per minute from the formula

$$U = \frac{BN}{12}$$

8) *Film thickness, h*. Found in inches from the formula

$$h = \sqrt{\frac{2.09 \times 10^{-9} i a^3 U Z}{W}}$$

9) *Depth of step, e*. According to the general parameter

$$e = 0.7h$$

10) *Friction power loss, P_f*. Found from the formula

$$P_f = \frac{7.35 \times 10^{-13} i a^2 U^2 Z}{h}$$

11) *Pad step length, b_2*. This distance, on the pitch line, from the leading edge of the pad to the step in inches is determined by the general parameters

$$b_2 = \frac{1.2b}{2.2}$$

12) *Hydrodynamic oil flow, Q*. Found in gallons per minute from the formula

$$Q = 6.65 \times 10^{-4} iahU$$

13) *Temperature rise, Δt*. Found in degrees F from the formula

$$\Delta t = \frac{42.4 P_f}{cQ}$$

If the flow is insufficient, as indicated by too high a temperature rise, chamfers can be added to provide adequate flow as in Steps 12–15 of the flat plate thrust bearing design.

Example: Design a step thrust bearing for positioning a ⅞-inch diameter shaft operating with a 25-pound thrust load and a speed of 5,000 rpm. The lubricating oil has a viscosity of

25 centipoises at the operating temperature of 160 deg. F and has a specific heat of 3.4 Btu per gal. per deg. F.

1) *Internal diameter.* Assumed to be 1 inch to clear the shaft.

2) *External diameter.* Because the example is a positioning bearing with low total load, unit load will be negligible and the external diameter is not established by using the formula given in Step 2 of the procedure, but a convenient size is taken to give the desired overall bearing proportions.

$$D_2 = 3 \text{ inches}$$

3) *Radial pad width.*

$$a = \frac{3-1}{2} = 1 \text{ inch}$$

4) *Pitch-line circumference.*

$$B = \frac{\pi(3+1)}{2} = 6.28 \text{ inches}$$

5) *Number of pads.* Assuming a minimum groove width of 0.062 inch,

$$i_{\text{app}} = \frac{6.28}{1+0.062} = 5.9$$

Take $i = 6$.

6) *Length of pad.*

$$b = \frac{6.28}{6} - 0.062 = 0.985$$

7) *Pitch-line velocity.*

$$U = \frac{6.28 \times 5{,}000}{12} = 2{,}620 \text{ fpm}$$

8) *Film thickness.*

$$h = \sqrt{\frac{2.09\times10^{-9} \times 6 \times 1^3 \times 2{,}620 \times 25}{25}} = 0.0057 \text{ inch}$$

9) *Depth of step.*

$$e = 0.7 \times 0.0057 = 0.004 \text{ inch}$$

10) *Power loss.*

$$P_f = \frac{7.35\times10^{-13} \times 6 \times 1^2 \times 2{,}620^2 \times 25}{0.0057} = 0.133 \text{ hp}$$

11) *Pad step length.*

$$b_2 = \frac{1.2 \times 0.985}{2.2} = 0.537 \text{ inch}$$

12) *Total hydrodynamic oil flow.*

$$Q = 6.65\times10^{-4} \times 6 \times 1 \times 0.0057 \times 2{,}620 = 0.060 \text{ gpm}$$

13) *Temperature rise.*

$$\Delta t = \frac{42.4 \times 0.133}{3.4 \times 0.060} = 28° \text{ F}$$

Tapered Land Thrust Bearing Design.—The following steps define the performance of a tapered land thrust bearing, one section of which is shown in Fig. 25. Although each bearing section is wedge shaped, as shown in Fig. 25, right, for the purposes of design calculation, it is considered to be a rectangle with a length b equal to the circumferential length along the pitch line of the section being considered and a width a equal to the difference in the external and internal radii.

General Parameters: Usually, the taper extends to only 80 per cent of the pad length with the remainder being flat, thus: $b_2 = 0.8b$ and $b_1 = 0.2b$.

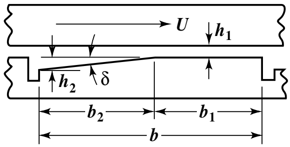


Fig. 25. Basic elements of tapered land thrust bearing.*

1) *Inside diameter, D_1.* Determined by shaft size and clearance.

2) *Outside diameter, D_2.* Calculated by the formula

$$D_2 = \left(\frac{4W}{\pi K_g P_a} + D_1^2\right)^{1/2}$$

where K_g = 0.8 or 0.9 and W = applied load, pounds
P_a = assumed unit load from Table 7, page 2219

3) *Radial pad width, a.* Equal to one-half the difference between the inside and outside diameters.

$$a = \frac{D_2 - D_1}{2}$$

4) *Pitch-line circumference, B.* Found from the mean diameter:

$$B = \frac{\pi(D_1 + D_2)}{2}$$

5) *Number of pads, i.* Assume an oil groove width, s, and find the approximate number of pads, assuming the pad length is equal to a.

$$i_{\text{app}} = \frac{B}{a + s}$$

Then i is taken as the nearest even number.

6) *Length of pad, b.* Readily determined because the number of pads and groove width are known.

$$b = \frac{B - is}{i}$$

7) *Taper values, δ_1 and δ_2.* Can be taken from Table 8.

8) *Actual bearing unit load, p.* Calculated in pounds per square inch from the formula

$$p = \frac{W}{iab}$$

9) *Pitch-line velocity, U.* Found in feet per minute at the pitch circle from the formula

$$U = \frac{BN}{12}$$

where N = rpm

10) *Oil leakage factor, Y_L.* Found either from Fig. 26 which shows curves for Y_L as functions of the pad width a and length of land b or from the formula

$$Y_L = \frac{b}{1 + (\pi^2 b^2 / 12a^2)}$$

11) *Film thickness factor, K.* Calculated using the formula

* See footnote on page 2219.

$$K = \frac{5.75 \times 10^6 p}{U Y_L Z}$$

12) *Minimum film thickness, h*. Using the value of K just determined and the selected taper values δ_1 and δ_2, h is found from Fig. 27. In general, h should be 0.001 inch for small bearings and 0.002 inch for larger and high-speed bearings.

13) *Friction power loss, P_f*. Using the film thickness h, the coefficient J can be obtained from Fig. 28. The friction loss in horsepower is then calculated from the formula

$$P_f = 8.79 \times 10^{-13} iabJU^2Z$$

14) *Required oil flow, Q*. May be estimated in gallons per minute for a given temperature rise Δ_t from the formula

$$Q = \frac{42.4 P_f}{c \Delta t}$$

where c = specific heat of the oil in Btu/gal/°F

Note: A Δt of 50°F is an acceptable maximum.

15) *Shape factor, Y_s*. Needed to compute the actual oil flow and calculated from

$$Y_S = \frac{8ab}{D^2{}_2 - D_1^2}$$

16) *Oil flow factor, Y_G*. Found from Fig. 29 using Y_s and D_1/D_2.

17) *Actual oil film flow, Q_F*. The amount of oil in gallons per minute that the bearing film will pass is calculated from the formula

$$Q_F = \frac{8.9 \times 10^{-4} i \delta_2 D_2^3 N Y_G Y_S^2}{D_2 - D_1}$$

18) If the flow is insufficient, the tapers can be increased or chamfers calculated to provide adequate flow, as in Steps 12–15 of the flat plate thrust bearing design procedure.

Example: Design a tapered land thrust bearing for 70,000 pounds at 3600 rpm. The shaft diameter is 6.5 inches. The oil inlet temperature is 110°F at 20 psi.

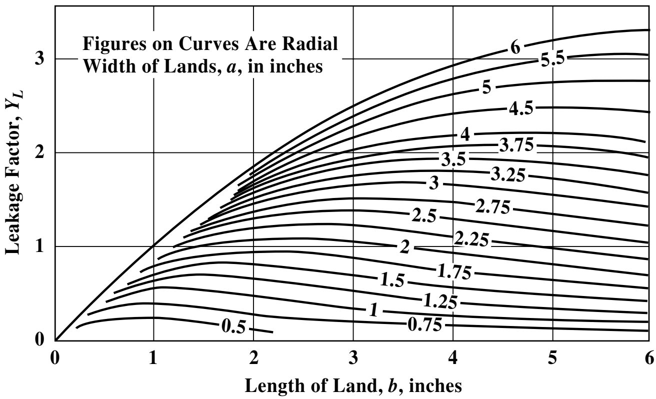


Fig. 26. Leakage factor, Y_L, vs. pad dimensions a and b—tapered land thrust bearings.*

* See footnote on page 2219.

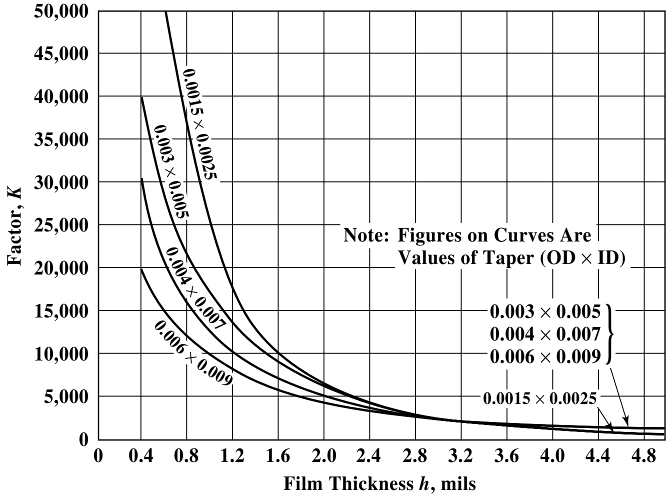


Fig. 27. Thickness, *h*, vs. factor *K*—tapered land thrust bearings.*

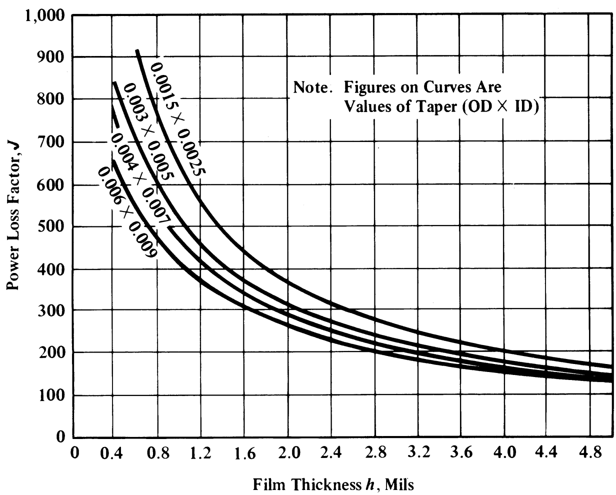


Fig. 28. Power-loss coefficient. *J*, vs. film thickness, *h*—tapered land thrust bearings.*

A maximum temperature rise of 50°F is acceptable and results in a viscosity of 18 centipoises. Use values of $K_g = 0.9$ and $c = 3.5$ Btu/gal/°F.

* See footnote on page 2219.

1) *Internal diameter.* Assume $D_1 = 7$ inches to clear shaft.

2) *External diameter.* Assume a unit bearing load p_a of 400 pounds per square inch from Table 7, then

$$D_2 = \sqrt{\frac{4 \times 70{,}000}{3.14 \times 0.9 \times 400} + 7^2} = 17.2 \text{ inches}$$

Round off to 17 inches.

3) *Radial pad width.*

$$a = \frac{17 - 7}{2} = 5 \text{ inches}$$

4) *Pitch-line circumference.*

$$B = \frac{3.14(17 + 7)}{2} = 37.7 \text{ inches}$$

5) *Number of pads.* Assume groove width of 0.5 inch, then

$$i_{\text{app}} = \frac{37.7}{5 + 0.5} = 6.85$$

Take $i = 6$.

6) *Length of pad.*

$$b = \frac{37.7 - 6 \times 0.5}{6} = 5.78 \text{ inches}$$

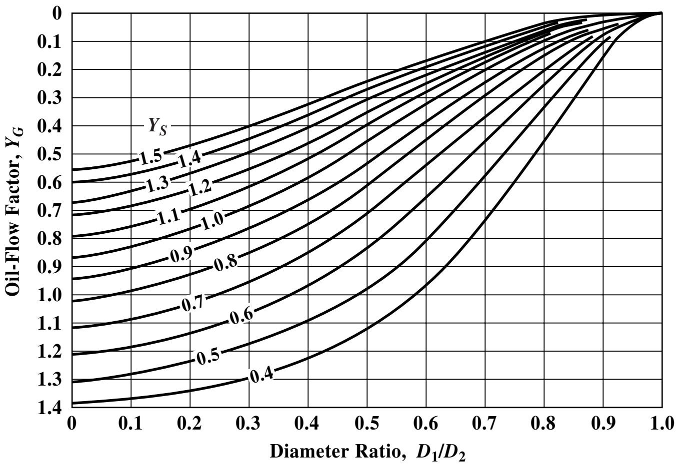


Fig. 29. Oil-flow factor, Y_G, vs. diameter ratio D_1/D_2—tapered land bearings.*

7) *Taper values.* Interpolate in Table 8 to obtain

$$\delta_1 = 0.008 \text{ inch} \qquad \text{and} \qquad \delta_2 = 0.005 \text{ inch}$$

8) *Actual bearing unit load.*

$$p = \frac{70{,}000}{6 \times 5 \times 5.78} = 404 \text{ psi}$$

* See footnote on page 2219.

9) *Pitch-line velocity.*

$$U = \frac{37.7 \times 3600}{12} = 11{,}300 \text{ ft per min}$$

10) *Oil leakage factor.*

$$\text{From Fig. 26, } Y_L = 2.75$$

11) *Film-thickness factor.*

$$K = \frac{5.75 \times 10^6 \times 404}{11,300 \times 2.75 \times 18} = 4150$$

12) *Minimum film thickness.*

$$\text{From Fig. 27, } h = 2.2 \text{ mils}$$

13) *Friction power loss.* From Fig. 28, $J = 260$, then

$$P_f = 8.79 \times 10^{-13} \times 6 \times 5 \times 5.78 \times 260 \times 11{,}300^2 \times 18 = 91 \text{ hp}$$

14) *Required oil flow.*

$$Q = \frac{42.4 \times 91}{3.5 \times 50} = 22.0 \text{ gpm}$$

See footnote on page 2219.

15) *Shape factor.*

$$Y_S = \frac{8 \times 5 \times 5.78}{17^2 - 7^2} = 0.963$$

16) *Oil-flow factor.*

$$\text{From Fig. 29, } Y_G = 0.61$$

where $D_1/D_2 = 0.41$

17) *Actual oil film flow.*

$$Q_F = \frac{8.9 \times 10^{-4} \times 6 \times 0.005 \times 17^3 \times 3600 \times 0.61 \times 0.963^2}{17 - 7} = 26.7 \text{gpm}$$

Because calculated film flow exceeds required oil flow, chamfers are not necessary. However, if film flow were less than required, suitable chamfers would be needed.

Table 8. Taper Values for Tapered Land Thrust Bearings

Pad Dimensions, Inches	Taper, Inch	
$a \times b$	$\delta_1 = h_2 - h_1$ (at ID)	$\delta_2 = h_2 - h_1$ (at OD)
½ × ½	0.0025	0.0015
1 × 1	0.005	0.003
3 × 3	0.007	0.004
7 × 7	0.009	0.006

Tilting Pad Thrust Bearing Design.—The following steps define the performance of a tilting pad thrust bearing, one section of which is shown in Fig. 30. Although each bearing section is wedge shaped, as shown at the right below, for the purposes of design calculation, it is considered to be a rectangle with a length *b* equal to the circumferential length along the pitch line of the section being considered and a width *a* equal to the difference in the external and internal radii, as shown at left in Fig. 30. The location of the pivot shown in Fig. 30 is optimum. If shaft rotation in both directions is required, however, the pivot must be at the midpoint, which results in little or no detrimental effect on the performance.

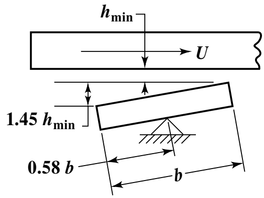


Fig. 30. Basic elements of tilting pad thrust bearing.*

1) *Inside diameter, D_1*. Determined by shaft size and clearance.

2) *Outside diameter, D_2*. Calculated from the formula

$$D_2 = \left(\frac{4W}{\pi K_g p} + D_1^2\right)^{1/2}$$

where W = applied load, pounds

$K_g = 0.8$

p = unit load from Table 7

3) *Radial pad width, a*. Equal to one-half the difference between the inside and outside diameters:

$$a = \frac{D_2 - D_1}{2}$$

4) *Pitch-line circumference, B*. Found from the mean diameter:

$$B = \pi\left(\frac{D_1 + D_2}{2}\right)$$

5) *Number of pads, i*. The number of pads may be estimated from the formula

$$i = \frac{BK_g}{a}$$

Select the nearest even number.

6) *Length of pad, b*. Found from the formula

$$b \cong \frac{BK_g}{i}$$

7) *Pitch-line velocity, U*. Calculated in feet per minute from the formula

$$U = \frac{BN}{12}$$

8) *Bearing unit load, p*. Calculated from the formula

$$p = \frac{W}{iab}$$

9) *Operating number, O*. Calculated from the formula

$$O = \frac{1.45 \times 10^{-7} Z_2 U}{5pb}$$

10) where Z_2 = viscosity of oil at outlet temperature (inlet temperature plus assumed temperature rise through the bearing).

* See footnote on page 2219.

11) *Minimum film thickness,* h_{min}. By using the operating number, the value of α = dimensionless film thickness is found from Fig. 31. Then the actual minimum film thickness is calculated from the formula:

$$h_{min} = \alpha b$$

In general, this value should be 0.001 inch for small bearings and 0.002 inch for larger and high-speed bearings.

12) *Coefficient of friction, f.* Found from Fig. 32.

13) *Friction power loss,* P_f. This horsepower loss now is calculated by the formula

$$P_f = \frac{fWU}{33,000}$$

14) *Actual oil flow, Q.* This flow over the pad in gallons per minute is calculated from the formula

$$Q = 0.0591\alpha iabU$$

15) *Temperature rise,* Δt. Found from the formula

$$\Delta t = 0.0217\frac{fp}{\alpha c}$$

16) where c = specific heat of oil in Btu/gal/°F

If the flow is insufficient, as indicated by too high a temperature rise, chamfers can be added to provide adequate flow, as in Steps 12–15 of the flat plate thrust bearing design.

Example: Design a tilting pad thrust bearing for 70,000 pounds thrust at 3600 rpm. The shaft diameter is 6.5 inches and a maximum OD of 15 inches is available. The oil inlet temperature is 110°F and the supply pressure is 20 pounds per square inch. A maximum temperature rise of 50°F is acceptable and results in a viscosity of 18 centipoises. Use a value of 3.5 Btu/gal/°F for c.

1) *Inside diameter.* Assume D_1, = 7 inches to clear shaft.

2) *Outside diameter.* Given maximum D_2 = 15 inches.

3) *Radial pad width.*

$$a = \frac{15-7}{2} = 4 \text{ inches}$$

4) *Pitch-line circumference.*

$$B = \pi\left(\frac{7+15}{2}\right) = 34.6 \text{ inches}$$

5) *Number of pads.*

$$i = \frac{34.6 \times 0.8}{4} = 6.9$$

Select 6 pads: $i = 6$.

6) *Length of pad.*

$$b = \frac{34.6 \times 0.8}{6} = 4.61 \text{ inches}$$

Make b = 4.75 inches.

7) *Pitch-line velocity.*

$$U = \frac{34.6 \times 3600}{12} = 10,400 \text{ ft/min}$$

8) *Bearing unit load.*

$$p = \frac{70,000}{6 \times 4 \times 4.75} = 614 \text{ psi}$$

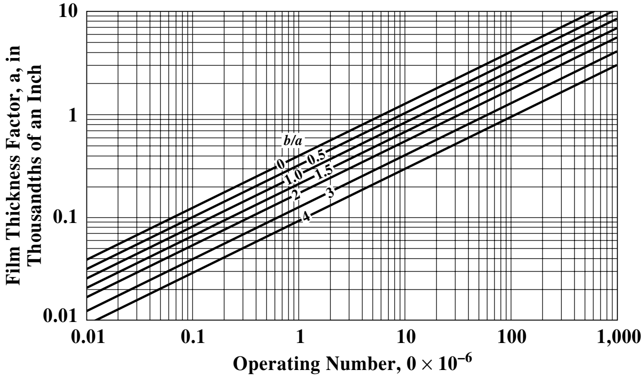


Fig. 31. Dimensionless minimum film thickness, α, vs. operating number, *O*—tilting pad thrust bearings.*

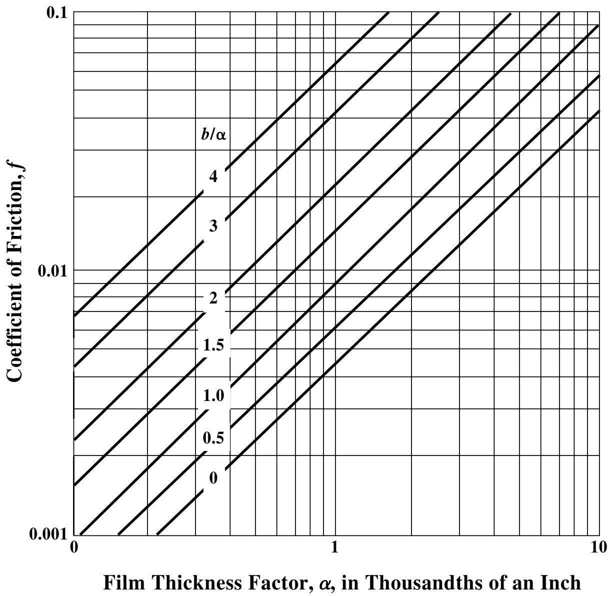


Fig. 32. Coefficient of friction *f* vs. dimensionless film thickness α for tilting pad thrust bearings with optimum pivot location.*

* See footnote on page 2219.

9) *Operating number.*

$$O = \frac{1.45\times10^{-7} \times 18 \times 10,400}{5 \times 614 \times 4.75} = 1.86\times10^{-6}$$

10) *Minimum film thickness.* From Fig. 31, $\alpha = 0.30 \times 10^{-3}$.

$$h_{\min} = 0.00030 \times 4.75 = 0.0014 \text{ inch}$$

11) *Coefficient of friction.* From Fig. 32, $f = 0.0036$.

12) *Friction power loss.*

$$P_f = \frac{0.0036 \times 70,000 \times 10,400}{33,000} = 79.4 \text{ hp}$$

13) *Oil flow.*

$$Q = 0.0591 \times 6 \times 0.30 \times 10^{-3} \times 4 \times 4.75 \times 10,400 = 21.02 \text{ gpm}$$

14) *Temperature rise.*

$$\Delta t = \frac{0.0217 \times 0.0036 \times 614}{0.30 \times 10^{-3} \times 3.5} = 45.7^{\circ}\text{F}$$

Because this temperature is less than the 50°F, which is considered as the acceptable maximum, the design is satisfactory.

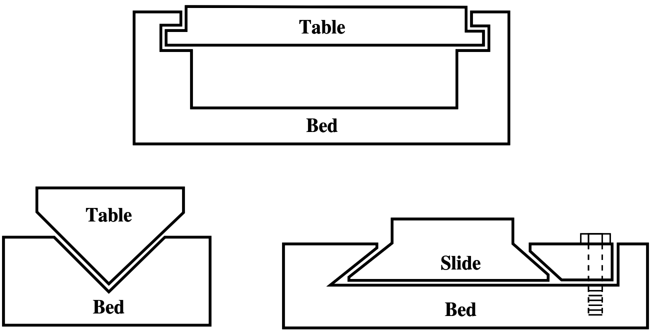


Fig. 33. Types of Guide Bearings

Guide Bearings

This type of bearing is generally used as a positioning device or as a guide to linear motion such as in machine tools. Fig. 33 shows several examples of guideway bearing designs. It is normal for this type of bearing to operate in the boundary lubrication region with either dry, dry film such as molybdenum disulfide (MoS_2) or tetrafluorethylene (TFE), grease, oil, or gaseous lubrication. Hydrostatic lubrication is often used to improve performance, reduce wear, and increase stability. This type of design uses pumps to supply air or gas under pressure to pockets designed to produce a bearing film and maintain complete separation of the sliding surfaces.

PLAIN BEARING MATERIALS

Materials used for sliding bearings cover a wide range of metals and nonmetals. To make the optimum selection requires a complete analysis of the specific application. The important general categories are: Babbitts, alkali-hardened lead, cadmium alloys, copper lead, aluminum bronze, silver, sintered metals, plastics, wood, rubber, and carbon graphite.

Properties of Bearing Materials.—For a material to be used as a plain bearing, it must possess certain physical and chemical properties that permit it to operate properly. If a material does not possess all of these characteristics to some degree, it will not function long as a bearing. It should be noted, however, that few, if any, materials are outstanding in all these characteristics. Therefore, the selection of the optimum bearing material for a given application is at best a compromise to secure the most desirable combination of properties required for that particular usage.

The seven properties generally acknowledged to be the most significant are: 1) Fatigue resistance; 2) Embeddability; 3) Compatibility; 4) Conformability; 5) Thermal conductivity; 6) Corrosion resistance; and 7) Load capacity.

These properties are described as follows:

1) *Fatigue resistance* is the ability of the bearing lining material to withstand repeated applications of stress and strain without cracking, flaking, or being destroyed by some other means.

2) *Embeddability* is the ability of the bearing lining material to absorb or embed within itself any of the larger of the small dirt particles present in a lubrication system. Poor embeddability permits particles circulating around the bearing to score both the bearing surface and the journal or shaft. Good embeddability will permit these particles to be trapped and forced into the bearing surface and out of the way where they can do no harm.

3) *Compatibility or antiscoring tendencies* permit the shaft and bearing to "get along" with each other. It is the ability to resist galling or seizing under conditions of metal-to-metal contact such as at startup. This characteristic is most truly a bearing property, because contact between the bearing and shaft in good designs occurs only at startup.

4) *Conformability* is defined as malleability or as the ability of the bearing material to creep or flow slightly under load, as in the initial stages of running, to permit the shaft and bearing contours to conform with each other or to compensate for nonuniform loading caused by misalignment.

5) *High thermal conductivity* is required to absorb and carry away the heat generated in the bearing. This conductivity is most important, not in removing frictional heat generated in the oil film, but in preventing seizures due to hot spots caused by local asperity breakthroughs or foreign particles.

6) *Corrosion resistance* is required to resist attack by organic acids that are sometimes formed in oils at operating conditions.

7) *Load capacity or strength* is the ability of the material to withstand the hydrodynamic pressures exerted upon it during operation.

Babbitt or White Metal Alloys.—Many different bearing metal compositions are referred to as babbitt metals. The exact composition of the original babbitt metal is not known; however, the ingredients were probably tin, copper, and antimony in approximately the following percentages: 89.3, 3.6, and 7.1. Tin and lead-base babbitts are probably the best known of all bearing materials. With their excellent embeddability and compatibility characteristics under boundary lubrication, babbitt bearings are used in a wide range of applications including household appliances, automobile and diesel engines, railroad cars, electric motors, generators, steam and gas turbines, and industrial and marine gear units.

Table 9. Bearing and Bushing Alloys—Composition, Forms, Characteristics, and Applications *SAE General Information*

SAE No.and Alloy Grouping		Nominal Composition, Per cent	Form of Use (1), Characteristics (2), and Applications (3)
Sn-Base Alloys	11	{Sn, 87.5; Sb, 6.75; Cu,5.75	(1) Cast on steel, bronze, or brass backs, or directly in the bearing housing. (2) Soft, corrosion-resistant with moderate fatigue resistance. (3) Main and connecting-rod bearings; motor bushings. Operates with either hard or soft journal.
	12	{ Sn, 89; Sb,7.5;Cu, 3.5	
Pb-Base Alloys	13	{ Pb, 84; Sb, 10; Sn, 6	(1) SAE 13 and 14 are cast on steel, bronze, or brass, or in the bearing housing; SAE 15 is cast on steel; and SAE 16 is cast into and on a porous sintered matrix, usually copper-nickel bonded to steel. (2) Soft, moderately fatigue-resistant, corrosion-resistant. (3) Main and connecting-rod bearings. Operates with hard or soft journal with good finish.
	14	{ Pb, 75; Sb, 15; Sn, 10	
	15	{Pb, 83; Sb,15 ;Sn,14 ; As,1	
	16	{ Pb, 92; Sb, 3.5;Sn, 4.5	
Pb-Sn Overlays	19	{ Pb, 90; Sn, 10	(1) Electrodeposited as a thin layer on copper-lead or silver bearings faces. (2) Soft, corrosion-resistant. Bearings so coated run satisfactorily against soft shafts throughout the life of the coating. (3) Heavy-duty, high-speed main and connecting-rod bearings.
	190	{ Pb, 93; Sn, 7	
Cu-Pb Alloys	49	{ Cu, 76; Pb, 24	(1) Cast or sintered on steel back with the exception of SAE 481, which is cast on steel back only. (2) Moderately hard. Somewhat subject to oil corrosion. Some oils minimize this; protection with overlay may be desirable. Fatigue resistance good to fairly good. Listed in order of decreasing hardness and fatigue resistance. (3) Main and connecting-rod bearings. The higher lead alloys can be used unplated against a soft shaft, although an overlay is helpful. The lower lead alloys may be used against a hard shaft, or with an overlay against a soft one.
	48	{ Cu, 70; Pb, 30	
	480	{ Cu, 65; Pb, 35	
	481	{ Cu, 60; Pb, 40	
Cu-Pb-Sn-Alloys	482	{ Cu, 67; Pb, 28; Sn, 5	(1) Steel-backed and lined with a structure combining sintered copper alloy matrix with corrosion-resistant lead alloy. (2) Moderately hard. Corrosion resistance improved over copper-leads of equal lead content without tin. Fatigue resistance fairly good. Listed in order of decreasing hardness and fatigue resistance. (3) Main and connecting-rod bearings. Generally used without overlay. SAE 484 and 485 may be used with hard or soft shaft, and a hardened or cast shaft is recommended for SAE 482.
	484	{ Cu, 55; Pb, 42; Sn, 3	
	485	{ Cu, 46; Pb, 51; Sn, 3	
Al-Base Alloys	770	{ Al, 91.75; Sn, 6.25; Cu, 1; Ni, 1	(1) SAE 770 cast in permanent molds; work-hardened to improve physical properties. SAE 780 and 782 usually bonded to steel back but is procurable in strip form without steel backing. SAE 781 usually bonded to steel back but can be produced as castings or wrought strip without steel back. (2) Hard, extremely fatigue-resistant, resistant to oil corrosion. (3) Main and connecting-rod bearings. Generally used with suitable overlay. SAE 781 and 782 also used for bushings and thrust bearings with or without overlay.
	780	{ Al, 91; Sn, 6; Si, 1.5; Cu, 1; Ni, 0.5	
	781	{ Al, 95; Si, 4; Cd, 1	
	782	{ Al, 95; Cu, 1;Ni, 1; Cd, 3	
Other Cu-Base Alloys	795	{ Cu, 90; Zn, 9.5; Sn, 0.5	(1) Wrought solid bronze, (2) Hard, strong, good fatigue resistance, (3) Intermediate-load oscillating motion such as tie-rods and brake shafts.
	791	{Cu, 88; Zn,4;Sn,4; Pb, 4	(1) SAE 791, wrought solid bronze; SAE 793, cast on steel back; SAE 798, sintered on steel back. (2) General-purpose bearing material, good shock and load capacity. Resistant to high temperatures. Hard shaft desirable. Less score-resistant than higher lead alloys. (3) Medium to high loads. Transmission bushings and thrust washers. SAE 791 also used for piston pin and 793 and 798 for chassis bushings.
	793	{ Cu, 84; Pb, 8; Sn, 4; Zn, 4	
	798	{Cu, 84; Pb, 8; Sn,4; Zn, 4	
Other Cu-Base Alloys	792	{ Cu, 80; Sn, 10; Pb, 10	(1) SAE 792, cast on steel back, SAE 797, sintered on steel back. (2) Has maximum shock and load-carrying capacity of conventional cast bearing alloys; hard, both fatigue- and corrosion-resistant. Hard shaft desirable. (3) Heavy loads with oscillating or rotating motion. Used for piston pins, steering knuckles, differential axles, thrust washers, and wear plates.
	797	{ Cu, 80; Sn, 10; Pb, 10	
	794	{ Cu, 73.5; Pb, 23; Sn, 3.5	(1) SAE 794, cast on steel back; SAE 799, sintered on steel back. (2) Higher lead content gives improved surface action for higher speeds but results in somewhat less corrosion resistance. (3) Intermediate load application for both oscillating and rotating shafts, that is, rocker-arm bushings, transmissions, and farm implements.
	799	{ Cu, 73.5,; Pb, 23; Sn, 3.5	

Table 10. White Metal Bearing Alloys—Composition and Properties
ASTM B23-83, reapproved 1988

ASTM Alloy Number	Nominal Composition, Per Cent				Compressive Yield Point,[a] psi		Ultimate Compressive Strength,[b] psi		Brinell Hardness[c]		Melting Point °F	Proper Pouring Temperature, °F
	Sn	Sb	Pb	Cu	68 °F	212 °F	68 °F	212 °F	68 °F	212 °F		
1	91.0	4.5	…	4.5	4400	2650	12,850	6950	17.0	8.0	433	825
2	89.0	7.5	…	3.5	6100	3000	14,900	8700	24.5	12.0	466	795
3	83.33	8.33	…	8.33	6600	3150	17,600	9900	27.0	14.5	464	915
4	75.0	12.0	10.0	3.0	5550	2150	16,150	6900	24.5	12.0	363	710
5	65.0	15.0	18.0	2.0	5050	2150	15,050	6750	22.5	10.0	358	690
6	20.0	15.0	63.5	1.5	3800	2050	14,550	8050	21.0	10.5	358	655
7[d]	10.0	15.0	bal.	…	3550	1600	15,650	6150	22.5	10.5	464	640
8[d]	5.0	15.0	bal.	…	3400	1750	15,600	6150	20.0	9.5	459	645
10	2.0	15.0	83.0	…	3350	1850	15,450	5750	17.5	9.0	468	630
11	…	15.0	85.0	…	3050	1400	12,800	5100	15.0	7.0	471	630
12	…	10.0	90.0	…	2800	1250	12,900	5100	14.5	6.5	473	625
15[e]	1.0	16.0	bal.	0.5	…	…	…	…	21.0	13.0	479	662
16	10.0	12.5	77.0	0.5	…	…	…	…	27.5	13.6	471	620
19	5.0	9.0	86.0	…	…	…	15,600	6100	17.7	8.0	462	620

[a] The values for yield point were taken from stress-strain curves at the deformation of 0.125 per cent reduction of gage.

[b] The ultimate strength values were taken as the unit load necessary to produce a deformation of 25 per cent of the length of the specimen.

[c] These values are the average Brinell number of three impressions on each alloy using a 10-mm ball and a 500-kg load applied for 30 seconds.

[d] Also nominal arsenic, 0.45 per cent.

[e] Also nominal arsenic, 1 per cent.

Data for ASTM alloys 1, 2, 3, 7, 8, and 15 appear in the Appendix of ASTM B23-83; the data for alloys 4, 5, 6, 10, 11, 12, 16, and 19 are given in ASTM B23-49. All values are for reference purposes only.

The compression test specimens were cylinders 1.5 inches in length and 0.5 inch in diameter, machined from chill castings 2 inches in length and 0.75 inch in diameter. The Brinell tests were made on the bottom face of parallel machined specimens cast in a 2-inch diameter by 0.625-inch deep steel mold at room temperature.

Both the Society of Automotive Engineers and American Society for Testing and Materials have classified white metal bearing alloys. Tables 9 and 10 give compositions and properties or characteristics for the two classifications.

In small bushings for fractional-horsepower motors and in automotive engine bearings, the babbitt is generally used as a thin coating over a flat steel strip. After forming oil distribution grooves and drilling required holes, the strip is cut to size, then rolled and shaped into the finished bearing. These bearings are available for shaft diameters from 0.5 to 5 inches. Strip bearings are turned out by the millions yearly in highly automated factories and offer an excellent combination of low cost with good bearing properties.

For larger bearings in heavy-duty equipment, a thicker babbitt is cast on a rigid backing of steel or cast iron. Chemical and electrolytic cleaning of the bearing shell, thorough rinsing, tinning, and then centrifugal casting of the babbitt are desirable for sound bonding of the babbitt to the bearing shell. After machining, the babbitt layer is usually $\frac{1}{2}$ to $\frac{1}{4}$ inch thick.

Compared to other bearing materials, babbitts generally have lower load-carrying capacity and fatigue strength, are a little higher in cost, and require a more complicated design. Also, their strength decreases rapidly with increasing temperature. These shortcomings can be avoided by using an intermediate layer of high-strength, fatigue-resistant material that is placed between a steel backing and a thin babbitt surface layer. Such composite bearings frequently eliminate any need for using alternate materials having poorer bearing characteristics.

Tin babbitt is composed of 80 to 90 per cent tin to which is added about 3 to 8 per cent copper and 4 to 14 per cent antimony. An increase in copper or antimony produces increased hardness and tensile strength and decreased ductility. However, if the percentages of these alloys are increased above those shown in Table 10, the resulting alloy will have decreased fatigue resistance. These alloys have very little tendency to cause wear to their journals because of their ability to embed dirt. They resist the corrosive effects of acids, are not prone to oil-film failure, and are easily bonded and cast. Two drawbacks are encountered from use of these alloys because they have low fatigue resistance and their hardness and strength drop appreciably at low temperatures.

Lead babbitt compositions generally range from 10 to 15 per cent antimony and up to 10 per cent tin in combination with the lead. Like tin-base babbitts, these alloys have little tendency to cause wear to their journals, embed dirt well, resist the corrosive effects of acids, are not prone to oil-film failure and are easily bonded and cast. Their chief disadvantages when compared with tin-base alloys are a rather lower strength and a susceptibility to corrosion.

Cadmium Base.—Cadmium alloy bearings have a greater resistance to fatigue than babbitt bearings, but their use is very limited due to their poor corrosion resistance. These alloys contain 1 to 15 per cent nickel, or 0.4 to 0.75 per cent copper, and 0.5 to 2.0 per cent silver. Their prime attribute is their high-temperature capability. The load-carrying capacity and relative basic bearing properties are shown in Tables 11 and 12.

Table 11. Properties of Bearing Alloys

Material	Recommended Shaft Hardness, Brinell	Load-Carrying Capacity, psi	Maximum Operating Temp., °F
Tin-Base Babbitt	150 or less	800–1500	300
Lead-Base Babbitt	150 or less	800–1200	300
Cadmium Base	200–250	1200–2000	500
Copper-Lead	300	1500–2500	350
Tin-Bronze	300–400	4000+	500+
Lead-Bronze	300	3000–4500	450–500
Aluminum	300	4000+	225–300
Silver-Overplate	300	4000+	500
Trimetal-Overplate	230 or less	2000–4000+	225–300

Table 12. Bearing Characteristics Ratings

Material	Compatibility	Conformability and Embeddability	Corrosion Resistance	Fatigue Strength
Tin-Base Babbitt	1	1	1	5
Lead-Base Babbitt	1	1	3	5
Cadmium Base	1	2	5	4
Copper-Lead	2	2	5	3
Tin-Bronze	3	5	2	1
Lead-Bronze	3	4	4	2
Aluminum	5	3	1	2
Silver Overplate	2	3	1	1
Trimetal-Overplate	1	2	2	3

Note: 1 is best; 5 is worst.

Copper-Lead.—Copper-lead bearings are a binary mixture of copper and lead containing from 20 to 40 per cent lead. Lead is practically insoluble in copper, so a cast microstructure consists of lead pockets in a copper matrix. A steel backing is commonly used with this material and high volume is achieved either by continuous casting or by powder metallurgy techniques. This material is very often used with an overplate such as lead-tin and lead-tin-copper to increase basic bearing properties. Tables 11 and 12 provide comparisons of material properties.

The combination of good fatigue strength, high-load capacity, and high-temperature performance has resulted in extensive use of this material for heavy-duty main and connecting-rod bearings as well as moderate-load and speed applications in turbines and electric motors.

Leaded Bronze and Tin-Bronze.—Leaded and tin-bronzes contain up to 25 per cent lead or approximately 10 per cent tin, respectively. Cast leaded bronze bearings offer good compatibility, excellent casting, and easy machining characteristics, low cost, good structural properties and high-load capacity, usefulness as a single material that requires neither a separate overlay nor a steel backing. Bronzes are available in standard bar stock, sand or permanent molds, investment, centrifugal or continuous casting. Leaded bronzes have better compatibility than tin-bronzes because the spheroids of lead smear over the bearing surface under conditions of inadequate lubrication. These alloys are generally a first choice at intermediate loads and speeds. Tables 11 and 12 provide comparisons of basic bearing properties of these materials.

Aluminum.—Aluminum bearings are either cast solid aluminum, aluminum with a steel backing, or aluminum with a suitable overlay. The aluminum is usually alloyed with small amounts of tin, silicon, cadmium, nickel, or copper, as shown in Table 9. An aluminum bearing alloy with 20 to 30 per cent tin alloy and up to 3 per cent copper has shown promise as a substitute for bronzes in some industrial applications.

These bearings are best suited for operation with hard journals. Owing to the high thermal expansion of the metal (resulting in diametral contraction when it is confined as a bearing in a rigid housing), large clearances are required, which tend to make the bearing noisy, especially on starting. Overlays of lead-tin, lead, or lead-tin-copper may be applied to aluminum bearings to facilitate their use with soft shafts.

Aluminum alloys are available with properties specifically designed for bearing applications, such as high load-carrying capacity, fatigue strength, and thermal conductivity, in addition to excellent corrosion resistance and low cost.

Silver.—Silver bearings were developed for and have an excellent record in heavy-duty applications such as aircraft master rod and diesel engine main bearings. Silver has a higher fatigue rating than any of the other bearing materials; the steel backing used with this material may show evidence of fatigue before the silver. The advent of overlays, or more commonly called overplates, made it possible for silver to be used as a bearing material. Silver by itself does not possess any of the desirable bearing qualities except high fatigue resistance and high thermal conductivity. The overlays such as lead, lead-tin, or lead-indium improve the embeddability and antiscoring properties of silver. The relative basic properties of this material, when used as an overplate, are shown in Tables 11 and 12.

Cast Iron.—Cast iron is an inexpensive bearing material capable of operation at light loads and low speeds, i.e., up to 130 ft/min and 150 lb/in.^{2} These bearings must be well lubricated and have a rather large clearance so as to avoid scoring from particles torn from the cast iron that ride between bearing and journal. A journal hardness of between 150 and 250 Brinell has been found to be best when using cast-iron bearings.

Porous Metals.—Porous metal self-lubricating bearings are usually made by sintering metals such as plain or leaded bronze, iron, and stainless steel. The sintering produces a spongelike structure capable of absorbing fairly large quantities of oil, usually 10–35 per cent of the total volume. These bearings are used where lubrication supply is difficult, inadequate, or infrequent. This type of bearing should be flooded from time to time to resaturate the material. Another use of these porous materials is to meter a small quantity of oil to the bearings such as in drip feed systems. The general design operating characteristics of this class of materials are shown in Table 13.

Table 13. Application Limits — Sintered Metal and Nonmetallic Bearings

Bearing Material	Load Capacity (psi)	Maximum Temperature (°F)	Surface Speed, V_{max} (max. fpm)	PV Limit P = psi load V = surface ft/min
Acetal	1000	180	1000	3000
Graphite (dry)	600	750	2500	15,000
Graphite (lubricated)	600	750	2500	150,000
Nylon, Polycarbonate	1000	200	1000	3000
Nylon composite	…	400	…	16,000
Phenolics	6000	200	2500	15,000
Porous bronze	4500	160	1500	50,000
Porous iron	8000	160	800	50,000
Porous metals	4000–8000	150	1500	50,000
Virgin Teflon (TFE)	500	500	50	1000
Reinforced Teflon	2500	500	1000	10,000–15,000
TFE fabric	60,000	500	150	25,000
Rubber	50	150	4000	15,000
Maple & Lignum Vitae	2000	150	2000	15,000

Table 14 gives the chemical compositions, permissible loads, interference fits, and running clearances of bronze-base and iron-base metal-powder sintered bearings that are specified in the ASTM specifications for oil-impregnated metal-powder sintered bearings (B438-83a and B439-83).

Plastics Bearings.—Plastics are finding increased use as bearing materials because of their resistance to corrosion, quiet operation, ability to be molded into many configurations, and their excellent compatibility, which minimizes or eliminates the need for lubrication. Many plastics are capable of operating as bearings, especially phenolic, tetrafluoroethylene (TFE), and polyamide (nylon) resins. The general application limits for these materials are shown in Table 13.

Laminated Phenolics: These composite materials consist of cotton fabric, asbestos, or other fillers bonded with phenolic resin. They have excellent compatibility with various fluids as well as strength and shock resistance. However, precautions must be taken to maintain adequate bearing cooling because the thermal conductivity of these materials is low.

Nylon: This material has the widest use for small, lightly loaded applications. It has low frictional properties and requires no lubrication.

Teflon: This material, with its exceptional low coefficient of friction, self-lubricating characteristics, resistance to attack by almost any chemicals, and its wide temperature range, is one of the most interesting of the plastics for bearing use. High cost combined with low load capacity cause Teflon to be selected mostly in modified form, where other less expensive materials have proved inadequate for design requirements.

Bearings made of laminated phenolics, nylon, or Teflon are all unaffected by acids and alkalies except if highly concentrated and therefore can be used with lubricants containing dilute acids or alkalies. Water is used to lubricate most phenolic laminate bearings but oil, grease, and emulsions of grease and water are also used. Water and oil are used as lubricants for nylon and Teflon bearings. Almost all types of plastic bearings absorb water and oil to some extent. In some the dimensional change caused by the absorption may be as much as three per cent in one direction. This means that bearings have to be treated before use so that proper clearances will be kept. This may be done by boiling in water, for water lubricated bearings. Boiling in water makes bearings swell the maximum amount. Clearances for phenolic bearings are kept at about 0.001 inch per inch of diameter on treated bearings. Partially lubricated or dry nylon bearings are given a clearance of 0.004 to 0.006 inches for a one-inch diameter bearing.

Table 14. Copper- and Iron-Base Sintered Bearings (Oil Impregnated) — *ASTM B438-83a (R1989) , B439-83 (R1989), and Appendices*

Chemical Requirements								
Alloying Elements[a]	Percentage Composition							
	Copper-Base Bearings				Iron-Base Bearings			
	Grade 1		Grade 2		Grades			
	Class A	Class B	Class A	Class B	1	2	3	4
Cu	87.5–99.5	87.5–90.5	87.5–90.5	87.5–90.5	…	…	7.0–11.0	18.0–22.0
Sn	9.5–10.5	9.5–10.5	9.5–10.5	9.5–10.5	…	…	…	…
Graphite	0.1 max.	1.75 max.	0.1 max.	1.75 max.	…	…	…	…
Pb	…	…	2.0–4.0	2.0–4.0	…	…	…	…
Fe	1.0 max.	1.0 max.	1.0 max.	1.0 max.	96.25 min.	95.9 min.	Balance[b]	Balance[b]
Comb. C[c]	…	…	…	…	0.25 max.	0.25–0.60	…	…
Si, max.	…	…	…	…	0.3	0.3	…	…
Al, max.	…	…	…	…	0.2	0.2	…	…
Others	0.5 max.	0.5 max.	1.0 max.	1.0 max.	3.0 max.	3.0 max.	3.0 max.	3.0 max.

[a] Abbreviations used for the alloying elements are as follows: Cu, copper; Fe, iron; Sn, tin; Pb, lead; Zn, zinc; Ni, nickel; Sb, antimony; Si, silicon; Al, aluminum; and C, carbon.

[b] Total of iron plus copper shall be 97 per cent, minimum.

[c] Combined carbon (on basis of iron only) may be a metallographic estimate of the carbon in the iron.

Permissible Loads						
Copper-Base Bearings				Iron-Base Bearings		
	Grades 1 & 2					
	Type 1	Type 2	Types 3 & 4		Grades 1 & 2	Grades 3 & 4
Shaft Velocity, fpm	Max. Load, psi			Shaft Velocity, fpm	Max. Load, psi	
Slow and intermittent	3200	4000	4000	Slow and intermittent	3600	8000
25	2000	2000	2000	25	1800	3000
50 to 100	500	500	550	50 to 100	450	700
Over 100 to 150	365	325	365	Over 100 to 150	300	400
Over 150 to 200	280	250	280	Over 150 to 200	225	300
Over 200		[a]	[a]	[a]	Over 200	[a]

[a] For shaft velocities over 200 fpm, the permissible loads may be calculated as follows: $P = 50{,}000/V$; where P = safe load, psi of projected area; and V = shaft velocity, fpm. With a shaft velocity of less than 50 fpm and a permissible load greater than 1,000 psi, an extreme pressure lubricant should be used; with heat dissipation and removal techniques, higher *PV* ratings can be obtained.

Clearances						
Press-Fit Clearances			Running Clearances[a]			
Copper- and Iron-Base			Copper-Base		Iron-Base	
Bearing OD	Min.	Max.	Shaft Size	Min. Clearance	Shaft Size	Min. Clearance
Up to 0.760	0.001	0.003	Up to 0.250	0.0003	Up to 0.760	0.0005
0.761–1.510	0.0015	0.004	0.250–0.760	0.0005	0.761–1.510	0.001
1.511–2.510	0.002	0.005	0.760–1.510	0.0010	1.511–2.510	0.0015
2.511–3.010	0.002	0.006	1.510–2.510	0.0015	Over 2.510	0.002
Over 3.010	0.002	0.007	Over 2.510	0.0020		

[a] Only minimum recommended clearances are listed. It is assumed that ground steel shafting will be used and that all bearings will be oil-impregnated.

Table 15. Copper- and Iron-Base Sintered Bearings (Oil Impregnated) — *ASTM B438-83a (R1989) , B439-83 (R1989), and Appendices*

Commercial Dimensional Tolerances[a,b]							
Diameter Tolerance		Length Tolerance		Diameter Tolerance		Length Tolerance	
Copper Base				Iron Base			
Inside or Outside Diameter	Total Diameter Tolerances	Length	Total Length Tolerances	Inside or Outside Diameter	Total Diameter Tolerances	Length	Total Length Tolerances
Up to 1	0.001	Up to 1.5	0.01	Up to 0.760	−0.001	Up to 1.495	0.01
1 to 1.5	0.0015	1.5 to 3	0.01	0.761–1.510	−0.0015	1.496–1.990	0.02
1.5 to 2	0.002	3 to 4.5	0.02	1.511–2.510	−0.002	1.991–2.990	0.02
2 to 2.5	0.0025	…	…	2.511–3.010	−0.003	2.991–4.985	0.03
2.5 to 3	0.003	…	…	3.011–4.010	−0.005	…	…
…	…	…	…	4.011–5.010	−0.005	…	…
…	…	…	…	5.011–6.010	−0.006	…	…

[a] For copper-base bearings with 4-to-1 maximum-length-diameter ratio and a 24-to-1 maximum-length-to-wall-thickness ratio; bearings with greater ratios are not covered here.

[b] For iron-base bearings with a 3-to-1 maximum-length-to-inside diameter ration and a 20-to-1 maximum-length-to-wall-thickness ratio; bearings with greater ratios are not covered here.

Concentricity Tolerance[a,b,a]		
Iron Base		
Outside Diameter	Max. Wall Thickness	Concentricity Tolerance
Up to 1.510	Up to 0.355	0.003
1.511 to 2.010	Up to 0.505	0.004
2.011 to 4.010	Up to 1.010	0.005
4.011 to 5.010	Up to 1.510	0.006
5.011 to 6.010	Up to 2.010	0.007
Copper Base		
Outside Diameter	Length	Concentricity Tolerance
Up to 1	0 to 1	0.00.
	1 to 2	0.004
	2 to 3	0.005
1 to 2	0 to 1	0.004
	1 to 2	0.005
	2 to 3	0.006
2 to 3	0 to 1	0.005
	1 to 2	0.006
	2 to 3	0.007

[a] Total indicator reading.

Table 16. Copper- and Iron-Base Sintered Bearings (Oil Impregnated) — *ASTM B438-83a (R1989), B439-83 (R1989), and Appendices*

Flange and Thrust Bearings, Diameter, and Thickness Tolerances[a]				
Diameter Range	Flange Diameter Tolerance		Flange Thickness Tolerance	
	Standard	Special	Standard	Special
0 to 1½	±0.005	±0.0025	±0.005	±0.0025
Over 1½ to 3	±0.010	±0.005	±0.010	±0.007
Over 3 to 6	±0.025	±0.010	±0.015	±0.010
Diameter Range	Parallellism on Faces, max.			
	Copper Base		Iron Base	
	Standard	Special	Standard	Special
0 to 1½	0.003	0.002	0.005	0.003
Over 1½ to 3	0.004	0.003	0.007	0.005
Over 3 to 6	0.005	0.004	0.010	0.007

[a] Standard and special tolerances are specified for diameters, thicknesses, and parallelism. Special tolerances should not be specified unless required because they require additional or secondary operations and, therefore, are costlier. Thrust bearings (¼ inch thickness, max.) have a standard thickness tolerance of ±0.005 inch and a special thickness tolerance of ±0.0025 inch for all diameters.

All dimensions in inches except where otherwise noted.

Wood: Bearings made from such woods as lignum vitae, rock maple, or oak offer self-lubricating properties, low cost, and clean operation. However, they have frequently been displaced in recent years by various plastics, rubber and sintered-metal bearings. General applications are shown in Table 10.

Rubber: Rubber bearings give excellent performance on propeller shafts and rudders of ships, hydraulic turbines, pumps, sand and gravel washers, dredges and other industrial equipment that handle water or slurries. The resilience of rubber helps to isolate vibration and provide quiet operation, allows running with relatively large clearances and helps to compensate for misalignment. In these beatings a fluted rubber structure is supported by a metal shell. The flutes or scallops in the rubber form a series of grooves through which lubricant or, as generally used, water and foreign material such as sand may pass through the beating.

Carbon-Graphite.—Bearings of molded and machined carbon-graphite are used where regular maintenance and lubrication cannot be given. They are dimensionally stable over a wide range of temperatures, may be lubricated if desired, and are not affected by chemicals. These bearings may be used up to temperatures of 700 to 750 degrees F. in air or 1200 degrees F. in a non-oxidizing atmosphere, and generally are operated at a maximum load of 20 pounds per square inch. In some instances a metal or metal alloy is added to the carbon-graphite composition to improve such properties as compressive strength and density. The temperature limitation depends upon the melting point of the metal or alloy and the maximum load is generally 350 pounds per square inch when used with no lubrication or 600 pounds per square inch when used with lubrication.

Normal running clearances for both types of carbon-graphite bearings used with steel shafts and operating at a temperature of less than 200 degrees F. are as follows: 0.001 inch for bearings of 0.187 to 0.500-inch inside diameter, 0.002 inch for bearings of 0.501 to 1.000-inch inside diameter, 0.003 inch for bearings of 1.001 to 1.250-inch inside diameter, 0.004 inch for bearings of 1.251 to 1.500-inch diameter, and 0.005 inch for bearings of 1.501 to 2.000-inch inside diameter. Speeds depend upon too many variables to list specifically so it can only be stated here that high loads require a low number of rpm and low loads permit a high number of rpm. Smooth journals are necessary in these bearings as rough ones tend to abrade the bearings quickly. Cast iron and hard chromium-plate steel shafts of 400 Brinell and over, and phosphor-bronze shafts over 135 Brinell are recommended.

BALL, ROLLER, AND NEEDLE BEARINGS

Rolling contact bearings substitute a rolling element, ball or roller, for a hydrodynamic or hydrostatic fluid film to carry an impressed load without wear and with reduced friction. Because of their greatly reduced starting friction, when compared to the conventional journal bearing, they have acquired the common designation of "anti-friction" bearings. Although normally made with hardened rolling elements and races, and usually utilizing a separator to space the rolling elements and reduce friction, many variations are in use throughout the mechanical and electrical industries. The most common anti-friction bearing application is that of the deep-groove ball bearing with ribbon-type separator and sealed-grease lubrication used to support a shaft with radial and thrust loads in rotating equipment. This shielded or sealed bearing has become a standard and commonplace item ordered from a supplier's catalogue in much the same manner as nuts and bolts. Because of the simple design approach and the elimination of a separate lubrication system or device, this bearing is found in as many installations as the wick-fed or impregnated porous plain bushing.

Currently, a number of manufacturers produce a complete range of ball and roller bearings in a fully interchangeable series with standard dimensions, tolerances and fits as specified in Anti-Friction Bearing Manufacturers Association (AFBMA) Standards. Except for deep-groove ball bearings, performance standards are not so well defined and sizing and selection must be done in close conformance with the specific manufacturer's catalogue requirements. In general, desired functional features should be carefully gone over with the vendor's representatives.

Rolling contact bearings are made to high standards of accuracy and with close metallurgical control. Balls and rollers are normally held to diametral tolerances of .0001 inch or less within one bearing and are often used as "gage" blocks in routine toolroom operations. This accuracy is essential to the performance and durability of rolling-contact bearings and in limiting runout, providing proper radial and axial clearances, and ensuring smoothness of operation.

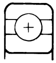
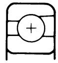
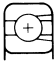
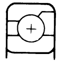
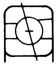
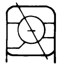
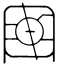
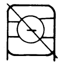
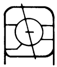
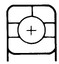
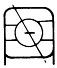
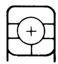
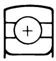
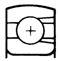
Because of their low friction, both starting and running, rolling-contact bearings are utilized to reduce the complexity of many systems that normally function with journal bearings. Aside from this advantage and that of precise radial and axial location of rotating elements, however, they also are desirable because of their reduced lubrication requirements and their ability to function during brief interruptions in normal lubrication.

In applying rolling-contact bearings it is well to appreciate that their life is limited by the fatigue life of the material from which they are made and is modified by the lubricant used. In rolling-contact fatigue, precise relationships among life, load, and design characteristics are not predictable, but a statistical function described as the "probability of survival" is used to relate them according to equations recommended by the AFBMA. Deviations from these formulas result when certain extremes in applications such as speed, deflection, temperature, lubrication, and internal geometry must be dealt with.

Types of Anti-friction Bearings.—The general types are usually determined by the shape of the rolling element, but many variations have been developed that apply conventional elements in unique ways. Thus it is well to know that special bearings can be procured with races adapted to specific applications, although this is not practical for other than high volume configurations or where the requirements cannot be met in a more economical manner. "Special" races are appreciably more expensive. Quite often, in such situations, races are made to incorporate other functions of the mechanism, or are "submerged" in the surrounding structure, with the rolling elements supported by a shaft or housing that has been hardened and finished in a suitable manner. Typical anti-friction bearing types are shown in Table 1.

Types of Ball Bearings.—Most types of ball bearings originate from three basic designs: the single-row radial, the single-row angular contact, and the double-row angular contact.

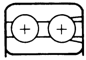
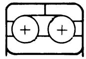
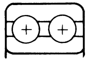
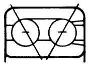
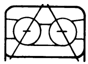
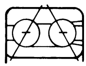
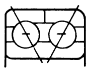
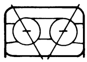
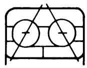
Table 1. Types of Rolling Element Bearings and Their Symbols

BALL BEARINGS, SINGLE ROW, RADIAL CONTACT			
Symbol	Description	Symbol	Description
BC	Non-filling slot assembly 	BH	Non-separable counter-bore assembly 
BL	Filling slot assembly 	BM	Separable assembly 
BALL BEARINGS, SINGLE ROW, ANGULAR CONTACT[a]			
Symbol	Description	Symbol	Description
BN	Non-separable Nominal contact angle: from above 10° to and including 22° 	BAS	Separable inner ring Nominal contact angle: from above 22° to and including 32° 
BNS	Separable outer ring Nominal contact angle: from above 10° to and including 22° 	BT	Non-separable Nominal contact angle: from above 32° to and including 45° 
BNT	Separable inner ring Nominal contact angle: from above 10° to and including 22° 	BY	Two-piece outer ring 
BA	Non-separable Nominal contact angle: from above 22° to and including 32° 	BZ	Two-piece inner ring 
BALL BEARINGS, SINGLE ROW, RADIAL CONTACT, SPHERICAL OUTSIDE SURFACE			
Symbol	Description	Symbol	Description
BCA	Non-filling slot assembly 	BLA	Filling slot assembly 

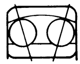
[a] A line through the ball contact points forms an acute angle with a perpendicular to the bearing axis.

Single-row Radial, Non-filling Slot: This is probably the most widely used ball bearing and is employed in many modified forms. It is also known as the "Conrad" type or "Deep-groove" type. It is a symmetrical unit capable of taking combined radial and thrust loads in which the thrust component is relatively high, but is not intended for pure thrust loads, however. Because this type is not self-aligning, accurate alignment between shaft and housing bore is required.

Single-row Radial, Filling Slot: This type is designed primarily to carry radial loads. Bearings of this type are assembled with as many balls as can be introduced by eccentric displacement of the rings, as in the non-filling slot type, and then several more balls are inserted through the loading slot, aided by a slight spreading of the rings and heat expansion of the outer ring, if necessary. This type of bearing will take a certain degree of thrust when in combination with a radial load but is not recommended where thrust loads exceed 60 per cent of the radial load.

BALL BEARINGS, DOUBLE ROW, RADIAL CONTACT			
Symbol	Description	Symbol	Description
BF	Filling slot assembly 	BHA	Non-separable two-piece outer ring 
BK	Non-filling slot assembly 		
BALL BEARINGS, DOUBLE ROW, ANGULAR CONTACT[a]			
Symbol	Description	Symbol	Description
BD	Filling slot assembly Vertex of contact angles inside bearing 	BG	Non-filling slot assembly Vertex of contact angles outside bearing 
BE	Filling slot assembly Vertex of contact angles outside bearing 	BAA	Non-separable Vertex of contact angles inside bearing Two-piece outer ring 
BJ	Non-filling slot assembly Vertex of contact angles inside bearing 	BVV	Separable Vertex of contact angles outside bearing Two-piece inner ring 

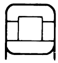
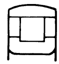
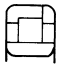
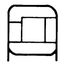
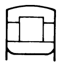
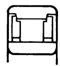
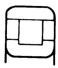
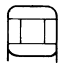
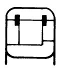
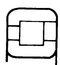
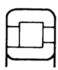
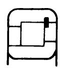
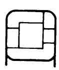
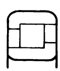
[a] A line through the ball contact points forms an acute angle with a perpendicular to the bearing axis.

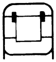
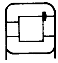
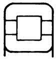
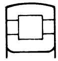
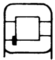
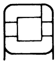
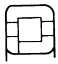
BALL BEARINGS, DOUBLE ROW, SELF-ALIGNING[a]			
	Symbol	Description	
	BS	Raceway of outer ring spherical	

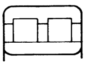
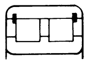
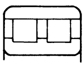
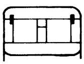
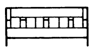
Single-row Angular-contact: This type is designed for combined radial and thrust loads where the thrust component may be large and axial deflection must be confined within very close limits. A high shoulder on one side of the outer ring is provided to take the thrust, while the shoulder on the other side is only high enough to make the bearing non-separable. Except where used for a pure thrust load in one direction, this type is applied either in pairs (duplex) or one at each end of the shaft, opposed.

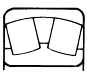
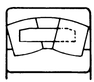
Double-row Bearings: These are, in effect, two single-row angular-contact bearings built as a unit with the internal fit between balls and raceway fixed at the time of bearing assembly. This fit is therefore not dependent upon mounting methods for internal rigidity. These bearings usually have a known amount of internal preload built in for maximum resistance to deflection under combined loads with thrust from either direction. Thus, with balls and races under compression before an external load is applied, due to this internal preload, the bearings are very effective for radial loads where bearing deflection must be minimized.

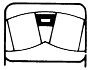
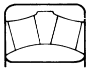
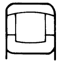
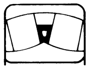
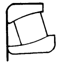
Other Types: Modifications of these basic types provide arrangements for self-sealing, location by snap ring, shielding, etc., but the fundamentals of mounting are not changed. A special type is the *self-aligning* ball bearing which can be used to compensate for an appreciable degree of misalignment between shaft and housing due to shaft deflections, mounting inaccuracies, or other causes commonly encountered. With a single row of balls, alignment is provided by a spherical outer surface on the outer ring; with a double row of balls, alignment is provided by a spherical raceway on the outer ring. Bearings in the wide series have a considerable amount of thrust capacity.

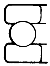
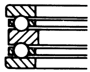
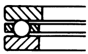
CYLINDRICAL ROLLER BEARING, SINGLE ROW, NON-LOCATING TYPE			
Symbol	Description	Symbol	Description
RU	Inner ring without ribs Double-ribbed outer-ring Inner ring separable 	RNS	Double-ribbed inner ring Outer ring without ribs Outer ring separable Spherical outside surface 
RUP	Inner ring without ribs Double-ribbed outer ring with oneloose rib Both rings separable 	RAB	Inner ring without ribs Single-ribbedouter ring Both rings separable 
RUA	Inner ring without ribs Double-ribbed outer ring Inner ring separable Spherical outside surface 	RM	Inner ring without ribs Rollers located by cage, end-rings or internal snap rings recesses in outer ring Inner ring separable 
RN	Double-ribbed inner ring Outer ring without ribs Outer ring separable 	RNU	Inner ring without ribs Outer ring without ribs Both rings separable 
CYLINDRICAL ROLLER BEARINGS, SINGLE ROW, ONE-DIRECTION-LOCATING TYPE			
Symbol	Description	Symbol	Description
RR	Single-ribbed inner-ring Outer ring with two internal snap rings Inner ring separable 	RF	Double-ribbed inner ring Single-ribbed outer ring Outer ring separable 
RJ	Single-ribbedinner ring Double-ribbed outer ring Inner ring separable 	RS	Single-ribbed inner ring Outer ring with one rib and one internal snap ring Inner ring separable 
RJP	Single-ribbed inner ring Double-ribbed outer ring with one loose rib Both rings separable 	RAA	Single-ribbed inner ring Single-ribbed outer ring Both rings separable 

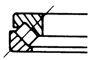
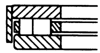
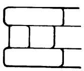
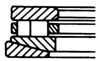
CYLINDRICAL ROLLER BEARINGS, SINGLE ROW, TWO-DIRECTION-LOCATING TYPE			
Symbol	Description	Symbol	Description
RK	Double-ribbed inner ring Outer ring with two internal snap rings Non-separable 	RY	Double-ribbed inner ring Outer ring with one rib and one internal snap ring Non-separable 
RC	Doble-ribbed inner ring Double-ribbed outer ring Non-separable 	RCS	Double-ribbed inner ring Double-ribbed outer ring Non-separable Spherical outside surface 
RG	Inner ring, with one rib and one snap ring Double-ribbed outer ring Non-separable 		
RP	Double-ribbed inner ring Double-ribbed outer ring with one loose rib Outer ring separable 	RT	Double-ribbed inner ring with one loose rib Double-ribbed outer ring Inner ring separable 

CYLINDRICAL ROLLER BEARINGS			
Double Row Non-Locating Type		Double Row Two-Direction-Locating Type	
Symbol	Description	Symbol	Description
RA	Inner ring without ribs Three integral ribs on outer ring Inner ring separable 	RB	Three integral ribs on inner ring Outer ring without ribs, with two internal snap rings Non-separable 
RD	Three integral ribs on inner ring Outer ring without ribs Outer ring separable 	Multi-Row Non-Locating Type	
		Symbol	Description
RE	Inner ring without ribs Outer rings without ribs, with two internal snap rings Inner ring separable 	RV	Inner ring without ribs Double-ribbed outer ring (loose ribs) Both rings separable 

SELF-ALIGNING ROLLER BEARINGS, DOUBLE ROW			
Symbol	Description	Symbol	Description
SD	Three integral ribs on inner ring Raceway of outer ring spherical 	SL	Raceway of outer ring spherical Rollers guided by the cage Two integral ribs on inner ring 

SELF-ALIGNING ROLLER BEARINGS, DOUBLE ROW			
Symbol	Description	Symbol	Description
SE	Raceway of outer ring spherical Rollers guided by separate center guide ring in outer ring 	SELF-ALIGNING ROLLER BEARINGS SINGLE ROW	
		Symbol	Description
SW	Raceway of inner ring spherical 	SR	Inner ring with ribs Raceway of outer ring spherical Radial contact 
SC	Raceway of outer ring spherical Rollers guided by separate axially floating guide ring on inner ring 	SA	Raceway of outer ring spherical Angular contact 
		SB	Raceway of inner ring spherical Angular contact 

THRUST BALL BEARINGS			
Symbol	Description	Symbol	Description
TA TB[a]	Single direction, grooved raceways, flat seats 	TDA	Double direction, washers with grooved raceways, flat seats 
TBF[a]	Single direction, flat washers, flat seats 		

THRUST ROLLER BEARINGS			
Symbol	Description	Symbol	Description
TS	Single direction, aligning flat seats, spherical rollers 	TPC[a]	Single direction, flat seats, flat races, outside band, cylindrical rollers 
TP	Single direction, flat seats, cylindrical rollers 	TR[a]	Single direction, flat races, aligning seat with aligning washer, cylindrical rollers 

[a] Inch dimensioned only.

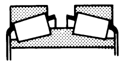
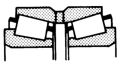
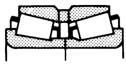
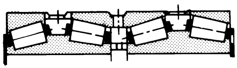
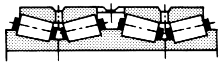
Types of Roller Bearings.—Types of roller bearings are distinguished by the design of rollers and raceways to handle axial, combined axial and thrust, or thrust loads.

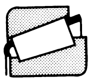
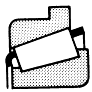
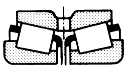
Cylindrical Roller: These bearings have solid or helically wound hollow cylindrical rollers. The free ring may have a restraining flange to provide some restraint to endwise movement in one direction or may be without a flange so that the bearing rings may be displaced axially with respect to each other. Either rolls or roller path on the races may be slightly crowned to prevent edge loading under slight shaft misalignment. Low friction makes this type suitable for relatively high speeds.

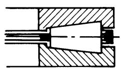
Barrel Roller: These bearings have rollers that are barrel-shaped and symmetrical. They are furnished in both single- and double-row mountings. As with cylindrical roller bearings, the single-row mounting type has a low thrust capacity, but angular mounting of rolls in the double-row type permits its use for combined axial and thrust loads.

Spherical Roller: These bearings are usually furnished in a double-row, self-aligning mounting. Both rows of rollers have a common spherical outer raceway. The rollers are barrel-shaped with one end smaller than the other to provide a small thrust to keep the rollers in contact with the center guide flange. This type of roller bearing has a high radial and thrust load carrying capacity with the ability to maintain this capacity under some degree of misalignment of shaft and bearing housing.

Tapered Roller: In this type, straight tapered rollers are held in accurate alignment by means of a guide flange on the inner ring. The basic characteristic of these bearings is that the apexes of the tapered working surfaces of both rollers and races, if extended, would coincide on the bearing axis. These bearings are separable. They have a high radial and thrust carrying capacity.

TAPERED ROLLER BEARINGS — INCH			
Symbol	Description	Symbol	Description
TS	Single row 	TDI	Two row, double-cone single cups 
TDO	Two row, double-cup single-cone adjustable 	TNA	Two row, double-cup single cone nonadjustable 
TQD, TQI	Four row, cup adjusted 		

TAPERED ROLLER BEARINGS — METRIC			
Symbol	Description	Symbol	Description
TS	Single row, straight bore 	TSF	Single row, straight bore, flanged cup 
TDO	Double row, straight bore, two single cones, one double cup with lubrication hole and groove 	2TS	Double row, straight bore, two single cones, two single cups 

THRUST TAPERED ROLLER BARINGS	
Symbol	Description
TT	Thrust bearings 

Types of Ball and Roller Thrust Bearings.—Are designed to take thrust loads alone or in combination with radial loads.

One-direction Ball Thrust: These bearings consist of a shaft ring and a flat or spherical housing ring with a single row of balls between. They are capable of carrying pure thrust loads in one direction only. They cannot carry any radial load.

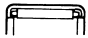
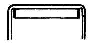
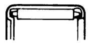
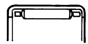
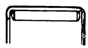
Two-direction Ball Thrust: These bearings consist of a shaft ring with a ball groove in either side, two sets of balls, and two housing rings so arranged that thrust loads in either direction can be supported. No radial loads can be carried.

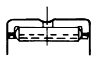
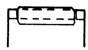
Spherical Roller Thrust: This type is similar in design to the radial spherical roller bearing except that it has a much larger contact angle. The rollers are barrel shaped with one end smaller than the other. This type of bearing has a very high thrust load carrying capacity and can also carry radial loads.

Tapered Roller Thrust: In this type the rollers are tapered and several different arrangements of housing and shaft are used.

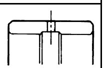
Roller Thrust: In this type the rollers are straight and several different arrangements of housing and shaft are used.

NEEDLE ROLLER BEARINGS, DRAWN CUP

Symbol[a]	Description	Symbol[a]	Description
NIB NB	Needle roller bearing, full complement, drawn cup, without inner ring 	NIYM NYM	Needle roller bearing, full complement, rollers retained by lubricant, drawn cup, closed end, without inner ring 
NIBM NBM	Needle roller bearing, full complement, drawn cup, closed end, without inner ring 	NIH NH	Needle roller bearing, with cage, drawn cup, without inner ring 
NIY NY	Needle roller bearing, full complement, rollers retained by lubricant, drawn cup, without inner ring 	NIHM NHM	Needle roller bearing, with cage, drawn cup, closed end, without inner ring 

NEEDLE ROLLER BEARINGS		NEEDLE ROLLER AND CAGE ASSEMBLIES	
Symbol[a]	Description	Symbol[a]	Description
NIA NA	Needle roller bearing, with cage, machined ring lubrication hole and groove in OD, without inner ring 	NIM NM	Needle roller and cage assembly 

NEEDLE ROLLER BEARING INNER RINGS

Symbol[a]	Description
NIR NR	Needle roller bearing inner ring, lubrication hole and groove in bore 

Machined Ring Needle Roller Bearings Type NIA may be used with inch dimensioned inner rings, Type NIR, and Type NA may be used with metric dimensional inner rings, Type NR.

[a] Symbols with I, as NIB, are inch-dimensioned, and those without the I, as NB, are metric dimensioned.

Types of Needle Bearings.—Needle bearings are characterized by their relatively small size rollers, usually not above $\frac{1}{4}$ inch in diameter, and a relatively high ratio of length to diameter, usually ranging from about 3 to 1 and 10 to 1. Another feature that is characteristic of several types of needle bearings is the absence of a cage or separator for retaining the individual rollers. Needle bearings may be divided into three classes: loose-roller, outer race and retained roller, and non-separable units.

Loose-roller: This type of bearing has no integral races or retaining members, the needles being located directly between the shaft and the outer bearing bore. Usually both shaft and outer bore bearing surfaces are hardened and retaining members that have smooth unbroken surfaces are provided to prevent endwise movement. Compactness and high radial load capacity are features of this type.

Outer Race and Retained Roller: There are two types of outer race and retained roller bearings. In the *Drawn Shell* type, the needle rollers are enclosed by a hardened shell that acts as a retaining member and as a hardened outer race. The needles roll directly on the shaft, the bearing surface of which should be hardened. The capacity for given roller length and shaft diameter is about two-thirds that of the loose roller type. It is mounted in the housing with a press fit.

In the *Machined Race* type, the outer race consists of a heavy machined member. Various modifications of this type provide heavy ends or faces for end location of the needle rollers, or open end construction with end washers for roller retention, or a cage that maintains alignment of the rollers and is itself held in place by retaining rings. An auxiliary outer member with spherical seat that holds the outer race may be provided for self-alignment. This type is applicable where split housings occur or where a press fit of the bearing into the housing is not possible.

Non-separable: This type consists of a non-separable unit of outer race, rollers and inner race. These bearings are used where high static or oscillating motion loads are expected as in certain aircraft components and where both outer and inner races are necessary.

Special or Unconventional Types.—Rolling contact bearings have been developed for many highly specialized applications. They may be constructed of non-corrosive materials, non-magnetic materials, plastics, ceramics, and even wood. Although the materials are chosen to adapt more conventional configurations to difficult applications or environments, even greater ingenuity has been applied in utilizing rolling contact for solving particular problems. Thus, linear or recirculating bearings are available to provide low friction, accurate location, and simplified lubrication features to such applications as machine ways, axial motion devices, jack-screws, steering linkages, collets, and chucks. This type of bearing utilizes the “full-complement” style of loading the rolling elements between “races” or ways without a cage and with each element advancing by the action of “races” in the loaded areas and by contact with the adjacent element in the unloaded areas. The “races” may not be cylindrical or bodies of revolution but plane surfaces, with suitable interruptions to free the rolling elements so that they can follow a return trough or slot back to the entry-point at the start of the “race” contact area. Combinations of radial and thrust bearings are available for the user with special requirements.

Plastics Bearings.—A more recent development has been the use of Acetal resin rollers and balls in applications where abrasive, corrosive and difficult-to-lubricate conditions exist. Although these bearings do not have the load carrying capacity nor the low friction factor of their hard steel counterparts, they do offer freedom from indentation, wear, and corrosion, while at the same time providing significant weight savings.

Of additional value are: 1) their resistance to indentation from shock loads or oscillation; and 2) their self-lubricating properties.

Usually these bearings are not available from stock, but must be designed and produced in accordance with the data made available by the plastics processor.

Pillow Block and Flanged Housing Bearings.—Of great interest to the shop man and particularly adaptable to “line-shafting” applications are a series of ball and roller bearings supplied with their own housings, adapters, and seals. Often called pre-mounted bearings, they come with a wide variety of flange mountings permitting location on faces parallel to or perpendicular to the shaft axis.

Inner races can be mounted directly on ground shafts, or can be adapter-mounted to “drill-rod” or to commercial shafting. For installations sensitive to imbalance and vibration, the use of accurately ground shaft seats is recommended.

Most pillow block designs incorporate self-aligning types of bearings so they do not require the precision mountings utilized with more normal bearing installations.

Conventional Bearing Materials.—Most rolling contact bearings are made with all load carrying members of full hard steel, either through- or case-hardened. For greater reliability this material is controlled and selected for cleanliness and alloying practices in conformity with rigid specifications in order to reduce anomalies and inclusions that could limit the useful fatigue life. Magnaflux inspection is employed to ensure that elements are free from both material defects and cracks. Likewise, a light etch is employed between rough and finish grinding to allow detection of burns due to heavy stock removal and associated decarburization in finished pieces.

Cage Materials.—Standard bearings are normally made with cages of free-machining brass or low carbon sulfurized steel. In high-speed applications or where lubrication may be intermittent or marginal, special materials may be employed. Iron-silicon-bronze, laminated phenolics, silver-plating, over-lays, solid-film baked-on coatings, carbon-graphite inserts, and, in extreme cases, sintered or even impregnated materials are used in separators.

Commercial bearings usually rely on stamped steel with or without a phosphate treatment; some low cost varieties are found with snap-in plastic or metallic cages.

So long as lubrication is adequate and speeds are both reasonable and steady, the materials and design of the cage are of secondary importance when compared with those of the rolling elements and their contacts with the races. In spite of this tolerance, a good portion of all rolling bearing failures encountered can be traced to cage failures resulting from inadequate lubrication. It can never be overemphasized that *no bearing can be designed to run continuously without lubrication!*

Standard Method of Bearing Designation.—The Anti-Friction Bearing Manufacturers Association has adopted a standard identification code that provides a specific designation for each different ball, roller, and needle bearing. Thus, for any given bearing, a uniform designation is provided for manufacturer and user alike, so that the confusion of different company designations can be avoided.

In this identification code there is a "basic number" for each bearing that consists of three elements: a one- to four-digit number to indicate the size of the bore in numbers of millimeters (metric series); a two- or three-letter symbol to indicate the type of bearing; and a two-digit number to identify the dimension series to which the bearing belongs.

In addition to this "basic number" other numbers and letters are added to designate type of tolerance, cage, lubrication, fit up, ring modification, addition of shields, seals, mounting accessories, etc. Thus, a complete designating symbol might be *50BC02JPXE0A10,* for example. The basic number is *50BC02* and the remainder is the supplementary number. For a radial bearing, this latter consists of up to four letters to indicate modification of design, one or two digits to indicate internal fit and tolerances, a letter to indicate lubricants and preservatives, and up to three digits to indicate special requirements.

For a thrust bearing the supplementary number would consist of two letters to indicate modifications of design, one digit to indicate tolerances, one letter to indicate lubricants and preservatives, and up to three digits to indicate special requirements.

For a needle bearing the supplementary number would consist of up to three letters indicating cage material or integral seal information or whether the outer ring has a crowned outside surface and one letter to indicate lubricants or preservatives.

Dimension Series: Annular ball, cylindrical roller, and self-aligning roller bearings are made in a series of different outside diameters for every given bore diameter and in a series of different widths for every given outside diameter. Thus, each of these bearings belongs to a dimension series that is designated by a two-digit number such as or, 23, 93, etc. The first digit (8, 0, 1, 2, 3, 4, 5, 6 and 9) indicates the *width series* and the second digit (7, 8, 9, 0, 1, 2, 3, and 4) the *diameter series* to which the bearing belongs. Similar types of identification codes are used for ball and roller thrust bearings and needle roller bearings.

Table 1. ABEC-1 and RBEC-1 Tolerance Limits for Metric Ball and Roller Bearings *ANSI/ABMA 20-1987*

Basic Inner Ring Bore Diameter, d		V_{dp},[a] max			Δ_{dmp}[b]		K_{ia}[c]
mm		Diameter Series					
Over	Incl.	7,8,9	0,1	2,3,4	High	Low	max
2.5	10	10	8	6	0	−8	10
10	18	10	8	6	0	−8	10
18	30	13	10	8	0	−10	13
30	50	15	12	9	0	−12	15
50	80	19	19	11	0	−15	20
80	120	25	25	15	0	−20	25
120	180	31	31	19	0	−25	30
180	250	38	38	23	0	−30	40
250	315	44	44	26	0	−35	50
315	400	50	50	30	0	−40	60

[a] Bore diameter variation in a single radial plane.
[b] Single plane mean bore diameter deviation from basic. (For a basically tapered bore, Δ_{dmp} refers only to the theoretical small end of the bore.)
[c] Radial runout of assembled bearing inner ring.

Basic Outer Ring Outside Outerside Diameter, D		V_{Dp},[a] max				Δ_{Dmp}[b]		K_{ea}[c]
		Open Bearings			Capped Bearings[d]			
mm		Diameter Series						
Over	Incl.	7,8,9	0,1	2,3,4	2,3,4	High	Low	max
6	18	10	8	6	10	0	−8	15
18	30	12	9	7	12	0	−9	15
30	50	14	11	8	16	0	−11	20
50	80	16	13	10	20	0	−13	25
80	120	19	19	11	26	0	−15	35
120	150	23	23	14	30	0	−18	40
150	180	31	31	19	38	0	−25	45
180	250	38	38	23	…	0	−30	50
250	315	44	44	26	…	0	−35	60
315	400	50	50	50	…	0	−40	70

[a] Outside diameter variation in a single radial plane. Applies before mounting and after removal of internal or external snap ring.
[b] Single plane mean outside diameter deviation from basic.
[c] Radial runout of assembled bearing outer ring.
[d] No values have been established for diameters series 7, 8, 9, 0, and 1.

Width Tolerances									
d		Δ_{Bs}[a]			d		Δ_{Bs}[a]		
mm		All	Normal	Modified[b]	mm		All	Normal	Modified[b]
Over	Incl.	High	Low		Over	Incl.	High	Low	
2.5	10	0	−120	−250	80	120	0	−200	−380
10	18	0	−120	−250	120	180	0	−250	−500
18	30	0	−120	−250	180	250	0	−300	−500
30	50	0	−120	−250	250	315	0	−350	−500
50	80	0	−150	−380	315	400	0	−400	−630

[a] Single inner ring width deviation from basic. Δ_{Cs} (single outer ring width deviation from basic) is identical to Δ_{Bs} of inner ring of same bearing.
[b] Refers to the rings of single bearings made for paired or stack mounting.

All units are micrometers, unless otherwise indicated.For sizes beyond range of this table, see Standard.This table does not cover tapered roller bearings.

Table 2. ABEC-3 AND RBEC-3 Tolerance Limits for Metric Ball and Roller Bearings *ANSI/ABMA 20-1987*

Basic Inner Ring Bore Diameter, d		V_{dp},[a] max			Δ_{dmp}[b]		K_{ia}[c]
mm		Diameter Series					
Over	Incl.	7, 8, 9	0, 1	2, 3, 4	High	Low	max
2.5	10	9	7	5	0	−7	6
10	18	9	7	5	0	−7	7
18	30	10	8	6	0	−8	8
30	50	13	10	8	0	−10	10
50	80	15	15	9	0	−12	10
80	120	19	19	11	0	−15	13
120	180	23	23	14	0	−18	18
180	250	28	28	17	0	−22	20
250	315	31	31	19	0	−25	25
315	400	38	38	23	0	−30	30

[a] Bore diameter variation in a single radial plane.

[b] Single plane mean bore diameter deviation from basic. (For a basically tapered bore, Δ_{dmp} refers only to the theoretical small end of the bore.)

[c] Radial runout of assembled bearing inner ring.

Basic Outer Ring Outside Outerside Diameter, D		V_{Dp},[a] max				Δ_{Dmp}[b]		K_{ea}[c]
		Open Bearings			Capped Bearings[d]			
mm		Diameter Series						
Over	Incl.	7,8,9	0,1	2,3,4	2,3,4	High	Low	max
6	18	9	7	5	9	0	−7	8
18	30	10	8	6	10	0	−8	9
30	50	11	9	7	13	0	−9	10
50	80	14	11	8	16	0	−11	13
80	120	16	16	10	20	0	−13	18
120	150	19	19	11	25	0	−15	20
150	180	23	23	14	30	0	−18	23
180	250	25	25	15	…	0	−20	25
250	315	31	31	19	…	0	−25	30
315	400	35	35	21	…	0	−28	35

[a] Outside diameter variation in a single radial plane. Applies before mounting and after removal of internal or external snap ring.

[b] Single plane mean outside diameter deviation from basic.

[c] Radial runout of assembled bearing outer ring.

[d] No values have been established for diameter series 7, 8, 9, 0, and 1.

Width Tolerances									
d		Δ_{Bs}[a]			d		Δ_{Bs}[a]		
mm		All	Normal	Modified[b]	mm		All	Normal	Modified[b]
Over	Incl.	High	Low		Over	Incl.	High	Low	
2.5	10	0	−120	−250	80	120	0	−200	−380
10	18	0	−120	−250	120	180	0	−250	−500
18	30	0	−120	−250	180	250	0	−300	−500
30	50	0	−120	−250	250	315	0	−350	−500
50	80	0	−150	−380	315	400	0	−400	−630

[a] Single inner ring width deviation from basic. Δ_{Cs} (single outer ring width deviation from basic) is identical to Δ_{Bs} of inner ring of same bearing.

[b] Refers to the rings of single bearings made for paired or stack mounting.

All units are micrometers, unless otherwise indicated.For sizes beyond range of this table, see Standard.This table does not cover tapered roller bearings.

Table 3. ABEC-5 and RBEC-5 Tolerance Limits for Metric Ball and Roller Bearings *ANSI/ABMA 20-1987*

INNER RING												
Inner Ring BoreBasic Dia., d		V_{dp},[a] max		Δ_{dmp}[b]		Radial Runout K_{ia}	Ref. Face Runout with Bore S_d	Axial Runout of Assembled Bearing with Inner Ring S_{ia}[c]	Width			
mm		Diameter Series							Δ_{Bs}[d]			V_{Bs}[e]
Over	Incl.	7, 8, 9	0, 1, 2, 3, 4	High	Low	max	max	max	All	Normal	Modified[f]	max
2.5	10	5	4	0	−5	4	7	7	0	−40	−250	5
10	18	5	4	0	−5	4	7	7	0	−80	−250	5
18	30	6	5	0	−6	4	8	8	0	−120	−250	5
30	50	8	6	0	−8	5	8	8	0	−120	−250	5
50	80	9	7	0	−9	5	8	8	0	−150	−250	6
80	120	10	8	0	−10	6	9	9	0	−200	−380	7
120	180	13	10	0	−13	8	10	10	0	−250	−380	8
180	250	15	12	0	−15	10	11	13	0	−300	−500	10

[a] Bore (V_{dp}) and outside diameter (V_{Dp}) variation in a single radial plane.

[b] Single plane mean bore (Δ_{dmp}) and outside diameter (Δ_{Dmp}) deviation from basic. (For a basically tapered bore, Δ_{dmp} refers only to the theoretical small end of the bore.)

[c] Applies to groove-type ball bearings only.

[d] Single bore (Δ_{Bs}) and outer ring (Δ_{Cs}) width variation.

[e] Inner (V_{Bs}) and outer (V_{Cs}) ring width deviation from basic.

[f] Applies to the rings of single bearings made for paired or stack mounting.

OUTER RING											
Basic Outer Ring Outside Dia., D		V_{Dp},[aa] max		Δ_{Dmp}[b]		Radial Runout K_{ea}	Outside Cylindrical Surface Runout with Outer Ring Ref. Face S_D	Axial Runout of Assembled Bearing with Outer Ring S_{ea}	Width		
mm		Diameter Series							Δ_{Cs}[d]		V_{Cs}[e]
Over	Incl.	7, 8, 9	0, 1, 2, 3, 4	High	Low	max	max	max	High	Low	max
6	18	5	4	0	−5	5	8	8			5
18	30	6	5	0	−6	6	8	8			5
30	50	7	5	0	−7	7	8	8	Identical		5
50	80	9	7	0	−9	8	8	10	to Δ_{Bs}		6
80	120	10	8	0	−10	10	9	11	of inner ring		8
120	150	11	8	0	−11	11	10	13	of same bearing		8
150	180	13	10	0	−13	13	10	14			8
180	250	15	11	0	−15	15	11	15			10

[a] No values have been established for capped bearings.

All units are micrometers, unless otherwise indicated.For sizes beyond range of this table, see Standard.This table does not cover instrument bearings and tapered roller bearings.

Table 4. ABEC-7 Tolerance Limits for Metric Ball and Roller Bearings *ANSI/ABMA 20-1987*

INNER RING																
Inner Ring Bore Basic Dia., d		V_{dp},[a] max		Δ_{dmp}[b]		Δ_{ds}[c]		Radial Runout K_{ia}	Ref. Face Runout with Bore S_d	Axial Runout of AssembledBearing with Inner Ring S_{ia}[d]	Width			
mm		Diameter Series									Δ_{Bs}[e]			V_{Bs}[f]
Over	Incl.	7, 8, 9	0, 1, 2, 3, 4	High	Low	High	Low	max	max	max	All	Normal	Modified[g]	max
2.5	10	4	3	0	−4	0	−4	2.5	3	3	0	−40	−250	2.5
10	18	4	3	0	−4	0	−4	2.5	3	3	0	−80	−250	2.5
18	30	5	4	0	−5	0	−5	3	4	4	0	−120	−250	2.5
30	50	6	5	0	−6	0	−6	4	4	4	0	−120	−250	3
50	80	7	5	0	−7	0	−7	4	5	5	0	−150	−250	4
80	120	8	6	0	−8	0	−8	5	5	5	0	−200	−380	4
120	180	10	8	0	−10	0	−10	6	6	7	0	−250	−380	5
180	250	12	9	0	−12	0	−12	8	7	8	0	−300	−500	6

[a] Bore (V_{dp}) and outside diameter (V_{Dp}) variation in a single radial plane.

[b] Single plane mean bore (Δ_{dmp}) and outside diameter (Δ_{Dmp}) deviation from basic. (For a basically tapered bore, Δ_{dmp} refers only to the theoretical small end of the bore.)

[c] Single bore (Δ_{ds}) and outside diameter (Δ_{Ds}) deviations from basic. These deviations apply to diameter series 0, 1, 2, 3, and 4 only.

[d] Applies to groove-type ball bearings only.

[e] Single bore (Δ_{Bs}) and outer ring (Δ_{Cs}) width deviation from basic.

[f] Inner (V_{Bs}) and outer (V_{Cs}) ring width variation.

[g] Applies to the rings of single bearings made for paired or stack mounting.

OUTER RING														
Basic Outer Ring Outside Dia., D		V_{Dp},[aa] max		Δ_{Dmp}[b]		Δ_{Ds}[c]		Radial Runout K_{ea}	Outside Cylindrical Surface Runout Outer Ring Ref. Face S_D	Axial Runout of Assembled Bearing Outer Ring S_{ea}[d]	Width		
mm		Diameter Series									Δ_{Cs}[e]		V_{Cs}[f]
Over	Incl.	7, 8, 9	0, 1, 2, 3, 4	High	low	High	Low	max	max	max	High	Low	max
6	18	4	3	0	−4	0	−4	3	4	5			2.5
18	30	5	4	0	−5	0	−5	4	4	5			2.5
30	50	6	5	0	−6	0	−6	5	4	5	Identical		2.5
50	80	7	5	0	−7	0	−7	5	4	5	to Δ_{Bs}		3
80	120	8	6	0	−8	0	−8	6	5	6	of inner ring		4
120	150	9	7	0	−9	0	−9	7	5	7	of same bearing		5
150	180	10	8	0	−10	0	−10	8	5	8			5
180	250	11	8	0	−11	0	−11	19	7	10			7

[a] No values have been established for capped bearings.

All units are micrometers, unless otherwise indicated. For sizes beyond range of this table, see Standard. This table does not cover instrument bearings.

Table 5. ABEC-9 Tolerance Limits for Metric Ball and Roller Bearing *ANSI/ABMA 20-1987*

INNER RING												
Inner Ring Bore Basic Dia., d		V_{dp},[a] max	Δ_{dmp}[b]		Δ_{ds}[c]		Radial Runout K_{ia}	Ref. Face Runout with Bore S_d	Axial Runout of Assembled Bearing with Inner Ring S_{ia}[d]	Width		
mm										Δ_{Bs}[e]		V_{Bs}[f]
Over	Incl.	max	High	Low	High	Low	max	max	max	High	Low	max
2.5	10	2.5	0	−2.5	0	−2.5	1.5	1.5	1.5	0	−40	1.5
10	18	2.5	0	−2.5	0	−2.5	1.5	1.5	1.5	0	−80	1.5
18	30	2.5	0	−2.5	0	−2.5	2.5	1.5	2.5	0	−120	1.5
30	50	2.5	0	−2.5	0	−2.5	2.5	1.5	2.5	0	−120	1.5
50	80	4	0	−4	0	−4	2.5	1.5	2.5	0	−150	1.5
50	80	4	0	−4	0	−4	2.5	1.5	2.5	0	−150	1.5
80	120	5	0	−5	0	−5	2.5	2.5	2.5	0	−200	2.5
120	150	7	0	−7	0	−7	2.5	2.5	2.5	0	−250	2.5
150	180	7	0	−7	0	−7	5	4	5	0	−300	4
180	250	8	0	−8	0	−8	5	5	5	0	−350	5

[a] Bore (V_{dp}) and outside diameter (V_{Dp}) variation in a single radial plane.

[b] Single plane mean bore (Δ_{dmp}) and outside diameter (Δ_{Dmp}) deviation from basic. (For a basically tapered bore, Δ_{dmp} refers to the theoretical small end of the bore.)

[c] Single bore diameter (Δ_{ds}) and outside diameter (Δ_{Ds}) deviation from basic.

[d] Applies to groove-type ball bearings only.

[e] Single bore (Δ_{Bs}) and outer ring (Δ_{Cs}) width variation from basic.

[f] Inner (V_{Bs}) and outer (V_{Cs}) ring width variation.

OUTER RING												
Basic Outside Diameter, D		V_{Dp}[aa]	Δ_{Dmp}[b]		Δ_{Ds}[c]		Radial Runout K_{ea}	Outside Cylindrical Surface Runout with Outer Ring S_D	Axial Runout of Assembled Bearing with Outer Ring S_{ea}	Width		
mm										Δ_{Cs}[e]		V_{Cs}[f]
Over	Incl.	max	High	Low	High	Low	max	max	max	High	Low	max
6	18	2.5	0	−2.5	0	−2.5	1.5	1.5	1.5	Identical to Δ_{Bs} of inner ring of same bearing		1.5
18	30	4	0	−4	0	−4	2.5	1.5	2.5			1.5
30	50	4	0	−4	0	−4	2.5	1.5	2.5			1.5
50	80	4	0	−4	0	−4	4	1.5	4			1.5
80	120	5	0	−5	0	−5	5	2.5	5			1.5
120	150	5	0	−5	0	−5	5	2.5	5			1.5
150	180	7	0	−7	0	−7	5	2.5	5			2.5
180	250	8	0	−8	0	−8	7	4	7			4
250	315	8	0	−8	0	−8	7	5	7			5

[a] No values have been established for capped bearings.

All units are micrometers, unless otherwise indicated.For sizes beyond range of this table, see Standard.This table does not cover instrument bearings.

Bearing Tolerances.—In order to provide standards of precision for proper application of ball or roller bearings in all types of equipment, five classes of tolerances have been established by the Anti-Friction Bearing Manufacturers Association for ball bearings, three for cylindrical roller bearings and one for spherical roller bearings. These tolerances are given in Tables 1, 2, 3, 4, and 5. They are designated as ABEC-1, ABEC-3, ABEC-5, ABEC-7 and ABEC-9 for ball bearings, the ABEC-9 being the most precise, RBEC-1, RBEC-3, and RBEC-5 for roller bearings. In general, bearings to specifications closer than ABEC-1 or RBEC-1 are required because of the need for very precise fits on shaft or housing, to reduce eccentricity or runout of shaft or supported part, or to permit operation at very high speeds. All five classes include tolerances for bore, outside diameter, ring width, and radial runouts of inner and outer rings. ABEC-5, ABEC-7 and ABEC-9 provide added tolerances for parallelism of sides, side runout and groove parallelism with sides.

Thrust Bearings: Anti-Friction Bearing Manufacturers Association and American National Standard tolerance limits for metric single direction thrust ball and roller bearings are given in Table 6. Tolerance limits for single direction thrust ball bearings, inch dimensioned are given in Table 7, and for cylindrical thrust roller bearings, inch dimensioned in Table 8.

Table 6. AFBMA and American National Standard Tolerance Limits for Metric Single Direction Thrust Ball (Type TA) and Roller Type (Type TS) Bearings
ANSI/ABMA 24.1-1989

Bore Dia.of Shaft Washer, d		Δd_{mp}[a]		S_i, S_e[b]	ΔT_s[c]			Outside Dia. of Housing Washer, D		ΔD_{mp}[a]	
mm								mm			
Over	Incl.	High	Low	Max	Max	Min Type TA	MinType TS	Over	Incl.	High	Low
18	30	0	−10	10	20	−250	…	10	18	0	−11
30	**50**	**0**	**−12**	10	**20**	−250	−300	18	30	0	−13
50	**80**	**0**	**−15**	10	**20**	−300	−400	30	50	0	−16
80	**120**	**0**	**−20**	15	**25**	−300	−400	**50**	**80**	**0**	**−19**
120	**180**	**0**	**−25**	15	**25**	−400	−500	**80**	**120**	**0**	**−22**
180	**250**	**0**	**−30**	20	**30**	−400	−500	**120**	**180**	**0**	**−25**
250	**315**	**0**	**−35**	25	**40**	−400	−700	**180**	**250**	**0**	**−30**
315	**400**	**0**	**−40**	30	**40**	−500	−700	**250**	**315**	**0**	**−35**
400	**500**	**0**	**−45**	30	**50**	−500	−900	**315**	**400**	**0**	**−40**
500	**630**	**0**	**−50**	35	**60**	−600	−1200	**400**	**500**	**0**	**−45**

[a] Single plane mean bore diameter deviation of central shaft washer (Δd_{mp}) and outside diameter (ΔD_{mp}) variation.

[b] Raceway parallelism with the face, housing-mounted (S_e) and boremounted (S_i) race or washer.

[c] Deviation of the actual bearing height.

All dimensions in micrometers, unless otherwise indicated.

Tolerances are for normal tolerance class only. For sizes beyond the range of this table and for other tolerance class values, see Standard. All entries apply to type TA bearings; boldface entries also apply to type TS bearings.

Table 7. Tolerance Limits for Single Direction Ball Thrust Bearings—Inch Design
ANSI/ABMA 24.2-1998

Bore Diameter[a] d, Inches		Single Plane Mean Bore Dia. Variation, d, Inch		Outside Diameter D, Inches		Single Plane Mean O.D. Variation, D, Inch	
Over	Incl.	High	Low	Over	Incl.	High	Low
0	6.7500	+0.005	0	0	5.3125	+0	−0.002
6.7500	20.0000	+0.010	0	5.3125	17.3750	+0	−0.003
…	…	…	…	17.3750	39.3701	+0	−0.004

[a] Bore tolerance limits are: For bore diameters over 0 to 1.8125 inches, inclusive, +0.005, −0.005; over 1.8125 to 12.000 inches, inclusive, +0.010, −0.010; over 12.000 to 20.000, inclusive, +0.0150, −0.0150.

Table 8. Tolerance Limits for Cylindrical Roller Thrust Bearings—Inch Design
ANSI/ABMA 24.2-1998

Basic Bore Dia., d		Δd_{mp} [a]		ΔT_s [b]		Basic Outside dia., D		ΔD_{mp} [c]	
Over	Incl.	Low	High	High	Low	Over	Incl.	High	Low
EXTRA LIGHT SERIES—TYPE TP									
0	0.9375	+.0040	+.0060	+.0050	−.0050	0	4.7188	+0	−.0030
0.9375	1.9375	+.0050	+.0070	+.0050	−.0050	4.7188	5.2188	+0	−.0030
1.9375	3.0000	+.0060	+.0080	+.0050	−.0050	…	…	…	…
3.0000	3.5000	+.0080	+.0100	−.0100	−.0100	…	…	…	…

[a] Single plane mean bore diameter deviation.
[b] Deviation of the actual bearing height, single direction bearing.
[c] Single plane mean outside diameter deviation.

Basic Bore Diameter, d		Δd_{mp} [a]		Basic Outside Diameter, D		Outside Dia., D Tolerance Limits		Basic Bore Diameter, d		ΔT_s	
Over	Inc.	High	Low	Over	Incl.	High	Low	Over	Incl.	High	Low
LIGHT SERIES—TYPE TP											
0	1.1870	+0	−.0005	0	2.8750	+.0005	−0	0	2.0000	+0	−.006
1.1870	1.3750	+0	−.0006	2.8750	3.3750	+.0007	−0	2.0000	3.0000	+0	−.008
1.3750	1.5620	+0	−.0007	3.3750	3.7500	+.0009	−0	3.0000	6.0000	+0	−.010
1.5620	1.7500	+0	−.0008	3.7500	4.1250	+.0011	−0	6.0000	10.0000	+0	−.015
1.7500	1.9370	+0	−.0009	4.1250	4.7180	+.0013	−0	10.0000	18.0000	+0	−.020
1.9370	2.1250	+0	−.0010	4.7180	5.2180	+.0015	−0	18.0000	30.0000	+0	−.025
2.1250	2.5000	+0	−.0011	…	…	…	…	…	…	…	…
2.2500	3.0000	+0	−.0012	…	…	…	…	…	…	…	…
3.0000	3.5000	+0	−.0013	…	…	…	…	…	…	…	…
HEAVY SERIES—TYPE TP											
2.0000	3.0000	+0	−.0010	5.0000	10.0000	+.0015	−0	0	2.000	+0	−.006
3.0000	3.5000	+0	−.0012	10.0000	18.0000	+.0020	−0	2.000	3.000	+0	−.008
3.5000	9.0000	+0	−.0015	18.0000	26.0000	+.0025	−0	3.000	6.000	+0	−.010
9.0000	12.0000	+0	−.0018	26.0000	34.0000	+.0030	−0	6.000	10.000	+0	−.015
12.0000	18.0000	+0	−.0020	34.0000	44.0000	+.0040	−0	10.000	18.000	+0	−.020
18.0000	22.0000	+0	−.0025	…	…	…	…	18.000	30.000	+0	−.025
22.0000	30.0000	+0	−.003	…	…	…	…	…	…	…	…
TYPE TPC											
0	2.0156	+.010	−0	2.5000	4.0000	+.005	−.005	0	2.0156	+0	−.008
2.0156	3.0156	+.010	−.020	4.0000	6.0000	+.006	−.006	2.0156	3.0156	+0	−.010
3.0156	6.0156	+.015	−.020	6.0000	10.0000	+.010	−.010	3.0156	6.0156	+0	−.015
6.0156	10.1560	+.015	−.050	10.0000	18.0000	+.012	−.012	6.0156	10.1560	+0	−.020

All dimensions are in inches.

For Type TR bearings, see Standard.

Only one class of tolerance limits is established for metric thrust bearings.

Radial Needle Roller Bearings: Tolerance limits for needle roller bearings, drawn cup, without inner ring, inch types NIB, NIBM, NIY, NIYM, NIH, and NIHM are given in Table 9 and for metric types NB, NBM, NY, NYM, NH and NHM in Table 10. Standard tolerance limits for needle roller bearings, with cage, machined ring, without inner ring, inch type NIA are given in Table 11 and for needle roller bearings inner rings, inch type NIR in Table 12.

Table 9. AFBMA and American National Standard Tolerance Limits for Needle Roller Bearings, Drawn Cup, Without Inner Ring — Inch Types NIB, NIBM, NIY, NIYM, NIH, and NIHM *ANSI/ABMA 18.2-1982 (R1993)*

Ring Gage Bore Diameter[a]		
Basic Outside Diameter, *D* Inch		Deviation from *D* Inch
Over	Incl.	
0.1875	0.9375	+0.0005
0.9375	4.0000	−0.0005

For fitting and mounting practice see Table 18.

Basic Bore Diameter under Needle Rollers, F_w		Allowable Deviation from F_w[a]		Allowable Deviation from Width, *B*	
Inch		Inch		Inch	
Over	Incl.	Low	High	High	Low
0.1875	0.6875	+0.0015	+0.0024	+0	−0.0100
0.6875	1.2500	+0.0005	+0.0014	+0	−0.0100
1.2500	1.3750	+0.0005	+0.0015	+0	−0.0100
1.3750	1.6250	+0.0005	+0.0016	+0	−0.0100
1.6250	1.8750	+0.0005	+0.0017	+0	−0.0100
1.8750	2.0000	+0.0006	+0.0018	+0	−0.0100
2.0000	2.5000	+0.0006	+0.0020	+0	−0.0100
2.5000	3.5000	+0.0010	+0.0024	+0	−0.0100

[a] The bore diameter under needle rollers can be measured only when bearing is pressed into a ring gage, which rounds and sizes the bearing.

Table 10. AFBMA and American National Standard Tolerance Limits for Needle Roller Bearings, Drawn Cup, Without Inner Ring — Metric Types NB, NBM, NY, NYM, NH, and NHM *ANSI/ABMA 18.1-1982 (R1994)*

Ring Gage Bore Diameter[a]			Basic Bore Diameter underNeedle Rollers, F_w		Allowable Deviation from F_w[a]		Allowable Deviation from Width, *B*	
Basic Outside Diameter, *D*		Deviation from *D*						
mm			mm		Micrometers		Micrometers	
Over	Incl.	Micrometers	Over	Incl.	Low	High	High	Low
6	10	−16	3	6	+10	+28	+0	−250
10	18	−20	6	10	+13	+31	+0	−250
30	50	−28	18	30	+20	+41	+0	−250
50	80	−33	30	50	+25	+50	+0	−250
…	…	…	50	70	+30	+60	+0	−250

[a] The bore diameter under needle rollers can be measured only when bearing is pressed into a ring gage, which rounds and sizes the bearing.

For fitting and mounting practice, see Table 18.

Table 11. AFBMA and American National Standard Tolerance Limits for Needle Roller Bearings, With Cage, Machined Ring, Without Inner Ring—Inch Type NIA *ANSI/ABMA 18.2-1982 (R1993)*

Basic Outside Diameter, *D*		Allowable Deviation From *D* of Single Mean Diameter, D_{mp}		Basic Bore Diameter under Needle Rollers, F_w		Allowable Deviation from F_w		Allowable Deviation from Width, *B*	
Inch		Inch		Inch		Inch		Inch	
Over	Incl.	High	Low	Over	Incl.	Low	High	High	Low
0.7500	2.0000	+0	−0.0005	0.3150	0.7087	+0.0008	+0.0017	+0	−0.0050
2.0000	3.2500	+0	−0.0006	0.7087	1.1811	+0.0009	+0.0018	+0	−0.0050
3.2500	4.7500	+0	−0.0008	1.1811	1.6535	+0.0010	+0.0019	+0	−0.0050
4.7500	7.2500	+0	−0.0010	1.6535	1.9685	+0.0010	+0.0020	+0	−0.0050
				1.9685	2.7559	+0.0011	+0.0021	+0	−0.0050
7.2500	10.2500	+0	−0.0012	2.7559	3.1496	+0.0011	+0.0023	+0	−0.0050
10.2500	11.1250	+0	−0.0014	3.1496	4.0157	+0.0012	+0.0024	+0	−0.0050
…	…	…	…	4.0157	4.7244	+0.0012	+0.0026	+0	−0.0050
…	…	…	…	4.7244	6.2992	+0.0013	+0.0027	+0	−0.0050
…	…	…	…	6.2992	7.0866	+0.0013	+0.0029	+0	−0.0050
…	…	…	…	7.0866	7.8740	+0.0014	+0.0030	+0	−0.0050
…	…	…	…	7.8740	9.2520	+0.0014	+0.0032	+0	−0.0050

For fitting and mounting practice, see Table 19.

Table 12. AFBMA and American National Standard Tolerance Limits for Needle Roller Bearing Inner Rings—Inch Type NIR *ANSI/ABMA 18.2-1982 (R1993)*

Basic Outside Diameter, F		Allowable Deviation From F of Single Mean Diameter, F_{mp}		Basic Bore Diameter d		Allowable Deviation from d of Single Mean Diameter, d_{mp}		Allowable Deviation from Width, B	
Inch		Inch		Inch		Inch		Inch	
Over	Incl.	High	Low	Over	Incl.	High	Low	High	Low
0.3937	0.7087	−0.0005	−0.0009	0.3125	0.7500	+0	−0.0004	+0.0100	+0.0050
0.7087	1.0236	−0.0007	−0.0012	0.7500	2.0000	+0	−0.0005	+0.0100	+0.0050
1.0236	1.1811	−0.0009	−0.0014	2.0000	3.2500	+0	−0.0006	+0.0100	+0.0050
1.1811	1.3780	−0.0009	−0.0015	3.2500	4.2500	+0	−0.0008	+0.0100	+0.0050
1.3780	1.9685	−0.0010	−0.0016	4.2500	4.7500	+0	−0.0008	+0.0150	+0.0100
1.9685	3.1496	−0.0011	−0.0018	4.7500	7.0000	+0	−0.0010	+0.0150	+0.0100
3.1496	3.9370	−0.0013	−0.0022	7.0000	8.0000	+0	−0.0012	+0.0150	+0.0100
3.9370	4.7244	−0.0015	−0.0024	…	…	…	…	…	…
4.7244	5.5118	−0.0015	−0.0025	…	…	…	…	…	…
5.5118	7.0866	−0.0017	−0.0027	…	…	…	…	…	…
7.0866	8.2677	−0.0019	−0.0031	…	…	…	…	…	…
8.2677	9.2520	−0.0020	−0.0032	…	…	…	…	…	…

For fitting and mounting practice, see Table 20.

Metric Radial Ball and Roller Bearing Shaft and Housing Fits.—To select the proper fits, it is necessary to consider the type and extent of the load, bearing type, and certain other design and performance requirements.

The required shaft and housing fits are indicated in Tables 13 and 14. The terms "Light," "Normal," and "Heavy" loads refer to radial loads that are generally within the following limits, with some overlap (C being the Basic Load Rating computed in accordance with AFBMA-ANSI Standards):

Bearing Type	Radial Load		
	Light	Normal	Heavy
Ball	Up to 0.075C	From 0.075C to 0.15C	Over 0.15C
Cylindrical Roller	Up to 0.075C	From 0.075C to 0.2C	Over 0.15C
Spherical Roller	Up to 0.075C	From 0.070C to 0.25C	Over 0.15C

Shaft Fits: Table 13 indicates the initial approach to shaft fit selection. Note that for most normal applications where the shaft rotates and the radial load direction is constant, an interference fit should be used. Also, the heavier the load, the greater is the required interference. For stationary shaft conditions and constant radial load direction, the inner ring may be moderately loose on the shaft.

For pure thrust (axial) loading, heavy interference fits are not necessary; only a moderately loose to tight fit is needed.

The upper part of Table 15 shows how the shaft diameters for various ANSI shaft limit classifications deviate from the basic bore diameters.

Table 16 gives metric values for the shaft diameter and housing bore tolerance limits given in Table 15.

The lower parts of Tables 15 and 16 show how housing bores for various ANSI hole limit classifications deviate from the basic shaft outside diameters.

Table 13. Selection of Shaft Tolerance Classifications for Metric Radial Ball and Roller Bearings of ABEC-1 and RBEC-1 Tolerance Classes *ANSI/ABMA 7-1995*

Operating Conditions			Nominal Shaft Diameter						
			Ball Bearings		Cylindrical Roller Bearings		Spherical Roller Bearings		Tolerance Symbol[a]
			mm	Inch	mm	Inch	mm	Inch	
Inner ring stationary in relation to the direction of the load.	All loads	Inner ring has to be easily displaceable	All diameters	All diameters	All diameters	All diameters	All diameters	All diameters	g6
		Inner ring does not have to be easily displaceable	All diameters	All diameters	All diameters	All diameters	All diameters	All diameters	h6
Direction of load indeterminate or the inner ring rotating in relation to the direction of the load.	Radial load:		≤18	≤0.71					h5
	LIGHT		>18	>0.71	≤40	≤1.57	≤40	≤1.57	j6[b]
					(40)-140	(1.57)–5.51	(40)–100	(1.57)–3.94	k6[b]
					(140)–320	(5.51)–12.6	(100)–320	(3.94)–12.6	m6[b]
					(320)–500	(126)–19.7	(320)–500	(126)–19.7	n6
					>500	>19.7	>500	>19.7	p6
	NORMAL		≤18	≤0.71					j5
			>18	>0.71	≤40	≤1.57	≤40	≤1.57	k5
					(40)–100	(1.57)–3.94	(40)–65	(1.57)–2.56	m5
					(100)–140	(3.94)–5.51	(65)–100	(2.56)–3.94	m6
					(140)–320	(5.51)–12.6	(100)–140	(3.94)–5.51	n6
					(320)–500	(12.6)–19.7	(140)–280	(5.51)–11.0	p6
					>500	>19.7	(280)–500	(11.0)–19.7	r6
							>500	>19.7	r7
	HEAVY		(18)–100	(0.71)–3.94					k5
			>100	>3.94	≤40	≤1.57	≤40	≤1.57	m5
					(40)-65	(1.57)–2.56	(40)–65	(1.57)–2.56	m6[b]
					(65)–140	(2.56)–5.51	(65)–100	(2.56)–3.94	n6[b]
					(140)–200	(5.51)–7.87	(100)–140	(3.94)–5.51	p6[b]
					(200)–500	(7.87)–19.7	(140)–200	(5.51)–7.87	r6[b]
					>500	>19.7	>200	>7.87	r7[b]
Pure Thrust Load			All diams.	All diams.	Consult Bearing Manufacturer				j6

[a] For solid steel shafts. For hollow or nonferrous shafts, tighter fits may be needed.

[b] When greater accuracy is required, use j5, k5, and m5 instead of j6, k6, and m6, respectively.

Numerical values are given in Tables 15 and 16.

Table 14. Selection of Housing Tolerance Classifications for Metric Radial Ball and Roller Bearings of ABEC-1 and RBEC-1 Tolerance Classes

Design and Operating Conditions				
Rotational Conditions	Loading	Outer Ring Axial Displacement Limitations	Other Conditions	Tolerance Classification[a]
Outer ring stationary in relation to load direction	Light Normal and Heavy	Outer ring must be easily axially displaceable	Heat input through shaft	G7
			Housing split axially	H7[b]
			Housing not split axially	H6[b]
	Shock with temporary complete unloading	Transitional Range[c]		J6[b]
Load direction is indeterminate	Light and normal			
	Normal and Heavy			
	Heavy Shock		split housing not recommended	K6[b]
Outer ring rotating in relation to load direction	Light			M6[b]
	Normal and Heavy	Outer ring need not be axially displaceable		N6[b]
	Heavy		Thin wall housing not split	P6[b]

[a] For cast iron or steel housings. For housings of nonferrous alloys tighter fits may be needed.

[b] Where wider tolerances are permissible, use tolerance classifications P7, N7, M7, K7, J7, and H7, in place of P6, N6, M6, K6, J6, and H6, respectively.

[c] The tolerance zones are such that the outer ring may be either tight or loose in the housing.

Table 15. AFBMA and American National Standard Shaft Diameter and Housing Bore Tolerance Limits *ANSI/ABMA 7-1995*

Allowable Deviations of Shaft Diameter from Basic Bore Diameter, Inch																
Inches		mm														
Over	Incl.	Over	Incl.													
Base Bore Diameter				g6	h6	h5	j5	j6	k5	k6	m5	m6	n6	p6	r6	r7
0.2362	0.3937	6	10	−.0002 −.0006	0 −.0004	0 −.0002	+.0002 −.0001	+.0003 −.0001	+.0003 0		+.0005 +.0002					
0.3937	0.7087	10	18	−.0002 −.0007	0 −.0004	0 −.0003	+.0002 −.0001	+.0003 −.0001	+.0004 0		+.0006 +.0003					
0.7087	1.1811	18	30	−.0003 −.0008	0 −.0005		+.0002 −.0002	+.0004 −.0002	+.0004 +.0001		+.0007 +.0003					
1.1811	1.9685	30	50	−.0004 −.0010	0 −.0006		+.0002 −.0002	+.0004 −.0002	+.0005 +.0001	+.0007 +.0001	+.0008 +.0004	+.0010 +.0004				
1.9685	3.1496	50	80	−.0004 −.0011	0 −.0007		+.0002 −.0003	+.0005 −.0003	+.0006 +.0001	+.0008 +.0001	+.0009 +.0004	+.0012 +.0004	+.0018 +.0009			
3.1496	4.7244	80	120	−.0005 −.0013	0 −.0009		+.0002 −.0004	+.0005 −.0004	+.0007 +.0001	+.0010 +.0001	+.0011 +.0005	+.0014 +.0005	+.0019 +.0010	+.0023 +.0015		
Allowable Deviations of Housing Bore from Basic Outside Diameter of Shaft, Inch																
Basic Outside Diameter				G7	H7	H6	J7	J6	K6	K7	M6	M7	N6	N7	P6	P7
0.7087	1.1811	18	30	+.0003 +.0011	0 +.0008	0 +.0005	−.0004 +.0005	−.0002 +.0003	−.0004 +.0001	−.0006 +.0002	−.0007 +.0002	−.0008 0	−.0009 −.0004	−.0011 −.0003	−.0012 −.0007	−.0014 −.0006
1.1811	1.9685	30	50	+.0004 +.0013	0 +.0010	0 +.0006	−.0004 +.0006	−.0002 +.0004	−.0005 +.0001	−.0007 +.0003	−.0008 −.0002	−.0010 0	−.0011 −.0005	−.0013 −.0003	−.0015 −.0008	−.0017 −.0007
1.9685	3.1496	50	80	+.0004 +.0016	0 +.0012	0 +.0007	−.0005 +.0007	−.0002 +.0005	−.0006 +.0002	−.0008 +.0004	−.0009 −.0002	−.0012 0	−.0013 −.0006	−.0015 −.0004	−.0018 −.0010	−.0020 −.0008
3.1496	4.7244	80	120	+.0005 +.0019	0 +.0014	0 +.0009	−.0005 +.0009	−.0002 +.0006	−.0007 +.0002	−.0010 +.0004	−.0011 −.0002	−.0014 0	−.0015 −.0006	−.0018 −.0004	−.0020 −.0012	−.0023 −.0009
4.7244	7.0866	120	180	+.0006 +.0021	0 +.0016	0 +.0010	−.0006 +.0010	−.0003 +.0007	−.0008 +.0002	−.0011 +.0005	−.0013 −.0003	−.0016 0	−.0018 −.0008	−.0020 −.0005	−.0024 −.0014	−.0027 −.0011
7.0866	9.8425	180	250	+.0006 +.0024	0 +.0018	0 +.0011	−.0006 +.0012	−.0003 +.0009	−.0009 +.0002	−.0013 +.0005	−.0015 −.0003	−.0018 0	−.0020 −.0009	−.0024 −.0006	−.0028 −.0016	−.0031 −.0013

Based on ANSI B4.1-1967 (R1994) Preferred Limits and Fits for Cylindrical Parts. Symbols g6, h6, etc., are shaft and G7, H7, etc., hole limits designations. For larger diameters and metric values see AFBMA Standard 7.

Table 16. AFBMA and American National Standard Shaft Diameter and Housing Bore Tolerance Limits *ANSI/ABMA 7-1995*

Allowable Deviations of Shaft Diameter from Basic Bore Diameter, mm																
Inches		mm														
Over	Incl.	Over	Incl.													
Base Bore Diameter				g6	h6	h5	j5	j6	k5	k6	m5	m6	n6	p6	r6	r7
0.2362	0.3937	6	10	−.005 −.014	0 −.009	0 −.006	+.004 −.002	+.007 −.002	+.007 −.001		+.012 +.006					
0.3937	0.7087	10	18	−.006 −.017	0 −.011	0 −.008	+.005 −.003	+.008 −.003	+.009 +.001		+.015 +.007					
0.7087	1.1811	18	30	−.007 −.020	0 −.013		+.005 −.004	+.009 −.004	+.011 +.002		+.017 +.008					
1.1811	1.9685	30	50	−.009 −.025	0 −.016		+.006 −.005	+.011 −.005	+.013 +.002	+.018 +.002	+.020 +.009	+.025 +.009				
1.9685	3.1496	50	80	−.010 −.029	0 −.019		+.006 −.007	+.012 −.007	+.015 +.002	+.021 +.002	+.024 +.011	+.030 +.011	+.039 +.020			
3.1496	4.7244	80	120	−.012 −.034	0 −.022		+.006 −.009	+.013 −.009	+.018 +.003	+.025 +.003	+.028 +.013	+.035 +.013	+.045 +.023	+.059 +.037		
Allowable Deviations of Housing Bore from Basic Outside Diameter of Shaft, mm																
Basic Outside Diameter				G7	H7	H6	J7	J6	K6	K7	M6	M7	N6	N7	P6	P7
.7086	1.1811	18	30	+.007 +.028	0 +.021	0 +.013	−.009 +.012	−.005 +.008	−.011 +.002	−.015 +.006	−.017 −.004	−.021 0	−.024 −.011	−.028 −.007	−.031 −.018	−.035 −.014
1.1811	1.9685	30	50	+.009 +.034	0 +.025	0 +.016	−.011 +.014	−.006 +.010	−.013 +.003	−.018 +.007	−.020 −.004	−.025 0	−.028 −.012	−.033 −.008	−.037 −.021	−.042 −.017
1.9685	3.1496	50	80	+.010 +.040	0 +.030	0 +.019	−.012 +.018	−.006 +.013	−.015 +.004	−.021 +.009	−.024 −.005	−.030 0	−.033 −.014	−.039 −.009	−.045 −.026	−.051 −.021
3.1496	4.7244	80	120	+.012 +.047	0 +.035	0 +.022	−.013 +.022	−.006 +.016	−.018 +.004	−.025 +.010	−.028 −.006	−.035 0	−.038 −.016	−.045 −.010	−.052 −.030	−.059 −.024
4.7244	7.0866	120	180	+.014 +.054	0 +.040	0 +.025	−.014 +.026	−.007 +.018	−.021 +.004	−.028 +.012	−.033 −.008	−.040 0	−.045 −.020	−.052 −.012	−.061 −.036	−.068 −.028
7.0866	9.8425	180	250	+.015 +.061	0 +.046	0 +.029	−.016 +.030	−.007 +.022	−.024 +.005	−.033 +.013	−.037 −.008	−.046 0	−.051 −.022	−.060 −.014	−.070 −.041	−.079 −.033

Based on ANSI B4.1-1967 (R1994) Preferred Limits and Fits for Cylindrical Parts. Symbols g6, h6, etc., are shaft and G7, H7, etc., hole limits designations. For larger diameters and metric values see AFBMA Standard 7.

Design and Installation Considerations.—Interference fitting will reduce bearing radial internal clearance, so it is recommended that prospective users consult bearing manufacturers to make certain that the required bearings are correctly specified to satisfy all mounting, environmental and other operating conditions and requirements. This check is particularly necessary where heat sources in associated parts may further diminish bearing clearances in operation.

Standard values of radial internal clearances of radial bearings are listed in AFBMA-ANSI Standard 20.

Allowance for Axial Displacement.—Consideration should be given to axial displacement of bearing components owing to thermal expansion or contraction of associated parts. Displacement may be accommodated either by the internal construction of the bearing or by allowing one of the bearing rings to be axially displace-able. For unusual applications consult bearing manufacturers.

Needle Roller Bearing Fitting and Mounting Practice.—The tolerance limits required for shaft and housing seat diameters for needle roller beatings with inner and outer rings as well as limits for raceway diameters where inner or outer rings or both are omitted and rollers operate directly upon these surfaces are given in Tables 17 through 20, inclusive. Unusual design and operating conditions may require a departure from these practices. In such cases, bearing manufacturers should be consulted.

Needle Roller Bearings, Drawn Cup: These beatings without inner ring, Types NIB, NB, NIBM, NBM, NIY, NY, NIYM, NYM, NIH, NH, NIHM, NHM, and Inner Rings, Type NIR depend on the housings into which they are pressed for their size and shape. Therefore, the housings must not only have the proper bore dimensions but also must have sufficient strength. Tables 17 and 18, show the bore tolerance limits for rigid housings such as those made from cast iron or steel of heavy radial section equal to or greater than the ring gage section given in AFBMA Standard 4, 1984. The bearing manufacturers should be consulted for recommendations if the housings must be of lower strength materials such as aluminum or even of steel of thin radial section. The shape of the housing bores should be such that when the mean bore diameter of a housing is measured in each of several radial planes, the maximum difference between these mean diameters should not exceed 0.0005 inch (0.013 mm) or one-half the housing bore tolerance limit, if smaller. Also, the radial deviation from circular form should not exceed 0.00025 inch (0.006 mm). The housing bore surface finish should not exceed 125 micro-inches (3.2 micrometers) arithmetical average.

Table 17. AFBMA and American National Standard Tolerance Limits for Shaft Raceway and Housing Bore Diameters—Needle Roller Bearings, Drawn Cup, Without Inner Ring, Inch Types NIB, NIBM, NIY, NIYM, NIH, and NIHM

ANSI/ABMA 18.2-1982 (R1993)

Basic Bore Diameter under Needle Rollers, F_w		Shaft Raceway Diameter[a] Allowable Deviation from F_w		Basic Outside Diameter, D		Housing Bore Diameter[a] Allowable Deviation from D	
Inch		Inch		Inch		Inch	
Over	Incl.	High	Low	Over	Incl.	Low	High
OUTER RING STATIONARY RELATIVE TO LOAD							
0.1875	1.8750	+0	−0.0005	0.3750	4.0000	−0.0005	+0.0005
1.8750	3.5000	+0	−0.0006	…	…	…	…
OUTER RING ROTATING RELATIVE TO LOAD							
0.1875	1.8750	−0.0005	−0.0010	0.3750	4.0000	−0.0010	+0
1.8750	3.5000	−0.0005	−0.0011	…	…	…	…

[a] See text for additional requirements.

For bearing tolerances, see Table 9.

Table 18. AFBMA and American National Standard Tolerance Limits for Shaft Raceway and Housing Bore Diameters—Needle Roller Bearings, Drawn Cup, Without Inner Ring, Metric Types NB, NBM, NY, NYM, NH, and NHM

ANSI/ABMA 18.1-1982 (R1994)

Basic Bore Diameter Under Needle Rollers, F_w				Shaft Raceway Diameter-[a]Allowable Deviation from F_w		Basic Outside Diameter, D				Housing Bore Diameter-[a]Allowable Deviation from D	
OUTER RING STATIONARY RELATIVE TO LOAD											
mm		Inch		ANSI h6, Inch		mm		Inch		ANSI N7, Inch	
Over	Incl.	Over	Incl.	High	Low	Over	Incl.	Over	Incl.	Low	High
3	6	0.1181	0.2362	+0	−0.0003	6	10	0.2362	0.3937	−0.0007	−0.0002
6	10	0.2362	0.3937	+0	−0.0004	10	18	0.3937	0.7087	−0.0009	−0.0002
10	18	0.3937	0.7087	+0	−0.0004	18	30	0.7087	1.1811	−0.0011	−0.0003
18	30	0.7087	1.1811	+0	−0.0005	30	50	1.1811	1.9685	−0.0013	−0.0003
30	50	1.1811	1.9685	+0	−0.0006	50	80	1.9685	3.1496	−0.0015	−0.0004
50	80	1.9685	3.1496	+0	−0.0007	…	…	…	…	…	…
OUTER RING ROTATING RELATIVE TO LOAD											
mm		Inch		ANSI f6, Inch		mm		Inch		ANSI R7, Inch	
Over	Incl.	Over	Incl.	High	Low	Over	Incl.	Over	Incl.	Low	High
3	6	0.1181	0.2362	−0.0004	−0.0007	6	10	0.2362	0.3937	−0.0011	−0.0005
6	10	0.2362	0.3937	−0.0005	−0.0009	10	18	0.3937	0.7087	−0.0013	−0.0006
10	18	0.3937	0.7087	−0.0006	−0.0011	18	30	0.7087	1.1811	−0.0016	−0.0008
18	30	0.7087	1.1811	−0.0008	−0.0013	30	50	1.1811	1.9685	−0.0020	−0.0010
30	50	1.1811	1.9685	−0.0010	−0.0016	50	65	1.9685	2.5591	−0.0024	−0.0012
50	80	1.9685	3.1496	−0.0012	−0.0019	65	80	2.5591	3.1496	−0.0024	−0.0013

For bearing tolerances, see Table 10.

Table 19. AFBMA and American National Standard Tolerance Limits for Shaft Raceway and Housing Bore Diameters—Needle Roller Bearings, With Cage, Machined Ring, Without Inner Ring, Inch Type NIA

ANSI/ABMA 18.2-1982 (R1993)

Basic Bore Diameter under Needle Rollers, F_w		Shaft Raceway Diameter[a] Allowable Deviation from F_w		Basic Outside Diameter, D		Housing Bore Diameter[a] Allowable Deviation from D	
OUTER RING STATIONARY RELATIVE TO LOAD							
Inch		ANSI h6, Inch		Inch		ANSI H7, Inch	
Over	Incl.	High	Low	Over	Incl.	Low	High
0.2362	0.3937	+0	−0.0004	0.3937	0.7087	+0	+0.0007
0.3937	0.7087	+0	−0.0004	0.7087	1.1811	+0	+0.0008
0.7087	1.1811	+0	−0.0005	1.1811	1.9685	+0	+0.0010
1.1811	1.9685	+0	−0.0006	1.9685	3.1496	+0	+0.0012
1.9685	3.1496	+0	−0.0007	3.1496	4.7244	+0	+0.0014
3.1496	4.7244	+0	−0.0009	4.7244	7.0866	+0	+0.0016
4.7244	7.0866	+0	−0.0010	7.0866	9.8425	+0	+0.0018
7.0866	9.8425	+0	−0.0011	9.8425	12.4016	+0	+0.0020
OUTER RING ROTATING RELATIVE TO LOAD							
Inch		ANSI f6, Inch		Inch		ANSI N7, Inch	
Over	Incl.	High	Low	Over	Incl.	Low	High
0.2362	0.3937	−0.0005	−0.0009	0.3937	0.7087	−0.0009	−0.0002
0.3937	0.7087	−0.0006	−0.0011	0.7087	1.1811	−0.0011	−0.0003
0.7087	1.1811	−0.0008	−0.0013	1.1811	1.9685	−0.0013	−0.0003
1.1811	1.9685	−0.0010	−0.0016	1.9685	3.1496	−0.0015	−0.0004
1.9685	3.1496	−0.0012	−0.0019	3.1496	4.7244	−0.0018	−0.0004
3.1496	4.7244	−0.0014	−0.0023	4.7244	7.0866	−0.0020	−0.0005
4.7244	7.0866	−0.0016	−0.0027	7.0866	9.8425	−0.0024	−0.0006
7.0866	9.8425	−0.0020	−0.0031	9.8425	11.2205	−0.0026	−0.0006

[a] See text for additional requirements.

For bearing tolerances, see Table 11.

Table 20. AFBMA and American National Standard Tolerance Limits for Shaft Diameters—Needle Roller Bearing Inner Rings, Inch Type NIR (Used with Bearing Type NIA) *ANSI/ABMA 18.2-1982 (R1993)*

Basic Bore, *d*		Shaft Diameter[a]			
		Shaft Rotating Relative to Load, Outer Ring Stationary Relative to Load Allowable Deviation from *d*		Shaft Stationary Relative to Load, Outer Ring Rotating Relative to Load Allowable Deviation from *d*	
Inch		ANSI m5, Inch		ANSI g6, Inch	
Over	Incl.	High	Low	High	Low
0.2362	0.3937	+0.0005	+0.0002	−0.0002	−0.0006
0.3937	0.7087	+0.0006	+0.0003	−0.0002	−0.0007
0.7087	1.1811	+0.0007	+0.0003	−0.0003	−0.0008
1.1811	1.9685	+0.0008	+0.0004	−0.0004	−0.0010
1.9685	3.1496	+0.0009	+0.0004	−0.0004	−0.0011
3.1496	4.7244	+0.0011	+0.0005	−0.0005	−0.0013
4.7244	7.0866	+0.0013	+0.0006	−0.0006	−0.0015
7.0866	9.8425	+0.0015	+0.0007	−0.0006	−0.0017

[a] See text for additional requirements.

For inner ring tolerance limits, see Table 12.

Most needle roller bearings do not use inner rings, but operate directly on the surfaces of shafts. When shafts are used as inner raceways, they should be made of bearing quality steel hardened to Rockwell C 58 minimum. Tables 14 and 18 show the shaft raceway tolerance limits and Table 20 shows the shaft seat tolerance limits when inner rings are used. However, whether the shaft surfaces are used as inner raceways or as seats for inner rings, the mean outside diameter of the shaft surface in each of several radial planes should be determined. The difference between these mean diameters should not exceed 0.0003 inch (0.008 mm) or one-half the diameter tolerance limit, if smaller. The radial deviation from circular form should not exceed 0.0001 inch (0.0025 mm), for diameters up to and including 1 in. (25.4 mm). Above one inch the allowable deviation is 0.0001 times the shaft diameter. The surface finish should not exceed 16 micro-inches (0.4 micrometer) arithmetical average. The housing bore and shaft diameter tolerance limits depend upon whether the load rotates relative to the shaft or the housing.

Needle Roller Bearing With Cage, Machined Ring, Without Inner Ring: The following covers needle roller bearings Type NIA and inner rings Type NIR. The shape of the housing bores should be such that when the mean bore diameter of a housing is measured in each of several radial planes, the maximum difference between these mean diameters does not exceed 0.0005 inch (0.013 mm) or one-half the housing bore tolerance limit, if smaller. Also, the radial deviation from circular form should not exceed 0.00025 inch (0.006 mm). The housing bore surface finish should not exceed 125 micro-inches (3.2 micrometers) arithmetical average. Table 20 shows the housing bore tolerance limits.

When shafts are used as inner raceways their requirements are the same as those given above for Needle Roller Bearings, Drawn Cup. Table 19 shows the shaft raceway tolerance limits and Table 20 shows the shaft seat tolerance limits when inner rings are used.

Needle Roller and Cage Assemblies, Types NIM and NM: For information concerning boundary dimensions, tolerance limits, and fitting and mounting practice, reference should be made to ANSI/ABMA 18.1-1982 (R1994) and ANSI/ABMA 18.2-1982 (R1993).

Bearing Mounting Practice.—Because of their inherent design and material rigidity, rolling contact bearings must be mounted with careful control of their alignment and runout. Medium-speed or slower (400,000 *DN* values or less where *D* is the bearing bore in millimeters and N is the beating speed in revolutions per minute), and medium to light load (C/P values of 7 or greater where C is the beating specific dynamic capacity in pounds and P is the average beating load in pounds) applications can endure misalignments equivalent to those acceptable for high-capacity, precision journal beatings utilizing hard bearing materials such as silver, copper-lead, or aluminum. In no case, however, should the maximum shaft deflection exceed .001 inch per inch for well-crowned roller bearings, and .003 inch per inch for deep-groove ball-beatings. Except for self-aligning ball-bearings and spherical or barrel roller bearings, all other types require shaft alignments with deflections no greater than .0002 inch per inch. With preloaded ball bearings, this same limit is recommended as a maximum. Close-clearance tapered bearings or thrust beatings of most types require the same shaft alignment also.

Of major importance for all bearings requiring good reliability, is the location of the races on the shaft and in the housing.

Assembly methods must insure: 1) that the faces are square, before the cavity is closed;

2) that the cover face is square to the shoulder and pulled in evenly; and 3) that it will be located by a face parallel to it when finally seated against the housing.

These requirements are shown in the accompanying figure. In applications not controlled by automatic tooling with closely controlled fixtures and bolt torquing mechanisms, races should be checked for squareness by sweeping with a dial indicator mounted as shown below. For commercial applications with moderate life and reliability requirements, outer race runouts should be held to .0005 inch per inch of radius and inner race runout to .0004 inch per inch of radius. In preloaded and precision applications, these tolerances must be cut in half. In regard to the question of alignment, it must be recognized that rolling-contact bearings, being made of fully-hardened steel, do not wear in as may certain journal bearings when carefully applied and initially operated. Likewise, rolling contact bearings absorb relatively little deflection when loaded to C/P values of 6 or less. At such stress levels the rolling element-race deformation is generally not over .0002 inch. Consequently, proper mounting and control of shaft deflections are imperative for reliable bearing performance. Aside from inadequate lubrication, these factors are the most frequent causes of premature bearing failures.

Mountings for Precision and Quiet-running Applications.—In applications of rolling-element bearings where vibration or smoothness of operation is critical, special precautions must be taken to eliminate those conditions which can serve to initiate radial and axial motions. These exciting forces can result in shaft excursions which are in resonance with shaft or housing components over a range of frequencies from well below shaft speed to as much as 100 times above it. The more sensitive the configuration, the greater is the need for precision bearings and mountings to be used.

Precision bearings are normally made to much closer tolerances than standard and therefore benefit from better finishing techniques. Special inspection operations are required, however, to provide races and rolling elements with smoothness and runouts compatible with the needs of the application. Similarly, shafts and housings must be carefully controlled.

Among the important elements to be controlled are shaft, race, and housing roundness; squareness of faces, diameters, shoulders, and rolling paths. Though not readily appreciated, grinding chatter, lobular and compensating out-of-roundness, waviness, and flats of less than .0005 inch deviation from the average or mean diameter can cause significant roughness. To detect these and insure the selection of good pieces, three-point electronic indicator inspection must be made. For ultra-precise or quiet applications, pieces are often checked on a “Talyrond” or a similar continuous recording instrument capable of measuring to within a few millionths of an inch. Though this may seem extreme, it has been found that shaft deformities will be reflected through inner races shrunk onto them. Similarly, tight-fit outer races pick up significant deviations in housings. In many instrument and in missile guidance applications, such deviations and deformities may have to be limited to less than .00002 inch.

In most of these precision applications, bearings are used with rolling elements controlled to less than 5 millionths of an inch deviation from roundness and within the same range for diameter.

Special attention is required both in housing design and in assembly of the bearing to shaft and housing. Housing response to axial excursions forced by bearing wobble (which in itself is a result of out-of-square mounting) has been found to be a major source of small electric and other rotating equipment noise and howl. Stiffer, more massive housings and careful alignment of bearing races can make significant improvements in applications where noise or vibration has been found to be objectionable.

Commercial Application Alignment Tolerances

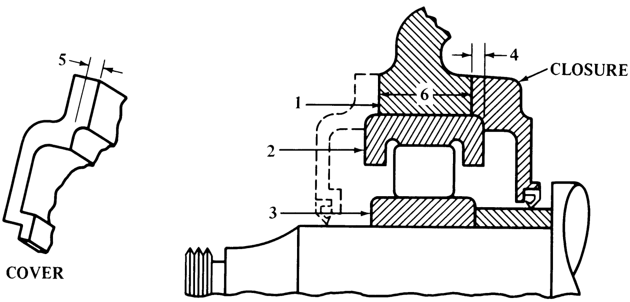
1. **Housing Face Runout** — Square to shaft center within .0004 inch/inch of radius full indicator-eading.

2. **Outer Race Face Runout** — Square to shaft center within .0004 inch/inch of radius full indicator reading and complementary to the housing runout (not opposed).

3. **Inner Race Face Runout** — Square to shaft center within .0003 inch/inch of radius full indicator reading.

4. and 5. **Cover and Closure Mounting Face Parallelism** — Parallel within .001.

6. **Housing Mounting Face Parallelism** — Parallel within .001



Squareness and Alignment.—In addition to the limits for roundness and wall variation of the races and their supports, squareness of end faces and shoulders must be closely controlled. Tolerances of .0001 inch full indicator reading per inch of diameter are normally required for end faces and shoulders, with appropriately selected limits for fillet eccentricities. The latter must also fall within specified limits for radii tolerances to prevent interference and the resulting cocking of the race. Reference should be made to the bearing dimension tables which list corner radii for typical bearings. Shoulders must also be of a sufficient height to insure proper support for the races, since they are of hardened steel and are less capable of absorbing shock loads and abuse. The general subject of squareness and alignment is of primary importance to the life of rolling element bearings.

The following recommendations for shaft and housing design are given by the New Departure Division of General Motors Corporation:*

"As a rule, there is little trouble experienced with inaccuracies in shafts. Bearings seats and locating shoulders are turned and ground to size with the shaft held on centers and, with ordinary care, there is small chance for serious out-of-roundness or taper. Shaft shoul-

* New Departure Handbook. Vol. II — 1951.

ders should present sufficient surface in contact with the bearing face to assure positive and accurate location.

"Where an undercut must be made for wheel runout in grinding a bearing seat, care should be exercised that no sharp corners are left, for it is at such points that fatigue is most likely to result in shaft breakage. It is best to undercut as little as possible and to have the undercut end in a fillet instead of a sharp corner.

"Where clamping nuts are to be used, it is important to cut the threads as true and square as possible in order to insure even pressure at all points on the bearing inner ring faces when the nuts are set up tight. It is also important not to cut threads so far into the bearing seat as to leave part of the inner ring unsupported or carried on the threads. Excessive deflection is usually the result of improperly designed or undersized machine parts. With a weak shaft, it is possible to seriously affect bearing operation through misalignment due to shaft deflection. Where shafts are comparatively long, the diameter between bearings must be great enough to properly resist bending. In general, the use of more than two bearings on a single shaft should be avoided, owing to the difficulty of securing accurate alignment. With bearings mounted close to each other, this can result in extremely heavy bearing loads.

"Design is as important as careful machining in construction of accurate bearing housings. There should be plenty of metal in the wall sections and large, thin areas should be avoided as much as possible, since they are likely to permit deflection of the boring tool when the housing is being finish-machined.

"Wherever possible, it is best to design a housing so that the radial load placed on the bearing is transmitted as directly as possible to the wall or rib supporting the housing. Diaphragm walls connecting an offset housing to the main wall or side of a machine are apt to deflect unless made thick and well braced.

"When two bearings are to be mounted opposed, but in separate housings, the housings should be so reinforced with fins or webs as to prevent deflection due to the axial load under which the bearings are opposed.

"Where housings are deep and considerable overhang of the boring tool is required, there is a tendency to produce out-of-roundness and taper, unless the tool is very rigid and light finishing cuts are taken. In a too roughly bored housing there is a possibility for the ridges of metal to peen down under load, thus eventually resulting in too loose a fit for the bearing outer ring."

Soft Metal and Resilient Housings.—In applications relying on bearing housings made of soft materials (aluminum, magnesium, light sheet metal, etc.) or those which lose their fit because of differential thermal expansion, outer race mounting must be approached in a cautious manner. Of first importance is the determination of the possible consequences of race loosening and turning. In conjunction with this, the type of loading must be considered for it may serve to magnify the effect of race loosening. It must be remembered that generally, balancing processes do not insure zero unbalance at operating speeds, but rather an "acceptable" maximum. This force exerted by the rotating element on the outer race can initiate a precession which will aggravate the race loosening problem by causing further attrition through wear, pounding, and abrasion. Since this force is generally of an order greater than the friction forces in effect between the outer race, housing, and closures (retaining nuts also), no foolproof method can be recommended for securing outer races in housings which deform significantly under load or after appreciable service wear. Though many such "fixes" are offered, the only sure solution is to press the race into a housing of sufficient stiffness with the heaviest fit consistent with the installed and operating clearances. In many cases, inserts, or liners of cast iron or steel are provided to maintain the desired fit and increase useful life of both bearing and housing.

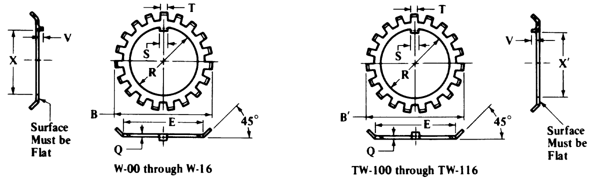
Quiet or Vibration-free Mountings.—In seeming contradiction is the approach to bearing mountings in which all shaft or rotating element excursions must be isolated from the frame, housing, or supporting structure. Here bearing outer races are often supported on elastomeric or metallic springs. Fundamentally, this is an isolation problem and must be approached with caution to insure solution of the primary bearing objective — location and restraint of the rotating body, as well as the reduction or elimination of the dynamic problem. Again, the danger of skidding rolling elements must be considered and reference to the resident engineers or sales engineers of the numerous bearing companies is recommended, as this problem generally develops requirements for special, or non-catalog-type bearings.

General Mounting Precautions.—Since the last operations involving the bearing application — mounting and closing — have such important effects on bearing performance, durability, and reliability, it must be cautioned that more bearings are abused or “killed” in this early stage of their life than wear out or “die” under conditions for which they were designed. Hammer and chisel “mechanics” invariably handle bearings as though no blow could be too hard, no dirt too abrasive, and no misalignment of any consequence. Proper tools, fixtures, and techniques are a must for rolling bearing application, and it is the responsibility of the design engineer to provide for this in his design, advisory notes, mounting instructions, and service manuals. Nicks, dents, scores, scratches, corrosion staining, and dirt must be avoided if reliability, long life, and smooth running are to be expected of rolling bearings. All manufacturers have pertinent service instructions available for the bearing user. These should be followed for best performance. In a later section, methods for inspecting bearings and descriptions of most common bearing deficiencies will be given.

Seating Fits for Bearings.—Anti-Friction Bearing Manufacturers Association (AFBMA) standard shaft and housing bearing seat tolerances are given in Tables 12 through 17, inclusive.

Clamping and Retaining Methods.—Various methods of clamping bearings to prevent axial movement on the shaft are employed, one of the most common being a nut screwed on the end of the shaft and held in place by a tongued lock washer (see Table 21). The shaft thread for the clamping nut (see Table 22) should be cut in accurate relation to bearing seats and shoulders if bearing stresses are to be avoided. The threads used are of American National Form, Class 3; special diameters and data for these are given in Tables 23 and 24. Where somewhat closer than average accuracy is required, the washers and locknut faces may be obtained ground for closer alignment with the threads. For a high degree of accuracy the shaft threads are ground and a more precise clamping means is employed. Where a bearing inner ring is to be clamped, it is important to provide a sufficiently high shoulder on the shaft to locate the bearing positively and accurately. If the difference between bearing bore and maximum shaft diameter gives a low shoulder which would enter the corner of the radius of the bearing, a shoulder ring that extends above the shoulder and well into the shaft corner is employed. A shoulder ring with snap wire fitting into a groove in the shaft is sometimes used where no locating shaft shoulder is present. A snap ring fitting into a groove is frequently employed to prevent endwise movement of the bearing away from the locating shoulder where tight clamping is not required. Such a retaining ring should not be used where a slot in the shaft surface might lead to fatigue failure. Snap rings are also used to locate the outer bearing ring in the housing. Dimensions of snap rings used for this latter purpose are given in AFBMA and ANSI standards.

Table 21. AFBMA Standard Lockwashers (Series W-00) for Ball Bearings and Cylindrical and Spherical Roller Bearings and (Series TW-100) for Tapered Roller Bearings. Inch Design.



Type W No.	Q	Type TW No.	Q	Tangs No.	Tangs Width[a] T	Key Project.[a] V	Key Width S Min.	Key Width S Max.	Key X Min.	Key X Max.	Key X′ Min.	Key X′ Max.	Bore R Min.	Bore R Max.	Diameter E	Diameter Tol.	Dia. Over Tangs. Max. B	Dia. Over Tangs. Max. B′
W-00	.032	TW-100	.032	9	.120	.031	.110	.120	.334	.359	.334	.359	.406	0.421	0.625	+.015	0.875	0.891
W-01	.032	TW-101	.032	9	.120	.031	.110	.120	.412	.437	.412	.437	.484	.499	0.719	+.015	1.016	1.031
W-02	.032	TW-102	.048	11	.120	.031	.110	.120	.529	.554	.513	.538	.601	.616	0.813	+.015	1.156	1.156
W-03	.032	TW-103	.048	11	.120	.031	.110	.120	.607	.632	.591	.616	.679	.694	0.938	+.015	1.328	1.344
W-04	.032	TW-104	.048	11	.166	.031	.156	.176	.729	.754	.713	.738	.801	.816	1.125	+.015	1.531	1.563
W-05	.040	TW-105	.052	13	.166	.047	.156	.176	.909	.939	.897	.927	.989	1.009	1.281	+.015	1.719	1.703
W-06	.040	TW-106	.052	13	.166	.047	.156	.176	1.093	1.128	1.081	1.116	1.193	1.213	1.500	+.015	1.922	1.953
		TW-065	.052	15	.166	…	.156	.176	…	…	1.221	1.256	1.333	1.353	1.813	+.015	…	2.234
W-07	.040	TW-107	.052	15	.166	.047	.156	.176	1.296	1.331	1.284	1.319	1.396	1.416	1.813	+.015	2.250	2.250
W-08	.048	TW-108	.062	15	.234	.047	.250	.290	1.475	1.510	1.461	1.496	1.583	1.603	2.000	+.030	2.469	2.484
W-09	.048	TW-109	.062	17	.234	.062	.250	.290	1.684	1.724	1.670	1.710	1.792	1.817	2.281	+.030	2.734	2.719
W-10	.048	TW-110	.062	17	.234	.062	.250	.290	1.884	1.924	1.870	1.910	1.992	2.017	2.438	+.030	2.922	2.922
W-11	.053	TW-111	.062	17	.234	.062	.250	.290	2.069	2.109	2.060	2.100	2.182	2.207	2.656	±.030	3.109	3.094
W-12	.053	TW-112	.072	17	.234	.062	.250	.290	2.267	2.307	2.248	2.288	2.400	2.425	2.844	+.030	3.344	3.328
W-13	.053	TW-113	.072	19	.234	.062	.250	.290	2.455	2.495	2.436	2.476	2.588	2.613	3.063	+.030	3.578	3.563
W-14	.053	TW-114	.072	19	.234	.094	.250	.290	2.658	2.698	2.639	2.679	2.791	2.816	3.313	+.030	3.828	3.813
W-15	.062	TW-115	.085	19	.328	.094	.250	.290	2.831	2.876	2.808	2.853	2.973	3.003	3.563	+.030	4.109	4.047
W-16	.062	TW-116	.085	19	.328	.094	.313	.353	3.035	3.080	3.012	3.057	3.177	3.207	3.844	+.030	4.375	4.391

[a] *Tolerances:* On width, *T*, −.010 inch for Types W-00 to W-03 and TW-100 to TW-103; −.020 inch for W-04 to W-07 and TW-104 to TW-107; −.030 inch for all others shown. On Projection *V*, +.031 inch for all sizes up through W-13 and TW-113; +.062 inch for all others shown.

All dimensions in inches. For dimensions in millimeters, multiply inch values by 25.4 and round result to two decimal places.

Data for sizes larger than shown are given in ANSI/AFBMA Standard 8.2-1991.

Table 22. AFBMA Standard Locknuts (Series N-00) for Ball Bearings and Cylindrical and Spherical Roller Bearings and (Series TN-00) for Tapered Roller Bearings. Inch Design.

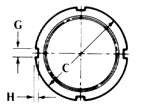
Runout and parallelism of faces measured on a tight fitting threaded arbor.

N-00 to N-06 = .002 Max.
N-07 to AN-15 = .004 Max.

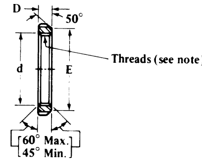
TN-065 to TAN-15 = .002 Max.

Surface Finish Note

TN-065 to TN-11, 100μ in., max.
TN-12 to TAN-15, 120μ in., max.



N-00 through AN-15
TN-065 through TAN-15



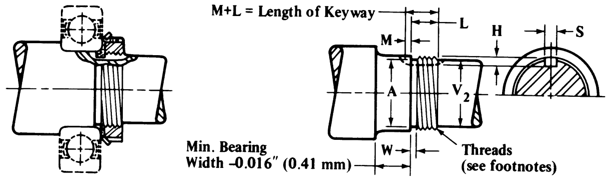
BB & RB Nut No.	TRB Nut No.	Thds. per Inch	Thread Minor Deam.		Thread Pitch Dia.		Thd. Major Dia. *d*	Outside Dia. *C*	Face Dia. *E*		Slot dimension Width *G*		Slot dimension Height *H*	Thickness *D*	
			Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Max.	Min.	Max.
N-00	—	32	0.3572	0.3606	0.3707	0.3733	0.391	0.755	.605	.625	.120	.130	.073	.209	.229
N-01	—	32	0.4352	0.4386	0.4487	0.4513	0.469	0.880	.699	.719	.120	.130	.073	.303	.323
N-02	—	32	0.5522	0.5556	0.5657	0.5687	0.586	1.005	.793	.813	.120	.130	.104	.303	.323
N-03	—	32	0.6302	0.6336	0.6437	0.6467	0.664	1.130	.918	.938	.120	.130	.104	.334	.354
N-04	—	32	0.7472	0.7506	0.7607	0.7641	0.781	1.380	1.105	1.125	.178	.198	.104	.365	.385
N-05	—	32	0.9352	0.9386	0.9487	0.9521	0.969	1.568	1.261	1.281	.178	.198	.104	.396	.416
N-06	—	18	1.1129	1.1189	1.1369	1.1409	1.173	1.755	1.480	1.500	.178	.198	.104	.396	.416
	TN-065	18	1.2524	1.2584	1.2764	1.2804	1.312	2.068	1.793	1.813	.178	.198	.104	.428	.448
N-07	TN-07	18	1.3159	1.3219	1.3399	1.3439	1.376	2.068	1.793	1.813	.178	.198	.104	.428	.448
N-08	TN-08	18	1.5029	1.5089	1.5269	1.5314	1.563	2.255	1.980	2.000	.240	.260	.104	.428	.448
N-09	TN-09	18	1.7069	1.7129	1.7309	1.7354	1.767	2.536	2.261	2.281	.240	.260	.104	.428	.448
N-10	TN-10	18	1.9069	1.9129	1.9309	1.9354	1.967	2.693	2.418	2.438	.240	.260	.104	.490	.510
N-11	TN-11	18	2.0969	2.1029	2.1209	2.1260	2.157	2.974	2.636	2.656	.240	.260	.135	.490	.510
N-12	TN-12	18	2.2999	2.3059	2.3239	2.3290	2.360	3.161	2.824	2.844	.240	.260	.135	.521	.541
N-13	TN-13	18	2.4879	2.4949	2.5119	2.5170	2.548	3.380	3.043	3.063	.240	.260	.135	.553	.573
N-14	TN-14	18	2.6909	2.6969	2.7149	2.7200	2.751	3.630	3.283	3.313	.240	.260	.135	.553	.573
AN-15	TAN-15	12	2.8428	2.8518	2.8789	2.8843	2.933	3.880	3.533	3.563	.360	.385	.135	.584	.604

All dimensions in inches. For dimensions in millimeters, multiply inch values, except thread diameters, by 25.4 and round result to two decimal places.

Threads are American National form, Class 3.

Typical steels for locknuts are: AISI, C1015, C1018, C1020, C1025, C1035, C1117, C1118, C1212, C1213, and C1215. Minimum hardness, tensile strength, yield strength and elongation are given in ANSI/ABMA 8.2-1991 which also lists larger sizes of locknuts.

Table 23. AFBMA Standard for Shafts for Locknuts (series N-00) for Ball Bearings and Cylindrical and Spherical Roller Bearings. Inch Design.

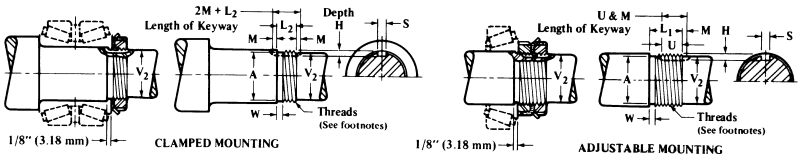


Locknut Number	Bearing Bore	V_2	Threads[a]					Relief		Keyway		
			No. per inch	Major Dia.	Pitch Dia.	Minor Dia.	Length L	Dia. A	Width W	Depth H	Width S	M
		Max.		Max.	Max.	Max.	Max.	Max.	Max.	Min.	Min.	Min.
N-00	0.3937	0.312	32	0.391	0.3707	0.3527	0.297	0.3421	0.078	0.062	0.125	0.094
N-01	0.4724	0.406	32	0.469	0.4487	0.4307	0.391	0.4201	0.078	0.062	0.125	0.094
N-02	0.5906	0.500	32	0.586	0.5657	0.5477	0.391	0.5371	0.078	0.078	0.125	0.094
N-03	0.6693	0.562	32	0.664	0.6437	0.6257	0.422	0.6151	0.078	0.078	0.125	0.094
N-04	0.7874	0.719	32	0.781	0.7607	0.7427	0.453	0.7321	0.078	0.078	0.188	0.094
N-05	0.9843	0.875	32	0.969	0.9487	0.9307	0.484	0.9201	0.078	0.094	0.188	0.125
N-06	1.1811	1.062	18	1.173	1.1369	1.1048	0.484	1.0942	0.109	0.094	0.188	0.125
N-07	1.3780	1.250	18	1.376	1.3399	1.3078	0.516	1.2972	0.109	0.094	0.188	0.125
N-08	1.5748	1.469	18	1.563	1.5269	1.4948	0.547	1.4842	0.109	0.094	0.312	0.125
N-09	1.7717	1.688	18	1.767	1.7309	1.6988	0.547	1.6882	0.141	0.094	0.312	0.156
N-10	1.9685	1.875	18	1.967	1.9309	1.8988	0.609	1.8882	0.141	0.094	0.312	0.156
N-11	2.1654	2.062	18	2.157	2.1209	2.0888	0.609	2.0782	0.141	0.125	0.312	0.156
N-12	2.3622	2.250	18	2.360	2.3239	2.2918	0.641	2.2812	0.141	0.125	0.312	0.156
N-13	2.5591	2.438	18	2.548	2.5119	2.4798	0.672	2.4692	0.141	0.125	0.312	0.156
N-14	2.7559	2.625	18	2.751	2.7149	2.6828	0.672	2.6722	0.141	0.125	0.312	0.250
AN-15	2.9528	2.781	12	2.933	2.8789	2.8308	0.703	2.8095	0.172	0.125	0.312	0.250
AN-16	3.1496	3.000	12	3.137	3.0829	3.0348	0.703	3.0135	0.172	0.125	0.375	0.250

[a] Threads are American National form Class 3.

All dimensions in inches. For dimensions in millimeters, multiply inch values, except thread diameters, by 25.4 and round result to two decimal places. See footnote to Table 24 for material other than stteel.For sizes larger than shown, see ANSI/ABMA 8.2-1991.

Table 24. AFBMA Standard for Shafts for Tapered Roller Bearing Locknuts. Inch Design.



Locknut Number	Bearing Bore	V_2	Threads[a]						Relief		Keyway			
			No. per inch	Major Dia.	Pitch Dia.	Minor Dia.	Length L_1	Length L_2	Dia. A	Width W	Depth H	Width S	M	U
		Max.		Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Min.	Min.
N-00	0.3937	0.312	32	0.391	0.3707	0.3527	0.609	0.391	0.3421	0.078	0.094	0.125	0.094	0.469
N-01	0.4724	0.406	32	0.469	0.4487	0.4307	0.797	0.484	0.4201	0.078	0.094	0.125	0.094	0.562
N-02	0.5906	0.500	32	0.586	0.5657	0.5477	0.828	0.516	0.5371	0.078	0.094	0.125	0.094	0.594
N-03	0.6693	0.562	32	0.664	0.6437	0.6257	0.891	0.547	0.6151	0.078	0.078	0.125	0.094	0.625
N-04	0.7874	0.703	32	0.781	0.7607	0.7427	0.922	0.547	0.7321	0.078	0.094	0.188	0.094	0.625
N-05	0.9843	0.875	32	0.969	0.9487	0.9307	1.016	0.609	0.9201	0.078	0.125	0.188	0.125	0.719
N-06	1.1811	1.062	18	1.173	1.1369	1.1048	1.016	0.609	1.0942	0.109	0.125	0.188	0.125	0.719
TN-065	1.3750	1.188	18	1.312	1.2764	1.2443	1.078	0.641	1.2337	0.109	0.125	0.188	0.125	0.750
TN-07	1.3780	1.250	18	1.376	1.3399	1.3078	1.078	0.641	1.2972	0.109	0.125	0.188	0.125	0.750
TN-08	1.5748	1.438	18	1.563	1.5269	1.4948	1.078	0.641	1.4842	0.109	0.125	0.312	0.125	0.750
TN-09	1.7717	1.656	18	1.767	1.7309	1.6988	1.078	0.641	1.6882	0.141	0.125	0.312	0.156	0.781
TN-10	1.9685	1.859	18	1.967	1.9309	1.8988	1.203	0.703	1.882	0.141	0.125	0.312	0.156	0.844
TN-11	2.1654	2.047	18	2.157	2.1209	2.0888	1.203	0.703	2.0782	0.141	0.125	0.312	0.156	0.844
TN-12	2.3622	2.250	18	2.360	2.3239	2.2918	1.297	0.766	2.2812	0.141	0.156	0.312	0.156	0.906
TN-13	2.5591	2.422	18	2.548	2.5119	2.4798	1.359	0.797	2.4692	0.141	0.156	0.312	0.156	0.938
TN-14	2.7559	2.625	18	2.751	2.7149	2.6828	1.359	0.797	2.6722	0.141	0.156	0.312	0.250	1.000
TAN-15	2.9528	2.781	12	2.933	2.8789	2.8308	1.422	0.828	2.8095	0.172	0.188	0.312	0.250	1.031
TAN-16	3.1496	3.000	12	3.137	3.0829	3.0348	1.422	0.828	3.0135	0.172	0.188	0.375	0.250	1.031

[a] Threads are American National form Class 3.

All dimensions in inches. For dimensions in millimeters, multiply inch values, except thread diameters, by 25.4 and round results to two decimal places.These data apply to steel. When either the nut or the shaft is made of stainless steel, aluminum, or other material having a tendency to seize, it is recommended that the maximum thread diameter of the shaft, both major and pitch, be reduced by 20 per cent of the pitch diameter tolerance listed in the Standard.For sizes larger than shown, see ANSI/ABMA 8.2-1991.

Bearing Closures.—Shields, seals, labyrinths, and slingers are employed to retain the lubricant in the bearing and to prevent the entry of dirt, moisture, or other harmful substances. The type selected for a given application depends upon the lubricant, shaft, speed, and the atmospheric conditions in which the unit is to operate. The shields or seals may be located in the bearing itself. Shields differ from seals in that they are attached to one bearing race but there is a definite clearance between the shield and the other, usually the inner, race. When a shielded bearing is placed in a housing in which the grease space has been filled, the bearing in running will tend to expel excess grease past the shields or to accept grease from the housing when the amount in the bearing itself is low.

Seals of leather, rubber, cork, felt, or plastic composition may be used. Since they must bear against the rotating member, excessive pressure should be avoided and some lubricant must be allowed to flow into the area of contact in order to prevent seizing and burning of the seal and scoring of the rotating member. Some seals are made up in the form of cartridges which can be pressed into the end of the bearing housing.

Leather seals may be used over a wide range of speeds. Although lubricant is best retained with a leather cupped inward toward the bearing, this arrangement is not suitable at high speeds due to danger of burning the leather. At high speeds where abrasive dust is present, the seal should be arranged with the leather cupped outward to lead some lubricant into the contact area. Only light pressure of leather against the shaft should be maintained.

Bearing Fits.—The slipping or creeping of a bearing ring on a rotating shaft or in a rotating housing occurs when the fit of the ring on the shaft or in the housing is loose. Such slipping or creeping action may cause rapid wear of both shaft and bearing ring when the surfaces are dry and highly loaded. To prevent this action the bearing is customarily mounted with the rotating ring a press fit and the stationary ring a push fit, the tightness or looseness depending upon the service intended. Thus, where shock or vibratory loads are to be encountered, fits should be made somewhat tighter than for ordinary service. The stationary ring, if correctly fitted, is allowed to creep very slowly so that prolonged stressing of one part of the raceway is avoided.

To facilitate the assembly of a bearing on a shaft it may become necessary to expand the inner ring by heating. This should be done in clean oil or in a temperature-controlled furnace at a temperature of between 200 and 250°F. The utmost care must be used to make sure that the temperature does not exceed 250°F. as overheating will tend to reduce the hardness of the rings. Prelubricated bearings should not be mounted by this method.

Friction Losses in Rolling Element Bearings.—The static and kinematic torques of rolling element bearings are generally small and in many applications are not significant. Bearing torque is a measure of the frictional resistance of the bearing to rotation and is the sum of three components: the torque due to the applied load; the torque due to viscous forces in lubricated rolling element bearings; and the torque due to roller end motions, for example, thrust loads against flanges. The friction or torque data may be used to calculate power absorption or heat generation within the bearing and can be utilized in efficiency or system-cooling studies.

Empirical equations have been developed for each of the torque components. These equations are influenced by such factors as bearing load, lubrication environment, and bearing design parameters. These design parameters include sliding friction from contact between the rolling elements and separator surfaces or between adjacent rolling elements; rolling friction from material deformations during the passage of the rolling elements over the race path; skidding or sliding of the Hertzian contact; and windage friction as a function of speed.

Starting or breakaway torques are also of interest in some situations. Breakaway torques tend to be between 1.5 and 1.8 times the running or kinetic torques.

When evaluating the torque requirements of a system under design, it should be noted that other components of the bearing package, such as seals and closures, can increase the overall system torque significantly. Seal torques have been shown to vary from a fraction of the bearing torque to several times that torque. In addition, the torque values given can vary significantly when load, speed of rotation, temperature, or lubrication are outside normal ranges.

For small instrument bearings friction torque has implications more critical than for larger types of bearings. These bearings have three operating friction torques to consider: starting torque, normal running torque, and peak running torque. These torque levels may vary between manufacturers and among lots from a given manufacturer.

Instrument bearings are even more critically dependent on design features — radial play, retainer type, and race conformity — than larger bearings. Typical starting torque values for small bearings are given in the accompanying table, extracted from the New Departure General Catalog.

Finally, if accurate control of friction torque is critical to a particular application, tests of the selected bearings should be conducted to evaluate performance.

Starting Torque — ABEC7

Bearing Bore (in.)	Max. Starting Torque (g cm)	Thrust Load (g)	Minimum Radial Play Range (inches)	
			High Carbon Chrome Steel and All Miniatures	Stainless Steel Except Miniatures
0.125	0.10	75	0.0003–0.0005	—
	0.14	75	0.0002–0.0004	0.0004–0.0006
	0.18	75	0.0001–0.0003	0.0003–0.0005
	0.22	75	0.0001–0.0003	0.0001–0.0003
0.1875–0.312	0.40	400	0.0005–0.0008	—
	0.45	400	0.0004–0.0006	0.0005–0.0008
	0.50	400	0.0003–0.0005	0.0003–0.0005
	0.63	400	0.0001–0.0003	0.0002–0.0004
0.375	0.50	400	0.0005–0.0008	0.0008–0.0011
	0.63	400	0.0004–0.0006	0.0005–0.0008
	0.75	400	0.0003–0.0005	0.0004–0.0006
	0.95	400	0.0002–0.0004	0.0003–0.0005

Selection of Ball and Roller Bearings.—As compared with sleeve bearings, ball and roller bearings offer the following advantages: 1) Starting friction is low; 2) Less axial space is required; 3) Relatively accurate shaft alignment can be maintained; 4) Both radial and axial loads can be carried by certain types; 5) Angle of load application is not restricted; 6) Replacement is relatively easy; 7) Comparatively heavy overloads can be carried momentarily; 8) Lubrication is simple; and 9) Design and application can be made with the assistance of bearing supplier engineers.

In selecting a ball or roller bearing for a specific application five choices must be made:

1) the bearing series; 2) the type of bearing; 3) the size of bearing; 4) the method of lubrication; and 5) the type of mounting.

Naturally these considerations are modified or affected by the anticipated operating conditions, expected life, cost, and overhaul philosophy.

It is well to review the possible history of the bearing and its function in the machine it will be applied to, thus: 1) Will it be expected to endure removal and reapplication?; 2) Must it be free from maintenance attention during its useful life?; 3) Can wear of the housing or shaft be tolerated during the overhaul period?; 4) Must it be adjustable to take up wear, or to change shaft location?; 5) How accurately can the load spectrum be estimated? and; and 6) Will it be relatively free from abuse in operation?.

Though many cautions could be pointed out, it should always be remembered that inadequate design approaches limit the utilization of rolling element bearings, reduce customer satisfaction, and reduce reliability. Time spent in this stage of design is the most rewarding effort of the bearing engineer, and here again he can depend on the bearing manufacturers' field organization for assistance.

Type: Where loads are low, ball bearings are usually less expensive than roller bearings in terms of unit-carrying capacity. Where loads are high, the reverse is usually true.

For a purely radial load, almost any type of radial bearing can be used, the actual choice being determined by other factors. To support a combination of thrust and radial loads, several types of bearings may be considered. If the thrust load component is large, it may be most economical to provide a separate thrust bearing. When a separate thrust bearing cannot be used due to high speed, lack of space, or other factors, the following types may be considered: angular contact ball bearing, deep groove ball bearing without filling slot, tapered roller bearing with steep contact angle, and self-aligning bearing of the wide type. If movement or deflection in an axial direction must be held to a minimum, then a separate thrust bearing or a preloaded bearing capable of taking considerable thrust load is required. To minimize deflection due to a moment in an axial plane, a rigid bearing such as a double row angular contact type with outwardly converging load lines is required. In such cases, the resulting stresses must be taken into consideration in determining the proper size of the bearing.

For shock loads or heavy loads of short duration, roller bearings are usually preferred.

Special bearing designs may be required where accelerations are usually high as in planetary or crank motions.

Where the problem of excessive shaft deflection or misalignment between shaft and housing is present, a self-aligning type of bearing may be a satisfactory solution.

It should be kept in mind that a great deal of difficulty can be avoided if standard types of bearings are used in preference to special designs, wherever possible.

Size: The size of bearing required for a given application is determined by the loads that are to be carried and, in some cases, by the amount of rigidity that is necessary to limit deflection to some specified amount.

The forces to which a bearing will be subjected can be calculated by the laws of engineering mechanics from the known loads, power, operating pressure, etc. Where loads are irregular, varying, or of unknown magnitude, it may be difficult to determine the actual forces. In such cases, empirical determination of such forces, based on extensive experience in bearing design, may be needed to attack the problem successfully. Where such experience is lacking, the bearing manufacturer should be consulted or the services of a bearing expert obtained.

If a ball or roller bearing is to be subjected to a combination of radial and thrust loads, an *equivalent radial load* is computed in the case of radial or angular type bearings and an *equivalent thrust load* is computed in the case of thrust bearings.

Method of Lubrication.—If speeds are high, relubrication difficult, the shaft angle other than horizontal, the application environment incompatible with normal lubrication, leakage cannot be tolerated; if other elements of the mechanism establish the lubrication requirements, bearing selection must be made with these criteria as controlling influences. Modern bearing types cover a wide selection of lubrication means. Though the most popular type is the “cartridge” type of sealed grease ball bearing, many applications have requirements which dictate against them. Often, operating environments may subject bearings to temperatures too high for seals utilized in the more popular designs. If minute leakage or the accumulation of traces of dirt at seal lips cannot be tolerated by the application (as in baking industry machinery), then the selections of bearings must be made with other sealing and lubrication systems in mind.

High shaft speeds generally dictate bearing selection based on the need for cooling, the suppression of churning or aeration of conventional lubricants, and most important of all, the inherent speed limitations of certain bearing types. An example of the latter is the effect of cage design and of the roller-end thrust-flange contact on the lubrication requirements in commercial taper roller bearings, which limit the speed they can endure and the thrust load they can carry. Reference to the manufacturers' catalog and application-design manuals is recommended before making bearing selections.

Type of Mounting.—Many bearing installations are complicated because the best adapted type was not selected. Similarly, performance, reliability, and maintenance operations are restricted because the mounting was not thoroughly considered. There is no universally adaptable bearing for all needs. Careful reviews of the machine requirements should be made before designs are implemented. In many cases complicated machining, redundant shaft and housings, and use of an oversize bearing can be eliminated if the proper bearing in a well-thought-out mounting is chosen.

Advantage should be taken of the many race variations available in "standard" series of bearings. Puller grooves, tapered sleeves, ranged outer races, split races, fully demountable rolling-element and cage assemblies, flexible mountings, hydraulic removal features, relubrication holes and grooves, and many other innovations are available beyond the obvious advantages which are inherent in the basic bearing types.

Radial and Axial Clearance.—In designing the bearing mounting, a major consideration is to provide running clearances consistent with the requirements of the application. Race fits must be expected to absorb some of the original bearing clearance so that allowance should be made for approximately 80 per cent of the actual interference showing up in the diameter of the race. This will increase for heavy, stiff housings or for extra light series races shrunk onto solid shafts, while light metal housings (aluminum, magnesium, or sheet metal) and tubular shafts with wall sections less than the race wall thickness will cause a lesser change in the race diameter.

Where the application will impose heat losses through housing or shaft, or where a temperature differential may be expected, allowances must be made in the proper direction to insure proper operating clearance. Some compromises are required in applications where the indicated modification cannot be fully accommodated without endangering the bearing performance at lower speeds, during starting, or under lower temperature conditions than anticipated. Some leeway can be relied on with ball bearings since they can run with moderate preloads (.0005 inch, max.) without affecting bearing life or temperature rise. Roller bearings, however, have a lesser tolerance for preloading, and must be carefully controlled to avoid overheating and resulting self-destruction.

In all critical applications axial and radial clearances should be checked with feeler gages or dial indicators to insure mounted clearances within tolerances established by the design engineer. Since chips, scores, race misalignment, shaft or housing denting, housing distortion, end cover (closure) off-squareness, and mismatch of rotor and housing axial dimensions can rob the bearing of clearance, careful checks of running clearance is recommended.

For precision applications, taper-sleeve mountings, opposed ball or tapered-roller bearings with adjustable or shimmed closures are employed to provide careful control of radial and/or axial clearances. This practice requires skill and experience as well as the initial assistance of the bearing manufacturer's field engineer.

Tapered bore bearings are often used in applications such as these, again requiring careful and well worked-out assembly procedures. They can be assembled on either tapered shafts or on adapter sleeves. Advancement of the inner race over the tapered shaft can be done either by controlled heating (to expand the race as required) or by the use of a hydraulic jack. The adapter sleeve is supplied with a lock-nut which is used to advance the race on the tapered sleeve. With the heavier fits normally required to effect the clearance changes compatible with such mountings, hydraulic removal devices are normally recommended.

For the conventional application, with standard fits, clearances provided in the standard bearing are suitable for normal operation. To insure that the design conditions are "normal," a careful review of the application requirements, environments, operating speed range, anticipated abuses, and design parameters must be made.

General Bearing Handling Precautions.—To insure that rolling element bearings are capable of achieving their design life and that they perform without objectionable noise, temperature rise, or shaft excursions, the following precautions are recommended:

1) Use the best bearing available for the application, consistent with the value of the application. Remember, the cost of the best bearing is generally small compared to the replacement costs of the rotating components that can be destroyed if a bearing fails or malfunctions.

2) If questions arise in designing the bearing application, seek out the assistance of the bearing manufacturer's representative.

3) Handle bearings with care, keeping them in the sealed, original container until ready to use.

4) Follow the manufacturer's instructions in handling and assembling the bearings.

5) Work with clean tools, clean dry hands, and in clean surroundings.

6) Do not wash or wipe bearings prior to installation unless special instructions or requirements have been established to do so.

7) Place unwrapped bearings on clean paper and keep them similarly covered until applied, if they cannot be kept in the original container.

8) Don't use wooden mallets, brittle or chipped tools, or dirty fixtures and tools in mounting bearings.

9) Don't spin uncleaned bearings, nor spin *any* bearing with an air blast.

10) Use care not to scratch or nick bearings.

11) Don't strike or press on race flanges.

12) Use adapters for mounting which provide uniform steady pressure rather than hammering on a drift or sleeve.

13) Insure that races are started onto shafts and into housings evenly so as to prevent cocking.

14) Inspect shafts and housings before mounting beating to insure that proper fits will be maintained.

15) When removing beatings, clean housings, covers, and shafts before exposing the bearings. All dirt can be considered an abrasive, dangerous to the reuse of any rolling bearing.

16) Treat used beatings, which may be reused, as new ones.

17) Protect dismantled bearings from dirt and moisture.

18) Use clean, lint-free rags if bearings are wiped.

19) Wrap beatings in clean, oil-proof paper when not in use.

20) Use clean filtered, water-free Stoddard's solvent or flushing oil to clean bearings.

21) In heating beatings for mounting onto shafts, follow manufacturer's instructions.

22) In assembling bearings onto shafts *never* strike the outer race, or press on it to force the inner race. Apply the pressure on the inner race only. In dismantling follow the same precautions.

23) Do not press, strike, or otherwise force the seal or shield on factory-sealed beatings.

Bearing Failures, Deficiencies, and Their Origins.—The general classifications of failures and deficiencies requiting bearing removal are:

1) Overheating a) Inadequate or insufficient lubrication; b) Excessive lubrication;

c) Grease liquefaction or aeration; d) Oil foaming; e) Abrasive or corrosive action due to contaminants in beating; f) Distortion of housing due to warping, or out-of-round;

g) Seal rubbing or failure; h) Inadequate or blocked scavenge oil passages; i) Inadequate beating-clearance or bearing-preload; j) Race turning; k) Cage wear; 1) and

a) Shaft expansion — loss of bearing or seal clearance..

2) Vibration a) Dirt or chips in bearing; b) Fatigued race or rolling elements; c) Race turning; d) Rotor unbalance; e) Out-of-round shaft; f) Race misalignment; g) Housing resonance; h) Cage wear; i) Flats on races or rolling elements; j) Excessive clearance;

k) Corrosion; l) False-brinelling or indentation of races; m) Electrical discharge (similar to corrosion effects); n) Mixed rolling element diameters; 3) and a) Out-of-square rolling paths in races.

4) Turning on shaft a) Growth of race due to overheating; b) Fretting wear; c) Improper initial fit; d) Excessive shaft deflection; e) Initially coarse shaft finish; 5) and a) Seal rub on inner race.

6) Binding of the shaft a) Lubricant breakdown; b) Contamination by abrasive or corrosive matter; c) Housing distortion or out-of-round pinching bearing; d) Uneven shimming of housing with loss of clearance; e) Tight rubbing seals; f) Preloaded beatings;

g) Cocked races; h) Loss of clearance due to excessive tightening of adapter; i) Thermal expansion of shaft or housing; 7) and a) Cage failure.

8) Noisy bearing a) Lubrication breakdown, inadequate lubrication, stiff grease;

b) Contamination; c) Pinched beating; d) Seal rubbing; e) Loss of clearance and preloading; f) Bearing slipping on shaft or in housing; g) Flatted roller or ball; h) Brinelling due to assembly abuse, handling, or shock loads; i) Variation in size of rolling elements;

j) Out-of-round or lobular shaft; k) Housing bore waviness; 9) and a) Chips or scores under beating race seat.

10) Displaced shaft a) Bearing wear; b) Improper housing or closure assembly;

c) Overheated and shifted bearing; d) Inadequate shaft or housing shoulder; e) Lubrication and cage failure permitting rolling elements to bunch; f) Loosened retainer nut or adapter; g) Excessive heat application in assembling inner race, causing growth and shifting on shaft; 11) and a) Housing pounding out.

12) Lubricant leakage a) Overfilling of lubricant; b) Grease churning due to use of too soft a consistency; c) Grease deterioration due to excessive operating temperature;

d) Operating life longer than grease life (grease breakdown, aeration, and purging);

e) Seal wear; f) Wrong shaft attitude (bearing seals designed for horizontal mounting only); g) Seal failure; h) Clogged breather; i) Oil foaming due to churning or air flow through housing; j) Gasket (O-ring) failure or misapplication; k) Porous housing or closure; 13) and a) Lubricator set at wrong flow rate.

Load Ratings and Fatigue Life

Ball and Roller Bearing Life.—The performance of ball and roller bearings is a function of many variables. These include the bearing design, the characteristics of the material from which the bearings are made, the way in which they are manufactured, as well as many variables associated with their application. The only sure way to establish the satisfactory operation of a bearing selected for a specific application is by actual performance in the application. As this is often impractical, another basis is required to estimate the suitability of a particular bearing for a given application. Two factors are taken into consideration: the beating fatigue life, and its ability to withstand static loading.

Life Criterion: Even if a ball or roller bearing is properly mounted, adequately lubricated, protected from foreign matter and not subjected to extreme operating conditions, it can ultimately fatigue. Under ideal conditions, the repeated stresses developed in the contact areas between the balls or rollers and the raceways eventually can result in the fatigue of the material which manifests itself with the spalling of the load-carrying surfaces. In most applications the fatigue life is the maximum useful life of a bearing.

Static Load Criterion: A static load is a load acting on a non-rotating bearing. Permanent deformations appear in balls or rollers and raceways under a static load of moderate magnitude and increase gradually with increasing load. The permissible static load is, therefore, dependent upon the permissible magnitude of permanent deformation. It has been found that for ball and roller bearings suitably manufactured from hardened alloy steel, deformations occurring under maximum contact stress of 4,000 megapascals (580,000 pounds per square inch) acting at the center of contact (in the case of roller beatings, of a uniformly loaded roller) do not greatly impair smoothness or friction. Depending on requirements for smoothness of operation, friction, or sound level, higher or lower static load limits may be tolerated.

Ball Bearing Types Covered.—AFBMA and American National Standard ANSI/ABMA 9-1990 sets forth the method of determining ball bearing Rating Life and Static Load Rating and covers the following types:

1) *Radial, deep groove and angular contact ball bearings* whose inner ring race-ways have a cross-sectional radius not larger than 52 percent of the ball diameter and whose outer ring raceways have a cross-sectional radius not larger than 53 percent of the ball diameter.

2) *Radial, self-aligning ball bearings* whose inner ring raceways have cross-sectional radii not larger than 53 percent of the ball diameter.

3) *Thrust ball bearings* whose washer raceways have cross-sectional radii not larger than 54 percent of the ball diameter.

4) *Double row, radial and angular contact ball bearings* and double direction thrust ball bearings are presumed to be symmetrical.

Limitations for Ball Bearings.—The following limitations apply:

1) *Truncated contact area.* This standard* may not be safely applied to ball bearings subjected to loading which causes the contact area of the ball with the raceway to be truncated by the raceway shoulder. This limitation depends strongly on details of bearing design which are not standardized.

2) *Material.* This standard applies only to ball bearings fabricated from hardened good quality steel.

3) *Types.* The f_c factors specified in the basic load rating formulas are valid only for those ball bearing types specified above.

4) *Lubrication.* The Rating Life calculated according to this standard is based on the assumption that the bearing is adequately lubricated. The determination of adequate lubrication depends upon the bearing application.

5) *Ring support and alignment.* The Rating Life calculated according to this standard assumes that the bearing inner and outer rings are rigidly supported and the inner and outer ring axes are properly aligned.

6) *Internal clearance.* The radial ball bearing Rating Life calculated according to this standard is based on the assumption that only a nominal interior clearance occurs in the mounted bearing at operating speed, load and temperature.

7) *High speed effects.* The Rating Life calculated according to this standard does not account for high speed effects such as ball centrifugal forces and gyroscopic moments. These effects tend to diminish fatigue life. Analytical evaluation of these effects frequently

* All references to "standard" are to AFBMA and American National Standard "Load Ratings and Fatigue Life for Ball Bearings"ANSI/ABMA 9-1990.

requires the use of high speed digital computation devices and hence is not covered in the standard.

8) *Groove radii.* If groove radii are smaller than those specified in the bearing types covered, the ability of a bearing to resist fatigue is not improved: however, it is diminished by the use of larger radii.

Ball Bearing Rating Life.—According to the Anti-Friction Bearing Manufacturers Association standards the Rating Life L_{10} of a group of apparently identical ball bearings is the life in millions of revolutions that 90 percent of the group will complete or exceed. For a single bearing, L_{10} also refers to the life associated with 90 percent reliability.

Radial and Angular Contact Ball Bearings: The magnitude of the Rating Life L_{10} in millions of revolutions, for a radial or angular contact ball bearing application is given by the formula:

$$L_{10} = \left(\frac{C}{P}\right)^3 \tag{1}$$

where C = basic load rating, newtons (pounds). See Formulas (2), (3a) and (3b)

P = equivalent radial load, newtons (pounds). See Formula (4)

Table 25. Values of f_c for Radial and Angular Contact Ball Bearings

$\frac{D \cos\alpha}{d_m}$	Single Row Radial Contact; Single and Double Row Angular Contact, Groove Type[a]		Double Row Radial Contact Groove Type		Self-Aligning	
	Values of f_c					
	Metric[b]	Inch[c]	Metric[b]	Inch[c]	Metric[b]	Inch[c]
0.05	46.7	3550	44.2	3360	17.3	1310
0.06	49.1	3730	46.5	3530	18.6	1420
0.07	51.1	3880	48.4	3680	19.9	1510
0.08	52.8	4020	50.0	3810	21.1	1600
0.09	54.3	4130	51.4	3900	22.3	1690
0.10	55.5	4220	52.6	4000	23.4	1770
0.12	57.5	4370	54.5	4140	25.6	1940
0.14	58.8	4470	55.7	4230	27.7	2100
0.16	59.6	4530	56.5	4290	29.7	2260
0.18	59.9	4550	56.8	4310	31.7	2410
0.20	59.9	4550	56.8	4310	33.5	2550
0.22	59.6	4530	56.5	4290	35.2	2680
0.24	59.0	4480	55.9	4250	36.8	2790
0.26	58.2	4420	55.1	4190	38.2	2910
0.28	57.1	4340	54.1	4110	39.4	3000
0.30	56.0	4250	53.0	4030	40.3	3060
0.32	54.6	4160	51.8	3950	40.9	3110
0.34	53.2	4050	50.4	3840	41.2	3130
0.36	51.7	3930	48.9	3730	41.3	3140
0.38	50.0	3800	47.4	3610	41.0	3110
0.40	48.4	3670	45.8	3480	40.4	3070

[a] A. When calculating the basic load rating for a unit consisting of two similar, single row, radial contact ball bearings, in a duplex mounting, the pair is considered as one, double row, radial contact ball bearing.
B. When calculating the basic load rating for a unit consisting of two, similar, single row, angular contact ball bearings in a duplex mounting, "face-to-face" or "back-to-back," the pair is considered as one, double row, angular contact ball bearing.
C. When calculating the basic load rating for a unit consisting of two or more similar, single angular contact ball bearings mounted "in tandem," properly manufactured and mounted for equal load distribution, the rating of the combination is the number of bearings to the 0.7 power times the rating of a single row ball bearing. If the unit may be treated as a number of individually interchangeable single row bearings, this footnote "C" does not apply.

[b] Use to obtain *C* in newtons when *D* is given in mm.

[c] Use to obtain *C* in pounds when *D* is given in inches.

Table 26. Values of *X* and *Y* for Computing Equivalent Radial Load *P* of Radial and Angular Contact Ball Bearings

Contact Angle, α	Table Entering Factors[a]				Single Row Bearings[b]		Double Row Bearings			
					$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$	
RADIAL CONTACT GROOVE BEARINGS										
		F_a/iZD^2 Metric Units	F_a/iZD^2 Inch Units							
	F_a/C_o	Metric Units	Inch Units	*e*	*X*	*Y*	*X*	*Y*	*X*	*Y*
	0.014	0.172	25	0.19		2.30				2.30
	0.028	0.345	50	0.22		1.99				1.99
	0.056	0.689	100	0.26		1.71				1.71
	0.084	1.03	150	0.28		1.56				1.55
0°	0.11	1.38	200	0.30	0.56	1.45	1	0	0.56	1.45
	0.17	2.07	300	0.34		1.31				1.31
	0.28	3.45	500	0.38		1.15				1.15
	0.42	5.17	750	0.42		1.04				1.04
	0.56	6.89	1000	0.44		1.00				1.00
ANGULAR CONTACT GROOVE BEARINGS										
		F_a/ZD^2 Metric Units	F_a/ZD^2 Inch Units							
	iF_a/C_o	Metric Units	Inch Units	*e*	*X*	*Y*	*X*	*Y*	*X*	*Y*
	0.014	0.172	25	0.23	For this type use the *X*, *Y*, and *e* values applicable to single row radial contact bearings			2.78	0.78	3.74
	0.028	0.345	50	0.26				2.40		3.23
	0.056	0.689	100	0.30				2.07		2.78
	0.085	1.03	150	0.34				1.87		2.52
5°	0.11	1.38	200	0.36			1	1.75		2.36
	0.17	2.07	300	0.40				1.58		2.13
	0.28	3.45	500	0.45				1.39		1.87
	0.42	5.17	750	0.50				1.26		1.69
	0.56	6.89	1000	0.52				1.21		1.63
	0.014	0.172	25	0.29		1.88		2.18		3.06
	0.029	0.345	50	0.32		1.71		1.98		2.78
	0.057	0.689	100	0.36		1.52		1.76		2.47
	0.086	1.03	150	0.38		1.41		1.63		2.20
10°	0.11	1.38	200	0.40	0.46	1.34	1	1.55	0.75	2.18
	0.17	2.07	300	0.44		1.23		1.42		2.00
	0.29	3.45	500	0.49		1.10		1.27		1.79
	0.43	5.17	750	0.54		1.01		1.17		1.64
	0.57	6.89	1000	0.54		1.00		1.16		1.63

Table 26. *(Continued)* **Values of *X* and *Y* for Computing Equivalent Radial Load *P* of Radial and Angular Contact Ball Bearings**

	0.015	0.172	25	0.38		1.47		1.65		2.39
	0.029	0.345	50	0.40		1.40		1.57		2.28
	0.058	0.689	100	0.43		1.30		1.46		2.11
	0.087	1.03	150	0.46		1.23		1.38		2.00
15°	0.12	1.38	200	0.47	0.44	1.19	1	1.34	0.72	1.93
	0.17	2.07	300	0.50		1.12		1.26		1.82
	0.29	3.45	500	0.55		1.02		1.14		1.66
	0.44	5.17	750	0.56		1.00		1.12		1.63
	0.58	6.89	1000	0.56		1.00		1.12		1.63
20°	…	…	…	0.57	0.43	1.00	1	1.09	0.70	1.63
25°	…	…	…	0.68	0.41	0.87	1	0.92	0.67	1.41
30°	…	…	…	0.80	0.39	0.76	1	0.78	0.63	1.24
35°	…	…	…	0.95	0.37	0.66	1	0.66	0.60	1.07
40°	…	…	…	1.14	0.35	0.57	1	0.55	0.57	0.98
Self-aligning Ball Bearings				1.5 tan α	0.40	0.4 cot α	1	0.42 cot α	0.65	0.65 cot α

[a] Symbol definitions are given on the following page.

[b] For single row bearings when $F_a/F_r \leq e$, use $X = 1$, $Y = 0$. Two similar, single row, angular contact ball bearings mounted face-to-face or back-to-back are considered as one double row, angular contact bearing.

Values of X, Y, and e for a load or contact angle other than shown are obtained by linear interpolation.Values of X, Y, and e do not apply to filling slot bearings for applications in which ball-raceway contact areas project substantially into the filling slot under load.Symbol Definitions: F_a is the applied axial load in newtons (pounds); C_o is the static load rating in newtons (pounds) of the bearing under consideration and is found by Formula (20); i is the number of rows of balls in the bearing; Z is the number of balls per row in a radial or angular contact bearing or the number of balls in a single row, single direction thrust bearing; D is the ball diameter in millimeters (inches); and F_r is the applied radial load in newtons (pounds).

For radial and angular contact ball bearings with balls not larger than 25.4 mm (1 inch) in diameter, C is found by the formula:

$$C = f_c(i\cos\alpha)^{0.7}Z^{2/3}D^{1.8} \quad (2)$$

and with balls larger than 25.4 mm (1 inch) in diameter C is found by the formula:

$$C = 3.647f_c(i\cos\alpha)^{0.7}Z^{2/3}D^{1.4} \quad \text{(metric)} \quad (3a)$$

$$C = f_c(i\cos\alpha)^{0.7}Z^{2/3}D^{1.4} \quad \text{(inch)} \quad (3b)$$

where f_c = a factor which depends on the geometry of the bearing components, the accuracy to which the various bearing parts are made and the material. Values of f_c, are given in Table 25

i = number of rows of balls in the bearing

α = nominal contact angle, degrees

Z = number of balls per row in a radial or angular contact bearing

D = ball diameter, mm (inches)

The magnitude of the equivalent radial load, P, in newtons (pounds) for radial and angular contact ball bearings, under combined constant radial and constant thrust loads is given by the formula:

$$P = XF_r + YF_a \quad (4)$$

where F_r = the applied radial load in newtons (pounds)

F_a = the applied axial load in newtons (pounds)

X = radial load factor as given in Table 28

Y = axial load factor as given in Table 28

Thrust Ball Bearings: The magnitude of the Rating Life L_{10} in millions of revolutions for a thrust ball bearing application is given by the formula:

$$L_{10} = \left(\frac{C_a}{P_a}\right)^3 \quad (5)$$

where C_a = the basic load rating, newtons (pounds). See Formulas (6) to (10)

P_a = equivalent thrust load, newtons (pounds). See Formula (11)

For single row, single and double direction, thrust ball bearing with balls not larger than 25.4 mm (1 inch) in diameter, C_a is found by the formulas:

$$\text{for } \alpha = 90\text{ degrees,} \qquad C_a = f_c Z^{2/3} D^{1.8} \quad (6)$$

$$\text{for } \alpha \neq 90 \text{ degrees,} \qquad C_a = f_c(\cos\alpha)^{0.7} Z^{2/3} D^{1.8} \tan\alpha \quad (7)$$

and with balls larger than 25.4 mm (1 inch) in diameter, C_a is found by the formulas:

$$\text{for } \alpha = 90 \text{ degrees,} \qquad C_a = 3.647 f_c Z^{2/3} D^{1.4} \quad \text{(metric)} \quad (8a)$$

$$C_a = f_c Z^{2/3} D^{1.4} \quad \text{(inch)} \quad (8b)$$

$$\text{for } \alpha \neq 90 \text{ degrees,} \qquad C_a = 3.647 f_c(\cos\alpha)^{0.7} Z^{2/3} D^{1.4} \tan\alpha \quad \text{(metric)} \quad (9a)$$

$$C_a = f_c(\cos\alpha)^{0.7} Z^{2/3} D^{1.4} \tan\alpha \quad \text{(inch)} \quad (9b)$$

where f_c = a factor which depends on the geometry of the bearing components, the accuracy to which the various bearing parts are made, and the material. Values of f_c are given in Table 27

Z = number of balls per row in a single row, single direction thrust ball bearing

D = ball diameter, mm (inches)

α = nominal contact angle, degrees

Table 27. Values of f_c for Thrust Ball Bearings

$\frac{D}{d_m}$	α = 90°		$D \cos\alpha$	α = 45°		α = 60°		α = 75°	
	Metric[a]	Inch[b]		Metric[a]	Inch[b]	Metric[a]	Inch[b]	Metric[a]	Inch[b]
0.01	36.7	2790	0.01	42.1	3200	39.2	2970	37.3	2840
0.02	45.2	3430	0.02	51.7	3930	48.1	3650	45.9	3490
0.03	51.1	3880	0.03	58.2	4430	54.2	4120	51.7	3930
0.04	55.7	4230	0.04	63.3	4810	58.9	4470	56.1	4260
0.05	59.5	4520	0.05	67.3	5110	62.6	4760	59.7	4540
0.06	62.9	4780	0.06	70.7	5360	65.8	4990	62.7	4760
0.07	65.8	5000	0.07	73.5	5580	68.4	5190	65.2	4950
0.08	68.5	5210	0.08	75.9	5770	70.7	5360	67.3	5120
0.09	71.0	5390	0.09	78.0	5920	72.6	5510	69.2	5250
0.10	73.3	5570	0.10	79.7	6050	74.2	5630	70.7	5370
0.12	77.4	5880	0.12	82.3	6260	76.6	5830	…	…
0.14	81.1	6160	0.14	84.1	6390	78.3	5950	…	…
0.16	84.4	6410	0.16	85.1	6470	79.2	6020	…	…
0.18	87.4	6640	0.18	85.5	6500	79.6	6050	…	…
0.20	90.2	6854	0.20	85.4	6490	79.5	6040	…	…
0.22	92.8	7060	0.22	84.9	6450	…	…	…	…
0.24	95.3	7240	0.24	84.0	6380	…	…	…	…
0.26	97.6	7410	0.26	82.8	6290	…	…	…	…
0.28	99.8	7600	0.28	81.3	6180	…	…	…	…
0.30	101.9	7750	0.30	79.6	6040	…	…	…	…
0.32	103.9	7900	…	…	…	…	…	…	…
0.34	105.8	8050	…	…	…	…	…	…	…

[a] **Use to obtain C_a in newtons when D is given in mm.**

[b] Use to obtain C_a in pounds when D is given in inches.

For thrust ball bearings with two or more rows of similar balls carrying loads in the same direction, the basic load rating, C_a, in newtons (pounds) is found by the formula:

$$C_a = (Z_1 + Z_2 + \dots Z_n)\left[\left(\frac{Z_1}{C_{a1}}\right)^{10/3} + \left(\frac{Z_2}{C_{a2}}\right)^{10/3} + \dots\left(\frac{Z_n}{C_{an}}\right)^{10/3}\right]^{-0.3} \quad (10)$$

where $Z_1, Z_2 \dots Z_n$ = number of balls in respective rows of a single-direction multi-row thrust ball bearing

$C_{a1}, C_{a2} \dots C_{an}$ = basic load rating per row of a single-direction, multi-row thrust ball bearing, each calculated as a single-row bearing with $Z_1, Z_2 \dots Z_n$ balls, respectively

The magnitude of the equivalent thrust load, P_a, in newtons (pounds) for thrust ball bearings with $\alpha \neq 90$ degrees under combined constant thrust and constant radial loads is found by the formula:

$$P_a = XF_r + YF_a \quad (11)$$

where F_r = the applied radial load in newtons (pounds)

F_a = the applied axial load in newtons (pounds)

X = radial load factor as given in Table 28

Y = axial load factor as given in Table 28

Table 28. Values of *X* and *Y* for Computing Equivalent Thrust Load P_a for Thrust Ball Bearings

Contact Angle α	e	Single Direction Bearings $\frac{F_a}{F_r} > e$		Double Direction Bearings $\frac{F_a}{F_r} \leq e$		Double Direction Bearings $\frac{F_a}{F_r} > e$	
		X	Y	X	Y	X	Y
45°	1.25	0.66	1	1.18	0.59	0.66	1
60°	2.17	0.92	1	1.90	0.54	0.92	1
75°	4.67	1.66	1	3.89	0.52	1.66	1

For $\alpha = 90°$, $F_r = 0$ and $Y = 1$.

Roller Bearing Types Covered.—This standard* applies to *cylindrical, tapered and self-aligning radial and thrust roller bearings* and to *needle roller bearings*. These bearings are presumed to be within the size ranges shown in the AFBMA dimensional standards, of good quality and produced in accordance with good manufacturing practice.

Roller bearings vary considerably in design and execution. Since small differences in relative shape of contacting surfaces may account for distinct differences in load carrying ability, this standard does not attempt to cover all design variations, rather it applies to basic roller bearing designs.

The following limitations apply:

1) *Truncated contact area.* This standard may not be safely applied to roller bearings subjected to application conditions which cause the contact area of the roller with the raceway to be severely truncated by the edge of the raceway or roller.

2) *Stress concentrations.* A cylindrical, tapered or self-aligning roller bearing must be expected to have a basic load rating less than that obtained using a value of f_c taken from Table 29 or Table 30 if, under load, a stress concentration is present in some part of the roller-raceway contact. Such stress concentrations occur in the center of nominal point contacts, at the contact extremities for line contacts and at inadequately blended junctions of a rolling surface profile. Stress concentrations can also occur if the rollers are not accu-

* All references to "standard" are to AFBMA and American National Standard "Load Ratings and Fatigue Life for Roller Bearings" ANSI/AFBMA Std 11–1990.

rately guided such as in bearings without cages and bearings not having rigid integral flanges. Values of f_c given in Tables 29 and 30 are based upon bearings manufactured to achieve optimized contact. For no bearing type or execution will the factor f_c be greater than that obtained in Tables 29 and 30.

3) *Material.* This standard applies only to roller bearings fabricated from hardened, good quality steel.

4) *Lubrication.* Rating Life calculated according to this standard is based on the assumption that the bearing is adequately lubricated. Determination of adequate lubrication depends upon the bearing application.

5) *Ring support and alignment.* Rating Life calculated according to this standard assumes that the bearing inner and outer rings are rigidly supported, and that the inner and outer ring axes are properly aligned.

6) *Internal clearance.* Radial roller bearing Rating Life calculated according to this standard is based on the assumption that only a nominal internal clearance occurs in the mounted bearing at operating speed, load, and temperature.

7) *High speed effects.* The Rating Life calculated according to this standard does not account for high speed effects such as roller centrifugal forces and gyroscopic moments: These effects tend to diminish fatigue life. Analytical evaluation of these effects frequently requires the use of high speed digital computation devices and hence, cannot be included.

Table 29. Values of f_c for Radial Roller Bearings

$\frac{D\cos\alpha}{d_m}$	f_c Metric[a]	f_c Inch[b]	$\frac{D\cos\alpha}{d_m}$	f_c Metric[a]	f_c Inch[b]	$\frac{D\cos\alpha}{d_m}$	f_c Metric[a]	f_c Inch[b]
0.01	52.1	4680	0.18	88.8	7980	0.35	79.5	7140
0.02	60.8	5460	0.19	88.8	7980	0.36	78.6	7060
0.03	66.5	5970	0.20	88.7	7970	0.37	77.6	6970
0.04	70.7	6350	0.21	88.5	7950	0.38	76.7	6890
0.05	74.1	6660	0.22	88.2	7920	0.39	75.7	6800
0.06	76.9	6910	0.23	87.9	7890	0.40	74.6	6700
0.07	79.2	7120	0.24	87.5	7850	0.41	73.6	6610
0.08	81.2	7290	0.25	87.0	7810	0.42	72.5	6510
0.09	82.8	7440	0.26	86.4	7760	0.43	71.4	6420
0.10	84.2	7570	0.27	85.8	7710	0.44	70.3	6320
0.11	85.4	7670	0.28	85.2	7650	0.45	69.2	6220
0.12	86.4	7760	0.29	84.5	7590	0.46	68.1	6120
0.13	87.1	7830	0.30	83.8	7520	0.47	67.0	6010
0.14	87.7	7880	0.31	83.0	7450	0.48	65.8	5910
0.15	88.2	7920	0.32	82.2	7380	0.49	64.6	5810
0.16	88.5	7950	0.33	81.3	7300	0.50	63.5	5700
0.17	88.7	7970	0.34	80.4	7230	…	…	…

[a] For $\alpha = 0°$, $F_a = 0$ and $X = 1$.

[b] Use to obtain C in pounds when l_{eff} and D are given in inches.

Table 30. Values of f_c for Thrust Roller Bearings

$\frac{D\cos\alpha}{d_m}$	$45° < \alpha < 60°$ f_c Metric[a]	$45° < \alpha < 60°$ f_c Inch[b]	$60° < \alpha < 75°$ f_c Metric[a]	$60° < \alpha < 75°$ f_c Inch[b]	$75° \le \alpha < 90°$ f_c Metric[a]	$75° \le \alpha < 90°$ f_c Inch[b]	$\frac{D}{d_m}$	$\alpha = 90°$ f_c Metric[a]	$\alpha = 90°$ f_c Inch[b]
0.01	109.7	9840	107.1	9610	105.6	9470	0.01	105.4	9500
0.02	127.8	11460	124.7	11180	123.0	11030	0.02	122.9	11000
0.03	139.5	12510	136.2	12220	134.3	12050	0.03	134.5	12100
0.04	148.3	13300	144.7	12980	142.8	12810	0.04	143.4	12800
0.05	155.2	13920	151.5	13590	149.4	13400	0.05	150.7	13200
0.06	160.9	14430	157.0	14080	154.9	13890	0.06	156.9	14100

Table 30. *(Continued)* **Values of f_c for Thrust Roller Bearings**

$\frac{D\cos\alpha}{d_m}$	45° < α < 60°		60° < α < 75°		75° ≤ α < 90°		$\frac{D}{d_m}$	α = 90°	
	f_c							f_c	
	Metric[a]	Inch[b]	Metric[a]	Inch[b]	Metric[a]	Inch[b]		Metric[a]	Inch[b]
0.07	165.6	14850	161.6	14490	159.4	14300	0.07	162.4	14500
0.08	169.5	15200	165.5	14840	163.2	14640	0.08	167.2	15100
0.09	172.8	15500	168.7	15130	166.4	14930	0.09	171.7	15400
0.10	175.5	15740	171.4	15370	169.0	15160	0.10	175.7	15900
0.12	179.7	16120	175.4	15730	173.0	15520	0.12	183.0	16300
0.14	182.3	16350	177.9	15960	175.5	15740	0.14	189.4	17000
0.16	183.7	16480	179.3	16080	…	…	0.16	195.1	17500
0.18	184.1	16510	179.7	16120	…	…	0.18	200.3	18000
0.20	183.7	16480	179.3	16080	…	…	0.20	205.0	18500
0.22	182.6	16380	…	…	…	…	0.22	209.4	18800
0.24	180.9	16230	…	…	…	…	0.24	213.5	19100
0.26	178.7	16030	…	…	…	…	0.26	217.3	19600
0.28	…	…	…	…	…	…	0.28	220.9	19900
0.30	…	…	…	…	…	…	0.30	224.3	20100

[a] **Use to obtain C_a in newtons when l_{eff} and D are given in mm.**

[b] Use to obtain C_a in pounds when l_{eff} and D are given in inches.

Roller Bearing Rating Life.—The Rating Life L_{10} of a group of apparently identical roller bearings is the life in millions of revolutions that 90 percent of the group will complete or exceed. For a single bearing, L_{10} also refers to the life associated with 90 percent reliability.

Radial Roller Bearings: The magnitude of the Rating Life, L_{10}, in millions of revolutions, for a radial roller bearing application is given by the formula:

$$L_{10} = \left(\frac{C}{P}\right)^{10/3} \tag{12}$$

where C = the basic load rating in newtons (pounds), see Formula (13); and, P = equivalent radial load in newtons (pounds), see Formula (14).

For radial roller bearings, C is found by the formula:

$$C = f_c(il_{eff}\cos\alpha)^{7/9}Z^{3/4}D^{29/27} \tag{13}$$

where f_c = a factor which depends on the geometry of the bearing components, the accuracy to which the various bearing parts are made, and the material. Maximum values of f_c are given in Table 29

i = number of rows of rollers in the bearing

l_{eff} = effective length, mm (inches) α = nominal contact angle, degrees

Z = number of rollers per row in a radial roller bearing

D = roller diameter, mm (inches) (mean diameter for a tapered roller, major diameter for a spherical roller)

When rollers are longer than $2.5D$, a reduction in the f_c value must be anticipated. In this case, the bearing manufacturer may be expected to establish load ratings accordingly.

In applications where rollers operate directly on a shaft surface or a housing surface, such a surface must be equivalent in all respects to the raceway it replaces to achieve the basic load rating of the bearing.

When calculating the basic load rating for a unit consisting of two or more similar single-row bearings mounted "in tandem," properly manufactured and mounted for equal load distribution, the rating of the combination is the number of bearings to the 7/9 power times the rating of a single-row bearing. If, for some technical reason, the unit may be treated as a number of individually interchangeable single-row bearings, this consideration does not apply.

The magnitude of the equivalent radial load, P, in newtons (pounds), for radial roller bearings, under combined constant radial and constant thrust loads is given by the formula:

$$P = XF_r + YF_a \qquad (14)$$

where F_r = the applied radial load in newtons (pounds)

F_a = the applied axial load in newtons (pounds)

X = radial load factor as given in Table 31

Y = axial load factor as given in Table 31

Typical Bearing Life for Various Design Applications

Uses	Design life in hours	Uses	Design life in hours
Agricultural equipment	3000 – 6000	Gearing units	
Aircraft equipment	500 – 2000	Automotive	600 – 5000
Automotive		Multipurpose	8000 – 15000
Race car	500 – 800	Machine tools	20000
Light motor cycle	600 – 1200	Rail Vehicles	15000 – 25000
Heavy motor cycle	1000 – 2000	Heavy rolling mill	> 50000
Light cars	1000 – 2000	Machines	
Heavy cars	1500 – 2500	Beater mills	20000 – 30000
Light trucks	1500 – 2500	Briquette presses	20000 – 30000
Heavy trucks	2000 – 2500	Grinding spindles	1000 – 2000
Buses	2000 – 5000	Machine tools	10000 – 30000
Electrical		Mining machinery	4000 – 15000
Household appliances	1000 – 2000	Paper machines	50000 – 80000
Motors ≤ ½ hp	1000 – 2000	Rolling mills	
Motors ≤ 3 hp	8000 – 10000	Small cold mills	5000 – 6000
Motors, medium	10000 – 15000	Large multipurpose mills	8000 – 10000
Motors, large	20000 – 30000	Rail vehicle axle	
Elevator cables sheaves	40000 – 60000	Mining cars	5000
Mine ventillation fans	40000 – 50000	Motor rail cars	16000 – 20000
Propeller thrust bearings	15000 – 25000	Open–pit mining cars	20000 – 25000
Propeller shaft bearings	> 80000	Streetcars	20000 – 25000
Gear drives		Passenger cars	26000
Boat gearing units	3000 – 5000	Freight cars	35000
Gear drives	> 50000	Locomotive outer bearings	20000 – 25000
Ship gear drives	20000 – 30000	Locomotive inner bearings	30000 – 40000
Machinery for 8 hour service which are not always fully utilized	14000 – 20000	Machinery for short or intermittent opearation where service interruption is of minor importance	4000 – 8000
Machinery for 8 hour service which are fully utilized	20000 – 30000	Machinery for intermittent service where reliable opeara-tion is of great importance	8000 – 14000
Machinery for continuous 24 hour service	50000 – 60000	Instruments and apparatus in frequent use	0 – 500

Table 31. Values of *X* and *Y* for Computing Equivalent Radial Load *P* for Radial Roller Bearing

Bearing Type	$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$	
	X	Y	X	Y
Self-Aligning and Tapered Roller Bearings[a] $\alpha \neq 0°$	Single Row Bearings			
	1	0	0.4	0.4 cot α
	Double Row Bearings[a]			
	1	0.45 cot α	0.67	0.67 cot α

[a] For $\alpha = 0°$, $F_a = 0$ and $X = 1$.

$e = 1.5 \tan\alpha$

Roller bearings are generally designed to achieve optimized contact; however, they usually support loads other than the loading at which optimized contact is maintained. The 10⁄3 exponent in Rating Life Formulas (12) and (15) was selected to yield satisfactory Rating Life estimates for a broad spectrum from light to heavy loading. When loading exceeds that which develops optimized contact, e.g., loading greater than $C/4$ to $C/2$ or $C_a/4$ to $C_a/2$, the user should consult the bearing manufacturer to establish the adequacy of the Rating Life formulas for the particular application.

Thrust Roller Bearings: The magnitude of the Rating Life, L_{10}, in millions of revolutions for a thrust roller bearing application is given by the formula:

$$L_{10} = \left(\frac{C_a}{P_a}\right)^{10/3} \tag{15}$$

where C_a = basic load rating, newtons (pounds). See Formulas (16) to (18)

P_a = equivalent thrust load, newtons (pounds). See Formula (19)

For single row, single and double direction, thrust roller bearings, the magnitude of the basic load rating, C_a, in newtons (pounds), is found by the formulas:

$$\text{for } \alpha = 90^\circ,\ C_a = f_c l_{eff}^{7/9} Z^{3/4} D^{29/27} \tag{16}$$

$$\text{for } \alpha \neq 90^\circ,\ C_a = f_c (l_{eff} \cos\alpha)^{7/9} Z^{3/4} D^{29/27} \tan\alpha \tag{17}$$

where f_c = a factor which depends on the geometry of the bearing components, the accuracy to which the various parts are made, and the material. Values of f_c are given in Table

l_{eff} = effective length, mm (inches)

Z = number of rollers in a single row, single direction, thrust roller bearing

D = roller diameter, mm (inches) (mean diameter for a tapered roller, major diameter for a spherical roller)

α = nominal contact angle, degrees

For thrust roller bearings with two or more rows of rollers carrying loads in the same direction the magnitude of C_a is found by the formula:

$$C_a = (Z_1 l_{eff1} + Z_2 l_{eff2} \dots Z_n l_{effn}) \left\{ \left[\frac{Z_1 l_{eff1}}{C_{a1}}\right]^{9/2} + \left[\frac{Z_2 l_{eff2}}{C_{a2}}\right]^{9/2} + \dots \left[\frac{Z_n l_{effn}}{C_{an}}\right]^{9/2} \right\}^{-2/9} \tag{18}$$

Where $Z_1, Z_2 \ldots . Z_n$ = the number of rollers in respective rows of a single direction, multi-row bearing

$C_{a1}, C_{a2} \ldots . C_{an}$ = the basic load rating per row of a single direction, multi-row, thrust roller bearing, each calculated as a single row bearing with $Z_1, Z_2 \ldots Z_n$ rollers respectively

$l_{eff1}, l_{eff2} \ldots l_{effn}$ = effective length, mm (inches), or rollers in the respective rows

In applications where rollers operate directly on a surface supplied by the user, such a surface must be equivalent in all respects to the washer raceway it replaces to achieve the basic load rating of the bearing.

In case the bearing is so designed that several rollers are located on a common axis, these rollers are considered as one roller of a length equal to the total effective length of contact of the several rollers. Rollers as defined above, or portions thereof which contact the same washer-raceway area, belong to one row.

When the ratio of the individual roller effective length to the pitch diameter (at which this roller operates) is too large, a reduction of the f_c value must be anticipated due to excessive slip in the roller-raceway contact.

When calculating the basic load rating for a unit consisting of two or more similar single row bearings mounted "in tandem," properly manufactured and mounted for equal load distribution, the rating of the combination is defined by Formula (18). If, for some technical reason, the unit may be treated as a number of individually interchangeable single-row bearings, this consideration does not apply.

The magnitude of the equivalent thrust load, P_a, in pounds, for thrust roller bearings with α not equal to 90 degrees under combined constant thrust and constant radial loads is given by the formula:

$$P_a = XF_r + YF_a \tag{19}$$

where F_r = applied radial load, newtons (pounds)

F_a = applied axial load, newtons (pounds)

X = radial load factor as given in Table 32

Y = axial load factor as given in Table 32

Table 32. Values of X and Y for Computing Equivalent Thrust Load P_a for Thrust Roller Bearings

Bearing Type	Single Direction Bearings		Double Direction Bearings			
	$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$	
	X	Y	X	Y	X	Y
Self-Aligning Tapered Thrust Roller Bearings[a] $\alpha \neq 0$	tan α	1	1.5 tan α	0.67	tan α	1

[a] For $\alpha = 90°$, $F_r = 0$ and $Y = 1$.

$e = 1.5 \tan \alpha$

Life Adjustment Factors.—In certain applications of ball or roller bearings it is desirable to specify life for a reliability other than 90 per cent. In other cases the bearings may be fabricated from special bearing steels such as vacuum-degassed and vacuum-melted steels, and improved processing techniques. Finally, application conditions may indicate other than normal lubrication, load distribution, or temperature. For such conditions a series of life adjustment factors may be applied to the fatigue life formula. This is fully explained in AFBMA and American National Standard "Load Ratings and Fatigue Life for Ball Bearings"ANSI/AFBMA Std 9–1990 and AFBMA and American National Standard "Load Ratings and Fatigue Life for Roller Bearings"ANSI/AFBMA Std 11–1990. In addition to consulting these standards it may be advantageous to also obtain information from the bearing manufacturer.

Life Adjustment Factor for Reliability.—For certain applications, it is desirable to specify life for a reliability greater than 90 per cent which is the basis of the Rating Life.

To determine the bearing life of ball or roller bearings for reliability greater than 90 per cent, the Rating Life must be adjusted by a factor a_1 such that $L_n = a_1 L_{10}$. For a reliability of 95 per cent, designated as L_5, the life adjustment factor a_1 is 0.62; for 96 per cent, L_4, a_1 is 0.53; for 97 per cent, L_3, a_1 is 0.44; for 98 per cent, L_2, a_1 is 0.33; and for 99 per cent, L_1, a_1 is 0.21.

Life Adjustment Factor for Material.—For certain types of ball or roller bearings which incorporate improved materials and processing, the Rating Life can be adjusted by a factor a_2 such that $L_{10}' = a_2 L_{10}$. Factor a_2 depends upon steel analysis, metallurgical processes, forming methods, heat treatment, and manufacturing methods in general. Ball and roller bearings fabricated from consumable vacuum remelted steels and certain other special analysis steels, have demonstrated extraordinarily long endurance. These steels are of exceptionally high quality, and bearings fabricated from these are usually considered special manufacture. Generally, a_2 values for such steels can be obtained from the bearing manufacturer. However, all of the specified limitations and qualifications for the application of the Rating Life formulas still apply.

Life Adjustment Factor for Application Condition.—Application conditions which affect ball or roller bearing life include: 1) lubrication; 2) load distribution (including effects of clearance, misalignment, housing and shaft stiffness, type of loading, and thermal gradients); and 3) temperature.

Items 2 and 3 require special analytical and experimental techniques, therefore the user should consult the bearing manufacturer for evaluations and recommendations.

Operating conditions where the factor a_3 might be less than 1 include: D) exceptionally low values ofNd_m (rpm times pitch diameter, in mm); e.g.,$Nd_m < 10,000$; E) lubricant viscosity at less than 70 SSU for ball bearings and 100 SSU for roller bearings at operating temperature; and F) excessively high operating temperatures.

When a_3 is less than 1 it may not be assumed that the deficiency in lubrication can be overcome by using an improved steel. When this factor is applied, $L_{10}' = a_3 L_{10}$.

In most ball and roller bearing applications, lubrication is required to separate the rolling surfaces, i.e., rollers and raceways, to reduce the retainer-roller and retainer-land friction and sometimes to act as a coolant to remove heat generated by the bearing.

Factor Combinations.—A fatigue life formula embodying the foregoing life adjustment factors is $L_{10}' = a_1 a_2 a_3 L_{10}$. Indiscriminate application of the life adjustment factors in this formula may lead to serious overestimation of bearing endurance, since fatigue life is only one criterion for bearing selection. Care must be exercised to select bearings which are of sufficient size for the application.

Ball Bearing Static Load Rating.—For ball bearings suitably manufactured from hardened alloy steels, the static radial load rating is that uniformly distributed static radial bearing load which produces a maximum contact stress of 4,000 megapascals (580,000 pounds per square inch). In the case of a single row, angular contact ball bearing, the static radial load rating refers to the radial component of that load which causes a purely radial displacement of the bearing rings in relation to each other. The static axial load rating is that uniformly distributed static centric axial load which produces a maximum contact stress of 4,000 megapascals (580,000 pounds per square inch).

Radial and Angular Contact Groove Ball Bearings: The magnitude of the static load rating C_o in newtons (pounds) for radial ball bearings is found by the formula:

$$C_o = f_o i Z D^2 \cos\alpha \qquad (20)$$

where f_o = a factor for different kinds of ball bearings given in Table 33

i = number of rows of balls in bearing

Z = number of balls per row

D = ball diameter, mm (inches)

α = nominal contact angle, degrees

This formula applies to bearings with a cross sectional raceway groove radius not larger than 0.52 D in radial and angular contact groove ball bearing inner rings and 0.53 D in radial and angular contact groove ball bearing outer rings and self-aligning ball bearing inner rings.

The load carrying ability of a ball bearing is not necessarily increased by the use of a smaller groove radius but is reduced by the use of a larger radius than those indicated above.

Radial or Angular Contact Ball Bearing Combinations: The basic static load rating for two similar single row radial or angular contact ball bearings mounted side by side on the same shaft such that they operate as a unit (duplex mounting) in "back-to-back" or "face-to-face" arrangement is two times the rating of one single row bearing.

The basic static radial load rating for two or more single row radial or angular contact ball bearings mounted side by side on the same shaft such that they operate as a unit (duplex or stack mounting) in "tandem" arrangement, properly manufactured and mounted for equal load distribution, is the number of bearings times the rating of one single row bearing.

Thrust Ball Bearings: The magnitude of the static load rating C_{oa} for thrust ball bearings is found by the formula:

$$C_{oa} = f_o Z D^2 \cos\alpha \qquad (21)$$

where f_o = a factor given in Table 33

Z = number of balls carrying the load in one direction

D = ball diameter, mm (inches)

α = nominal contact angle, degrees

This formula applies to thrust ball bearings with a cross sectional raceway radius not larger than 0.54 D. The load carrying ability of a bearing is not necessarily increased by use of a smaller radius, but is reduced by use of a larger radius.

Roller Bearing Static Load Rating: For roller bearings suitably manufactured from hardened alloy steels, the static radial load rating is that uniformly distributed static radial bearing load which produces a maximum contact stress of 4,000 megapascals (580,000 pounds per square inch) acting at the center of contact of the most heavily loaded rolling element. The static axial load rating is that uniformly distributed static centric axial load which produces a maximum contact stress of 4,000 megapascals (580,000 pounds per square inch) acting at the center of contact of each rolling element.

Table 33. Values of f_o for Calculating Static Load Rating for Ball Bearings

$\frac{D \cos \alpha}{d_m}$	Radial and Angular Contact Groove Type		Radial Self-Aligning		Thrust	
	Metric[a]	Inch[b]	Metric[a]	Inch[b]	Metric[a]	Inch[b]
0.00	12.7	1850	1.3	187	51.9	7730
0.01	13.0	1880	1.3	191	52.6	7620
0.02	13.2	1920	1.3	195	51.7	7500
0.03	13.5	1960	1.4	198	50.9	7380
0.04	13.7	1990	1.4	202	50.2	7280
0.05	14.0	2030	1.4	206	49.6	7190
0.06	14.3	2070	1.5	210	48.9	7090
0.07	14.5	2100	1.5	214	48.3	7000
0.08	14.7	2140	1.5	218	47.6	6900
0.09	14.5	2110	1.5	222	46.9	6800
0.10	14.3	2080	1.6	226	46.4	6730
0.11	14.1	2050	1.6	231	45.9	6660
0.12	13.9	2020	1.6	235	45.5	6590
0.13	13.6	1980	1.7	239	44.7	6480
0.14	13.4	1950	1.7	243	44.0	6380
0.15	13.2	1920	1.7	247	43.3	6280
0.16	13.0	1890	1.7	252	42.6	6180
0.17	12.7	1850	1.8	256	41.9	6070
0.18	12.5	1820	1.8	261	41.2	5970
0.19	12.3	1790	1.8	265	40.4	5860
0.20	12.1	1760	1.9	269	39.7	5760
0.21	11.9	1730	1.9	274	39.0	5650
0.22	11.6	1690	1.9	278	38.3	5550
0.23	11.4	1660	2.0	283	37.5	5440
0.24	11.2	1630	2.0	288	37.0	5360
0.25	11.0	1600	2.0	293	36.4	5280
0.26	10.8	1570	2.1	297	35.8	5190
0.27	10.6	1540	2.1	302	35.0	5080
0.28	10.4	1510	2.1	307	34.4	4980
0.29	10.3	1490	2.1	311	33.7	4890
0.30	10.1	1460	2.2	316	33.2	4810
0.31	9.9	1440	2.2	321	32.7	4740
0.32	9.7	1410	2.3	326	32.0	4640
0.33	9.5	1380	2.3	331	31.2	4530
0.34	9.3	1350	2.3	336	30.5	4420
0.35	9.1	1320	2.4	341	30.0	4350
0.36	8.9	1290	2.4	346	29.5	4270
0.37	8.7	1260	2.4	351	28.8	4170
0.38	8.5	1240	2.5	356	28.0	4060
0.39	8.3	1210	2.5	361	27.2	3950
0.40	8.1	1180	2.5	367	26.8	3880
0.41	8.0	1160	2.6	372	26.2	3800
0.42	7.8	1130	2.6	377	25.7	3720
0.43	7.6	1100	2.6	383	25.1	3640
0.44	7.4	1080	2.7	388	24.6	3560
0.45	7.2	1050	2.7	393	24.0	3480
0.46	7.1	1030	2.8	399	23.5	3400
0.47	6.9	1000	2.8	404	22.9	3320
0.48	6.7	977	2.8	410	22.4	3240
0.49	6.6	952	2.9	415	21.8	3160
0.50	6.4	927	2.9	421	21.2	3080

[a] Use to obtain C_o or C_{oa} in newtons when D is given in mm.

[b] Use to obtain C_o or C_{oa} in pounds when D is given in inches.

Note: Based on modulus of elasticity = 2.07×10^5 megapascals (30×10^6 pounds per square inch) and Poisson's ratio = 0.3.

Radial Roller Bearings: The magnitude of the static load rating C_o in newtons (pounds) for radial roller bearings is found by the formulas:

$$C_o = 44\left(1 - \frac{D\cos\alpha}{d_m}\right) iZl_{eff}D\cos\alpha \qquad \text{(metric)} \qquad (22a)$$

$$C_o = 6430\left(1 - \frac{D\cos\alpha}{d_m}\right) iZl_{eff}D\cos\alpha \qquad \text{(inch)} \qquad (22b)$$

where D = roller diameter, mm (inches); mean diameter for a tapered roller and major diameter for a spherical roller

d_m = mean pitch diameter of the roller complement, mm (inches)

i = number of rows of rollers in bearing

Z = number of rollers per row

l_{eff} = effective length, mm (inches); overall roller length minus roller chamfers or minus grinding undercuts at the ring where contact is shortest

α = nominal contact angle, degrees

Radial Roller Bearing Combinations: The static load rating for two similar single row roller bearings mounted side by side on the same shaft such that they operate as a unit is two times the rating of one single row bearing.

The static radial load rating for two or more similar single row roller bearings mounted side by side on the same shaft such that they operate as a unit (duplex or stack mounting) in "tandem" arrangement, properly manufactured and mounted for equal load distribution, is the number of bearings times the rating of one single row bearing.

Thrust Roller Bearings: The magnitude of the static load rating C_{oa} in newtons (pounds) for thrust roller bearings is found by the formulas:

$$C_{oa} = 220\left(1 - \frac{D\cos\alpha}{d_m}\right) Zl_{eff}D\sin\alpha \qquad \text{(metric)} \qquad (23a)$$

$$C_{oa} = 32150\left(1 - \frac{D\cos\alpha}{d_m}\right) Zl_{eff}D\sin\alpha \qquad \text{(inch)} \qquad (23b)$$

where the symbol definitions are the same as for Formulas (22a) and (22b).

Thrust Roller Bearing Combination: The static axial load rating for two or more similar single direction thrust roller bearings mounted side by side on the same shaft such that they operate as a unit (duplex or stack mounting) in "tandem" arrangement, properly manufactured and mounted for equal load distribution, is the number of bearings times the rating of one single direction bearing. The accuracy of this formula decreases in the case of single direction bearings when $F_r > 0.44\ F_a \cot\alpha$ where F_r is the applied radial load in newtons (pounds) and F_a is the applied axial load in newtons (pounds).

Ball Bearing Static Equivalent Load.—For ball bearings the static equivalent radial load is that calculated static radial load which produces a maximum contact stress equal in magnitude to the maximum contact stress in the actual condition of loading. The static equivalent axial load is that calculated static centric axial load which produces a maximum contact stress equal in magnitude to the maximum contact stress in the actual condition of loading.

Radial and Angular Contact Ball Bearings: The magnitude of the static equivalent radial load P_o in newtons (pounds) for radial and angular contact ball bearings under combined thrust and radial loads is the greater of:

$$P_o = X_oF_r + Y_oF_a \qquad (24)$$

$$P_o = F_r \tag{25}$$

where X_o = radial load factor given in Table 34

Y_o = axial load factor given in Table 34

F_r = applied radial load, newtons (pounds)

F_a = applied axial load, newtons (pounds)

Table 34. Values of X_o and Y_o for Computing Static Equivalent Radial Load P_o of Ball Bearings

Contact Angle	Single Row Bearings[a]		Double Row Bearings	
	X_o	Y_o[b]	X_o	Y_o[b]
RADIAL CONTACT GROOVE BEARINGS[c,a]				
α = 0°	0.6	0.5	0.6	0.5
ANGULAR CONTACT GROOVE BEARINGS				
α = 15°	0.5	0.47	1	0.94
α = 20°	0.5	0.42	1	0.84
α = 25°	0.5	0.38	1	0.76
α = 30°	0.5	0.33	1	0.66
α = 35°	0.5	0.29	1	0.58
α = 40°	0.5	0.26	1	0.52
SELF-ALIGNING BEARINGS				
…	0.5	0.22 cot α	1	0.44 cot α

[a] P_o is always $\geq F_r$.

[b] Values of Y_o for intermediate contact angles are obtained by linear interpolation.

[c] Permissible maximum value of F_a/C_o (where F_a is applied axial load and C_o is static radial load rating) depends on the bearing design (groove depth and internal clearance).

Thrust Ball Bearings: The magnitude of the static equivalent axial load P_{oa} in newtons (pounds) for thrust ball bearings with contact angle $\alpha \neq 90°$ under combined radial and thrust loads is found by the formula:

$$P_{oa} = F_a + 2.3F_r \tan\alpha \tag{26}$$

where the symbol definitions are the same as for Formulas (24) and (25). This formula is valid for all load directions in the case of double direction ball bearings. For single direction ball bearings, it is valid where $F_r/F_a \leq 0.44 \cot\alpha$ and gives a satisfactory but less conservative value of P_{oa} for F_r/F_a up to $0.67 \cot\alpha$.

Thrust ball bearings with $\alpha = 90°$ can support axial loads only. The static equivalent load for this type of bearing is $P_{oa} = F_a$.

Roller Bearing Static Equivalent Load.—The static equivalent radial load for roller bearings is that calculated, static radial load which produces a maximum contact stress acting at the center of contact of a uniformly loaded rolling element equal in magnitude to the maximum contact stress in the actual condition of loading. The static equivalent axial load is that calculated, static centric axial load which produces a maximum contact stress acting at the center of contact of a uniformly loaded rolling element equal in magnitude to the maximum contact stress in the actual condition of loading.

Radial Roller Bearings: The magnitude of the static equivalent radial load P_o in newtons (pounds) for radial roller bearings under combined radial and thrust loads is the greater of:

$$P_o = X_o F_r + Y_o F_a \qquad (27)$$

$$P_o = F_r \qquad (28)$$

where X_o = radial factor given in Table 35

Y_o = axial factor given in Table 35

F_r = applied radial load, newtons (pounds)

F_a = applied axial load, newtons (pounds)

Table 35. Values of X_o and Y_o for Computing Static Equivalent Radial Load P_o for Self-Aligning and Tapered Roller Bearings

Bearing Type	Single Row[a]		Double Row	
	X_o	Y_o	X_o	Y_o
Self-Aligningand Tapered $\alpha \neq 0$	0.5	0.22 cot α	1	0.44 cot α

[a] P_o is always $\geq F_r$.

The static equivalent radial load for radial roller bearings with $\alpha = 0°$ and subjected to radial load only is $P_{or} = F_r$.

Note: The ability of radial roller bearings with $\alpha = 0°$ to support axial loads varies considerably with bearing design and execution. The bearing user should therefore consult the bearing manufacturer for recommendations regarding the evaluation of equivalent load in cases where bearings with $\alpha = 0°$ are subjected to axial load.

Radial Roller Bearing Combinations: When calculating the static equivalent radial load for two similar single row angular contact roller bearings mounted side by side on the same shaft such that they operate as a unit (duplex mounting) in "back-to-back" or "face-to-face" arrangement, use the X_o and Y_o values for a double row bearing and the F_r and F_a values for the total loads on the arrangement.

When calculating the static equivalent radial load for two or more similar single row angular contact roller bearings mounted side by side on the same shaft such that they operate as a unit (duplex or stack mounting) in "tandem" arrangement, use the X_o and Y_o values for a single row bearing and the F_r and F_a values for the total loads on the arrangement.

Thrust Roller Bearings: The magnitude of the static equivalent axial load P_{oa} in newtons (pounds) for thrust roller bearings with contact angle $\alpha \neq 90°$, under combined radial and thrust loads is found by the formula:

$$P_{oa} = F_a + 2.3 F_r \tan\alpha \qquad (29)$$

where F_a = applied axial load, newtons (pounds)

F_r = applied radial load, newtons (pounds)

α = nominal contact angle, degrees

The accuracy of this formula decreases for single direction thrust roller bearings when $F_r > 0.44\, F_a \cot\alpha$.

Thrust Roller Bearing Combinations: When calculating the static equivalent axial load for two or more thrust roller bearings mounted side by side on the same shaft such that they operate as a unit (duplex or stack mounting) in "tandem" arrangement, use the F_r and F_a values for the total loads acting on the arrangement.

STANDARD METAL BALLS

Standard Metal Balls.—American National Standard ANSI/AFBMA Std 10-1989 provides information for the user of metal balls permitting them to be described readily and accurately. It also covers certain measurable characteristics affecting ball quality.

On the following pages, tables taken from this Standard cover standard balls for bearings and other purposes by type of material, grade, and size range; preferred ball sizes; ball hardness corrections for curvature; various tolerances, marking increments, and maximum surface roughnesses by grades; total hardness ranges for various materials; and minimum case depths for carbon steel balls. The numbers of balls per pound and per kilogram for ferrous and nonferrous metals are also shown.

Definitions and Symbols.—The following definitions and symbols apply to American National Standard metal balls.

Nominal Ball Diameter, D_w: The diameter value that is used for the general identification of a ball size, e.g., ¼ inch, 6 mm, etc.

Single Diameter of a Ball, D_{ws}: The distance between two parallel planes tangent to the surface of a ball.

Mean Diameter of a Ball, D_{wm}: The arithmetical mean of the largest and smallest single diameters of a ball.

Ball Diameter Variation, V_{Dws}: The difference between the largest and smallest single diameters of one ball.

Deviation from Spherical Form, ΔR_w: The greatest radial distance in any radial plane between a sphere circumscribed around the ball surface and any point on the ball surface.

Lot: A definite quantity of balls manufactured under conditions that are presumed uniform, considered and identified as an entirety.

Lot Mean Diameter, D_{wmL}: The arithmetical mean of the mean diameter of the largest ball and that of the smallest ball in the lot.

Lot Diameter Variation, V_{DwL}: The difference between the mean diameter of the largest ball and that of the smallest ball in the lot.

Nominal Ball Diameter Tolerance: The maximum allowable deviation of any ball lot mean diameter from the Nominal Ball Diameter.

Container Marking Increment: The Standard unit steps in millionths of an inch or in micrometers used to express the Specific Diameter.

Specific Diameter: The amount by which the lot mean diameter (D_{wmL}) differs from the nominal diameter (D_w), accurate to the container marking increment for that grade; the specific diameter should be marked on the unit container.

Ball Gage Deviation, ΔS: The difference between the lot mean diameter and the sum of the nominal mean diameter and the ball gage.

Surface Roughness, R_a: Surface roughness consists of all those irregularities that form surface relief and are conventionally defined within the area where deviations of form and waviness are eliminated. (See Handbook Surface Texture Section.)

Ordering Specifications.—Unless otherwise agreed between producer and user, orders for metal balls should provide the following information: quantity, material, nominal ball diameter, grade, and ball gage. A *ball grade* embodies a specific combination of dimensional form, and surface roughness tolerances. A *ball gage(s)* is the prescribed small amount, expressed with the proper algebraic sign, by which the lot mean diameter (arithmetic mean of the mean diameters of the largest and smallest balls in the lot) should differ from the nominal diameter, this amount being one of an established series of amounts as shown in the table below. The 0 ball gage is commonly referred to as “OK”.

Preferred Ball Gages for Grades 3 to 200

Grade	Ball Gages (in 0.0001-in. units) Minus	OK	Plus	Ball Gages (in 1μm units) Minus	OK	Plus
3, 5	− 3 − 2 − 1	0	+ 1 + 2 + 3	− 8 − 7 − 6 − 5 − 4 − 3 − 2 − 1	0	+ 1 + 2 + 3 + 4 + 5 + 6 + 7 + 8
10, 16	− 4 − 3 − 2 − 1	0	+ 1 + 2 + 3 + 4	− 10 − 8 − 6 − 4 − 2	0	+ 2 + 4 + 6 + 8 + 10
24	− 5 − 4 − 3 − 2 − 1	0	+ 1 + 2 + 3 + 4 + 5	− 12 − 10 − 8 − 6 − 4 − 2	0	+ 2 + 4 + 6 + 8 + 10 + 12
48	− 6 − 4 − 2	0	+ 2 + 4 + 6	− 16 − 12 − 8 − 4	0	+ 4 + 8 + 12 + 16
100		0			0	
200		0			0	

Table 1. AFBMA Standard Balls — Tolerances for Individual Balls and for Lots of Balls

Grade	Allowable Ball Diameter Variation	Allowable Deviation from Spherical Form	Maximum Surface Roughness R_a	Allowable Lot Diameter Variation	Nominal Ball Diameter Tolerance (±)	Container Marking Increments
	For Individual Balls			For Lots of Balls		
	Millionths of an Inch					
3	3	3	0.5	5	a	10
5	5	5	0.8	10	a	10
10	10	10	1	20	a	10
16	16	16	1	32	a	10
24	24	24	2	48	a	10
48	48	48	3	96	a	50
100	100	100	5	200	500	a
200	200	200	8	400	1000	a
500	500	500	a	1000	2000	a
1000	1000	1000	a	2000	5000	a
	Micrometers					
3	0.08	0.08	0.012	0.13	a	0.25
5	0.13	0.13	0.02	0.25	a	0.25
10	0.25	0.25	0.025	0.5	a	0.25
16	0.4	0.4	0.025	0.8	a	0.25
24	0.6	0.6	0.05	1.2	a	0.25
48	1.2	1.2	0.08	2.4	a	1.25
100	2.5	2.5	0.125	5	12.5	a
200	5	5	0.2	10	25	a
500	13	13	a	25	50	a
1000	25	25	a	50	125	a

[a] Not applicable.

Allowable ball gage (see text) deviation is for Grade 3: + 0.000030, − 0.000030 inch (+0.75, − 0.75 μm); for Grades 5, 10, and 16: + 0.000050, − 0.000040 inch (+ 1.25, − 1 μm); and for Grade 24: + 0.000100, − 0.000100 inch (+ 2.5, − 2.5 μm). Other grades not given.

Examples: A typical order, in inch units, might read as follows: 80,000 pieces, chrome alloy steel, ¼-inch Nominal Diameter, Grade 16, and Ball Gage to be −0.0002 inch.

A typical order, in metric units, might read as follows: 80,000 pieces, chrome alloy steel, 6 mm Nominal Diameter, Grade 16, and Ball Gage to be −4 μm.

Package Marking: The ball manufacturer or supplier will identify packages containing each lot with information provided on the orders, as given above. In addition, the specific diameter of the contents shall be stated. Container marking increments are listed in Table 1.

Examples: Balls supplied to the order of the first of the previous examples would, if perfect size, be $D_{wmL} = 0.249800$ inch. In Grade 16 these balls would be acceptable with D_{wmL} from 0.249760 to 0.249850 inch. If they actually measured 0.249823 (which would be rounded off to 0.249820), each package would be marked: 5,000 Balls, Chrome Alloy Steel, 1/4″ Nominal Diameter, Grade 16, −0.0002 inch Ball Gage, and −0.000180 inch Specific Diameter.

Balls supplied to the order of the second of the two previous examples would, if perfect size, be $D_{wmL} = 5.99600$ mm. In Grade 16 these balls would be acceptable with a D_{wmL} from 5.99500 to 5.99725 mm. If they actually measured 5.99627 mm (which would be rounded off to 5.99625 mm), each package would be marked: 5,000 Balls, Chrome Alloy Steel, 6 mm Nominal Diameter, Grade 16, −4 μm Ball Gage, and −3.75 μm Specific Diameter.

Table 2. AFBMA Standard Balls — Typical Nominal Size Ranges by Material and Grade

Steel Balls[a]			
		Size Range[b]	
Material	Grade	Inch	mm
Chrome Alloy	3	1/32–1	0.8–25
	5,10, 16,24	1/64–1 1/2	0.3–38
	48, 100, 200, 500	1/32–2 7/8	0.8–75
	1000	3/8–4 1/2	10–115
AISI M-50	3 5,10,16	1/32–1/2	0.8–12
	24,48	1/32–1 5/8	0.8–40
Corrosion Resisting Hardened	3,5,10,16	1/64–3/4	0.3–19
	24	1/32–1	0.8–25
	48	1/32–2	0.8–50
	100,200	1/32–4 1/2	0.8–115
Corrosion-Resisting Unhardened	100,200, 500	1/16–3/4	1.5–19
Carbon Steel[c]	100,200, 500, 1000	1/16–1 1/2	1.5–38
Silicon Molybdenum	200	1/4–1 1/8	6.5–28

Non-Ferrous Balls[a]			
		Size Range[b]	
Material Grade	Grade	Inch	mm
Aluminum	200	1/16–1	1.5–25
Aluminum Bronze	200	13/16–4	20–100
Brass	100,200, 500, 1000	1/16–3/4	1.5–19
Bronze	200,500, 1000	1/16–3/4	1.5–19
Monel Metal 400	100,200, 500	1/16–3/4	1.5–19
K-Monel Metal 500	100	1/16–3/4	1.5–19
	200	1/16–1 11/16	1.5–45
Tungsten Carbide	5	3/64–1/2	1.2–12
	10	3/64–3/4	1.2–19
	16	3/64–1	1.2–25
	24	3/64–1 1/4	1.2–32

[a] For hardness rages see Table 3.
[b] For tolerances see Table 1.
[c] For minimum case depths refer to the Standard.

Table 3. AFBMA Standard Balls—Typical Hardness Ranges

Material	Common Standard	SAE Unified Number	Rockwell Value[a,b]
Steel—			
Alloy tool	AISI/SAE M50	T-11350	60–65 “C”[c,d]
Carbon[e]	AISI/SAE 1008	G-10080	60 Minimum “C”[b]
	AISI/SAE 1013	G-10130	60 Minimum “C”[b]
	AISI/SAE 1018	G-10180	60 Minimum “C”[b]
AISI/SAE 1022	G-10220	60 Minimum “C”[b]	
Chrome alloy	AISI/SAE E52100	G-5298660	60–67 “C”[c,d]
	AISI/SAE E51100	G-51986	60-67 “C”[c,d]
Corrosion-resisting			
	AISI/SAE 440C	S-44004	58–65 “C”[f,d]
hardened	AISI/SAE 440B	S-44003	55–62 “C”[f,d]
	AISI/SAE 420	S-42000	52 Minimum “C”[f,d]
	AISI/SAE 410	S-41000	97 “B”; 41 “C”[f,d]
	AISI/SAE 329	S-32900	45 Minimum “C”[f,d]
Corrosion-resisting			
unhardened	AISI/SAE 3025	S-30200	25–39 “C”[d,g]
	AISI/SAE 304	S-304000	25–39 “C”[d,g]
	AISI/SAE 305	S-305000	25–39 “C”[d,g]
	AISI/SAE 316	S-31600	25–39 “C”[d,g]
	AISI/SAE 430	S-43000	48–63 “A”[d]
Silicon molybdenum	AISI/SAE S2	T-41902	52–60 “C”[c]
Aluminum	AA-2017	A-92017	54–72 “B”
Aluminium bronze	CDA-624	C-62400	94–98 “B”
	CDA-630	C-63000	94–98 “B”
Brass	CDA-260	C-26000	75–87 “B”
Bronze	CDA-464	C-46400	75–98 “B”
Monel 400	AMS-4730	N-04400	85–95 “B”
Monel K-500	QA-N-286	N-05500	24 Minimum “C”
Tungsten carbide	JIC Carbide Classification	…	84-91.5 “A”

[a] Rockwell Hardness Tests shall be conducted on parallel flats in accordance with ASTM Standard E-18 unless otherwise specified.

[b] Hardness readings taken on spherical surfaces are subject to the corrections shown in Table 5. Hardness readings for carbon steel balls smaller than 5 mm ($\frac{1}{4}$ inch) shall be taken by the microhardness method (detailed in ANSI/AFBMA Std 10-1989) or as agreed between manufacturer and purchaser.

[c] Hardness of balls in any one lot shall be within 3 points on Rockwell C scale.

[d] When microhardness method (see ANSI/AFBMA Std 10-1989 is used, the Rockwell hardness values given are converted to DPH in accordance with ASTM Standard E 140, “Standard Hardness Conversion Tables for Metals.”

[e] Choice of carbon steels shown to be at ball manufacturer's option.

[f] Hardness of balls in any one lot shall be within 4 points on Rockwell C scale.

[g] Annealed hardness of 75-90 “B” is available when specified.

For complete details as to material requirements, quality specifications, quality assurance provisions, and methods of hardness testing, reference should be made to the Standard.

Table 4. Preferred Ball Sizes

Nominal Ball Sizes Metric	Diameter mm	Diameter Inches	Nominal Ball Sizes Inch	Nominal Ball Sizes Metric	Diameter mmb	Diameter Inches	Nominal Ball Sizes Inch
0.3	0.300 00	0.011 810			0.793 75	0.031 250	$^1/_{32}$
	0.396 88	0.015 625	$^1/_{64}$	0.8	0.800 00	0.031 496	
0.4	0.400 00	0.015 750		1	1.000 00	0.039 370	
0.5	0.500 00	0.019 680			1.190 63	0.046 875	$^3/_{64}$
	0.508 00	0.020 000	0.020	1.2	1.200 00	0.047 240	
0.6	0.600 00	0.023 620		1.5	1.500 00	0.059 060	
	0.635 00	0.025 000	0.025		1.587 50	0.062 500	$^1/_{16}$
0.7	0.700 00	0.027 560			1.984 38	0.078 125	$^5/_{64}$
2	2.000 00	0.078 740		21	21.000 000	0.826 770	
	2.381 25	0.093 750	$^3/_{32}$		21.431 25	0.843 750	$^{27}/_{32}$
2.5	2.500 00	0.098 420		22	22.000 00	0.866 140	
	2.778 00	0.109 375	$^7/_{64}$		22.225 00	0.875 000	$^7/_8$
3	3.000 00	0.118 110		23	23.000 00	0.905 510	
	3.175 00	0.125 000	$^1/_8$		23.018 75	0.906 250	$^{29}/_{32}$
3.5	3.500 00	0.137 800			23.812 50	0.937 500	$^{15}/_{16}$
	3.571 87	0.140 625	$^9/_{64}$	24	24.000 00	0.944 880	
	3.968 75	0.156 250	$^5/_{32}$		24.606 25	0.968 750	$^{31}/_{32}$
4	4.000 00	0.157 480		25	25.000 00	0.984 250	
	4.365 63	0.171 875	$^{11}/_{64}$		25.400 00	1.000 000	1
4.5	4.500 00	0.177 160		26	26.000 00	1.023 620	
	4.762 50	0.187 500	$^3/_{16}$		26.987 50	1.062 500	$1^1/_{16}$
5	5.000 00	0.196 850		28	28.000 00	1.102 360	
5.5	5.500 00	0.216 540			28.575 00	1.125 000	$1^1/_8$
	5.556 25	0.218 750	$^7/_{32}$	30	30.000 00	1.181 100	
	5.953 12	0.234 375	$^{15}/_{64}$		30.162 50	1.187 500	$1^3/_{16}$
6	6.000 00	0.236 220			31.750 00	1.250 000	$1^1/_4$
	6.350 00	0.250 000	$^1/_4$	32	32.000 00	1.259 840	
6.5	6.500 00	0.255 900			33.337 50	1.312 500	$1^5/_{16}$
	6.746 88	0.265 625	$^{17}/_{64}$	34	34.000 00	1.338 580	
7	7.000 00	0.275 590			34.925 00	1.375 000	$1^3/_8$
	7.143 75	0.281 250	$^9/_{32}$	35	35.000 00	1.377 950	
7.5	7.500 00	0.295 280		36	36.000 00	1.417 320	
	7.540 63	0.296 875	$^{19}/_{64}$		36.512 50	1.437 500	$1^7/_{16}$
	7.937 50	0.312 500	$^5/_{16}$	38	38.000 00	1.496 060	
8	8.000 00	0.314 960			38.100 00	1.500 000	$1^1/_2$
8.5	8.500 00	0.334 640			39.687 50	1.562 500	$1^9/_{16}$
	8.731 25	0.343 750	$^{11}/_{32}$	40	40.000 00	1.574 800	
9	9.000 00	0.354 330			41.275 00	1.625 000	$1^5/_8$
	9.128 12	0.359 375	$^{23}/_{64}$		42.862 50	1.687 500	$1^{11}/_{16}$
	9.525 00	0.375 000	$^3/_8$		44.450 00	1.750 000	$1^3/_4$
	9.921 87	0.390 625	$^{25}/_{64}$	45	45.000 00	1.771 650	
10	10.000 00	0.393 700			46.037 50	1.812 500	$1^{13}/_{16}$
	10.318 75	0.406 250	$^{13}/_{32}$		47.625 00	1.875 000	$1^7/_8$
11	11.000 00	0.433 070			49.212 50	1.937 500	$1^{15}/_{16}$
	11.112 50	0.437 500	$^7/_{16}$	50	50.000 00	1.968 500	
11.5	11.500 00	0.452 756			50.800 00	2.000 000	2
	11.509 38	0.453 125	$^{29}/_{64}$		53.975 00	2.125 000	$2^1/_8$

Table 4. *(Continued)* **Preferred Ball Sizes**

Nominal Ball Sizes Metric	Diameter mm	Diameter Inches	Nominal Ball Sizes Inch	Nominal Ball Sizes Metric	Diameter mmb	Diameter Inches	Nominal Ball Sizes Inch
	11.906 25	0.468 750	15⁄32	55	55.000 00	2.165 354	
12	12.000 00	0.472 440			57.150 00	2.250 000	2¼
	12.303 12	0.484 375	31⁄64	60	60.000 00	2.362 205	
	12.700 00	0.500 000	½		60.325 00	2.375 00	2⅜
13	13.000 00	0.511 810			63.500 00	2.500 000	2½
	13.493 75	0.531 250	17⁄32	65	65.000 00	2.559 055	
14	14.000 00	0.551 180			66.675 00	2.625 000	2⅝
	14.287 50	0.562 500	9⁄16		69.850 00	2.750 000	2¾
15	15.000 00	0.590 550			73.025 00	2.875 000	2⅞
	15.081 25	0.593 750	19⁄32		76.200 00	3.000 000	3
	15.875 00	0.625 000	⅝		79.375 00	3.125 000	3 ⅛
16	16.000 00	0.629 920			82.550 00	3.250 000	3¼
	16.668 75	0.656 250	21⁄32		85.725 00	3.375 00	3⅜
17	17.000 00	0.669 290			88.900 00	3.500 000	3½
	17.462 50	0.687 500	11⁄16		92.075 00	3.625 000	3⅝
18	18.000 00	0.708 660			95.250 00	3.750 000	3¾
	18.256 25	0.718 750	23⁄32		98.425 00	3.875 000	3⅞
19	19.000 00	0.748 030			101.600 00	4.000 000	4
	19.050 00	0.750 000	¾		104.775 00	4.125 000	4⅛
	19.843 75	0.781 250	25⁄32		107.950 00	4.250 000	4¼
20	20.000 00	0.787 400			111.125 00	4.375 000	4⅜
	20.637 50	0.812 500	13⁄16		114.300 00	4.500 000	4½

Table 5. Ball Hardness Corrections for Curvatures

Hardness Reading, Rockwell C	Ball Diameters, Inch						
	¼	5⁄16	⅜	½	⅝	¾	1
	Correction—Rockwell C						
20	12.1	9.3	7.7	6.1	4.9	4.1	3.1
25	11.0	8.4	7.0	5.5	4.4	3.7	2.7
30	9.8	7.5	6.2	4.9	3.9	3.2	2.4
35	8.6	6.6	5.5	4.3	3.4	2.8	2.1
40	7.5	5.7	4.7	3.6	2.9	2.4	1.7
45	6.3	4.9	4.0	3.0	2.4	1.9	1.4
50	5.2	4.0	3.2	2.4	1.9	1.5	1.1
55	4.1	3.1	2.5	1.8	1.4	1.1	0.8
60	2.9	2.2	1.8	1.2	0.9	0.7	0.4
65	1.8	1.3	1.0	0.5	0.3	0.2	0.1
20	12.8	9.3	7.6	6.6	5.2	4.0	3.2
25	11.7	8.4	6.9	5.9	4.6	3.5	2.8
30	10.5	7.5	6.1	5.2	4.1	3.1	2.4
35	9.4	6.6	5.4	4.6	3.6	2.7	2.1
40	8.0	5.7	4.5	3.8	3.0	2.2	1.8
45	6.7	4.9	3.8	3.2	2.5	1.8	1.4
50	5.5	4.0	3.0	2.6	2.0	1.4	1.1
55	4.3	3.1	2.3	1.9	1.5	1.0	0.8
60	3.0	2.2	1.7	1.2	1.0	0.6	0.4
65	1.9	1.3	0.9	0.6	0.4	0.2	0.1

Corrections to be added to Rockwell C readings obtained on spherical surfaces of chrome alloy steel, corrosion resisting hardened and unhardened steel, and carbon steel balls. For other ball sizes and hardness readings, interpolate between correction values shown.

Table 6. Number of Metal Balls per Pound

Nom. Dia.,[a] Inches	Material Density, Pounds per Cubic Inch												
	.101	.274	.277	.279	.283	.284	.286	.288	.301	.304	.306	.319	.540
1/32	620 000	228 000	226 000	224 000	221 000	220 000	219 000	217 000	208 000	206 000	205 000	196 000	116 000
1/16	77 500	28 600	28 200	28 000	27 600	27 500	27 400	27 200	26 000	25 700	25 600	24 500	14 500
3/32	22 900	8 460	8 370	8 310	8 190	8 160	8 100	8 050	7 700	7 620	7 570	7 270	4 290
1/8	9 680	3 570	3 530	3 500	3 460	3 440	3 420	3 400	3 250	3 220	3 200	3 070	1 810
5/32	4 960	1 830	1 810	1 790	1 770	1 760	1 750	1 740	1 660	1 650	1 640	1 570	927
3/16	2 870	1 060	1 050	1 040	1 020	1 020	1 010	1 010	963	953	947	908	537
7/32	1 810	666	659	654	645	642	638	634	606	600	596	572	338
1/4	1 210	446	441	438	432	430	427	424	406	402	399	383	226
9/32	850	313	310	308	303	302	300	298	285	282	281	269	159
5/16	620	228	226	224	221	220	219	217	208	206	205	196	116
11/32	466	172.	170.	169.	166.	166.	164.	163.	156.	155.	154.	147.	87.1
3/8	359	132.	131.	130.	128.	128.	127.	126.	120.	119.	118.	114.	67.1
13/32	282	104.	103.	102.	101.	100.	99.6	98.9	94.6	93.7	93.1	89.3	52.8
7/16	226	83.2	82.3	81.7	80.6	80.3	79.7	79.2	75.8	75.0	74.5	71.5	42.2
15/32	184	67.7	66.9	66.5	65.5	65.3	64.8	64.4	61.6	61.0	60.6	58.1	34.3
1/2	151.	55.8	55.2	54.8	54.0	53.8	53.4	53.1	50.8	50.3	49.9	47.9	28.3
17/32	126.	46.5	46.0	45.7	45.0	44.9	44.5	44.2	42.3	41.9	41.6	39.9	23.6
9/16	106.	39.2	38.7	38.5	37.9	37.8	37.5	37.3	35.7	35.3	35.1	33.6	19.9
19/32	90.3	33.3	32.9	32.7	32.2	32.1	31.9	31.7	30.3	30.0	29.8	28.6	16.9
5/8	77.5	28.6	28.2	28.0	27.6	27.5	27.4	27.2	26.0	25.7	25.6	24.5	14.5
21/32	66.9	24.7	24.4	24.2	23.9	23.8	23.6	23.5	22.5	22.2	22.1	21.2	12.5
11/16	58.2	21.5	21.2	21.1	20.8	20.7	20.6	20.4	19.5	19.3	19.2	18.4	10.9
23/32	50.9	18.8	18.6	18.4	18.2	18.1	18.0	17.9	17.1	16.9	16.8	16.1	9.53
3/4	44.8	16.5	16.3	16.2	16.0	15.9	15.8	15.7	15.0	14.9	14.8	14.2	8.38
25/32	39.7	14.6	14.5	14.4	14.2	14.1	14.0	13.9	13.3	13.2	13.1	12.6	7.42
13/16	35.3	13.0	12.9	12.8	12.6	12.5	12.5	12.4	11.8	11.7	11.6	11.2	6.59
27/32	31.5	11.6	11.5	11.4	11.2	11.2	11.1	11.0	10.6	10.5	10.4	9.97	5.89
7/8	28.2	10.4	10.3	10.2	10.1	10.0	9.97	9.90	9.47	9.38	9.32	8.94	5.28
29/32	25.4	9.37	9.26	9.20	9.07	9.04	8.97	8.91	8.53	8.44	8.39	8.04	4.75
15/16	22.9	8.46	8.37	8.31	8.19	8.16	8.10	8.05	7.70	7.62	7.57	7.27	4.29
31/32	20.8	7.67	7.58	7.53	7.42	7.40	7.35	7.29	6.98	6.91	6.87	6.59	3.89
1	18.9	6.97	6.89	6.85	6.75	6.72	6.68	6.63	6.35	6.28	6.24	5.99	3.54

[a] For sizes above 1 in. diameter, use the following formula: No. balls per pound = 1.91 ÷ [(nom. dia., in.)^3 × (material density, lbs. per cubic in.)].

Ball material densities in pounds per cubic inch: aluminum .101; aluminum bronze .274; corrosion resisting hardened steel .277; AISI M-50 and silicon molybdenum steels .279; chrome alloy steel .283; carbon steel .284; AISI 302 corrosion resisting unhardened steel .286; AISI 316 corrosion resisting unhardened steel .288; bronze .304; brass and K-Monel metal .306; Monel metal .319; and tungsten carbide .540.

Table 7. Number of Metal Balls per Kilogram

Nom.Dia.,[a] mm	Material Density, Grams per Cubic Centimeter												
	2.796	7.584	7.667	7.723	7.833	7.861	7.916	7.972	8.332	8.415	8.470	8.830	14.947
0.3	25 300 000	9 330 000	9 230 000	9 160 000	9 030 000	9 000 000	8 940 000	8 870 000	8 490 000	8 410 000	8 350 000	8 010 000	4 730 000
0.4	10 670 000	3 930 000	3 890 000	3 860 000	3 810 000	3 800 000	3 770 000	3 740 000	3 580 000	3 550 000	3 520 000	3 380 000	2 000 000
0.5	5 470 000	2 010 000	1 990 000	1 980 000	1 950 000	1 940 000	1 930 000	1 920 000	1 830 000	1 820 000	1 800 000	1 730 000	1 020 000
0.7	1 990 000	734 000	726 000	721 000	711 000	708 000	703 000	698 000	668 000	662 000	657 000	631 000	373 000
0.8	1 330 000	492 000	487 000	483 000	476 000	475 000	471 000	468 000	448 000	443 000	440 000	422 000	250 000
1.0	683 000	252 000	249 000	247 000	244 000	243 000	241 000	240 000	229 000	227 000	225 000	216 000	128 000
1.2	395 000	146 000	144 000	143 000	141 000	141 000	140 000	139 000	133 000	131 000	130 000	125 000	73 900
1.5	202 000	74 600	73 800	73 300	72 200	72 000	71 500	71 000	67 900	67 200	66 800	64 100	37 900
2.0	85 400	31 500	31 100	30 900	30 500	30 400	30 200	29 900	28 700	28 400	28 200	27 000	16 000
2.5	43 700	16 100	15 900	15 800	15 600	15 500	15 400	15 300	14 700	14 500	14 400	13 800	8 180
3.0	25 300	9 330	9 230	9 160	9 030	9 000	8 940	8 870	8 490	8 410	8 350	8 010	4 730
3.5	15 900	5 870	5 810	5 770	5 690	5 670	5 630	5 590	5 350	5 290	5 260	5 040	2 980
4.0	10 700	3 930	3 890	3 860	3 810	3 800	3 770	3 740	3 580	3 550	3 520	3 380	2 000
4.5	7 500	2 760	2 730	2 710	2 680	2 670	2 650	2 630	2 520	2 490	2 470	2 370	1 400
5.0	5 470	5 010	1 990	1 980	1 950	1 940	1 930	1 920	1 830	1 820	1 800	1 730	1 020
5.5	4 110	1 510	1 500	1 490	1 470	1 460	1 450	1 440	1 380	1 360	1 360	1 300	768
6.0	3 160	1 170	1 150	1 140	1 130	1 120	1 120	1 110	1 060	1 050	1 040	1 000	592
6.5	2 490	917	907	901	888	885	878	872	835	826	821	788	465
7.0	1 990	734	726	721	711	708	703	698	668	662	657	631	373
7.5	1 620	597	590	586	578	576	572	568	543	538	534	513	303
8.0	1 330	492	487	483	476	475	471	468	448	443	440	422	250
8.5	1 110	410	406	403	397	396	393	390	373	370	367	352	208
9.0	937	345	342	339	334	333	331	329	314	311	309	297	175
10.0	683	252	249	247	244	243	241	240	229	227	225	216	128
11.0	513.0	189.0	187.0	186.0	183.0	183.0	181.0	180.0	172.0	171.0	169.0	163.0	96.0
11.5	449.0	166.0	164.0	163.0	160.0	160.0	159.0	158.0	151.0	149.0	148.0	142.0	84.0
12.0	395.0	146.0	144.0	143.0	141.0	141.0	140.0	139.0	133.0	131.0	130.0	125.0	73.9
13.0	311.0	115.0	113.0	113.0	111.0	111.0	110.0	109.0	104.0	103.0	103.0	98.5	58.2
14.0	249.0	91.8	90.8	90.1	88.9	88.5	87.9	87.3	83.5	82.7	82.2	78.8	46.6
15.0	202.0	74.6	73.8	73.3	72.2	72.0	71.5	71.0	67.9	67.2	66.8	64.1	37.9
16.0	167.0	61.5	60.8	60.4	59.5	59.3	58.9	58.5	56.0	55.4	55.1	52.8	31.2
17.0	139.0	51.3	50.7	50.3	49.6	49.5	49.1	48.8	46.7	46.2	45.9	44.0	26.0

[a] For sizes above 17 mm diameter, use the following formula: No. balls per kilogram = 1,910,000 ÷ [(nom. dia., mm)^3 × (material density, grams per cu. cm)].

Ball material densities in grams per cubic centimeter: aluminum, 2.796; aluminum bronze, 7.584; corrosion-resisting hardened steel, 7.677; AISI M-50 and silicon molybdenum steel, 7.723; chrome alloy steel, 7.833; carbon steel, 7.861; AISI 302 corrosion-resisting unhardened steel, 7.916; AISI 316 corrosion-resisting unhardened steel, 7.972; bronze, 8.415; brass and K-Monel metal, 8.470; Monel metal, 8.830; tungsten carbide, 14.947.

LUBRICANTS AND LUBRICATION

A lubricant is used for one or more of the following purposes: to reduce friction; to prevent wear; to prevent adhesion; to aid in distributing the load; to cool the moving elements; and to prevent corrosion.

The range of materials used as lubricants has been greatly broadened over the years, so that in addition to oils and greases, many plastics and solids and even gases are now being applied in this role. The only limitations on many of these materials are their ability to replenish themselves, to dissipate frictional heat, their reaction to high environmental temperatures, and their stability in combined environments. Because of the wide selection of lubricating materials available, great care is advisable in choosing the material and the method of application. The following types of lubricants are available: petroleum fluids, synthetic fluids, greases, solid films, working fluids, gases, plastics, animal fat, metallic and mineral films, and vegetable oils.

Lubricating Oils.—The most versatile and best-known lubricant is mineral oil. When applied in well-designed applications that provide for the limitations of both mechanical and hydraulic elements, oil is recognized as the most reliable lubricant. Concurrently, it is offered in a wide selection of stocks, carefully developed to meet the requirements of the specific application.

Lubricating oils are seldom marketed without additives blended for a narrow range of applications. These "additive packages" are developed for particular applications, so it is advisable to consult the sales-engineering representatives of a reputable petroleum company on the proper selection for the conditions under consideration. The following are the most common types of additives: wear preventive, oxidation inhibitor, rust inhibitor, detergent-dispersant, viscosity index improver, defoaming agent, and pour-point depressant.

A more recent development in the field of additives is a series of organic compounds that leave no ash when heated to a temperature high enough to evaporate or burn off the base oil. Initially produced for internal-combustion-engine applications these additives have found ready acceptance in those other applications where metallic or mineral trace elements would promote catalytic, corrosive, deposition, or degradation effects on mechanism materials.

Additives usually are not stable over the entire temperature and shear-rate ranges considered acceptable for the base stock oil application. Because of this problem, additive type oils must be carefully monitored to ensure that they are not continued in service after their principal capabilities have been diminished or depleted. Of primary importance in this regard is the action of the detergent-dispersant additives that function so well to reduce and control degradation products that would otherwise deposit on the operating parts and oil cavity walls. Because the materials cause the oil to carry a higher than normal amount of the breakdown products in a fine suspension, they may cause an accelerated deposition rate or foaming when they have been depleted or degenerated by thermal or contamination action. Ingestion of water by condensation or leaking can cause markedly harmful effects.

Viscosity index improvers serve to modify oils so that their change in viscosity is reduced over the operating temperature range. These materials may be used to improve both a heavy or a light oil; however, the original stock will tend to revert to its natural state when the additive has been depleted or degraded due to exposure to high temperatures or to the high shear rates normally encountered in the load-carrying zones of bearings and gears. In heavy-duty installations, it is generally advisable to select a heavier or a more highly refined oil (and one that is generally more costly) rather than to rely on a less stable viscosity-index-improvement product. Viscosity-index-improved oils are generally used in applications where the shear rate is well below 1,000,000 reciprocal seconds, as determined by the following formula:

$$\text{Shear rate}(s^{-1}) = \frac{DN}{60t}$$

where D is the journal diameter in inches, N is the journal speed in rpm, and t is the film thickness in inches.

Types of Oils.—Aside from being aware of the many additives available to satisfy particular application requirements and improve the performance of fluids, the designer must also be acquainted with the wide variety of oils, natural and synthetic, which are also available. Each oil has its own special features that make it suitable for specific applications and limit its utility in others. Though a complete description of each oil and its application feasibility cannot be given here, reference to major petroleum and chemical company sales engineers will provide full descriptions and sound recommendations. In some applications, however, it must be accepted that the interrelation of many variables, including shear rate, load, and temperature variations, prohibit precise recommendations or predictions of fluid durability and performance. Thus, prototype and rig testing are often required to ensure the final selection of the most satisfactory fluid.

The following table lists the major classifications and properties of available commercial petroleum oils.

Properties of Commercial Petroleum Oils and Their Applications

Group A				Group B			
Type	Viscosity,Centistokes		Density, g/cc at 60°F	Type	Viscosity,Centistokes		Density, g/cc at 60°F
	100°F	210°F			100°F	210°F	
SAE 10 W	41	6.0	0.870	General Purpose	22	3.9	0.880
SAE 20 W	71	8.5	0.885		44	6.0	0.898
SAE 30	114	11.2	0.890		66	7.0	0.915
SAE 40	173	14.5	0.890		110	9.9	0.915
SAE 50	270	19.5	0.900		200	15.5	0.890
Group C				Group D			
SAE 75	47	7.0		Turbine			
SAE 80	69	8.0		Light	32	5.5	0.871
SAE 90	285	20.5	0.930, approx.	Medium	65	8.1	0.876
SAE 140	725	34.0		Heavy	99	10.7	0.885
SAE 250	1,220	47.0					
Group E				Group F			
	5	1.5	0.858		76	9.3	0.875
Aviation	10	2.5	0.864	Aviation	268	20.0	0.891
					369	25.0	0.892

Group A. Automotive. With increased additives, diesel and marine reciprocating engines.
Group B. Gear trains and transmissions. With E. P. additives, hypoid gears.
Group C. Machine tools and other industrial applications.
Group D. Marine propulsion and stationary power turbines.
Group E. Turbojet engines.
Group F. Reciprocating engines.

Viscosity.—As noted before, fluids used as lubricants are generally categorized by their viscosity at 100 and 210 deg. F. Absolute viscosity is defined as a fluid's resistance to shear or motion—its internal friction in other words. This property is described in several ways, but basically it is the force required to move a plane surface of unit area with unit speed parallel to a second plane and at unit distance from it. In the metric system, the unit of viscosity is called the "poise" and in the English system is called the "reyn." One reyn is equal to 68,950 poises. One poise is the viscosity of a fluid, such that one dyne force is required to move a surface of one square centimeter with a speed of one centimeter per second, the distance between surfaces being one centimeter. The range of kinematic viscosity for a series of typical fluids is shown in the table on page 2312. Kinematic viscosity is related directly to the flow time of a fluid through the viscosimeter capillary. By multiplying the kinematic viscosity by the density of the fluid at the test temperature, one can determine the absolute viscosity. Because, in the metric system, the mass density is equal to the specific gravity, the conversion from kinematic to absolute viscosity is generally made in this system and then converted to English units where required. The densities of typical lubricating fluids with comparable viscosities at 100 deg. F and 210 deg. F are shown in this same table.

The following conversion table may be found helpful.

Multiply	By	To Get
Centipoises, $Z, \frac{\text{dyne-s}}{100\ \text{cm}^2}$	1.45×10^{-7}	Reyns, $\mu, \frac{\text{lb force-s}}{\text{in.}^2}$
Centistokes, $\nu, \frac{\text{cm}^2}{100\ \text{s}}$	Density in g/cc	Centipoises, $Z, \frac{\text{dyne-s}}{100\ \text{cm}^2}$
Saybolt Universal Seconds, t_s	$0.22 t_s - \frac{180}{t_s}$	Centistokes, $\nu, \frac{\text{cm}^2}{100\ \text{s}}$

Finding Specific Gravity of Oils at Different Temperatures.—The standard practice in the oil industry is to obtain a measure of specific gravity at 60 deg. F on an arbitrary scale, in degrees API, as specified by the American Petroleum Institute. As an example, API gravity, ρ_{API}, may be expressed as 27.5 degrees at 60 deg. F.

The relation between gravity in API degrees and specific gravity (grams of mass per cubic centimeter) at 60 deg. F, ρ_{60}, is

$$\rho_{60} = \frac{141.5}{131.5 + \rho_{API}}$$

The specific gravity, ρ_T, at some other temperature, T, is found from the equation

$$\rho_T = \rho_{60} - 0.00035(T - 60)$$

Normal values of specific gravity for sleeve-bearing lubricants range from 0.75 to 0.95 at 60 deg. F. If the API rating is not known, an assumed value of 0.85 may be used.

Application of Lubricating Oils.—In the selection and application of lubricating oils, careful attention must be given to the temperature in the critical operating area and its effect on oil properties. Analysis of each application should be made with detailed attention given to cooling, friction losses, shear rates, and contaminants.

Many oil selections are found to result in excessive operating temperatures because of a viscosity that is initially too high, which raises the friction losses. As a general rule, the lightest-weight oil that can carry the maximum load should be used. Where it is felt that the load carrying capacity is borderline, lubricity improvers may be employed rather than an arbitrarily higher viscosity fluid. It is well to remember that in many mechanisms the thicker fluid may increase friction losses sufficiently to lower the operating viscosity into the range provided by an initially lighter fluid. In such situations also, improved cooling, such as may be accomplished by increasing the oil flow, can improve the fluid properties in the load zone.

Similar improvements can be accomplished in many gear trains and other mechanisms by reducing churning and aeration through improved scavenging, direction of oil jets, and elimination of obstacles to the flow of the fluid. Many devices, such as journal bearings, are extremely sensitive to the effects of cooling flow and can be improved by greater flow rates with a lighter fluid. In other cases it is well to remember that the load carrying capacity of a petroleum oil is affected by pressure, shear rate, and bearing surface finish as well as initial viscosity and therefore these must be considered in the selection of the fluid. Detailed explanation of these factors is not within the scope of this text; however the technical representatives of the petroleum companies can supply practical guides for most applications.

Other factors to consider in the selection of an oil include the following:

1) Compatibility with system materials
2) Water absorption properties
3) Break-in requirements
4) Detergent requirements

5) Corrosion protection
6) Low temperature properties
7) Foaming tendencies
8) Boundary lubrication properties
9) Oxidation resistance (high temperature properties)
10) Viscosity/temperature stability (Viscosity Temperature Index).

Generally, the factors listed above are those which are usually modified by additives as described earlier. Since additives are used in limited amounts in most petroleum products, blended oils are not as durable as the base stock and must therefore be used in carefully worked-out systems. Maintenance procedures must be established to monitor the oil so that it may be replaced when the effect of the additive is noted or expected to degrade. In large systems supervised by a lubricating engineer, sampling and associated laboratory analysis can be relied on, while in customer-maintained systems as in automobiles and reciprocating engines, the design engineer must specify a safe replacement period which takes into account any variation in type of service or utilization.

Some large systems, such as turbine-power units, have complete oil systems which are designed to filter, cool, monitor, meter, and replenish the oil automatically. In such facilities, much larger oil quantities are used and they are maintained by regularly assigned lubricating personnel. Here reliance is placed on conservatively chosen fluids with the expectation that they will endure many months or even years of service.

Centralized Lubrication Systems.— Various forms of centralized lubrication systems are used to simplify and render more efficient the task of lubricating machines. In general, a central reservoir provides the supply of oil, which is conveyed to each bearing either through individual lines of tubing or through a single line of tubing that has branches extending to each of the different bearings. Oil is pumped into the lines either manually by a single movement of a lever or handle, or automatically by mechanical drive from some revolving shaft or other part of the machine. In either case, all bearings in the central system are lubricated simultaneously. Centralized force-feed lubrication is adaptable to various classes of machine tools such as lathes, planers, and milling machines and to many other types of machines. It permits the use of a lighter grade of oil, especially where complete coverage of the moving parts is assured.

Gravity Lubrication Systems.— Gravity systems of lubrication usually consist of a small number of distributing centers or manifolds from which oil is taken by piping as directly as possible to the various surfaces to be lubricated, each bearing point having its own independent pipe and set of connections. The aim of the gravity system, as of all lubrication systems, is to provide a reliable means of supplying the bearing surfaces with the proper amount of lubricating oil. The means employed to maintain this steady supply of oil include drip feeds, wick feeds, and the wiping type of oiler. Most manifolds are adapted to use either or both drip and wick feeds.

Drip-feed Lubricators: A drip feed consists of a simple cup or manifold mounted in a convenient position for filling and connected by a pipe or duct to each bearing to be oiled. The rate of feed in each pipe is regulated by a needle or conical valve. A loose-fitting cover is usually fitted to the manifold in order to prevent cinders or other foreign matter from becoming mixed with the oil. When a cylinder or other chamber operating under pressure is to be lubricated, the oil-cup takes the form of a lubricator having a tight-fitting screw cover and a valve in the oil line. To fill a lubricator of this kind, it is only necessary to close the valve and unscrew the cover.

Operation of Wick Feeds: For a wick feed, the siphoning effect of strands of worsted yarn is employed. The worsted wicks give a regular and reliable supply of oil and at the same time act as filters and strainers. A wick composed of the proper number of strands is fitted into each oil-tube. In order to insure using the proper sizes of wicks, a study should be made of the oil requirements of each installation, and the number of strands necessary to meet the demands of bearings at different rates of speed should be determined. When the necessary data have been obtained, a table should be prepared showing the size of wick or the number of strands to be used for each bearing of the machine.

Oil-conducting Capacity of Wicks: With the oil level maintained at a point $\frac{3}{8}$ to $\frac{3}{4}$ inch below the top of an oil-tube, each strand of a clean worsted yarn will carry slightly more than one drop of oil a minute. A twenty-four-strand wick will feed approximately thirty drops a minute, which is ordinarily sufficient for operating a large bearing at high speed. The wicks should be removed from the oil-tubes when the machinery is idle. If left in place, they will continue to deliver oil to the bearings until the supply in the cup is exhausted, thus wasting a considerable quantity of oil, as well as flooding the bearing. When bearings require an extra supply of oil temporarily, it may be supplied by dipping the wicks or by pouring oil down the tubes from an oil-can or, in the case of drip feeds, by opening the needle valves. When equipment that has remained idle for some time is to be started up, the wicks should be dipped and the moving parts oiled by hand to insure an ample initial supply of oil. The oil should be kept at about the same level in the cup, as otherwise the rate of flow will be affected. Wicks should be lifted periodically to prevent dirt accumulations at the ends from obstructing the flow of oil.

How Lubricating Wicks are Made: Wicks for lubricating purposes are made by cutting worsted yarn into lengths about twice the height of the top of the oil-tube above the bottom of the oil-cup, plus 4 inches. Half the required number of strands are then assembled and doubled over a piece of soft copper wire, laid across the middle of the strands. The free ends are then caught together by a small piece of folded sheet lead, and the copper wire twisted together throughout its length. The lead serves to hold the lower end of the wick in place, and the wire assists in forcing the other end of the wick several inches into the tube. When the wicks are removed, the free end of the copper wire may be hooked over the tube end to indicate which tube the wick belongs to. Dirt from the oil causes the wick to become gummy and to lose its filtering effect. Wicks that have thus become clogged with dirt should be cleaned or replaced by new ones. The cleaning is done by boiling the wicks in soda water and then rinsing them thoroughly to remove all traces of the soda. Oil-pipes are sometimes fitted with openings through which the flow of oil can be observed. In some installations, a short glass tube is substituted for such an opening.

Wiper-type Lubricating Systems: Wiper-type lubricators are used for out-of-the-way oscillating parts. A wiper consists of an oil-cup with a central blade or plate extending above the cup, and is attached to a moving part. A strip of fibrous material fed with oil from a source of supply is placed on a stationary part in such a position that the cup in its motion scrapes along the fibrous material and wipes off the oil, which then passes to the bearing surfaces.

Oil manifolds, cups, and pipes should be cleaned occasionally with steam conducted through a hose or with boiling soda water. When soda water is used, the pipes should be disconnected, so that no soda water can reach the bearings.

Oil Mist Systems.— A very effective system for both lubricating and cooling many elements which require a limited quantity of fluid is found in a device which generates a mist of oil, separates out the denser and larger (wet) oil particles, and then distributes the mist through a piping or conduit system. The mist is delivered into the bearing, gear, or lubricated element cavity through a condensing or spray nozzle, which also serves to meter the flow. In applications which do not encounter low temperatures or which permit the use of visual devices to monitor the accumulation of solid oil, oil mist devices offer advantages in providing cooling, clean lubricant, pressurized cavities which prevent entrance of contaminants, efficient application of limited lubricant quantities, and near-automatic performance. These devices are supplied with fluid reservoirs holding from a few ounces up to several gallons of oil and with accommodations for either accepting shop air or working from a self-contained compressor powered by electricity. With proper control of the fluid temperature, these units can atomize and dispense most motor and many gear oils.

Lubricating Greases.—In many applications, fluid lubricants cannot be used because of the difficulty of retention, relubrication, or the danger of churning. To satisfy these and other requirements such as simplification, greases are applied. These formulations are usually petroleum oils thickened by dispersions of soap, but may consist of synthetic oils with soap or inorganic thickeners, or oil with silaceous dispersions. In all cases, the thickener, which must be carefully prepared and mixed with the fluid, is used to immobilize the oil, serving as a storehouse from which the oil bleeds at a slow rate. Though the thickener very often has lubricating properties itself, the oil bleeding from the bulk of the grease is the determining lubricating function. Thus, it has been shown that when the oil has been depleted to the level of 50 per cent of the total weight of the grease, the lubricating ability of the material is no longer reliable. In some applications requiring an initially softer and wetter material, however, this level may be as high as 60 per cent.

Grease Consistency Classifications.—To classify greases as to mobility and oil content, they are divided into Grades by the NLGI (National Lubricating Grease Institute). These grades, ranging from 0, the softest, up through 6, the stiffest, are determined by testing in a penetrometer, with the depth of penetration of a specific cone and weight being the controlling criterion. To insure proper averaging of specimen resistance to the cone, most specifications include a requirement that the specimen be worked in a sieve-like device before being packed into the penetrometer cup for the penetration test. Since many greases exhibit thixotropic properties (they soften with working, as they often do in an application with agitation of the bulk of the grease by the working elements or accelerations), this penetration of the worked specimen should be used as a guide to compare the material to the original manufactured condition of it and other greases, rather than to the exact condition in which it will be found in the application. Conversely, many greases are found to stiffen when exposed to high shear rates at moderate loads as in automatic grease dispensing equipment. The application of a grease, therefore must be determined by a carefully planned cut-and-try procedure. Most often this is done by the original equipment manufacturer with the aid of the petroleum company representatives, but in many cases it is advisable to include the bearing engineer as well. In this general area it is well to remember that shock loads, axial or thrust movement within or on the grease cavity can cause the grease to contact the moving parts and initiate softening due to the shearing or working thus induced. To limit this action, grease-lubricated bearing assemblies often utilize dams or dividers to keep the bulk of the grease contained and unchanged by this working. Successful application of a grease depends however, on a relatively small amount of mobile lubricant (the oil bled out of the bulk) to replenish that small amount of lubricant in the element to be lubricated. If the space between the bulk of the mobile grease and the bearing is too large, then a critical delay period (which will be regulated by the grease bleed rate and the temperature at which it is held) will ensue before lubricant in the element can be resupplied. Since most lubricants undergo some attrition due to thermal degradation, evaporation, shearing, or decomposition in the bearing area to which applied, this delay can be fatal.

To prevent this from leading to failure, grease is normally applied so that the material in the cavity contacts the bearing in the lower quadrants, insuring that the excess originally packed into it impinges on the material in the reservoir. With the proper selection of a grease which does not slump excessively, and a reservoir construction to prevent churning, the initial action of the bearing when started into operation will be to purge itself of excess grease, and to establish a flow path for bleed oil to enter the bearing. For this purpose, most greases selected will be of a grade 2 or 3 consistency, falling into the "channelling" variety or designation.

Types of Grease.—Greases are made with a variety of soaps and are chosen for many particular characteristics. Most popular today, however, are the lithium, or soda-soap grease and the modified-clay thickened materials. For high temperature applications (250 deg. F. and above) certain finely divided dyes and other synthetic thickeners are applied. For all-around use the lithium soap greases are best for moderate temperature applications (up to 225 deg. F.) while a number of soda-soap greases have been found to work well up to 285 deg. F. Since the major suppliers offer a number of different formulations for these temperature ranges it is recommended that the user contact the engineering representatives of a reputable petroleum company before choosing a grease. Greases also vary in volatility and viscosity according to the oil used. Since the former will affect the useful life of the bulk applied to the bearing and the latter will affect the load carrying capacity of the grease, they must both be considered in selecting a grease.

For application to certain gears and slow-speed journal bearings, a variety of greases are thickened with carbon, graphite, molybdenum disulfide, lead, or zinc oxide. Some of these materials are likewise used to inhibit fretting corrosion or wear in sliding or oscillating mechanisms and in screw or thread applications. One material used as a “gear grease” is a residual asphaltic compound which is known as a “Crater Compound.” Being extremely stiff and having an extreme temperature-viscosity relationship, its application must also be made with careful consideration of its limitations and only after careful evaluation in the actual application. Its oxidation resistance is limited and its low mobility in winter temperature ranges make it a material to be used with care. However, it is used extensively in the railroad industry and in other applications where containment and application of lubricants is difficult. In such conditions its ability to adhere to gear and chain contact surfaces far outweighs its limitations and in some extremes it is “painted” onto the elements at regular intervals.

Temperature Effects on Grease Life.—Since most grease applications are made where long life is important and relubrication is not too practical, operating temperatures must be carefully considered and controlled. Being a hydro-carbon, and normally susceptible to oxidation, grease is subject to the general rule that: Above a critical threshold temperature, each 15- to 18-deg. F. rise in temperature reduces the oxidation life of the lubricant by half. For this reason, it is vital that all elements affecting the operating temperature of the application be considered, correlated, and controlled. With sealed-for-life bearings, in particular, grease life must be determined for representative bearings and limits must be established for all subsequent applications.

Most satisfactory control can be established by measuring bearing temperature rise during a controlled test, at a consistent measuring point or location. Once a base line and limiting range are determined, all deviating bearings should be dismantled, inspected, and reassembled with fresh lubricant for retest. In this manner mavericks or faulty assemblies will be ferreted out and the reliability of the application established. Generally, a well lubricated grease packed bearing will have a temperature rise above ambient, as measured at the outer race, of from 10 to 50 deg. F. In applications where heat is introduced into the bearing through the shaft or housing, a temperature rise must be added to that of the frame or shaft temperature.

In bearing applications care must be taken not to fill the cavity too full. The bearing should have a practical quantity of grease worked into it with the rolling elements thoroughly coated and the cage covered, but the housing (cap and cover) should be no more than 75 per cent filled; with softer greases, this should be no more than 50 per cent. Excessive packing is evidenced by overheating, churning, aerating, and eventual purging with final failure due to insufficient lubrication. In grease lubrication, *never* add a bit more for good luck — hold to the prescribed amount and determine this with care on a number of representative assemblies.

Relubricating with Grease.—In some applications, sealed-grease methods are not applicable and addition of grease at regular intervals is required. Where this is recommended by the manufacturer of the equipment, or where the method has been worked out as part of a development program, the procedure must be carefully followed. *First,* use the proper lubricant — the same as recommended by the manufacturer or as originally applied (grease performance can be drastically impaired if contaminated with another lubricant). *Second,* clean the lubrication fitting thoroughly with materials which will not affect the mechanism or penetrate into the grease cavity. *Third,* remove the cap (and if applicable, the drain or purge plug). *Fourth*, clean and inspect the drain or scavenge cavity. *Fifth,* weigh the grease gun or calibrate it to determine delivery rate. *Sixth,* apply the directed quantity or fill until grease is detected coming out the drain or purge hole. *Seventh*, operate the mechanism with the drain open so that excess grease is purged. *Last,* continue to operate the mechanism while determining the temperature rise and insure that it is within limits. Where there is access to a laboratory, samples of the purged material may be analyzed to determine the deterioration of the lubricant and to search for foreign material which may be evidence of contamination or of bearing failure.

Normally, with modern types of grease and bearings, lubrication need only be considered at overhaul periods or over intervals of three to ten years.

Solid Film Lubricants.—Solids such as graphite, molybdenum disulfide, polytetrafluoroethylene, lead, babbit, silver, or metallic oxides are used to provide dry film lubrication in high-load, slow-speed or oscillating load conditions. Though most are employed in conjunction with fluid or grease lubricants, they are often applied as the primary or sole lubricant where their inherent limitations are acceptable. Of foremost importance is their inability to carry away heat. Second, they cannot replenish themselves, though they generally do lay down an oriented film on the contacting interface. Third, they are relatively immobile and must be bonded to the substrate by a carrier, by plating, fusing, or by chemical or thermal deposition.

Though these materials do not provide the low coefficient of friction associated with fluid lubrication, they do provide coefficients in the range of 0.4 down to 0.02, depending on the method of application and the material against which they rub. Polytetrafluoroethylene, in normal atmospheres and after establishing a film on both surfaces has been found to exhibit a coefficient of friction down to 0.02. However, this material is subject to cold flow and must be supported by a filler or on a matrix to continue its function. Since it can now be cemented in thin sheets and is often supplied with a fine glass fiber filler, it is practical in a number of installations where the speed and load do not combine to melt the bond or cause the material to sublime.

Bonded films of molybdenum disulfide, using various resins and ceramic combinations as binders, are deposited over phosphate treated steel, aluminum, or other metals with good success. Since its action produces a gradual wear of the lubricant, its life is limited by the thickness which can be applied (not over a thousandth or two in the conventional application). In most applications this is adequate if the material is used to promote break-in, prevent galling or pick-up, and to reduce fretting or abrasion in contacts otherwise impossible to separate.

In all applications of solid film lubricants, the performance of the film is limited by the care and preparation of the surface to which they are applied. If they can't adhere properly, they cannot perform, coming off in flakes and often jamming under flexible components. The best advice is to seek the assistance of the supplier's field engineer and set up a close control of the surface preparation and solid film application procedure. It should be noted that the functions of a good solid film lubricant cannot overcome the need for better surface finishing. Contacting surfaces should be smooth and flat to insure long life and minimum friction forces. Generally, surfaces should be finished to no more than 24 micro-inches AA with wariness no greater than 0.00002 inch.

Anti-friction Bearing Lubrication.—The limiting factors in bearing lubrication are the load and the linear velocity of the centers of the balls or rollers. Since these are difficult to evaluate, a speed factor which consists of the inner race bore diameter × RPM is used as a criterion. This factor will be referred to as S_i where the bore diameter is in inches and S_m where it is in millimeters.

In order to be suitable for use in anti-friction bearings, grease must have the following properties:

1) Freedom from chemically or mechanically active ingredients such as uncombined metals or oxides, and similar mineral or solid contaminants.

2) The slightest possible tendency of change in consistency, such as thickening, separation of oil, evaporation or hardening.

3) A melting point considerably higher than the operating temperatures.

The choice of lubricating oils is easier. They are more uniform in their characteristics and if resistant to oxidation, gumming and evaporation, can be selected primarily with regard to a suitable viscosity.

Grease Lubrication: Anti-friction bearings are normally grease lubricated, both because grease is much easier than oil to retain in the housing over a long period and because it acts to some extent as a seal against the entry of dirt and other contaminants into the bearings. For almost all applications, a No. 2 soda-base grease or a mixed-base grease with up to 5 per cent calcium soap to give a smoother consistency, blended with an oil of around 250 to 300 SSU (Saybolt Universal Seconds) at 100 degrees F. is suitable. In cases where speeds are high, say S_i is 5000 or over, a grease made with an oil of about 150 SSU at 100 degrees F. may be more suitable especially if temperatures are also high. In many cases where bearings are exposed to large quantities of water, it has been found that a standard soda-base ball-bearing grease, although classed as water soluble gives better results than water-insoluble types. Greases are available that will give satisfactory lubrication over a temperature range of −40 degrees to +250 degrees F.

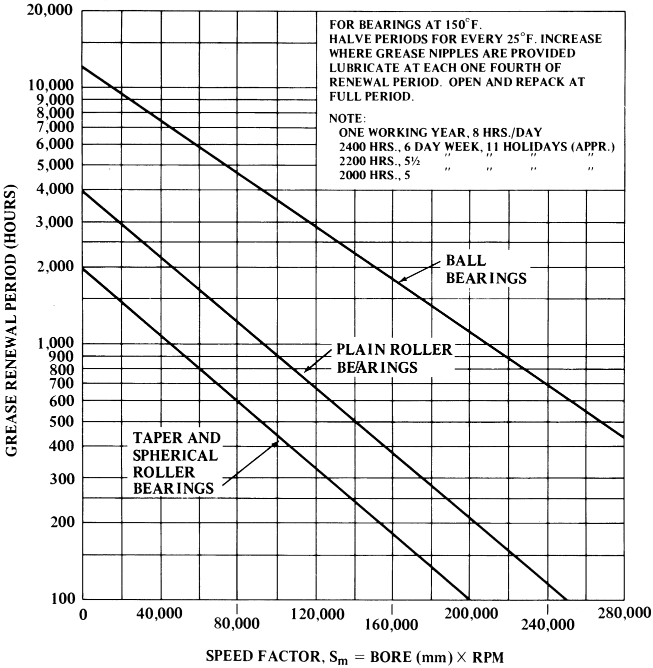
Conservative grease renewal periods will be found in the accompanying chart. Grease should not be allowed to remain in a bearing for longer than 48 months or if the service is very light and temperatures low, 60 months, irrespective of the number of hours' operation during that period as separation of the oil from the soap and oxidation continue whether the bearing is in operation or not.

Before renewing the grease in a hand-packed bearing, the bearing assembly should be removed and washed in clean kerosene, degreasing fluid or other solvent. As soon as the bearing is quite clean it should be washed at once in clean light mineral oil, preferably rust-inhibited. The bearing should *not be spun* before or while it is being oiled. Caustic solutions may be used if the old grease is hard and difficult to remove, but the best method is to soak the bearing for a few hours in light mineral oil, preferably warmed to about 130 degrees F., and then wash in cleaning fluid as described above. The use of chlorinated solvents is best avoided.

When replacing the grease, it should be forced with the fingers between the balls or rollers, dismantling the bearing, if convenient. The available space inside the bearing should be filled completely and the bearing then spun by hand. Any grease thrown out should be wiped off. The space on each side of the bearing in the housing should be not more than half-filled. Too much grease will result in considerable churning, high bearing temperatures and the possibility of early failure. Unlike any other kind of bearing, anti-friction bearings more often give trouble due to over-rather than to under-lubrication.

Grease is usually not very suitable for speed factors over 12,000 for S_i or 300,000 for S_m (although successful applications have been made up to an S_i of 50,000) or for temperatures much over 210 degrees F., 300 degrees F. being the extreme practical upper limit, even if synthetics are used. For temperatures above 210 degrees F., the grease renewal periods are very short.

Oil Lubrication: Oil lubrication is usually adopted when speeds and temperatures are high or when it is desired to adopt a central oil supply for the machine as a whole. Oil for anti-friction bearing lubrication should be well refined with high film strength and good resistance to oxidation and good corrosion protection. Anti-oxidation additives do no harm but are not really necessary at temperatures below about 200 degrees F. Anti-corrosion additives are always desirable. The accompanying table gives recommended viscosities of oil for ball bearing lubrication other than by an air-distributed oil mist. Within a given temperature and speed range, an oil towards the lighter end of the grade should be used, if convenient, as speeds increase. Roller bearings usually require an oil one grade heavier than do ball bearings for a given speed and temperature range. Cooled oil is sometimes circulated through an anti-friction bearing to carry off excess heat resulting from high speeds and heavy loads.



Oil Viscosities and Temperature Ranges for Ball Bearing Lubrication

Maximum Temperature Range Degrees F.	Optimum Temperature Range, Degrees F.	Speed Factor, S_i[a] Under 1000 — Viscosity	Speed Factor, S_i[a] Over 1000 — Viscosity
− 40 to + 100	− 40 to − 10	80 to 90 SSU[b]	70 to 80 SSU[b]
− 10 to + 100	− 10 to + 30	100 to 115 SSU[b]	80 to 100 SSU[b]
+ 30 to + 150	+ 30 to + 150	SAE 20	SAE 10
+ 30 to + 200	+ 150 to + 200	SAE 40	SAE 30
+ 50 to + 300	+ 200 to + 300	SAE 70	SAE 60

[a] Inner race bore diameter (inches) × RPM.

[b] At 100 deg. F.

Not applicable to air-distributed oil mist lubrication.

Aerodynamic Lubrication

A natural extension of hydrodynamic lubrication consists in using air or some other gas as the lubricant. The viscosity of air is 1,000 times smaller than that of a very thin mineral oil. Consequently, the viscous resistance to motion is very much less. However, the distance of nearest approach, i.e. the closest distance between the shaft and the bearing is also correspondingly smaller, so that special precautions must be taken.

To obtain full benefit from such aerodynamic lubrication, the surfaces must have a very fine finish, the alignment must be very good, the speeds must be high and the loading relatively low. If all these conditions are fulfilled extremely successful bearing system can be made to run at very low coefficients of friction. They may also operate at very high temperatures since chemical degradation of the lubricant need not occur. Furthermore, if air is used as the lubricant, it costs nothing. This type of lubrication mechanism is very important for oil-free compressors and gas turbines. Another area of growing application for aerodynamic bearings is in data recording heads for computers. Air is used as the lubricant for the recording heads which are designed to be separated from the magnetic recording disc by a thin air film. The need for high recording densities in magnetic discs necessitates the smallest possible air film thickness between the head and disc. A typical thickness is around 1μm.

The analysis of aerodynamic bearings is very similar to liquid hydrodynamic bearings. The main difference, however, is that the gas compressibility is now a distinctive feature and has to be incorporated into the analysis.

Elastohydrodynamic Lubrication.—In the arrangement of the shaft and bearing it is usually assumed that the surfaces are perfectly rigid and retain their geometric shape during operation. However, a question might be posed: what is the situation if the geometry or mechanical properties of the materials are such that appreciable elastic deformation of the surfaces occurs? Suppose a steel shaft rests on a rubber block. It deforms the block elastically and provides an approximation to a half-bearing (see Figure 1 a).

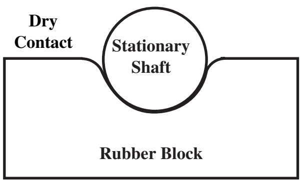


Fig. 1a.

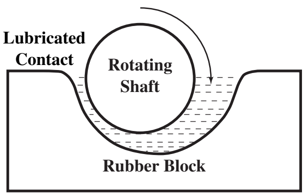


Fig. 1b.

If a lubricant is applied to the system it will be dragged into the interface and, if the conditions are right, it will form a hydrodynamic film. However, the pressures developed in the oil film will now have to match up with the elastic stresses in the rubber. In fact the shape of the rubber will be changed as indicated in Figure 1 b.

This type of lubrication is known as elastohydrodynamic lubrication. It occurs between rubber seals and shafts. It also occurs, rather surprisingly, in the contact between a windshield wiper blade and a windshield in the presence of rain. The geometry of the deformable member, its elastic properties, the load, the speed and the viscosity of the liquid and its dependence on the contact pressure are all important factors in the operation of elastohydrodynamic lubrication.

With conventional journals and bearings the average pressure over the bearing is of the order of 7×10^{-6} N/rn^2. With elastohydrodynamic bearings using a material such as rubber the pressures are perhaps 10 to 20 times smaller. At the other end of pressure spectrum, for instance in gear teeth, contact pressures of the order of 700x10^6 N/in^2 may easily be reached. Because the metals used for gears are very hard this may still be within the range of elastic deformation. With careful alignment of the engaging gear teeth and appropriate surface finish, gears can in fact run successfully under these conditions using an ordinary mineral oil as the lubricant. If the thickness of the elastohydrodynamic film formed at such pressures is calculated it will be found that it is less than an atomic diameter. Sincc even the smoothest metal surfaces are far rougher than this (a millionth of an inch is about 100 atomic diameters) it seems hard to understand why lubrication is effective in these circumstances.

The explanation was first provided by A.N.Grubin in 1949 and a little later (1958) by A.W.Crook. With most mineral oils the application of a high pressure can lead to an enormous increase in viscosity. For example, at a pressure of 700×10^6 N/rn^2 the viscosity may be increased 10,000-fold. The oil entering the gap between the gear teeth is trapped between the surfaces and at the high pressures existing in the contact region behaves virtually like a solid separating layer. This process explains why many mechanisms in engineering practice operate under much severer conditions than the classical theory would allow.

This type of elastohydrodynamic lubrication becomes apparent only when the film thickness is less than about 0.25 to 1 μm. To be exploited successfully it implies that the surfaces must be very smooth and very carefully aligned. If these conditions are met systems such as gears or cams and tappets can operate effectively at very high contact pressures without any metallic contact occurring. The coefficient of friction depends on the load, contact geometry, speed, etc., but generally it lies between about $\mu = 0.01$ at the lightest pressures and $\mu = 0.1$ at the highest pressures. The great success of elastohydrodynamic theory in explaining effective lubrication at very high contact pressures also raises a problem that has not yet been satisfactorily resolved: why do lubricants ever fail, since the harder they are squeezed the harder it is to extrude them? It is possible that high temperature flashes are responsible; alternatively the high rates of shear can actually fracture the lubricant film since when it is trapped between the surfaces it is, instantaneously, more like a wax than an oil.

It is clear that in this type of lubrication the effect of pressure on viscosity is a factor of major importance. It turns out that mineral oils have reasonably good pressure-viscosity characteristics. It appears that synthetic oils do not have satisfactory pressure-viscosity characteristics.

In engineering, two most frequently encountered types of contact are line contact and point contact.

The film thickness for line contact (gears, cam-tappet) can be estimated from:

$$h_o = 2.65 \frac{\alpha^{0.54} (\eta_o U)^{0.7} R_e^{0.43}}{w^{0.13} E_e^{0.03}}$$

In the case of point contact (ball bearings), the film thickness is given by:

$$h_o = 0.84 \alpha \eta_o U^{0.74} 0.41 R_e \left(\frac{E_e}{W} \right)^{0.074}$$

In the above equations the symbols used are defined as:

α = the pressure-viscosity coefficient. A typical value for a mineral oil is 1.8x1 0-8 m^2/N

ν = the viscosity of the lubricant at atmospheric pressure Ns/m^2

U = the entraining surface velocity, $U = (U_A + U_B)/2$ m/s, where the subscripts A and B refer to the velocities of bodies 'A' and 'B' respectively.

W = the load on the contact, N

w = the load per unit width of line contact, N/m

$E_O =$ the reduced Young's modulus $\frac{1}{E_e} = \frac{1}{2}\left(\frac{1-\upsilon_A^2}{E_A} + \frac{1-\upsilon_B^2}{E_B}\right)$ N/m^2 where 'ν_A and ν_B are the Poisson's ratios of the contacting bodies 'A' and 'B' respectively; E_A and E_B are the Young's moduli of the contacting bodies 'A' and 'B' respectively.

R_e = - is the reduced radius of curvature (meters) and is given by different equations for different contact configurations.

In ball bearings (see Figure 2) the reduced radius is given by:

- contact between the ball and inner race: $R_e = \frac{rR_1}{R_1 + r}$
- contact between the ball and outer race: $R_e = \frac{r(R_1 + 2r)}{R_1 + r}$

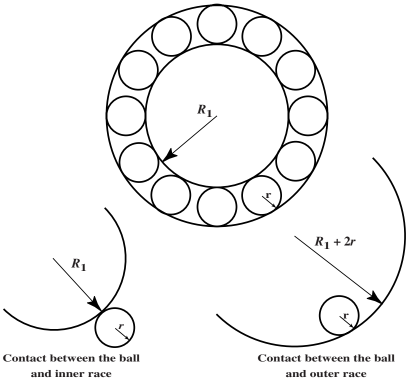


Fig. 2.

For involute gears it can readily be shown that the contact at a distance s from the pitch point can be represented by two cylinders of radii $R_{1,2}$ sin$\psi + s$ rotating with the angular velocity of the wheels (see Figure 3). In the expression below R_1 or R_2 represent pitch radii of the wheels and ψ is the pressure angle. Thus,

$$R_e = \frac{(R_1 \sin\psi + s)(R_2 \sin\psi + s)}{(R_1 + R_2)\sin\psi}$$

The thickness of the film developed in the contact zone between smooth surfaces must be related to the topography of the actual surfaces. The most commonly used parameter for this purpose is the specific film thickness defined as the ratio of the minimum film thickness for smooth surfaces (given by the above equations) to the roughness parameter of the contacting surfaces.

$$\lambda = -\frac{h_o}{\sqrt{R_{m1}^2 + R_{m2}^2}}$$

where $R_m = 1.11R_a$ is the root-mean-square height of surface asperities, and R_a is the centre-line-average height of surface asperities.

If λ is greater than 3 then it is usually assumed that there is full separation of contacting bodies by an elastohydrodynamic film.

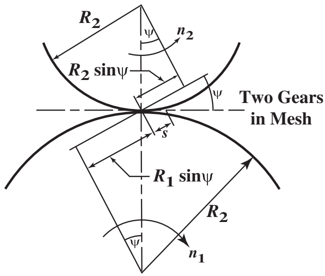


Fig. 3a.

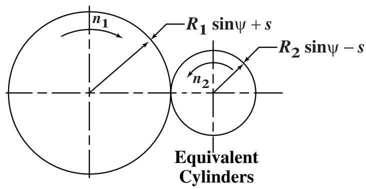


Fig. 3b.

Viscosity-pressure relationship.—Lubricant viscosity increases with pressure. For most lubricants this effect is considerably larger than the effect of temperature or shear when the pressure is appreciably above atmospheric. This is of fundamental importance in the lubrication of highly loaded concentrated contacts such as in rolling contact bearings, gears and cam-tappet systems.

The best known equation to calculate the viscosity of a lubricant at moderate pressures is the Barus equation.

$$\eta_p = \eta_o e^{\alpha p}$$

where η is the viscosity at pressure p (Ns/m^2), η_0 is the viscosity at atmospheric pressure (Ns/in^2), α is the pressure-viscosity coefficient (m^2/N) which can be obtained by plotting the natural logarithm of dynamic viscosity η measured at pressure p. The slope of the graph is α and p is the pressure of concern (N/m^2).

Values of dynamic viscosity η and pressure-viscosity coefficient α for most commonly used lubricants are given in Table 1.

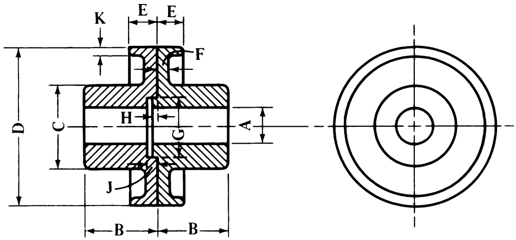
Table 1. Dynamic Viscosity η and Pressure-viscosity Coefficient α for Lubricants

Lubricant	Dynamic viscosity η measured at atmospheric pressure and room temperature $\eta \times 10^{-3}$ Ns/m^2	Pressure-viscosity coefficient α measured at room temperature $\alpha \times 10^{-3}$ m^2/N
Light machine oil	45	28
Heavy machine oil	153	23.7
Cylinder oil	810	34
Spindle oil	18.6	20
Medicinal whale oil	107	29.5
Castor oil	360	15.9
Glycerol (glycerine)	535	5.9

COUPLINGS AND CLUTCHES

Connecting Shafts.—For couplings to transmit up to about 150 horsepower, simple flange-type couplings of appropriate size, as shown in the table, are commonly used. The design shown is known as a safety flange coupling because the bolt heads and nuts are shrouded by the flange, but such couplings today are normally shielded by a sheet metal or other cover.

Safety Flange Couplings



A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	Bolts No.	Bolts Dia.
1	1 3/4	2 1/4	4	11/16	5/16	1 1/2	1/4	9/32	1/4	5	3/8
1 1/4	2 3/16	2 3/4	5	13/16	3/8	1 7/8	1/4	9/32	1/4	5	7/16
1 1/2	2 5/8	3 3/8	6	15/16	7/16	2 1/4	1/4	9/32	1/4	5	1/2
1 3/4	3 1/16	4	7	1 1/16	1/2	2 5/8	1/4	9/32	1/4	5	9/16
2	3 1/2	4 1/2	8	1 3/16	9/16	3	1/4	9/32	5/16	5	5/8
2 1/4	3 15/16	5 1/8	9	1 5/16	5/8	3 3/8	1/4	9/32	5/16	5	11/16
2 1/2	4 3/8	5 5/8	10	1 7/16	11/16	3 3/4	1/4	9/32	5/16	5	3/4
2 3/4	4 13/16	6 1/4	11	1 9/16	3/4	4 1/8	1/4	9/32	5/16	5	13/16
3	5 1/4	6 3/4	12	1 11/16	13/16	4 1/2	1/4	9/32	3/8	5	7/8
3 1/4	5 11/16	7 3/8	13	1 13/16	7/8	4 7/8	1/4	9/32	3/8	5	15/16
3 1/2	6 1/8	8	14	1 15/16	15/16	5 1/4	1/4	9/32	3/8	5	1
3 3/4	6 9/16	8 1/2	15	2 1/16	1	5 5/8	1/4	9/32	3/8	5	1 1/16
4	7	9	16	2 1/4	1 1/8	6	1/4	9/32	7/16	5	1 1/8
4 1/2	7 7/8	10 1/4	18	2 1/2	1 1/4	6 3/4	1/4	9/32	7/16	5	1 1/4
5	8 3/4	11 1/4	20	2 3/4	1 3/8	7 1/2	1/4	9/32	7/16	5	1 3/8
5 1/2	8 3/4	11 1/4	20	2 3/4	1 3/8	7 1/2	1/4	9/32	7/16	5	1 3/8
6	10 1/2	12 3/8	22	2 15/16	1 1/2	8 1/4	5/16	11/32	1/2	5	1 7/16
6 1/2	11 3/8	13 1/2	24	3 1/8	1 5/8	9	5/16	11/32	1/2	5	1 1/2
7	12 1/4	14 5/8	26	3 1/4	1 3/4	9 3/4	5/16	11/32	9/16	6	1 1/2
7 1/2	13 1/8	15 3/4	28	3 7/16	1 7/8	10 1/2	5/16	11/32	9/16	6	1 9/16
8	14	16 7/8	28	3 1/2	2	10 7/8	5/16	11/32	5/8	7	1 1/2
8 1/2	14 7/8	18	30	3 11/16	2 1/8	11 1/4	5/16	11/32	5/8	7	1 9/16
9	15 3/4	19 1/8	31	3 3/4	2 1/4	11 5/8	5/16	11/32	11/16	8	1 1/2
9 1/2	16 5/8	20 1/4	32	3 15/16	2 3/8	12	5/16	11/32	11/16	8	1 9/16
10	17 1/2	21 3/8	34	4 1/8	2 1/2	12 3/4	5/16	11/32	3/4	8	1 5/8
10 1/2	18 3/8	22 1/2	35	4 1/4	2 5/8	13 1/8	5/16	11/32	3/4	10	1 5/8
11	19 1/4	23 5/8	36	4 7/16	2 3/4	13 1/2	5/16	11/32	7/8	10	1 11/16
11 1/2	20 1/8	24 3/4	37	4 5/8	2 7/8	13 7/8	5/16	11/32	7/8	10	1 3/4
12	21	25 7/8	38	4 13/16	3	14 1/4	5/16	11/32	1	10	1 13/16

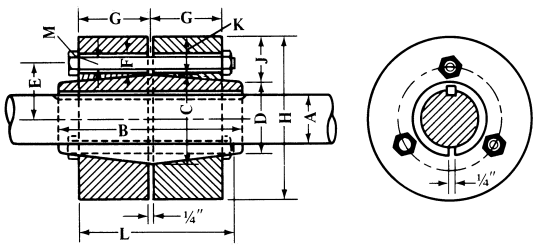
For small sizes and low power applications, a setscrew may provide the connection between the hub and the shaft, but higher power usually requires a key and perhaps two setscrews, one of them above the key. A flat on the shaft and some means of locking the setscrew(s) in position are advisable. In the AGMA Class I and II fits the shaft tolerances are –0.0005 inch from 1/2 to 1 1/2 inches diameter and -0.001 inch on larger diameters up to 7 inches.

Class I coupling bore tolerances are + 0.001 inch up to 1 1/2 inches diameter, then + 0.0015 inch to 7 inches diameter. Class II coupling bore tolerances are + 0.002 inch on sizes up to 3 inches diameter, + 0.003 inch on sizes from 3 1/4 through 3 3/4 inches diameter, and + 0.004 inch on larger diameters up to 7 inches.

Interference Fits.—Components of couplings transmitting over 150 horsepower often are made an interference fit on the shafts, which may reduce fretting corrosion. These couplings may or may not use keys, depending on the degree of interference. Keys may range in size from 1/8 inch wide by 1/16 inch high for 1/2-inch diameter shafts to 1 3/4 inches wide by 7/8 inch high for 7-inch diameter shafts. Couplings transmitting high torque or operating at high speeds or both may use two keys. Keys must be a good fit in their keyways to ensure good transmission of torque and prevent failure. AGMA standards provide recommendations for square parallel, rectangular section, and plain tapered keys, for shafts of 5/16 through 7 inches diameter, in three classes designated commercial, precision, and fitted. These standards also cover keyway offset, lead, parallelism, finish and radii, and face keys and splines. (See also ANSI and other Standards in Keys and Keyways section of this Handbook.)

Double-cone Clamping Couplings.—As shown in the table, double-cone clamping couplings are made in a range of sizes for shafts from 1 7/16 to 6 inches in diameter, and are easily assembled to shafts. These couplings provide an interference fit, but they usually cost more and have larger overall dimensions than regular flanged couplings.

Double-cone Clamping Couplings



A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	No. of Bolts	No. of Keys
1 7/16	5 1/4	2 3/4	2 1/8	1 5/8	5/8	2 1/8	4 3/4	1 1/8	1	5	1/2	3	1
1 15/16	7	3 1/2	2 7/8	2 1/8	5/8	2 3/4	6 1/4	1 1/8	1 3/8	6 1/4	1/2	3	1
2 7/16	8 3/4	4 5/16	3 5/8	3	3/4	3 1/2	7 13/16	1 7/8	1 3/4	7 7/8	5/8	3	1
3	10 1/2	5 1/2	4 3/32	3 1/2	3/4	4 3/16	9	2 1/4	2	9 1/2	5/8	3	1
3 1/2	12 1/4	7	5 3/8	4 3/8	7/8	5 1/16	11 1/4	2 5/8	2 1/8	11 1/4	3/4	4	1
4	14	7	5 1/2	4 3/4	7/8	5 1/2	12	3 3/4	2 1/2	12	3/4	4	1
4 1/2	15 1/2	8	6 7/8	5 1/4	7/8	6 3/4	13 1/2	3 3/4	2 3/4	14 1/2	3/4	4	1
5	17	9	7 1/4	5 3/4	7/8	7	15	3 3/4	3	15 1/4	3/4	4	1
5 1/2	17 1/2	9 1/2	7 3/4	6 1/4	1	7	15 1/2	3 3/4	3	15 1/4	7/8	4	1
6	18	10	8 1/4	6 3/4	1	7	16	3 3/4	3	15 1/4	7/8	4	2

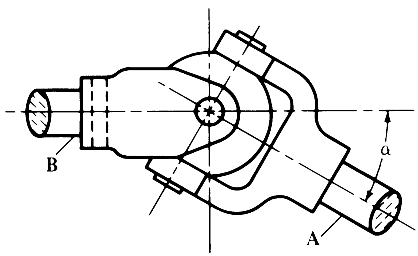
Flexible Couplings.—Shafts that are out of alignment laterally or angularly can be connected by any of several designs of flexible couplings. Such couplings also permit some degree of axial movement in one or both shafts. Some couplings use disks or diaphragms to transmit the torque. Another simple form of flexible coupling consists of two flanges connected by links or endless belts made of leather or other strong, pliable material. Alternatively, the flanges may have projections that engage spacers of molded rubber or other flexible materials that accommodate uneven motion between the shafts. More highly developed flexible couplings use toothed flanges engaged by correspondingly toothed elements, permitting relative movement. These couplings require lubrication unless one or more of the elements is made of a self-lubricating material. Other couplings use diaphragms or bellows that can flex to accommodate relative movement between the shafts.

The Universal Joint.—This form of coupling, originally known as a Cardan or Hooke's coupling, is used for connecting two shafts the axes of which are not in line with each other, but which merely intersect at a point. There are many different designs of universal joints or couplings, which are based on the principle embodied in the original design. One well-known type is shown by the accompanying diagram.

As a rule, a universal joint does not work well if the angle α (see illustration) is more than 45 degrees, and the angle should preferably be limited to about 20 degrees or 25 degrees, excepting when the speed of rotation is slow and little power is transmitted.

Variation in Angular Velocity of Driven Shaft: Owing to the angularity between two shafts connected by a universal joint, there is a variation in the angular velocity of one shaft during a single revolution, and because of this, the use of universal couplings is sometimes prohibited. Thus, the angular velocity of the driven shaft will not be the same at all points of the revolution as the angular velocity of the driving shaft. In other words, if the driving shaft moves with a uniform motion, then the driven shaft will have a variable motion and, therefore, the universal joint should not be used when absolute uniformity of motion is essential for the driven shaft.

Determining Maximum and Minimum Velocities: If shaft *A* (see diagram) runs at a constant speed, shaft *B* revolves at maximum speed when shaft A occupies the position shown in the illustration, and the minimum speed of shaft *B* occurs when the fork of the driving shaft *A* has turned 90 degrees from the position illustrated. The maximum speed of the driven shaft may be obtained by multiplying the speed of the driving shaft by the secant of angle α. The minimum speed of the driven shaft equals the speed of the driver multiplied by cosine α. Thus, if the driver rotates at a constant speed of 100 revolutions per minute and the shaft angle is 25 degrees, the maximum speed of the driven shaft is at a rate equal to $1.1034 \times 100 = 110.34$ R.P.M. The minimum speed rate equals $0.9063 \times 100 = 90.63$; hence, the extreme variation equals $110.34 - 90.63 = 19.71$ R.P.M.



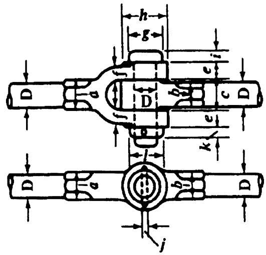
Use of Intermediate Shaft between Two Universal Joints.—The lack of uniformity in the speed of the driven shaft resulting from the use of a universal coupling, as previously explained, is objectionable for some forms of mechanisms. This variation may be avoided if the two shafts are connected with an intermediate shaft and two universal joints, provided the latter are properly arranged or located. Two conditions are necessary to obtain a constant speed ratio between the driving and driven shafts. First, the shafts must make the same angle with the intermediate shaft; second, the universal joint forks (assuming that the fork design is employed) on the intermediate shaft must be placed relatively so that when the plane of the fork at the left end coincides with the center lines of the intermediate shaft and the shaft attached to the left-hand coupling, the plane of the right-hand fork must also coincide with the center lines of the intermediate shaft and the shaft attached to the right-hand coupling; therefore the driving and the driven shafts may be placed in a variety of positions. One of the most common arrangements is with the driving and driven shafts parallel. The forks on the intermediate shafts should then be placed in the same plane.

This intermediate connecting shaft is frequently made telescoping, and then the driving and driven shafts can be moved independently of each other within certain limits in longitudinal and lateral directions. The telescoping intermediate shaft consists of a rod which enters a sleeve and is provided with a suitable spline, to prevent rotation between the rod and sleeve and permit a sliding movement. This arrangement is applied to various machine tools.

Knuckle Joints.—Movement at the joint between two rods may be provided by knuckle joints, for which typical proportions are seen in the table *Proportions of Knuckle Joints* that follows.

Friction Clutches.—Clutches which transmit motion from the driving to the driven member by the friction between the engaging surfaces are built in many different designs, although practically all of them can be classified under four general types, namely, conical clutches; radially expanding clutches; contracting-band clutches; and friction disk clutches in single and multiple types. There are many modifications of these general classes, some of which combine the features of different types. The proportions of various sizes of cone clutches are given in the table "Cast-iron Friction Clutches." The multicone friction clutch is a further development of the cone clutch. Instead of having a single cone-shaped surface, there is a series of concentric conical rings which engage annular grooves formed by corresponding rings on the opposite clutch member. The internal-expanding type is provided with shoes which are forced outward against an enclosing drum by the action of levers connecting with a collar free to slide along the shaft. The engaging shoes are commonly lined with wood or other material to increase the coefficient of friction. Disk clutches are based on the principle of multiple-plane friction, and use alternating plates or disks so arranged that one set engages with an outside cylindrical case and the other set with the shaft. When these plates are pressed together by spring pressure, or by other means, motion is transmitted from the driving to the driven members connected to the clutch. Some disk clutches have a few rather heavy or thick plates and others a relatively large number of thinner plates. Clutches of the latter type are common in automobile transmissions. One set of disks may be of soft steel and the other set of phosphor-bronze, or some other combination may be employed. For instance, disks are sometimes provided with cork inserts, or one set or series of disks may be faced with a special friction material such as asbestos-wire fabric, as in "dry plate" clutches, the disks of which are not lubricated like the disks of a clutch having, for example, the steel and phosphor-bronze combination. It is common practice to hold the driving and driven members of friction clutches in engagement by means of spring pressure, although pneumatic or hydraulic pressure may be employed.

Proportions of Knuckle Joints



For sizes not given below:

$a = 1.2\,D$ $\quad h = 2\,D$
$b = 1.1\,D$ $\quad i = 0.5\,D$
$c = 1.2\,D$ $\quad j = 0.25\,D$
$e = 0.75\,D$ $\quad k = 0.5\,D$
$f = 0.6\,D$ $\quad l = 1.5\,D$
$g = 1.5\,D$

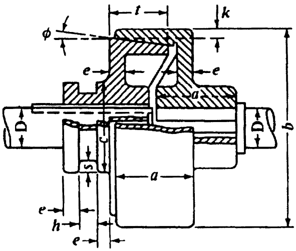
D	*a*	*b*	*c*	*e*	*f*	*g*	*h*	*i*	*j*	*k*	*l*
1/2	5/8	9/16	5/8	3/8	5/16	3/4	1	1/4	1/8	1/4	3/4
3/4	7/8	3/4	7/8	9/16	7/16	1 1/8	1 1/2	3/8	3/16	3/8	1 1/8
1	1 1/4	1 1/8	1 1/4	3/4	5/8	1 1/2	2	1/2	1/4	1/2	1 1/2
1 1/4	1 1/2	1 3/8	1 1/2	15/16	3/4	1 7/8	2 1/2	5/8	5/16	5/8	1 7/8
1 1/2	1 3/4	1 5/8	1 3/4	1 1/8	7/8	2 1/4	3	3/4	3/8	3/4	2 1/4
1 3/4	2 1/8	2	2 1/8	1 5/16	1 1/16	2 5/8	3 1/2	7/8	7/16	7/8	2 5/8
2	2 3/8	2 1/4	2 3/8	1 1/2	1 3/16	3	4	1	1/2	1	3
2 1/4	2 3/4	2 1/2	2 3/4	1 11/16	1 3/8	3 3/8	4 1/2	1 1/8	9/16	1 1/8	3 3/8
2 1/2	3	2 3/4	3	1 7/8	1 1/2	3 3/4	5	1 1/4	5/8	1 1/4	3 3/4
2 3/4	3 1/4	3	3 1/4	2 1/16	1 5/8	4 1/8	5 1/2	1 3/8	11/16	1 3/8	4 1/8
3	3 5/8	3 1/4	3 5/8	2 1/4	1 13/16	4 1/2	6	1 1/2	3/4	1 1/2	4 1/2
3 1/4	4	3 5/8	4	2 7/16	2	4 7/8	6 1/2	1 5/8	13/16	1 5/8	4 7/8
3 1/2	4 1/4	3 7/8	4 1/4	2 5/8	2 1/8	5 1/4	7	1 3/4	7/8	1 3/4	5 1/4
3 3/4	4 1/2	4 1/8	4 1/2	2 13/16	2 1/4	5 5/8	7 1/2	1 7/8	15/16	1 7/8	5 5/8
4	4 3/4	4 3/8	4 3/4	3	2 3/8	6	8	2	1	2	6
4 1/4	5 1/8	4 3/4	5 1/8	3 3/16	2 9/16	6 3/8	8 1/2	2 1/8	1 1/16	2 1/8	6 3/8
4 1/2	5 1/2	5	5 1/2	3 3/8	2 3/4	6 3/4	9	2 1/4	1 1/8	2 1/4	6 3/4
4 3/4	5 3/4	5 1/4	5 3/4	3 9/16	2 7/8	7 1/8	9 1/2	2 3/8	1 3/16	2 3/8	7 1/8
5	6	5 1/2	6	3 3/4	3	7 1/2	10	2 1/2	1 1/4	2 1/2	7 1/2

Power Transmitting Capacity of Friction Clutches.—When selecting a clutch for a given class of service, it is advisable to consider any overloads that may be encountered and base the power transmitting capacity of the clutch upon such overloads. When the load varies or is subject to frequent release or engagement, the clutch capacity should be greater than the actual amount of power transmitted. If the power is derived from a gas or gasoline engine, the horsepower rating of the clutch should be 75 or 100 per cent greater than that of the engine.

Power Transmitted by Disk Clutches.—The approximate amount of power that a disk clutch will transmit may be determined from the following formula, in which H = horsepower transmitted by the clutch; μ = coefficient of friction; r = mean radius of engaging surfaces; F = axial force in pounds (spring pressure) holding disks in contact; N = number of frictional surfaces; S = speed of shaft in revolutions per minute:

$$H = \frac{\mu r F N S}{63{,}000}$$

Cast-iron Friction Clutches



For sizes not given below:

$a = 2D$
$b = 4 \text{ to } 8D$
$c = 2\frac{1}{4}D$
$t = 1\frac{1}{2}D$
$e = \frac{3}{8}D$
$h = \frac{1}{2}D$
$s = \frac{5}{16}D$, nearly
$k = \frac{1}{4}D$

Note:— The angle ϕ of the cone may be from 4 to 10 degrees

D	*a*	*b*	*c*	*t*	*e*	*h*	*s*	*k*
1	2	4 – 8	$2\frac{1}{4}$	$1\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{1}{4}$
$1\frac{1}{4}$	$2\frac{1}{2}$	5–10	$2\frac{7}{8}$	$1\frac{7}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{5}{16}$
$1\frac{1}{2}$	3	6–12	$3\frac{3}{8}$	$2\frac{1}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
$1\frac{3}{4}$	$3\frac{1}{2}$	7–14	4	$2\frac{5}{8}$	$\frac{5}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{7}{16}$
2	4	8–16	$4\frac{1}{2}$	3	$\frac{3}{4}$	1	$\frac{5}{8}$	$\frac{1}{2}$
$2\frac{1}{4}$	$4\frac{1}{2}$	9–18	5	$3\frac{3}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{9}{16}$
$2\frac{1}{2}$	5	10–20	$5\frac{5}{8}$	$3\frac{3}{4}$	1	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{5}{8}$
$2\frac{3}{4}$	$5\frac{1}{2}$	11–22	$6\frac{1}{4}$	$4\frac{1}{8}$	1	$1\frac{3}{8}$	$\frac{7}{8}$	$\frac{11}{16}$
3	6	12–24	$6\frac{3}{4}$	$4\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{4}$
$3\frac{1}{4}$	$6\frac{1}{2}$	13–26	$7\frac{3}{8}$	$4\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{5}{8}$	1	$\frac{13}{16}$
$3\frac{1}{2}$	7	14–28	$7\frac{7}{8}$	$5\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{3}{4}$	1	$\frac{7}{8}$
$3\frac{3}{4}$	$7\frac{1}{2}$	15–30	$8\frac{1}{2}$	$5\frac{5}{8}$	$1\frac{3}{8}$	$1\frac{7}{8}$	$1\frac{1}{4}$	$\frac{15}{16}$
4	8	16–32	9	6	$1\frac{1}{2}$	2	$1\frac{1}{4}$	1
$4\frac{1}{4}$	$8\frac{1}{2}$	17–34	$9\frac{1}{2}$	$6\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{16}$
$4\frac{1}{2}$	9	18–36	$10\frac{1}{4}$	$6\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{8}$
$4\frac{3}{4}$	$9\frac{1}{2}$	19–38	$10\frac{3}{4}$	$7\frac{1}{8}$	$1\frac{3}{4}$	$2\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{16}$
5	10	20–40	$11\frac{1}{4}$	$7\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$
$5\frac{1}{4}$	$10\frac{1}{2}$	21–42	$11\frac{3}{4}$	$7\frac{7}{8}$	2	$2\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{5}{16}$
$5\frac{1}{2}$	11	22–44	$12\frac{3}{8}$	$8\frac{1}{4}$	2	$2\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{3}{8}$
$5\frac{3}{4}$	$11\frac{1}{2}$	23–46	13	$8\frac{5}{8}$	$2\frac{1}{4}$	$2\frac{7}{8}$	$1\frac{3}{4}$	$1\frac{7}{16}$
6	12	24–48	$13\frac{1}{2}$	9	$2\frac{1}{4}$	3	$1\frac{7}{8}$	$1\frac{1}{2}$

Frictional Coefficients for Clutch Calculations.—While the frictional coefficients used by designers of clutches differ somewhat and depend upon variable factors, the following values may be used in clutch calculations: For greasy leather on cast iron about 0.20 or 0.25, leather on metal that is quite oily 0.15; metal and cork on oily metal 0.32; the same on dry metal 0.35; metal on dry metal 0.15; disk clutches having lubricated surfaces 0.10.

Formulas for Cone Clutches.—In cone clutch design, different formulas have been developed for determining the horsepower transmitted. These formulas, at first sight, do not seem to agree, there being a variation due to the fact that in some of the formulas the friction clutch surfaces are assumed to engage without slip, whereas, in others, some allowance is made for slip. The following formulas include both of these conditions:

H.P. = horsepower transmitted

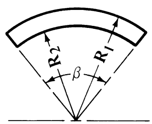
N = revolutions per minute
r = mean radius of friction cone, in inches
r_1 = large radius of friction cone, in inches
r_2 = small radius of friction cone, in inches
R_1 = outside radius of leather band, in inches
R_2 = inside radius of leather band, in inches
V = velocity of a point at distance r from the center, in feet per minute
F = tangential force acting at radius r, in pounds
P_n = total normal force between cone surfaces, in pounds
P_s = spring force, in pounds
α = angle of clutch surface with axis of shaft = 7 to 13 degrees
β = included angle of clutch leather, when developed, in degrees
f = coefficient of friction = 0.20 to 0.25 for greasy leather on iron
p = allowable pressure per square inch of leather band = 7 to 8 pounds
W = width of clutch leather, in inches

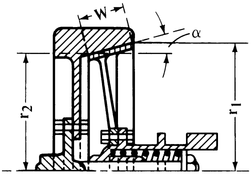
$$R_1 = \frac{r_1}{\sin\alpha} \qquad R_2 = \frac{r_2}{\sin\alpha}$$

$$\beta = \sin\alpha \times 360 \qquad r = \frac{r_1 + r_2}{2}$$

$$V = \frac{2\pi rN}{12}$$

DEVELOPMENT OF CLUTCH LEATHER





$$F = \frac{\mathrm{HP} \times 33{,}000}{V} \qquad W = \frac{P_n}{2\pi rp} \qquad \mathrm{HP} = \frac{P_n frN}{63{,}025}$$

For engagement with some slip:

$$P_n = \frac{P_s}{\sin\alpha} \qquad P_s = \frac{\mathrm{HP} \times 63{,}025 \sin\alpha}{frN}$$

For engagement without slip:

$$P_n = \frac{P_s}{\sin\alpha + f\cos\alpha} \qquad P_s = \frac{\mathrm{HP} \times 63{,}025(\sin\alpha + f\cos\alpha)}{frN}$$

Angle of Cone.—If the angle of the conical surface of the cone type of clutch is too small, it may be difficult to release the clutch on account of the wedging effect, whereas, if the angle is too large, excessive spring force will be required to prevent slipping. The minimum angle for a leather-faced cone is about 8 or 9 degrees and the maximum angle about 13 degrees. An angle of 12 ½ degrees appears to be the most common and is generally considered good practice. These angles are given with relation to the clutch axis and are one-half the included angle.

Magnetic Clutches.—Many disk and other clutches are operated electromagnetically with the magnetic force used only to move the friction disk(s) and the clutch disk(s) into or out of engagement against spring or other pressure. On the other hand, in a magnetic particle clutch, transmission of power is accomplished by magnetizing a quantity of metal particles enclosed between the driving and the driven components. forming a bond between them. Such clutches can be controlled to provide either a rigid coupling or uniform slip, useful in wire drawing and manufacture of cables.

Another type of magnetic clutch uses eddy currents induced in the input member which interact with the field in the output rotor. Torque transmitted is proportional to the coil current, so precise control of torque is provided. A third type of magnetic clutch relies on the hysteresis loss between magnetic fields generated by a coil in an input drum and a close-fitting cup on the output shaft, to transmit torque. Torque transmitted with this type of clutch also is proportional to coil current, so close control is possible.

Permanent-magnet types of clutches also are available, in which the engagement force is exerted by permanent magnets when the electrical supply to the disengagement coils is cut off. These types of clutches have capacities up to five times the torque-to-weight ratio of spring-operated clutches. In addition, if the controls are so arranged as to permit the coil polarity to be reversed instead of being cut off, the combined permanent magnet and electromagnetic forces can transmit even greater torque.

Centrifugal and Free-wheeling Clutches.—Centrifugal clutches have driving members that expand outward to engage a surrounding drum when speed is sufficient to generate centrifugal force. Free-wheeling clutches are made in many different designs and use balls, cams or sprags, ratchets, and fluids to transmit motion from one member to the other. These types of clutches are designed to transmit torque in only one direction and to take up the drive with various degrees of gradualness up to instantaneously.

Slipping Clutch/Couplings.—Where high shock loads are likely to be experienced, a slipping clutch or coupling or both should be used. The most common design uses a clutch plate that is clamped between the driving and driven plates by spring pressure that can be adjusted. When excessive load causes the driven member to slow, the clutch plate surfaces slip, allowing reduction of the torque transmitted. When the overload is removed, the drive is taken up automatically. Switches can be provided to cut off current supply to the driving motor when the driven shaft slows to a preset limit or to signal a warning or both. The slip or overload torque is calculated by taking 150 per cent of the normal running torque.

Wrapped-spring Clutches.—For certain applications, a simple steel spring sized so that its internal diameter is a snug fit on both driving and driven shafts will transmit adequate torque in one direction. The tightness of grip of the spring on the shafts increases as the torque transmitted increases. Disengagement can be effected by slight rotation of the spring, through a projecting tang, using electrical or mechanical means, to wind up the spring to a larger internal diameter, allowing one of the shafts to run free within the spring.

Normal running torque T_r in lb-ft = (required horsepower × 5250) ÷ rpm. For heavy shock load applications, multiply by a 200 per cent or greater overload factor. (See Motors, factors governing selection.)

The clutch starting torque T_c, in lb-ft, required to accelerate a given inertia in a specific time is calculated from the formula:

$$T_c = \frac{WR^2 \times \Delta N}{308t}$$

where WR^2 = total inertia encountered by clutch in lb-ft^2 (W = weight and R = radius of gyration of rotating part)

ΔN = final rpm — initial rpm; 308 = constant (see Motors, factors governing selection)

t = time to required speed in seconds

Example: If the inertia is 80 lb-ft^2, and the speed of the driven shaft is to be increased from 0 to 1500 rpm in 3 seconds, find the clutch starting torque in lb-ft.

$$T_c = \frac{80 \times 1500}{308 \times 3} = 130 \text{ lb-ft}$$

The heat E, in BTU, generated in one engagement of a clutch can be calculated from the formula:

$$E = \frac{T_c \times WR^2 \times (N_1^2 - N_2^2)}{(T_c - T_1) \times 4.7 \times 10^6}$$

where: WR^2 = total inertia encountered by clutch in lb-ft.^{2}
N_1 = final rpm
N_2 = initial rpm
Tc = clutch torque in lb-ft
T_1 = torque load in lb-ft

Example: Calculate the heat generated for each engagement under the conditions cited for the first example.

$$E = \frac{130 \times 80 \times (1500)^2}{(130 - 10) \times 4.7 \times 10^6} = 41.5 \text{ BTU}$$

The preferred location for a clutch is on the high- rather than on the low-speed shaft because a smaller-capacity unit, of lower cost and with more rapid dissipation of heat, can be used. However, the heat generated may also be more because of the greater slippage at higher speeds, and the clutch may have a shorter life. For light-duty applications, such as to a machine tool, where cutting occurs after the spindle has reached operating speed, the calculated torque should be multiplied by a safety factor of 1.5 to arrive at the capacity of the clutch to be used. Heavy-duty applications such as frequent starting of a heavily loaded vibratory-finishing barrel require a safety factor of 3 or more.

Positive Clutches.—When the driving and driven members of a clutch are connected by the engagement of interlocking teeth or projecting lugs, the clutch is said to be "positive" to distinguish it from the type in which the power is transmitted by frictional contact. The positive clutch is employed when a sudden starting action is not objectionable and when the inertia of the driven parts is relatively small. The various forms of positive clutches differ merely in the angle or shape of the engaging surfaces. The least positive form is one having planes of engagement which incline backward, with respect to the direction of motion. The tendency of such a clutch is to disengage under load, in which case it must be held in position by axial pressure.

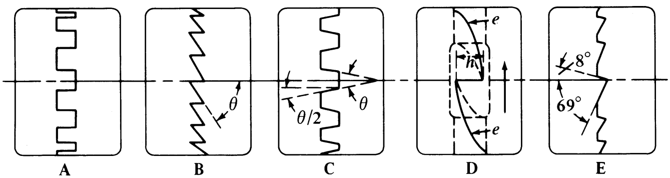


Fig. 1. Types of Clutch Teeth

This pressure may be regulated to perform normal duty, permitting the clutch to slip and disengage when over-loaded. Positive clutches, with the engaging planes parallel to the axis of rotation, are held together to obviate the tendency to jar out of engagement, but they provide no safety feature against over-load. So-called "under-cut" clutches engage more tightly the heavier the load, and are designed to be disengaged only when free from load. The teeth of positive clutches are made in a variety of forms, a few of the more common styles being shown in Fig. 1. Clutch *A* is a straight-toothed type, and *B* has angular or saw-shaped teeth. The driving member of the former can be rotated in either direction: the latter is adapted to the transmission of motion in one direction only, but is more readily engaged. The angle θ of the cutter for a saw-tooth clutch *B* is ordinarily 60 degrees. Clutch *C* is similar to *A*, except that the sides of the teeth are inclined to facilitate engagement and disengagement. Teeth of this shape are sometimes used when a clutch is required to run in either direction without backlash. Angle θ is varied to suit requirements and should not exceed 16 or 18 degrees. The straight-tooth clutch A is also modified to make the teeth engage more readily, by rounding the corners of the teeth at the top and bottom. Clutch *D* (commonly called a "spiral-jaw" clutch) differs from *B* in that the surfaces *e* are helicoidal. The driving member of this clutch can transmit motion in only one direction.

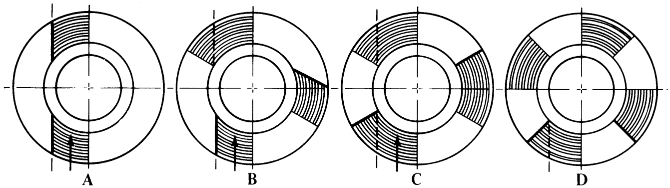


Fig. 2. Diagrammatic View Showing Method of Cutting Clutch Teeth

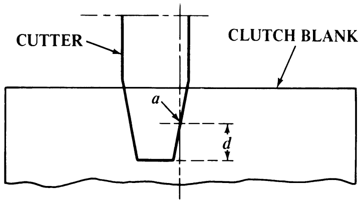


Fig. 3.

Clutches of this type are known as right- and left-hand, the former driving when turning to the right, as indicated by the arrow in the illustration. Clutch *E* is the form used on the back-shaft of the Brown & Sharpe automatic screw machines. The faces of the teeth are radial and incline at an angle of 8 degrees with the axis, so that the clutch can readily be disengaged. This type of clutch is easily operated, with little jar or noise. The 2-inch diameter size has 10 teeth. Height of working face, $\frac{1}{8}$ inch.

Cutting Clutch Teeth.—A common method of cutting a straight-tooth clutch is indicated by the diagrams *A*, *B* and *C*, Fig. 2, which show the first, second and third cuts required for forming the three teeth. The work is held in the chuck of a dividing-head, the latter being set at right angles to the table. A plain milling cutter may be used (unless the corners of the teeth are rounded), the side of the cutter being set to exactly coincide with the center-line. When the number of teeth in the clutch is odd, the cut can be taken clear across the blank as shown, thus finishing the sides of two teeth with one passage of the cutter. When the number of teeth is even, as at *D*, it is necessary to mill all the teeth on one side and then set the cutter for finishing the opposite side. Therefore, clutches of this type commonly have an odd number of teeth. The maximum width of the cutter depends upon the width of the space at the narrow ends of the teeth. If the cutter must be quite narrow in order to pass the narrow ends, some stock may be left in the tooth spaces, which must be removed by a separate cut. If the tooth is of the modified form shown at *C*, Fig. 1, the cutter should be set as indicated in Fig. 3; that is, so that a point a on the cutter at a radial distance *d* equal to one-half the depth of the clutch teeth lies in a radial plane. When it is important to eliminate all backlash, point *a* is sometimes located at a radial distance *d* equal to six-tenths of the depth of the tooth, in order to leave clearance spaces at the bottoms of the teeth; the two clutch members will then fit together tightly. Clutches of this type must be held in mesh.

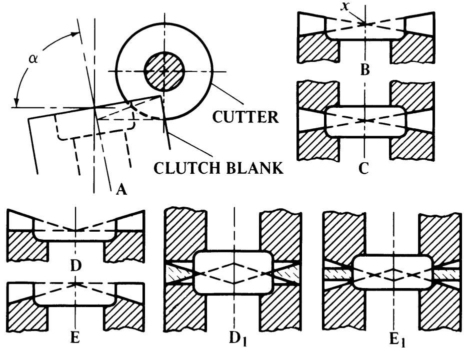
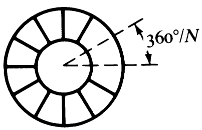


Fig. 4.

Angle of Dividing-head for Milling V-shaped Teeth with Single-angle Cutter



$$\cos\alpha = \frac{\tan(360°/N) \times \cot\theta}{2}$$

α is the angle shown in Fig. 4 and is the angle shown by the graduations on the dividing head. θ is the included angle of a single cutter, see Fig. 1.

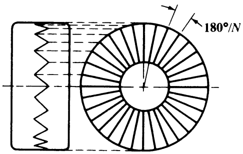
No. of Teeth, N	Angle of Single-angle Cutter, θ			No. of Teeth, N	Angle of Single-angle Cutter, θ		
	60°	70°	80°		60°	70°	80°
	Dividing Head Angle, α				Dividing Head Angle, α		
5	27° 19.2′	55° 56.3′	74° 15.4′	18	83° 58.1′	86° 12.1′	88° 9.67′
6	60	71 37.6	81 13	19	84 18.8	86 25.1	88 15.9
7	68 46.7	76 48.5	83 39.2	20	84 37.1	86 36.6	88 21.5
8	73 13.3	79 30.9	84 56.5	21	84 53.5	86 46.9	88 26.5
9	75 58.9	81 13	85 45.4	22	85 8.26	86 56.2	88 31
10	77 53.6	82 24.1	86 19.6	23	85 21.6	87 4.63	88 35.1
11	79 18.5	83 17	86 45.1	24	85 33.8	87 12.3	88 38.8
12	80 24.4	83 58.1	87 4.94	25	85 45	87 19.3	88 42.2
13	81 17.1	84 31.1	87 20.9	26	85 55.2	87 25.7	88 45.3
14	82 .536	84 58.3	87 34	27	86 4.61	87 31.7	88 48.2
15	82 36.9	85 21.2	87 45	28	86 13.3	87 37.2	88 50.8
16	83 7.95	85 40.6	87 54.4	29	86 21.4	87 42.3	88 53.3
17	83 34.7	85 57.4	88 2.56	30	86 28.9	87 47	88 55.6

Cutting Saw-tooth Clutches: When milling clutches having angular teeth as shown at *B*, Fig. 1, the axis of the clutch blank should be inclined a certain angle α as shown at *A* in Fig. 4. If the teeth were milled with the blank vertical, the tops of the teeth would incline towards the center as at *D*, whereas, if the blank were set to such an angle that the tops of the teeth were square with the axis, the bottoms would incline upwards as at *E*. In either case, the two clutch members would not mesh completely: the engagement of the teeth cut as shown at *D* and *E* would be as indicated at D_1 and E_1 respectively. As will be seen, when the outer points of the teeth at D_1 are at the bottom of the grooves in the opposite member, the inner ends are not together, the contact area being represented by the dotted lines. At E_1 the inner ends of the teeth strike first and spaces are left between the teeth around the outside of the clutch. To overcome this objectionable feature, the clutch teeth should be cut as indicated at *B*, or so that the bottoms and tops of the teeth have the same inclination, converging at a central point *x*. The teeth of both members will then engage across the entire width as shown at *C*. The angle α required for cutting a clutch as at *B* can be determined by the following formula in which α equals the required angle, *N* = number of teeth, θ = cutter angle, and 360°/*N* = angle between teeth:

$$\cos\alpha = \frac{\tan(360°/N) \times \cot\theta}{2}$$

The angles α for various numbers of teeth and for 60-, 70- or 80-degree single-angle cutters are given in the table on page 2335. The following table is for double-angle cutters used to cut V-shaped teeth.

Angle of Dividing-head for Milling V-shaped Teeth with Double-angle Cutter



$$\cos\alpha = \frac{\tan(180°/N) \times \cot(\theta/2)}{2}$$

This is the angle (α, Fig. 4) shown by graduations on the dividing-head. θ is the included angle of a double-angle cutter, see Fig. 1.

No. of Teeth, *N*	Included Angle of Cutter, θ: 60°		90°		No. of Teeth, *N*	Included Angle of Cutter, θ: 60°		90°	
	Dividing Head Angle, α					Dividing Head Angle, α			
10	73°	39.4′	80°	39′	31	84°	56.9′	87°	5.13′
11	75	16.1	81	33.5	32	85	6.42	87	10.6
12	76	34.9	82	18	33	85	15.4	87	15.8
13	77	40.5	82	55.3	34	85	23.8	87	20.7
14	78	36	83	26.8	35	85	31.8	87	25.2
15	79	23.6	83	54	36	85	39.3	87	29.6
16	80	4.83	84	17.5	37	85	46.4	87	33.7
17	80	41	84	38.2	38	85	53.1	87	37.5
18	81	13	84	56.5	39	85	59.5	87	41.2
19	81	41.5	85	12.8	40	86	5.51	87	44.7
20	82	6.97	85	27.5	41	86	11.3	87	48
21	82	30	85	40.7	42	86	16.7	87	51.2
22	82	50.8	85	52.6	43	86	22	87	54.2
23	83	9.82	86	3.56	44	86	26.9	87	57
24	83	27.2	86	13.5	45	86	31.7	87	59.8
25	83	43.1	86	22.7	46	86	36.2	88	2.4
26	83	57.8	26	31.2	47	86	40.6	88	4.91
27	84	11.4	86	39	48	86	44.8	88	7.32
28	84	24	86	46.2	49	86	48.8	88	9.63
29	84	35.7	86	53	50	86	52.6	88	11.8
30	84	46.7	86	59.3	51	86	56.3	88	14

The angles given in the table above are applicable to the milling of V-shaped grooves in brackets, etc., which must have toothed surfaces to prevent the two members from turning relative to each other, except when unclamped for angular adjustment

FRICTION BRAKES

Formulas for Band Brakes.—In any band brake, such as shown in Fig. 1, in the tabulation of formulas, where the brake wheel rotates in a clockwise direction, the tension in that part of the band marked x equals $P\frac{1}{e^{\mu\theta}-1}$

The tension in that part marked y equals $P\frac{e^{\mu\theta}}{e^{\mu\theta}-1}$.

P = tangential force in pounds at rim of brake wheel

e = base of natural logarithms = 2.71828

μ = coefficient of friction between the brake band and the brake wheel

θ = angle of contact of the brake band with the brake wheel expressed in radians $\left(\text{one radian} = \frac{180\text{ deg.}}{\pi\text{ radians}} = 57.296\frac{\text{deg.}}{\text{radian}}\right)$.

For simplicity in the formulas presented, the tensions at x and y (Fig. 1) are denoted by T_1 and T_2 respectively, for clockwise rotation. When the direction of the rotation is reversed, the tension in x equals T_2, and the tension in y equals T_1, which is the reverse of the tension in the clockwise direction.

The value of the expression $e^{\mu\theta}$ in these formulas may be most easily found by using a hand-held calculator of the scientific type; that is, one capable of raising 2.71828 to the power $\mu\theta$. The following example outlines the steps in the calculations.

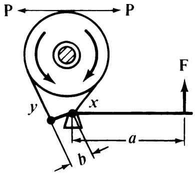
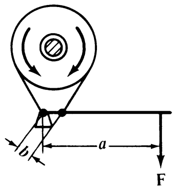
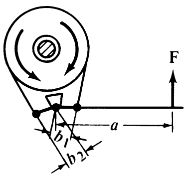
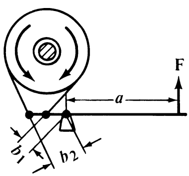
Table of Values of $e^{\mu\theta}$

Proportion of Contact to Whole Circumference, $\frac{\theta}{2\pi}$	Steel Band on Cast Iron, $\mu = 0.18$	Leather Belt on Wood: Slightly Greasy; $\mu = 0.47$	Leather Belt on Cast Iron: Very Greasy; $\mu = 0.12$	Leather Belt on Cast Iron: Slightly Greasy; $\mu = 0.28$	Leather Belt on Cast Iron: Damp; $\mu = 0.38$
0.1	1.12	1.34	1.08	1.19	1.27
0.2	1.25	1.81	1.16	1.42	1.61
0.3	1.40	2.43	1.25	1.69	2.05
0.4	1.57	3.26	1.35	2.02	2.60
0.425	1.62	3.51	1.38	2.11	2.76
0.45	1.66	3.78	1.40	2.21	2.93
0.475	1.71	4.07	1.43	2.31	3.11
0.5	1.76	4.38	1.46	2.41	3.30
0.525	1.81	4.71	1.49	2.52	3.50
0.55	1.86	5.07	1.51	2.63	3.72
0.6	1.97	5.88	1.57	2.81	4.19
0.7	2.21	7.90	1.66	3.43	5.32
0.8	2.47	10.60	1.83	4.09	6.75
0.9	2.77	14.30	1.97	4.87	8.57
1.0	3.10	19.20	2.12	5.81	10.90

Formulas for Simple and Differential Band Brakes

F = force in pounds at end of brake handle; P = tangential force in pounds at rim of brake wheel; e = base of natural logarithms = 2.71828; μ = coefficient of friction between the brake band and the brake wheel; θ = angle of contact of the brake band with the brake wheel, expressed in radians (one radian = 57.296 degrees).

$$T_1 = P\frac{1}{e^{\mu\theta}-1} \qquad T_2 = P\frac{e^{\mu\theta}}{e^{\mu\theta}-1}$$

 Fig. 1.	Simple band brake. For clockwise rotation: $F = \frac{bT_2}{a} = \frac{Pb}{a}\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right)$ For counter clockwise rotation: $F = \frac{bT_1}{a} = \frac{Pb}{a}\left(\frac{1}{e^{\mu\theta}-1}\right)$
 Fig. 2.	Simple band brake. For clockwise rotation: $F = \frac{bT_1}{a} = \frac{Pb}{a}\left(\frac{1}{e^{\mu\theta}-1}\right)$ For counter clockwise rotation: $F = \frac{bT_2}{a} = \frac{Pb}{a}\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right)$
 Fig. 3.	Differential band brake. For clockwise rotation: $F = \frac{b_2T_2 - b_1T_1}{a} = \frac{P}{a}\left(\frac{b_2e^{\mu\theta} - b_1}{e^{\mu\theta}-1}\right)$ For counter clockwise rotation: $F = \frac{b_2T_1 - b_1T_2}{a} = \frac{P}{a}\left(\frac{b_2 - b_1e^{\mu\theta}}{e^{\mu\theta}-1}\right)$ In this case, if b_2 is equal to, or less than, $b_1e^{\mu\theta}$, the force F will be 0 or negative and the band brake works automatically.
 Fig. 4.	Differential band brake. For clockwise rotation: $F = \frac{b_2T_2 + b_1T_1}{a} = \frac{P}{a}\left(\frac{b_2e^{\mu\theta} + b_1}{e^{\mu\theta}-1}\right)$ For counter clockwise rotation: $F = \frac{b_1T_2 + b_2T_1}{a} = \frac{P}{a}\left(\frac{b_1e^{\mu\theta} + b_2}{e^{\mu\theta}-1}\right)$ If $b_2 = b_1$, both of the above formulas reduce to $F = \frac{Pb_1}{a}\left(\frac{e^{\mu\theta}+1}{e^{\mu\theta}-1}\right)$. In this case, the same force F is required for rotation in either direction.

In a band brake of the type in Fig. 1, dimension a = 24 inches, and b = 4 inches; force P = 100 pounds; coefficient μ = 0.2, and angle of contact = 240 degrees, or

$$\theta = \frac{240}{180} \times \pi = 4.18$$

The rotation is clockwise. Find force F required.

$$F = \frac{Pb}{a}\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right)$$
$$= \frac{100\times 4}{24}\left(\frac{2.71828^{0.2\times 4.18}}{2.71828^{0.2\times 4.18}-1}\right)$$
$$= \frac{400}{24}\times\frac{2.71828^{0.836}}{2.71828^{0.836}-1}$$
$$= 16.66\times\frac{2.31}{2.31-1} = 29.4$$

If a hand-held calculator is not used, determining the value of $e^{\mu\theta}$ is rather tedious, and the table on page 2337 will save calculations.

Coefficient of Friction in Brakes.—The coefficients of friction that may be assumed for friction brake calculations are as follows: Iron on iron, 0.25 to 0.3 leather on iron, 0.3; cork on iron, 0.35. Values somewhat lower than these should be assumed when the velocities exceed 400 feet per minute at the beginning of the braking operation.

For brakes where wooden brake blocks are used on iron drums, poplar has proved the best brake-block material. The best material for the brake drum is wrought iron. Poplar gives a high coefficient of friction, and is little affected by oil. The average coefficient of friction for poplar brake blocks and wrought-iron drums is 0.6; for poplar on cast iron, 0.35 for oak on wrought iron, 0.5; for oak on cast iron, 0.3; for beech on wrought iron, 0.5; for beech on cast iron, 0.3; for elm on wrought iron, 0.6; and for elm on cast iron, 0.35. The objection to elm is that the friction decreases rapidly if the friction surfaces are oily. The coefficient of friction for elm and wrought iron, if oily, is less than 0.4.

Calculating Horsepower from Dynamometer Tests.—When a dynamometer is arranged for measuring the horsepower transmitted by a shaft, as indicated by the diagrammatic view in the illustration on page 2340, the horsepower may be obtained by the formula:

$$HP = \frac{2\pi LPN}{33,000}$$

in which H.P. = horsepower transmitted; N = number of revolutions per minute; L = distance (as shown in illustration) from center of pulley to point of action of weight P, in feet; P = weight hung on brake arm or read on scale.

By adopting a length of brake arm equal to 5 feet 3 inches, the formula may be reduced to the simple form:

$$HP = \frac{NP}{1000}$$

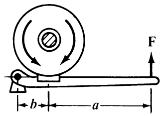
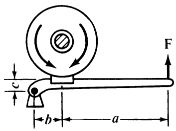
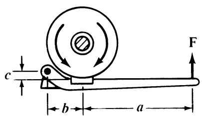
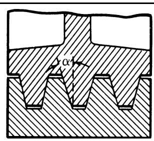
If a length of brake arm equal to 2 feet 7½ inches is adopted as a standard, the formula takes the form:

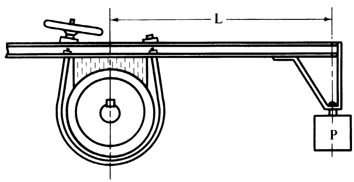
$$HP = \frac{NP}{2000}$$

The *transmission* type of dynamometer measures the power by transmitting it through the mechanism of the dynamometer from the apparatus in which it is generated, or to the apparatus in which it is to be utilized. Dynamometers known as *indicators* operate by simultaneously measuring the pressure and volume of a confined fluid. This type may be used for the measurement of the power generated by steam or gas engines or absorbed by refrigerating machinery, air compressors, or pumps. An electrical dynamometer is for measuring the power of an electric current, based on the mutual action of currents flowing in two coils. It consists principally of one fixed and one movable coil, which, in the normal position, are at right angles to each other. Both coils are connected in series, and, when a current traverses the coils, the fields produced are at right angles; hence, the coils tend to take up a parallel position. The movable coil with an attached pointer will be deflected, the deflection measuring directly the electric current.

Formulas for Block Brakes

F = force in pounds at end of brake handle;
P = tangential force in pounds at rim of brake wheel;
μ = coefficient of friction between the brake block and brake wheel.

 Fig. 1.	Block brake. For rotation in either direction: $F = P\frac{b}{a+b} \times \frac{1}{\mu} = \frac{Pb}{a+b}\left(\frac{1}{\mu}\right)$
 Fig. 2.	Block brake. For clockwise rotation: $F = \frac{\frac{Pb}{\mu} - Pc}{a+b} = \frac{Pb}{a+b}\left(\frac{1}{\mu} - \frac{c}{b}\right)$ For counter clockwise rotation: $F = \frac{\frac{Pb}{\mu} + Pc}{a+b} = \frac{Pb}{a+b}\left(\frac{1}{\mu} + \frac{c}{b}\right)$
 Fig. 3.	Block brake. For clockwise rotation: $F = \frac{\frac{Pb}{\mu} + Pc}{a+b} = \frac{Pb}{a+b}\left(\frac{1}{\mu} + \frac{c}{b}\right)$ For counter clockwise rotation: $F = \frac{\frac{Pb}{\mu} - Pc}{a+b} = \frac{Pb}{a+b}\left(\frac{1}{\mu} - \frac{c}{b}\right)$
 Fig. 4.	The brake wheel and friction block of the block brake are often grooved as shown in Fig. 4. In this case, substitute for μ in the above equations the value $\frac{\mu}{\sin\alpha + \mu\cos\alpha}$ where α is one-half the angle included by the facts of the grooves.



Friction Wheels for Power Transmission

When a rotating member is driven intermittently and the rate of driving does not need to be positive, friction wheels are frequently used, especially when the amount of power to be transmitted is comparatively small. The driven wheels in a pair of friction disks should always be made of a harder material than the driving wheels, so that if the driven wheel should be held stationary by the load, while the driving wheel revolves under its own pressure, a flat spot may not be rapidly worn on the driven wheel. The driven wheels, therefore, are usually made of iron, while the driving wheels are made of or covered with, rubber, paper, leather, wood or fiber. The safe working force per inch of face width of contact for various materials are as follows: Straw fiber, 150; leather fiber, 240; tarred fiber, 240; leather, 150; wood, 100 to 150; paper, 150. Coefficients of friction for different combinations of materials are given in the following table. Smaller values should be used for exceptionally high speeds, or when the transmission must be started while under load.

Horsepower of Friction Wheels.—Let D = diameter of friction wheel in inches; N = Number of revolutions per minute; W = width of face in inches; f = coefficient of friction; P = force in pounds, per inch width of face. Then:

$$\text{H.P.} = \frac{3.1416 \times D \times N \times P \times W \times f}{33{,}000 \times 12}$$

Assume

$$\frac{3.1416 \times P \times f}{33{,}000 \times 12} = C$$

then,

for $P = 100$ and $f = 0.20$, $C = 0.00016$

for $P = 150$ and $f = 0.20$, $C = 0.00024$

for $P = 200$ and $f = 0.20$, $C = 0.00032$

Working Values of Coefficient of Friction

Materials	Coefficient of Friction	Materials	Coefficient of Friction
Straw fiber and cast iron	0.26	Tarred fiber and aluminum	0.18
Straw fiber and aluminum	0.27	Leather and cast iron	0.14
Leather fiber and cast iron	0.31	Leather and aluminum	0.22
Leather fiber and aluminum	0.30	Leather and typemetal	0.25
Tarred fiber and cast iron	0.15	Wood and metal	0.25
Paper and cast iron	0.20		

The horsepower transmitted is then:

$$\text{HP} = D \times N \times W \times C$$

Example: Find the horsepower transmitted by a pair of friction wheels; the diameter of the driving wheel is 10 inches, and it revolves at 200 revolutions per minute. The width of the wheel is 2 inches. The force per inch width of face is 150 pounds, and the coefficient of friction 0.20.

$$\text{HP} = 10 \times 200 \times 2 \times 0.00024 = 0.96 \text{ horsepower}$$

Horsepower Which May be Transmitted by Means of a Clean Paper Friction Wheel of One-inch Face when Run Under a Force of 150 Pounds (Rockwood Mfg. Co.)

Dia. of Friction Wheel	Revolutions per Minute										
	25	50	75	100	150	200	300	400	600	800	1000
4	0.023	0.047	0.071	0.095	0.142	0.190	0.285	0.380	0.571	0.76	0.95
6	0.035	0.071	0.107	0.142	0.214	0.285	0.428	0.571	0.856	1.14	1.42
8	0.047	0.095	0.142	0.190	0.285	0.380	0.571	0.761	1.142	1.52	1.90
10	0.059	0.119	0.178	0.238	0.357	0.476	0.714	0.952	1.428	1.90	2.38
14	0.083	0.166	0.249	0.333	0.499	0.666	0.999	1.332	1.999	2.66	3.33
16	0.095	0.190	0.285	0.380	0.571	0.761	1.142	1.523	2.284	3.04	3.80
18	0.107	0.214	0.321	0.428	0.642	0.856	1.285	1.713	2.570	3.42	4.28
24	0.142	0.285	0.428	0.571	0.856	1.142	1.713	2.284	3.427	4.56	5.71
30	0.178	0.357	0.535	0.714	1.071	1.428	2.142	2.856	4.284	5.71	7.14
36	0.214	0.428	0.642	0.856	1.285	1.713	2.570	3.427	5.140	6.85	8.56
42	0.249	0.499	0.749	0.999	1.499	1.999	2.998	3.998	5.997	7.99	9.99
48	0.285	0.571	0.856	1.142	1.713	2.284	3.427	4.569	6.854	9.13	11.42
50	0.297	0.595	0.892	1.190	1.785	2.380	3.570	4.760	7.140	9.52	11.90

KEYS AND KEYSEATS

ANSI Standard Keys and Keyseats.—American National Standard, B17.1 Keys and Keyseats, based on current industry practice, was approved in 1967, and reaffirmed in 1989. This standard establishes a uniform relationship between shaft sizes and key sizes for parallel and taper keys as shown in Table 1. Other data in this standard are given in Tables 2 and 3 through 7. The sizes and tolerances shown are for single key applications only.

The following definitions are given in the standard:

Key: A demountable machinery part which, when assembled into keyseats, provides a positive means for transmitting torque between the shaft and hub.

Keyseat: An axially located rectangular groove in a shaft or hub.

This standard recognizes that there are two classes of stock for parallel keys used by industry. One is a close, plus toleranced key stock and the other is a broad, negative toleranced bar stock. Based on the use of two types of stock, two classes of fit are shown:

Class 1: A clearance or metal-to-metal side fit obtained by using bar stock keys and keyseat tolerances as given in Table 4. This is a relatively free fit and applies only to parallel keys.

Class 2: A side fit, with possible interference or clearance, obtained by using key stock and keyseat tolerances as given in Table 4. This is a relatively tight fit.

Class 3: This is an interference side fit and is not tabulated in Table 4 since the degree of interference has not been standardized. However, it is suggested that the top and bottom fit range given under Class 2 in Table 4, for parallel keys be used.

Table 1. Key Size Versus Shaft Diameter *ANSI B17.1-1967 (R1998)*

Nominal Shaft Diameter		Nominal Key Size			Normal Keyseat Depth	
			Height, *H*		*H*/2	
Over	To (Incl.)	Width, *W*	Square	Rectangular	Square	Rectangular
5⁄16	7⁄16	3⁄32	3⁄32	…	3⁄64	…
7⁄16	9⁄16	1⁄8	1⁄8	3⁄32	1⁄16	3⁄64
9⁄16	7⁄8	3⁄16	3⁄16	1⁄8	3⁄32	1⁄16
7⁄8	1 1⁄4	1⁄4	1⁄4	3⁄16	1⁄8	3⁄32
1 1⁄4	1 3⁄8	5⁄16	5⁄16	1⁄4	5⁄32	1⁄8
1 3⁄8	1 3⁄4	3⁄8	3⁄8	1⁄4	3⁄16	1⁄8
1 3⁄4	2 1⁄4	1⁄2	1⁄2	3⁄8	1⁄4	3⁄16
2 1⁄4	2 3⁄4	5⁄8	5⁄8	7⁄16	5⁄16	7⁄32
2 3⁄4	3 1⁄4	3⁄4	3⁄4	1⁄2	3⁄8	1⁄4
3 1⁄4	3 3⁄4	7⁄8	7⁄8	5⁄8	7⁄16	5⁄16
3 3⁄4	4 1⁄2	1	1	3⁄4	1⁄2	3⁄8
4 1⁄2	5 1⁄2	1 1⁄4	1 1⁄4	7⁄8	5⁄8	7⁄16
5 1⁄2	6 1⁄2	1 1⁄2	1 1⁄2	1	3⁄4	1⁄2
Square Keys preferred for shaft diameters above this line; rectangular keys, below						
6 1⁄2	7 1⁄2	1 3⁄4	1 3⁄4	1 1⁄2[a]	7⁄8	3⁄4
7 1⁄2	9	2	2	1 1⁄2	1	3⁄4
9	11	2 1⁄2	2 1⁄2	1 3⁄4	1 1⁄4	7⁄8

[a] Some key standards show 1 1⁄4 inches; preferred height is 1 1⁄2 inches.

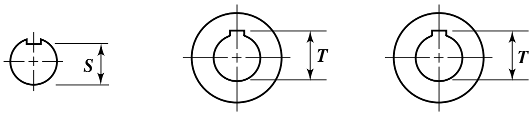
All dimensions are given in inches. For larger shaft sizes, see *ANSI Standard Woodruff Keys and Keyseats*.

Key Size vs. Shaft Diameter: Shaft diameters are listed in Table 1 for identification of various key sizes and are not intended to establish shaft dimensions, tolerances or selections. For a stepped shaft, the size of a key is determined by the diameter of the shaft at the point of location of the key. Up through 6½-inch diameter shafts square keys are preferred; rectangular keys are preferred for larger shafts.

If special considerations dictate the use of a keyseat in the hub shallower than the preferred nominal depth shown, it is recommended that the tabulated preferred nominal standard keyseat always be used in the shaft.

Keyseat Alignment Tolerances: A tolerance of 0.010 inch, max is provided for offset (due to parallel displacement of keyseat centerline from centerline of shaft or bore) of keyseats in shaft and bore. The following tolerances for maximum lead (due to angular displacement of keyseat centerline from centerline of shaft or bore and measured at right angles to the shaft or bore centerline) of keyseats in shaft and bore are specified: 0.002 inch for keyseat length up to and including 4 inches; 0.0005 inch per inch of length for keyseat lengths above 4 inches to and including 10 inches; and 0.005 inch for keyseat lengths above 10 inches. For the effect of keyways on shaft strength, see *Effect of Keyways on Shaft Strength* on page 283.

Table 2. Depth Control Values *S* and *T* for Shaft and Hub
ANSI B17.1-1967 (R1998)



Nominal Shaft Diameter	Parallel and Taper		Parallel		Taper	
	Square	Rectangular	Square	Rectangular	Square	Rectangular
	S	*S*	*T*	*T*	*T*	*T*
1/2	0.430	0.445	0.560	0.544	0.535	0.519
9/16	0.493	0.509	0.623	0.607	0.598	0.582
5/8	0.517	0.548	0.709	0.678	0.684	0.653
11/16	0.581	0.612	0.773	0.742	0.748	0.717
3/4	0.644	0.676	0.837	0.806	0.812	0.781
13/16	0.708	0.739	0.900	0.869	0.875	0.844
7/8	0.771	0.802	0.964	0.932	0.939	0.907
15/16	0.796	0.827	1.051	1.019	1.026	0.994
1	0.859	0.890	1.114	1.083	1.089	1.058
1 1/16	0.923	0.954	1.178	1.146	1.153	1.121
1 1/8	0.986	1.017	1.241	1.210	1.216	1.185
1 3/16	1.049	1.080	1.304	1.273	1.279	1.248
1 1/4	1.112	1.144	1.367	1.336	1.342	1.311
1 5/16	1.137	1.169	1.455	1.424	1.430	1.399
1 3/8	1.201	1.232	1.518	1.487	1.493	1.462
1 7/16	1.225	1.288	1.605	1.543	1.580	1.518
1 1/2	1.289	1.351	1.669	1.606	1.644	1.581
1 9/16	1.352	1.415	1.732	1.670	1.707	1.645
1 5/8	1.416	1.478	1.796	1.733	1.771	1.708
1 11/16	1.479	1.541	1.859	1.796	1.834	1.771
1 3/4	1.542	1.605	1.922	1.860	1.897	1.835
1 13/16	1.527	1.590	2.032	1.970	2.007	1.945
1 7/8	1.591	1.654	2.096	2.034	2.071	2.009
1 15/16	1.655	1.717	2.160	2.097	2.135	2.072
2	1.718	1.781	2.223	2.161	2.198	2.136
2 1/16	1.782	1.844	2.287	2.224	2.262	2.199
2 1/8	1.845	1.908	2.350	2.288	2.325	2.263
2 3/16	1.909	1.971	2.414	2.351	2.389	2.326
2 1/4	1.972	2.034	2.477	2.414	2.452	2.389
2 5/16	1.957	2.051	2.587	2.493	2.562	2.468
2 3/8	2.021	2.114	2.651	2.557	2.626	2.532

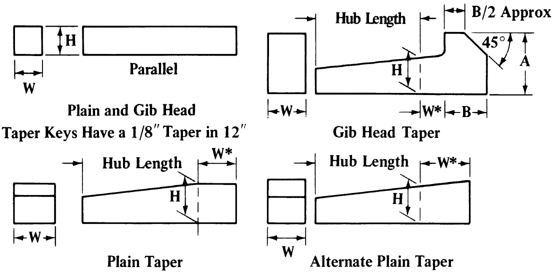
Table 2. *(Continued)* **Depth Control Values *S* and *T* for Shaft and Hub**
ANSI B17.1-1967 (R1998)

$2\frac{7}{16}$	2.084	2.178	2.714	2.621	2.689	2.596
$2\frac{1}{2}$	2.148	2.242	2.778	2.684	2.753	2.659
$2\frac{9}{16}$	2.211	2.305	2.841	2.748	2.816	2.723
$2\frac{5}{8}$	2.275	2.369	2.905	2.811	2.880	2.786
$2\frac{11}{16}$	2.338	2.432	2.968	2.874	2.943	2.849
$2\frac{3}{4}$	2.402	2.495	3.032	2.938	3.007	2.913
$2\frac{13}{16}$	2.387	2.512	3.142	3.017	3.117	2.992
$2\frac{7}{8}$	2.450	2.575	3.205	3.080	3.180	3.055
$2\frac{15}{16}$	2.514	2.639	3.269	3.144	3.244	3.119
3	2.577	2.702	3.332	3.207	3.307	3.182
$3\frac{1}{16}$	2.641	2.766	3.396	3.271	3.371	3.246
$3\frac{1}{8}$	2.704	2.829	3.459	3.334	3.434	3.309
$3\frac{3}{16}$	2.768	2.893	3.523	3.398	3.498	3.373
$3\frac{1}{4}$	2.831	2.956	3.586	3.461	3.561	3.436
$3\frac{5}{16}$	2.816	2.941	3.696	3.571	3.671	3.546
$3\frac{3}{8}$	2.880	3.005	3.760	3.635	3.735	3.610
$3\frac{7}{16}$	2.943	3.068	3.823	3.698	3.798	3.673
$3\frac{1}{2}$	3.007	3.132	3.887	3.762	3.862	3.737
$3\frac{9}{16}$	3.070	3.195	3.950	3.825	3.925	3.800
$3\frac{5}{8}$	3.134	3.259	4.014	3.889	3.989	3.864
$3\frac{11}{16}$	3.197	3.322	4.077	3.952	4.052	3.927
$3\frac{3}{4}$	3.261	3.386	4.141	4.016	4.116	3.991
$3\frac{13}{16}$	3.246	3.371	4.251	4.126	4.226	4.101
$3\frac{7}{8}$	3.309	3.434	4.314	4.189	4.289	4.164
$3\frac{15}{16}$	3.373	3.498	4.378	4.253	4.353	4.228
4	3.436	3.561	4.441	4.316	4.416	4.291
$4\frac{3}{16}$	3.627	3.752	4.632	4.507	4.607	4.482
$4\frac{1}{4}$	3.690	3.815	4.695	4.570	4.670	4.545
$4\frac{3}{8}$	3.817	3.942	4.822	4.697	4.797	4.672
$4\frac{7}{16}$	3.880	4.005	4.885	4.760	4.860	4.735
$4\frac{1}{2}$	3.944	4.069	4.949	4.824	4.924	4.799
$4\frac{3}{4}$	4.041	4.229	5.296	5.109	5.271	5.084
$4\frac{7}{8}$	4.169	4.356	5.424	5.236	5.399	5.211
$4\frac{15}{16}$	4.232	4.422	5.487	5.300	5.462	5.275
5	4.296	4.483	5.551	5.363	5.526	5.338
$5\frac{3}{16}$	4.486	4.674	5.741	5.554	5.716	5.529
$5\frac{1}{4}$	4.550	4.737	5.805	5.617	5.780	5.592
$5\frac{7}{16}$	4.740	4.927	5.995	5.807	5.970	5.782
$5\frac{1}{2}$	4.803	4.991	6.058	5.871	6.033	5.846
$5\frac{3}{4}$	4.900	5.150	6.405	6.155	6.380	6.130
$5\frac{15}{16}$	5.091	5.341	6.596	6.346	6.571	6.321
6	5.155	5.405	6.660	6.410	6.635	6.385
$6\frac{1}{4}$	5.409	5.659	6.914	6.664	6.889	6.639
$6\frac{1}{2}$	5.662	5.912	7.167	6.917	7.142	6.892
$6\frac{3}{4}$	5.760	[a]5.885	7.515	[a]7.390	7.490	[a]7.365
7	6.014	[a]6.139	7.769	[a]7.644	7.744	[a]7.619
$7\frac{1}{4}$	6.268	[a]6.393	8.023	[a]7.898	7.998	[a]7.873
$7\frac{1}{2}$	6.521	[a]6.646	8.276	[a]8.151	8.251	[a]8.126
$7\frac{3}{4}$	6.619	6.869	8.624	8.374	8.599	8.349
8	6.873	7.123	8.878	8.628	8.853	8.603
9	7.887	8.137	9.892	9.642	9.867	9.617
10	8.591	8.966	11.096	10.721	11.071	10.696
11	9.606	9.981	12.111	11.736	12.086	11.711
12	10.309	10.809	13.314	12.814	13.289	12.789
13	11.325	11.825	14.330	13.830	14.305	13.805
14	12.028	12.528	15.533	15.033	15.508	15.008
15	13.043	13.543	16.548	16.048	16.523	16.023

[a] $1\frac{3}{4} \times 1\frac{1}{2}$ inch key.

All dimensions are given in inches. See Table 4 for tolerances.

Table 3. ANSI Standard Plain and Gib Head Keys *ANSI B17.1-1967 (R1998)*



Key			Nominal Key Size Width *W* Over	To (Incl.)	Tolerance Width, *W*		Height, *H*	
Parallel	Square	Keystock	…	1 1/4	+0.001	−0.000	+0.001	−0.000
			1 1/4	3	+0.002	−0.000	+0.002	−0.000
			3	3 1/2	+0.003	−0.000	+0.003	−0.000
		Bar Stock	…	3/4	+0.000	−0.002	+0.000	−0.002
			3/4	1 1/2	+0.000	−0.003	+0.000	−0.003
			1 1/2	2 1/2	+0.000	−0.004	+0.000	−0.004
			2 1/2	3 1/2	+0.000	−0.006	+0.000	−0.006
	Rectangular	Keystock	…	1 1/4	+0.001	−0.000	+0.005	−0.005
			1 1/4	3	+0.002	−0.000	+0.005	−0.005
			3	7	+0.003	−0.000	+0.005	−0.005
		Bar Stock	…	3/4	+0.000	−0.003	+0.000	−0.003
			3/4	1 1/2	+0.000	−0.004	+0.000	−0.004
			1 1/2	3	+0.000	−0.005	+0.000	−0.005
			3	4	+0.000	−0.006	+0.000	−0.006
			4	6	+0.000	−0.008	+0.000	−0.008
			6	7	+0.000	−0.013	+0.000	−0.013
Taper	Plain or Gib Head Square or Rectangular		…	1 1/4	+0.001	−0.000	+0.005	−0.000
			1 1/4	3	+0.002	−0.000	+0.005	−0.000
			3	7	+0.003	−0.000	+0.005	−0.000

Gib Head Nominal Dimensions													
Nominal Key Size Width, *W*	Square *H*	Square *A*	Square *B*	Rectangular *H*	Rectangular *A*	Rectangular *B*	Nominal Key Size Width, *W*	Square *H*	Square *A*	Square *B*	Rectangular *H*	Rectangular *A*	Rectangular *B*
1/8	1/8	1/4	1/4	3/32	3/16	1/8	1	1	1 5/8	1 1/8	3/4	1 1/4	7/8
3/16	3/16	5/16	5/16	1/8	1/4	1/4	1 1/4	1 1/4	2	1 7/16	7/8	1 3/8	1
1/4	1/4	7/16	3/8	3/16	5/16	5/16	1 1/2	1 1/2	2 3/8	1 3/4	1	1 5/8	1 1/8
5/16	5/16	1/2	7/16	1/4	7/16	3/8	1 3/4	1 3/4	2 3/4	2	1 1/2	2 3/8	1 3/4
3/8	3/8	5/8	1/2	1/4	7/16	3/8	2	2	3 1/2	2 1/4	1 1/2	2 3/8	1 3/4
1/2	1/2	7/8	5/8	3/8	5/8	1/2	2 1/2	2 1/2	4	3	1 3/4	2 3/4	2
5/8	5/8	1	3/4	7/16	3/4	9/16	3	3	5	3 1/2	2	3 1/2	2 1/4
3/4	3/4	1 1/4	7/8	1/2	7/8	5/8	3 1/2	3 1/2	6	4	2 1/2	4	3
7/8	7/8	1 3/8	1	5/8	1	3/4	…	…	…	…	…	…	…

All dimensions are given in inches.

*For locating position of dimension *H*. Tolerance does not apply.

For larger sizes the following relationships are suggested as guides for establishing *A* and *B*: A = 1.8*H* and *B* = 1.2*H*.

Table 4. ANSI Standard Fits for Parallel and Taper Keys *ANSI B17.1-1967 (R1998)*

Type of Key	Key Width		Side Fit			Top and Bottom Fit			
			Width Tolerance			Depth Tolerance			
	Over	To (Incl.)	Key	Key-Seat	Fit Range[a]	Key	Shaft Key-Seat	Hub Key-Seat	Fit Range[a]
Class 1 Fit for Parallel Keys									
Square	…	½	+0.000	+0.002	0.004 CL	+0.000	+0.000	+0.010	0.032 CL
			−0.002	−0.000	0.000	−0.002	−0.015	−0.000	0.005 CL
	½	¾	+0.000	+0.003	0.005 CL	+0.000	+0.000	+0.010	0.032 CL
			−0.002	−0.000	0.000	−0.002	−0.015	−0.000	0.005 CL
	¾	1	+0.000	+0.003	0.006 CL	+0.000	+0.000	+0.010	0.033 CL
			−0.003	−0.000	0.000	−0.003	−0.015	−0.000	0.005 CL
	1	1½	+0.000	+0.004	0.007 CL	+0.000	+0.000	+0.010	0.033 CL
			−0.003	−0.000	0.000	−0.003	−0.015	−0.000	0.005 CL
	1½	2½	+0.000	+0.004	0.008 CL	+0.000	+0.000	+0.010	0.034 CL
			−0.004	−0.000	0.000	−0.004	−0.015	−0.000	0.005 CL
	2½	3½	+0.000	+0.004	0.010 CL	+0.000	+0.000	+0.010	0.036 CL
			−0.006	−0.000	0.000	−0.006	−0.015	−0.000	0.005 CL
Rectangular	…	½	+0.000	+0.002	0.005 CL	+0.000	+0.000	+0.010	0.033 CL
			−0.003	−0.000	0.000	−0.003	−0.015	−0.000	0.005 CL
	½	¾	+0.000	+0.003	0.006 CL	+0.000	+0.000	+0.010	0.033 CL
			−0.003	−0.000	0.000	−0.003	−0.015	−0.000	0.005 CL
	¾	1	+0.000	+0.003	0.007 CL	+0.000	+0.000	+0.010	0.034 CL
			−0.004	−0.000	0.000	−0.004	−0.015	−0.000	0.005 CL
	1	1½	+0.000	+0.004	0.008 CL	+0.000	+0.000	+0.010	0.034 CL
			−0.004	−0.000	0.000	−0.004	−0.015	−0.000	0.005 CL
	1½	3	+0.000	+0.004	0.009 CL	+0.000	+0.000	+0.010	0.035 CL
			−0.005	−0.000	0.000	−0.005	−0.015	−0.000	0.005 CL
	3	4	+0.000	+0.004	0.010 CL	+0.000	+0.000	+0.010	0.036 CL
			−0.006	−0.000	0.000	−0.006	−0.015	−0.000	0.005 CL
	4	6	+0.000	+0.004	0.012 CL	+0.000	+0.000	+0.010	0.038 CL
			−0.008	−0.000	0.000	−0.008	−0.015	−0.000	0.005 CL
	6	7	+0.000	+0.004	0.017 CL	+0.000	+0.000	+0.010	0.043 CL
			−0.013	−0.000	0.000	−0.013	−0.015	−0.000	0.005 CL
Class 2 Fit for Parallel and Taper Keys									
Parallel Square	…	1¼	+0.001	+0.002	0.002 CL	+0.001	+0.000	+0.010	0.030 CL
			−0.000	−0.000	0.001 INT	−0.000	−0.015	−0.000	0.004 CL
	1¼	3	+0.002	+0.002	0.002 CL	+0.002	+0.000	+0.010	0.030 CL
			−0.000	−0.000	0.002 INT	−0.000	−0.015	−0.000	0.003 CL
	3	3½	+0.003	+0.002	0.002 CL	+0.003	+0.000	+0.010	0.030 CL
			−0.000	−0.000	0.003 INT	−0.000	−0.015	−0.000	0.002 CL
Parallel Rectangular	…	1¼	+0.001	+0.002	0.002 CL	+0.005	+0.000	+0.010	0.035 CL
			−0.000	−0.000	0.001 INT	−0.005	−0.015	−0.000	0.000 CL
	1¼	3	+0.002	+0.002	0.002 CL	+0.005	+0.000	+0.010	0.035 CL
			−0.000	−0.000	0.002 INT	−0.005	−0.015	−0.000	0.000 CL
	3	7	+0.003	+0.002	0.002 CL	+0.005	+0.000	+0.010	0.035 CL
			−0.000	−0.000	0.003 INT	−0.005	−0.015	−0.000	0.000 CL
Taper	…	1¼	+0.001	+0.002	0.002 CL	+0.005	+0.000	+0.010	0.005 CL
			−0.000	−0.000	0.001 INT	−0.000	−0.015	−0.000	0.025 INT
	1¼	3	+0.002	+0.002	0.002 CL	+0.005	+0.000	+0.010	0.005 CL
			−0.000	−0.000	0.002 INT	−0.000	−0.015	−0.000	0.025 INT
	3	[b]	+0.003	+0.002	0.002 CL	+0.005	+0.000	+0.010	0.005 CL
			−0.000	−0.000	0.003 INT	−0.000	−0.015	−0.000	0.025 INT

[a] Limits of variation. CL = Clearance; INT = Interference.

[b] To (Incl.) 3½-inch Square and 7-inch Rectangular key widths.

All dimensions are given in inches. See also text on page 2342.

Table 5. Suggested Keyseat Fillet Radius and Key Chamfer
ANSI B17.1-1967 (R1998)

Keyseat Depth, *H*/2		Fillet Radius	45 deg. Chamfer	Keyseat Depth, *H*2		Fillet Radius	45 deg. Chamfer
Over	To (Incl.)			Over	To (Incl.)		
1⁄8	1⁄4	1⁄32	3⁄64	7⁄8	1 1⁄4	3⁄16	7⁄32
1⁄4	1⁄2	1⁄16	5⁄64	1 1⁄4	1 3⁄4	1⁄4	9⁄32
1⁄2	7⁄8	1⁄8	5⁄32	1 3⁄4	2 1⁄2	3⁄8	13⁄32

All dimensions are given in inches.

Table 6. ANSI Standard Keyseat Tolerances for Electric Motor and Generator Shaft Extensions *ANSI B17.1-1967 (R1998)*

Keyseat Width		Width Tolerance	Depth Tolerance
Over	To (Incl.)		
…	1⁄4	+0.001 −0.001	+0.000 −0.015
1⁄4	3⁄4	+0.000 −0.002	+0.000 −0.015
3⁄4	1 1⁄4	+0.000 −0.003	+0.000 −0.015

All dimensions are given in inches.

Table 7. Set Screws for Use Over Keys *ANSI B17.1-1967 (R1998)*

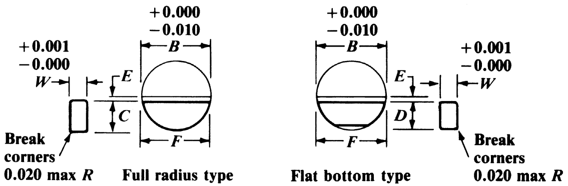
Nom. Shaft Diam.		Nom. Key Width	Set Screw Diam.	Nom. Shaft Diam.		Nom. Key Width	Set Screw Diam.
Over	To (Incl.)			Over	To (Incl.)		
5⁄16	7⁄16	3⁄32	No. 10	2 1⁄4	2 3⁄4	5⁄8	1⁄2
7⁄16	9⁄16	1⁄8	No. 10	2 3⁄4	3 1⁄4	3⁄4	5⁄8
9⁄16	7⁄8	3⁄16	1⁄4	3 1⁄4	3 3⁄4	7⁄8	3⁄4
7⁄8	1 1⁄4	1⁄4	5⁄16	3 3⁄4	4 1⁄2	1	3⁄4
1 1⁄4	1 3⁄8	5⁄16	3⁄8	4 1⁄2	5 1⁄2	1 1⁄4	7⁄8
1 3⁄8	1 3⁄4	3⁄8	3⁄8	5 1⁄2	6 1⁄2	1 1⁄2	1
1 3⁄4	2 1⁄4	1⁄2	1⁄2	…	…	…	…

All dimensions are given in inches.

These set screw diameter selections are offered as a guide but their use should be dependent upon design considerations.

ANSI Standard Woodruff Keys and Keyseats.—American National Standard B17.2 was approved in 1967, and reaffirmed in 1990. Data from this standard are shown in Tables 8, 9, and 10.

Table 8. ANSI Standard Woodruff Keys *ANSI B17.2-1967 (R1998)*

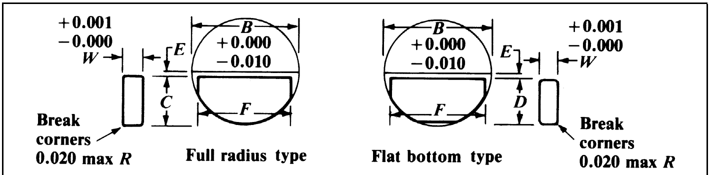


Key No.	Nominal Key Size W X B	Actual Length F +0.000 −0.010	Height of Key C Max.	C Min.	D Max.	D Min.	Distance Below Center E
202	1/16 × 1/4	0.248	0.109	0.104	0.109	0.104	1/64
202.5	1/16 × 5/16	0.311	0.140	0.135	0.140	0.135	1/64
302.5	3/32 × 5/16	0.311	0.140	0.135	0.140	0.135	1/64
203	1/16 × 3/8	0.374	0.172	0.167	0.172	0.167	1/64
303	3/32 × 3/8	0.374	0.172	0.167	0.172	0.167	1/64
403	1/8 × 3/8	0.374	0.172	0.167	0.172	0.167	1/64
204	1/16 × 1/2	0.491	0.203	0.198	0.194	0.188	3/64
304	3/32 × 1/2	0.491	0.203	0.198	0.194	0.188	3/64
404	1/8 × 1/2	0.491	0.203	0.198	0.194	0.188	3/64
305	3/32 × 5/8	0.612	0.250	0.245	0.240	0.234	1/16
405	1/8 × 5/8	0.612	0.250	0.245	0.240	0.234	1/16
505	5/32 × 5/8	0.612	0.250	0.245	0.240	0.234	1/16
605	3/16 × 5/8	0.612	0.250	0.245	0.240	0.234	1/16
406	1/8 × 3/4	0.740	0.313	0.308	0.303	0.297	1/16
506	5/32 × 3/4	0.740	0.313	0.308	0.303	0.297	1/16
606	3/16 × 3/4	0.740	0.313	0.308	0.303	0.297	1/16
806	1/4 × 3/4	0.740	0.313	0.308	0.303	0.297	1/16
507	5/32 × 7/8	0.866	0.375	0.370	0.365	0.359	1/16
607	3/16 × 7/8	0.866	0.375	0.370	0.365	0.359	1/16
707	7/32 × 7/8	0.866	0.375	0.370	0.365	0.359	1/16
807	1/4 × 7/8	0.866	0.375	0.370	0.365	0.359	1/16
608	3/16 × 1	0.992	0.438	0.433	0.428	0.422	1/16
708	7/32 × 1	0.992	0.438	0.433	0.428	0.422	1/16
808	1/4 × 1	0.992	0.438	0.433	0.428	0.422	1/16
1008	5/16 × 1	0.992	0.438	0.433	0.428	0.422	1/16
1208	3/8 × 1	0.992	0.438	0.433	0.428	0.422	1/16
609	3/16 × 1 1/8	1.114	0.484	0.479	0.475	0.469	5/64
709	7/32 × 1 1/8	1.114	0.484	0.479	0.475	0.469	5/64
809	1/4 × 1 1/8	1.114	0.484	0.479	0.475	0.469	5/64
1009	5/16 × 1 1/8	1.114	0.484	0.479	0.475	0.469	5/64
610	3/16 × 1 1/4	1.240	0.547	0.542	0.537	0.531	5/64
710	7/32 × 1 1/4	1.240	0.547	0.542	0.537	0.531	5/64
810	1/4 × 1 1/4	1.240	0.547	0.542	0.537	0.531	5/64
1010	5/16 × 1 1/4	1.240	0.547	0.542	0.537	0.531	5/64
1210	3/8 × 1 1/4	1.240	0.547	0.542	0.537	0.531	5/64
811	1/4 × 1 3/8	1.362	0.594	0.589	0.584	0.578	3/32
1011	5/16 × 1 3/8	1.362	0.594	0.589	0.584	0.578	3/32
1211	3/8 × 1 3/8	1.362	0.594	0.589	0.584	0.578	3/32
812	1/4 × 1 1/2	1.484	0.641	0.636	0.631	0.625	7/64
1012	5/16 × 1 1/2	1.484	0.641	0.636	0.631	0.625	7/64
1212	3/8 × 1 1/2	1.484	0.641	0.636	0.631	0.625	7/64

All dimensions are given in inches.

The Key numbers indicate normal 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 *W* in thirty-seconds of an inch.

Table 9. ANSI Standard Woodruff Keys *ANSI B17.2-1967 (R1998)*



Key No.	Nominal Key Size $W \times B$	Actual Length *F* +0.000 −0.010	Height of Key *C* Max.	Height of Key *C* Min.	Height of Key *D* Max.	Height of Key *D* Min.	Distance Below Center *E*
617-1	$\frac{3}{16} \times 2\frac{1}{8}$	1.380	0.406	0.401	0.396	0.390	$\frac{21}{32}$
817-1	$\frac{1}{4} \times 2\frac{1}{8}$	1.380	0.406	0.401	0.396	0.390	$\frac{21}{32}$
1017-1	$\frac{5}{16} \times 2\frac{1}{8}$	1.380	0.406	0.401	0.396	0.390	$\frac{21}{32}$
1217-1	$\frac{3}{8} \times 2\frac{1}{8}$	1.380	0.406	0.401	0.396	0.390	$\frac{21}{32}$
617	$\frac{3}{16} \times 2\frac{1}{8}$	1.723	0.531	0.526	0.521	0.515	$\frac{17}{32}$
817	$\frac{1}{4} \times 2\frac{1}{8}$	1.723	0.531	0.526	0.521	0.515	$\frac{17}{32}$
1017	$\frac{5}{16} \times 2\frac{1}{8}$	1.723	0.531	0.526	0.521	0.515	$\frac{17}{32}$
1217	$\frac{3}{8} \times 2\frac{1}{8}$	1.723	0.531	0.526	0.521	0.515	$\frac{17}{32}$
822-1	$\frac{1}{4} \times 2\frac{3}{4}$	2.000	0.594	0.589	0.584	0.578	$\frac{25}{32}$
1022-1	$\frac{5}{16} \times 2\frac{3}{4}$	2.000	0.594	0.589	0.584	0.578	$\frac{25}{32}$
1222-1	$\frac{3}{8} \times 2\frac{3}{4}$	2.000	0.594	0.589	0.584	0.578	$\frac{25}{32}$
1422-1	$\frac{7}{16} \times 2\frac{3}{4}$	2.000	0.594	0.589	0.584	0.578	$\frac{25}{32}$
1622-1	$\frac{1}{2} \times 2\frac{3}{4}$	2.000	0.594	0.589	0.584	0.578	$\frac{25}{32}$
822	$\frac{1}{4} \times 2\frac{3}{4}$	2.317	0.750	0.745	0.740	0.734	$\frac{5}{8}$
1022	$\frac{5}{16} \times 2\frac{3}{4}$	2.317	0.750	0.745	0.740	0.734	$\frac{5}{8}$
1222	$\frac{3}{8} \times 2\frac{3}{4}$	2.317	0.750	0.745	0.740	0.734	$\frac{5}{8}$
1422	$\frac{7}{16} \times 2\frac{3}{4}$	2.317	0.750	0.745	0.740	0.734	$\frac{5}{8}$
1622	$\frac{1}{2} \times 2\frac{3}{4}$	2.317	0.750	0.745	0.740	0.734	$\frac{5}{8}$
1228	$\frac{3}{8} \times 3\frac{1}{2}$	2.880	0.938	0.933	0.928	0.922	$\frac{13}{16}$
1428	$\frac{7}{16} \times 3\frac{1}{2}$	2.880	0.938	0.933	0.928	0.922	$\frac{13}{16}$
1628	$\frac{1}{2} \times 3\frac{1}{2}$	2.880	0.938	0.933	0.928	0.922	$\frac{13}{16}$
1828	$\frac{9}{16} \times 3\frac{1}{2}$	2.880	0.938	0.933	0.928	0.922	$\frac{13}{16}$
2028	$\frac{5}{8} \times 3\frac{1}{2}$	2.880	0.938	0.933	0.928	0.922	$\frac{13}{16}$
2228	$\frac{11}{16} \times 3\frac{1}{2}$	2.880	0.938	0.933	0.928	0.922	$\frac{13}{16}$
2428	$\frac{3}{4} \times 3\frac{1}{2}$	2.880	0.938	0.933	0.928	0.922	$\frac{13}{16}$

All dimensions are given in inches.

The key numbers indicate 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 *W* in thirty-seconds of an inch.

The key numbers with the −1 designation, while representing the nominal key size have a shorter length *F* and due to a greater distance below center *E* are less in height than the keys of the same number without the −1 designation.

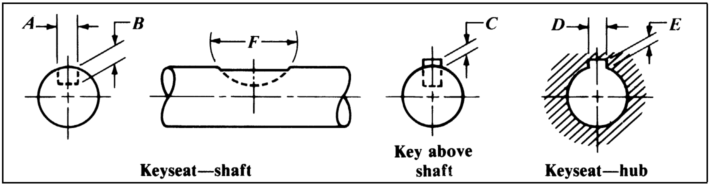


Table 10. ANSI Keyseat Dimensions for Woodruff Keys
ANSI B17.2-1967 (R1998)

Key No.	Nominal Size Key	Keyseat—Shaft					Key Above Shaft	Keyseat—Hub	
		Width A[a]		Depth B	Diameter F		Height C	Width D	Depth E
		Min.	Max.	+0.005 −0.000	Min.	Max.	+0.005 −0.005	+0.002 −0.000	+0.005 −0.000
202	1/16 × 1/4	0.0615	0.0630	0.0728	0.250	0.268	0.0312	0.0635	0.0372
202.5	1/16 × 5/16	0.0615	0.0630	0.1038	0.312	0.330	0.0312	0.0635	0.0372
302.5	3/32 × 5/16	0.0928	0.0943	0.0882	0.312	0.330	0.0469	0.0948	0.0529
203	1/16 × 3/8	0.0615	0.0630	0.1358	0.375	0.393	0.0312	0.0635	0.0372
303	3/32 × 3/8	0.0928	0.0943	0.1202	0.375	0.393	0.0469	0.0948	0.0529
403	1/8 × 3/8	0.1240	0.1255	0.1045	0.375	0.393	0.0625	0.1260	0.0685
204	1/16 × 1/2	0.0615	0.0630	0.1668	0.500	0.518	0.0312	0.0635	0.0372
304	3/32 × 1/2	0.0928	0.0943	0.1511	0.500	0.518	0.0469	0.0948	0.0529
404	1/8 × 1/2	0.1240	0.1255	0.1355	0.500	0.518	0.0625	0.1260	0.0685
305	3/32 × 5/8	0.0928	0.0943	0.1981	0.625	0.643	0.0469	0.0948	0.0529
405	1/8 × 5/8	0.1240	0.1255	0.1825	0.625	0.643	0.0625	0.1260	0.0685
505	5/32 × 5/8	0.1553	0.1568	0.1669	0.625	0.643	0.0781	0.1573	0.0841
605	3/16 × 5/8	0.1863	0.1880	0.1513	0.625	0.643	0.0937	0.1885	0.0997
406	1/8 × 3/4	0.1240	0.1255	0.2455	0.750	0.768	0.0625	0.1260	0.0685
506	5/32 × 3/4	0.1553	0.1568	0.2299	0.750	0.768	0.0781	0.1573	0.0841
606	3/16 × 3/4	0.1863	0.1880	0.2143	0.750	0.768	0.0937	0.1885	0.0997
806	1/4 × 3/4	0.2487	0.2505	0.1830	0.750	0.768	0.1250	0.2510	0.1310
507	5/32 × 7/8	0.1553	0.1568	0.2919	0.875	0.895	0.0781	0.1573	0.0841
607	3/16 × 7/8	0.1863	0.1880	0.2763	0.875	0.895	0.0937	0.1885	0.0997
707	7/32 × 7/8	0.2175	0.2193	0.2607	0.875	0.895	0.1093	0.2198	0.1153
807	1/4 × 7/8	0.2487	0.2505	0.2450	0.875	0.895	0.1250	0.2510	0.1310
608	3/16 × 1	0.1863	0.1880	0.3393	1.000	1.020	0.0937	0.1885	0.0997
708	7/32 × 1	0.2175	0.2193	0.3237	1.000	1.020	0.1093	0.2198	0.1153
808	1/4 × 1	0.2487	0.2505	0.3080	1.000	1.020	0.1250	0.2510	0.1310
1008	5/16 × 1	0.3111	0.3130	0.2768	1.000	1.020	0.1562	0.3135	0.1622
1208	3/8 × 1	0.3735	0.3755	0.2455	1.000	1.020	0.1875	0.3760	0.1935
609	3/16 × 1 1/8	0.1863	0.1880	0.3853	1.125	1.145	0.0937	0.1885	0.0997
709	7/32 × 1 1/8	0.2175	0.2193	0.3697	1.125	1.145	0.1093	0.2198	0.1153
809	1/4 × 1 1/8	0.2487	0.2505	0.3540	1.125	1.145	0.1250	0.2510	0.1310
1009	5/16 × 1 1/8	0.3111	0.3130	0.3228	1.125	1.145	0.1562	0.3135	0.1622
610	3/16 × 1 1/4	0.1863	0.1880	0.4483	1.250	1.273	0.0937	0.1885	0.0997
710	7/32 × 1 1/4	0.2175	0.2193	0.4327	1.250	1.273	0.1093	0.2198	0.1153
810	1/4 × 1 1/4	0.2487	0.2505	0.4170	1.250	1.273	0.1250	0.2510	0.1310
1010	5/16 × 1 1/4	0.3111	0.3130	0.3858	1.250	1.273	0.1562	0.3135	0.1622
1210	3/8 × 1 1/4	0.3735	0.3755	0.3545	1.250	1.273	0.1875	0.3760	0.1935
811	1/4 × 1 3/8	0.2487	0.2505	0.4640	1.375	1.398	0.1250	0.2510	0.1310
1011	5/16 × 1 3/8	0.3111	0.3130	0.4328	1.375	1.398	0.1562	0.3135	0.1622

Table 10. *(Continued)* **ANSI Keyseat Dimensions for Woodruff Keys**
ANSI B17.2-1967 (R1998)

Key No.	Nominal Size Key	Keyseat—Shaft					Key Above Shaft	Keyseat—Hub	
		Width *A*[a]		Depth *B*	Diameter *F*		Height *C*	Width *D*	Depth *E*
		Min.	Max.	+0.005 −0.000	Min.	Max.	+0.005 −0.005	+0.002 −0.000	+0.005 −0.000
1211	$\frac{3}{8} \times 1\frac{3}{8}$	0.3735	0.3755	0.4015	1.375	1.398	0.1875	0.3760	0.1935
812	$\frac{1}{4} \times 1\frac{1}{2}$	0.2487	0.2505	0.5110	1.500	1.523	0.1250	0.2510	0.1310
1012	$\frac{5}{16} \times 1\frac{1}{2}$	0.3111	0.3130	0.4798	1.500	1.523	0.1562	0.3135	0.1622
1212	$\frac{3}{8} \times 1\frac{1}{2}$	0.3735	0.3755	0.4485	1.500	1.523	0.1875	0.3760	0.1935
617-1	$\frac{3}{16} \times 2\frac{1}{8}$	0.1863	0.1880	0.3073	2.125	2.160	0.0937	0.1885	0.0997
817-1	$\frac{1}{4} \times 2\frac{1}{8}$	0.2487	0.2505	0.2760	2.125	2.160	0.1250	0.2510	0.1310
1017-1	$\frac{5}{16} \times 2\frac{1}{8}$	0.3111	0.3130	0.2448	2.125	2.160	0.1562	0.3135	0.1622
1217-1	$\frac{3}{8} \times 2\frac{1}{8}$	0.3735	0.3755	0.2135	2.125	2.160	0.1875	0.3760	0.1935
617	$\frac{3}{16} \times 2\frac{1}{8}$	0.1863	0.1880	0.4323	2.125	2.160	0.0937	0.1885	0.0997
817	$\frac{1}{4} \times 2\frac{1}{8}$	0.2487	0.2505	0.4010	2.125	2.160	0.1250	0.2510	0.1310
1017	$\frac{5}{16} \times 2\frac{1}{8}$	0.3111	0.3130	0.3698	2.125	2.160	0.1562	0.3135	0.1622
1217	$\frac{3}{8} \times 2\frac{1}{8}$	0.3735	0.3755	0.3385	2.125	2.160	0.1875	0.3760	0.1935
822-1	$\frac{1}{4} \times 2\frac{3}{4}$	0.2487	0.2505	0.4640	2.750	2.785	0.1250	0.2510	0.1310
1022-1	$\frac{5}{16} \times 2\frac{3}{4}$	0.3111	0.3130	0.4328	2.750	2.785	0.1562	0.3135	0.1622
1222-1	$\frac{3}{8} \times 2\frac{3}{4}$	0.3735	0.3755	0.4015	2.750	2.785	0.1875	0.3760	0.1935
1422-1	$\frac{7}{16} \times 2\frac{3}{4}$	0.4360	0.4380	0.3703	2.750	2.785	0.2187	0.4385	0.2247
1622-1	$\frac{1}{2} \times 2\frac{3}{4}$	0.4985	0.5005	0.3390	2.750	2.785	0.2500	0.5010	0.2560
822	$\frac{1}{4} \times 2\frac{3}{4}$	0.2487	0.2505	0.6200	2.750	2.785	0.1250	0.2510	0.1310
1022	$\frac{5}{16} \times 2\frac{3}{4}$	0.3111	0.3130	0.5888	2.750	2.785	0.1562	0.3135	0.1622
1222	$\frac{3}{8} \times 2\frac{3}{4}$	0.3735	0.3755	0.5575	2.750	2.785	0.1875	0.3760	0.1935
1422	$\frac{7}{16} \times 2\frac{3}{4}$	0.4360	0.4380	0.5263	2.750	2.785	0.2187	0.4385	0.2247
1622	$\frac{1}{2} \times 2\frac{3}{4}$	0.4985	0.5005	0.4950	2.750	2.785	0.2500	0.5010	0.2560
1228	$\frac{3}{8} \times 3\frac{1}{2}$	0.3735	0.3755	0.7455	3.500	3.535	0.1875	0.3760	0.1935
1428	$\frac{7}{16} \times 3\frac{1}{2}$	0.4360	0.4380	0.7143	3.500	3.535	0.2187	0.4385	0.2247
1628	$\frac{1}{2} \times 3\frac{1}{2}$	0.4985	0.5005	0.6830	3.500	3.535	0.2500	0.5010	0.2560
1828	$\frac{9}{16} \times 3\frac{1}{2}$	0.5610	0.5630	0.6518	3.500	3.535	0.2812	0.5635	0.2872
2028	$\frac{5}{8} \times 3\frac{1}{2}$	0.6235	0.6255	0.6205	3.500	3.535	0.3125	0.6260	0.3185
2228	$\frac{11}{16} \times 3\frac{1}{2}$	0.6860	0.6880	0.5893	3.500	3.535	0.3437	0.6885	0.3497
2428	$\frac{3}{4} \times 3\frac{1}{2}$	0.7485	0.7505	0.5580	3.500	3.535	0.3750	0.7510	0.3810

[a] These Width *A* values were set with the maximum keyseat (shaft) width as that figure which will receive a key with the greatest amount of looseness consistent with assuring the key's sticking in the keyseat (shaft). Minimum keyseat width is that figure permitting the largest shaft distortion acceptable when assembling maximum key in minimum keyseat. Dimensions *A*, *B*, *C*, *D* are taken at side intersection.

All dimensions are given in inches.

The following definitions are given in this standard:

Woodruff Key: A Remountable machinery part which, when assembled into key-seats, provides a positive means for transmitting torque between the shaft and hub.

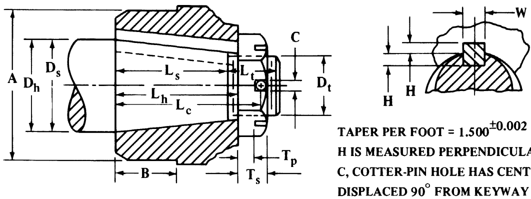
Woodruff Key Number: An identification number by which the size of key may be readily determined.

Woodruff Keyseat—Shaft: The circular pocket in which the key is retained.

Woodruff Keyseat—Hub: An axially located rectangular groove in a hub. (This has been referred to as a keyway.)

Woodruff Keyseat Milling Cutter: An arbor type or shank type milling cutter normally used for milling Woodruff keyseats in shafts.

Taper Shaft Ends with Slotted Nuts *SAE Standard*



TAPER PER FOOT = $1.500^{\pm 0.002}$ IN.
H IS MEASURED PERPENDICULAR TO KEY.
C, COTTER-PIN HOLE HAS CENTERLINE DISPLACED 90° FROM KEYWAY CENTER

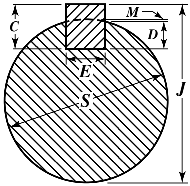
Nom. Diam.	Diam. of Shaft, D_s		Diam. of Hole, D_h		L_c	L_s	L_h	L_t	T_s	T_p	Nut Width, Flats
	Max.	Min.	Max.	Min.							
1/4	0.250	0.249	0.248	0.247	9/16	5/16	3/8	5/16	7/32	9/64	5/16
3/8	0.375	0.374	0.373	0.372	47/64	7/16	1/2	23/64	17/64	3/16	1/2
1/2	0.500	0.499	0.498	0.497	63/64	11/16	3/4	23/64	17/64	3/16	1/2
5/8	0.625	0.624	0.623	0.622	1 3/32	11/16	3/4	17/32	7/16	1/4	3/4
3/4	0.750	0.749	0.748	0.747	1 11/32	15/16	1	17/32	7/16	1/4	3/4
7/8	0.875	0.874	0.873	0.872	1 11/16	1 1/8	1 1/4	11/16	1/2	5/16	15/16
1	1.001	0.999	0.997	0.995	1 15/16	1 3/8	1 1/2	11/16	1/2	5/16	1 1/16
1 1/8	1.126	1.124	1.122	1.120	1 15/16	1 3/8	1 1/2	11/16	1/2	5/16	1 1/4
1 1/4	1.251	1.249	1.247	1.245	1 15/16	1 3/8	1 1/2	11/16	1/2	5/16	1 7/16
1 3/8	1.376	1.374	1.372	1.370	2 7/16	1 7/8	2	11/16	1/2	5/16	1 7/16
1 1/2	1.501	1.499	1.497	1.495	2 7/16	1 7/8	2	11/16	1/2	5/16	1 7/16
1 5/8	1.626	1.624	1.622	1.620	2 13/16	2 1/8	2 1/4	13/16	5/8	7/16	2 3/16
1 3/4	1.751	1.749	1.747	1.745	2 13/16	2 1/8	2 1/4	13/16	5/8	7/16	2 3/16
1 7/8	1.876	1.874	1.872	1.870	3 1/16	2 3/8	2 1/2	13/16	5/8	7/16	2 3/16
2	2.001	1.999	1.997	1.995	3 9/16	2 7/8	3	13/16	5/8	7/16	2 3/16
2 1/4	2.252	2.248	2.245	2.242	3 9/16	2 7/8	3	13/16	5/8	7/16	2 3/8
2 1/2	2.502	2.498	2.495	2.492	4 9/32	3 3/8	3 1/2	1 1/4	1	5/8	3 1/8
2 3/4	2.752	2.748	2.745	2.742	4 9/32	3 3/8	3 1/2	1 1/4	1	5/8	3 1/8
3	3.002	2.998	2.995	2.992	2 25/32	3 7/8	4	1 1/4	1	5/8	3 1/8
3 1/4	3.252	3.248	3.245	3.242	5 1/32	4 1/8	4 1/4	1 1/4	1	5/8	3 1/8
3 1/2	3.502	3.498	3.495	3.492	5 7/16	4 3/8	4 1/2	1 3/8	1 1/8	3/4	3 7/8
4	4.002	3.998	3.995	3.992	6 7/16	5 3/8	5 1/2	1 3/8	1 1/8	3/4	3 7/8

Nom. Diam.	D_t	Thds. per Inch	Keyway				Square Key		A	B	C
			W		H						
			Max.	Min.	Max.	Min.	Max.	Min.			
1/4	#10	40	0.0625	.0615	.037	.033	0.0635	0.0625	1/2	3/16	5/64
3/8	5/16	32	0.0937	.0927	.053	.049	0.0947	0.0937	11/16	1/4	5/64
1/2	5/16	32	0.1250	.1240	.069	.065	0.1260	0.1250	7/8	3/8	5/64
5/8	1/2	28	0.1562	.1552	.084	.080	0.1572	0.1562	1 1/16	3/8	1/8
3/4	1/2	28	0.1875	.1865	.100	.096	0.1885	0.1875	1 1/4	5/8	1/8
7/8	5/8	24	0.2500	.2490	.131	.127	0.2510	0.2500	1 1/2	3/4	5/32
1	3/4	20	0.2500	.2490	.131	.127	0.2510	0.2500	1 3/4	7/8	5/32
1 1/8	7/8	20	0.3125	.3115	.162	.158	0.3135	0.3125	2	7/8	5/32
1 1/4	1	20	0.3125	.3115	.162	.158	0.3135	0.3125	2 1/8	7/8	5/32
1 3/8	1	20	0.3750	.3740	.194	.190	0.3760	0.3750	2 1/4	1	5/32
1 1/2	1	20	0.3750	.3740	.194	.190	0.3760	0.3750	2 1/2	1	5/32
1 5/8	1 1/4	18	0.4375	.4365	.225	.221	0.4385	0.4375	2 3/4	1 1/4	5/32
1 3/4	1 1/4	18	0.4375	.4365	.225	.221	0.4385	0.4375	3	1 1/4	5/32
1 7/8	1 1/4	18	0.4375	.4365	.225	.221	0.4385	0.4375	3 1/8	1 1/4	5/32
2	1 1/4	18	0.5000	.4990	.256	.252	0.5010	0.5000	3 1/4	1 1/2	5/32
2 1/4	1 1/2	18	0.5625	.5610	.287	.283	0.5640	0.5625	3 1/2	1 1/2	5/32
2 1/2	2	16	0.6250	.6235	.319	.315	0.6265	0.6250	4	1 3/4	7/32
2 3/4	2	16	0.6875	.6860	.350	.346	0.6890	0.6875	4 3/8	1 3/4	7/32
3	2	16	0.7500	.7485	.381	.377	0.7515	0.7500	4 3/4	2	7/32
3 1/4	2	16	0.7500	.7485	.381	.377	0.7515	0.7500	5	2 1/8	7/32
3 1/2	2 1/2	16	0.8750	.8735	.444	.440	0.8765	0.8750	5 1/2	2 1/4	9/32
4	2 1/2	16	1.0000	.9985	.506	.502	1.0015	1.0000	6 1/4	2 3/4	9/32

All dimensions in inches except where otherwise noted.

Chamfered Keys and Filleted Keyseats.—In general practice, chamfered keys and filleted keyseats are not used. However, it is recognized that fillets in keyseats decrease stress concentration at corners. When used, fillet radii should be as large as possible without causing excessive bearing stresses due to reduced contact area between the key and its mating parts. Keys must be chamfered or rounded to clear fillet radii. Values in Table 5 assume general conditions and should be used only as a guide when critical stresses are encountered.

Table 11. Finding Depth of Keyseat and Distance from Top of Key to Bottom of Shaft



For milling keyseats, the total depth to feed cutter in from outside of shaft to bottom of keyseat is $M + D$, where D is depth of keyseat.

For checking an assembled key and shaft, caliper measurement J between top of key and bottom of shaft is used.

$$J = S - (M + D) + C$$

where C is depth of key. For Woodruff keys, dimensions C and D can be found in Tables 8 through 10. Assuming shaft diameter S is normal size, the tolerance on dimension J for Woodruff keys in keyslots are + 0.000, −0.010 inch.

Dia. of Shaft S, Inches	Width of Keyseat, E														
	1⁄16	3⁄32	1⁄8	5⁄32	3⁄16	7⁄32	1⁄4	5⁄16	3⁄8	7⁄16	1⁄2	9⁄16	5⁄8	11⁄16	3⁄4
	Dimension M, Inch														
0.3125	.0032	…	…	…	…	…	…	…	…	…	…	…	…	…	…
0.3437	.0029	.0065	…	…	…	…	…	…	…	…	…	…	…	…	…
0.3750	.0026	.0060	.0107	…	…	…	…	…	…	…	…	…	…	…	…
0.4060	.0024	.0055	.0099	…	…	…	…	…	…	…	…	…	…	…	…
0.4375	.0022	.0051	.0091	…	…	…	…	…	…	…	…	…	…	…	…
0.4687	.0021	.0047	.0085	.0134	…	…	…	…	…	…	…	…	…	…	…
0.5000	.0020	.0044	.0079	.0125	…	…	…	…	…	…	…	…	…	…	…
0.5625	…	.0039	.0070	.0111	.0161	…	…	…	…	…	…	…	…	…	…
0.6250	…	.0035	.0063	.0099	.0144	.0198	…	…	…	…	…	…	…	…	…
0.6875	…	.0032	.0057	.0090	.0130	.0179	.0235	…	…	…	…	…	…	…	…
0.7500	…	.0029	.0052	.0082	.0119	.0163	.0214	.0341	…	…	…	…	…	…	…
0.8125	…	.0027	.0048	.0076	.0110	.0150	.0197	.0312	…	…	…	…	…	…	…
0.8750	…	.0025	.0045	.0070	.0102	.0139	.0182	.0288	…	…	…	…	…	…	…
0.9375	…	…	.0042	.0066	.0095	.0129	.0170	.0263	.0391	…	…	…	…	…	…
1.0000	…	…	.0039	.0061	.0089	.0121	.0159	.0250	.0365	…	…	…	…	…	…
1.0625	…	…	.0037	.0058	.0083	.0114	.0149	.0235	.0342	…	…	…	…	…	…
1.1250	…	…	.0035	.0055	.0079	.0107	.0141	.0221	.0322	.0443	…	…	…	…	…
1.1875	…	…	.0033	.0052	.0074	.0102	.0133	.0209	.0304	.0418	…	…	…	…	…
1.2500	…	…	.0031	.0049	.0071	.0097	.0126	.0198	.0288	.0395	…	…	…	…	…
1.3750	…	…	…	.0045	.0064	.0088	.0115	.0180	.0261	.0357	.0471	…	…	…	…
1.5000	…	…	…	.0041	.0059	.0080	.0105	.0165	.0238	.0326	.0429	…	…	…	…
1.6250	…	…	…	.0038	.0054	.0074	.0097	.0152	.0219	.0300	.0394	.0502	…	…	…
1.7500	…	…	…	…	.0050	.0069	.0090	.0141	.0203	.0278	.0365	.0464	…	…	…
1.8750	…	…	…	…	.0047	.0064	.0084	.0131	.0189	.0259	.0340	.0432	.0536	…	…
2.0000	…	…	…	…	.0044	.0060	.0078	.0123	.0177	.0242	.0318	.0404	.0501	…	…
2.1250	…	…	…	…	…	.0056	.0074	.0116	.0167	.0228	.0298	.0379	.0470	.0572	.0684
2.2500	…	…	…	…	…	…	.0070	.0109	.0157	.0215	.0281	.0357	.0443	.0538	.0643
2.3750	…	…	…	…	…	…	…	.0103	.0149	.0203	.0266	.0338	.0419	.0509	.0608
2.5000	…	…	…	…	…	…	…	…	.0141	.0193	.0253	.0321	.0397	.0482	.0576
2.6250	…	…	…	…	…	…	…	…	.0135	.0184	.0240	.0305	.0377	.0457	.0547
2.7500	…	…	…	…	…	…	…	…	…	.0175	.0229	.0291	.0360	.0437	.0521
2.8750	…	…	…	…	…	…	…	…	…	.0168	.0219	.0278	.0344	.0417	.0498
3.0000	…	…	…	…	…	…	…	…	…	…	.0210	.0266	.0329	.0399	.0476

Depths for Milling Keyseats.—The above table has been compiled to facilitate the accurate milling of keyseats. This table gives the distance *M* (see illustration accompanying table) between the top of the shaft and a line passing through the upper corners or edges of the keyseat. Dimension *M* is calculated by the formula: $M = \frac{1}{2}(S - \sqrt{S^2 - E^2})$ where *S* is diameter of shaft, and *E* is width of keyseat. A simple approximate formula that gives *M* to within 0.001 inch is $M = E_2 \div 4S$.

Cotters.—A cotter is a form of key that is used to connect rods, etc., that are subjected either to tension or compression or both, the cotter being subjected to shearing stresses at two transverse cross-sections. When taper cotters are used for drawing and holding parts together, if the cotter is held in place by the friction between the bearing surfaces, the taper should not be too great. Ordinarily a taper varying from ¼ to ½ inch per foot is used for plain cotters. When a set-screw or other device is used to prevent the cotter from backing out of its slot, the taper may vary from 1 ½ to 2 inches per foot.

British Keys and Keyways

British Standard Metric Keys and Keyways.—This British Standard, BS 4235:Part 1:1972 (1986), covers square and rectangular parallel keys and keyways, and square and rectangular taper keys and keyways. Plain and gib-head taper keys are specified. There are three classes of fit for the square and rectangular parallel keys and keyways, designated free, normal, and close. A *free fit* is applied when the application requires the hub of an assembly to slide over the key; a *normal fit* is employed when the key is to be inserted in the keyway with the minimum amount of fitting, as may be required in mass-production assembly work; and a *close fit* is applied when accurate fitting of the key is required under maximum material conditions, which may involve selection of components.

The Standard does not provide for misalignment or offset greater than can be accommodated within the dimensional tolerances. If an assembly is to be heavily stressed, a check should be made to ensure that the cumulative effect of misalignment or offset, or both, does not prevent satisfactory bearing on the key. Radii and chamfers are not normally provided on keybar and keys as supplied, but they can be produced during manufacture by agreement between the user and supplier.

Unless otherwise specified, keys in compliance with this Standard are manufactured from steel made to BS 970 having a tensile strength of not less than 550 MN/m^2 in the finished condition. BS 970, Part 1, lists the following steels and maximum section sizes, respectively, that meet this tensile strength requirement: 070M20, 25 × 14 mm; 070M26, 36 × 20 mm; 080M30, 90 × 45 mm; and 080M40, 100 × 50 mm.

At the time of publication of this Standard, the demand for metric keys was not sufficient to enable standard ranges of lengths to be established. The lengths given in the accompanying table are those shown as standard in ISO Recommendations R773: 1969, "Rectangular or Square Parallel Keys and their Corresponding Keyways (Dimensions in Millimeters)," and R 774: 1969, "Taper Keys and their Corresponding Keyways—with or without Gib Head (Dimensions in Millimeters)."

Tables 1 through 4 on the following pages cover the dimensions and tolerances of square and rectangular keys and keyways, and square and rectangular taper keys and keyways.

Table 1. British Standard Metric Keyways for Square and Rectangular Parallel Keys *BS 4235:Part 1:1972 (1986)*

Shaft		Key	Keyway											
Nonimal Diameter d		Size, $b \times h$	Width, b						Depth				Radius r	
			Nom.	Free Fit		Normal Fit		Close Fit	Shaft t_1		Hub t_2			
Over	Up to and Incl.			Shaft (H9)	Hub (D10)	Shaft (N9)	Hub (J_S9)[a]	Shaft and Hub (P9)	Nom.	Tol.	Nom.	Tol.	Max.	Min.
				Tolerances										
Keyways for Square Parallel Keys														
6	8	2 × 2	2	+0.025 0	+0.060 +0.020	−0.004 −0.029	+0.012 −0.012	−0.006 −0.031	1.2	+0.1 0	1	+0.1 0	0.16	0.08
8	10	3 × 3	3						1.8		1.4		0.16	0.08
10	12	4 × 4	4	+0.030 0	+0.078 +0.030	0 −0.030	+0.015 −0.015	−0.012 −0.042	2.5		1.8		0.16	0.08
12	17	5 × 5	5						3		2.3		0.25	0.16
17	22	6 × 6	6						3.5		2.8		0.25	0.16

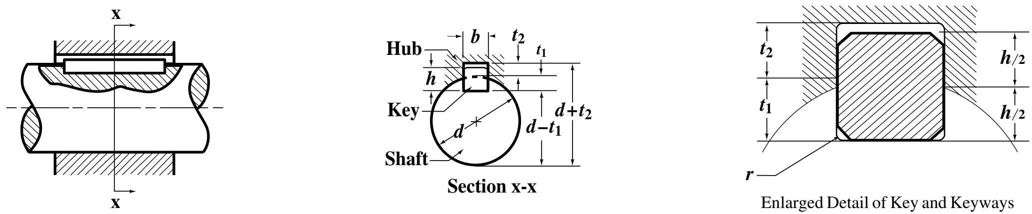
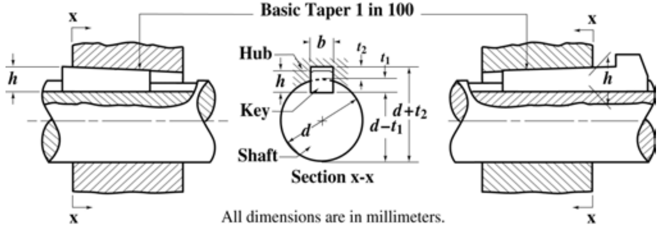


Table 1. *(Continued)* **British Standard Metric Keyways for Square and Rectangular Parallel Keys** *BS 4235:Part 1:1972 (1986)*

Shaft: Nonimal Diameter d Over	Shaft: Nonimal Diameter d Up to and Incl.	Key: Size, $b \times h$	Keyway Width, b: Nom.	Free Fit Shaft (H9) Tolerances	Free Fit Hub (D10) Tolerances	Normal Fit Shaft (N9) Tolerances	Normal Fit Hub (J_S9)[a] Tolerances	Close Fit Shaft and Hub (P9) Tolerances	Depth Shaft t_1 Nom.	Depth Shaft t_1 Tol.	Depth Hub t_2 Nom.	Depth Hub t_2 Tol.	Radius r Max.	Radius r Min.
Keyways for Rectangular Parallel Keys														
22	30	8 × 7	8	+0.036	+0.098	0	+0.018	−0.015	4		3.3		0.25	0.16
30	38	10 × 8	10	0	+0.040	−0.036	−0.018	−0.051	5		3.3		0.40	0.25
38	44	12 × 8	12						5		3.3		0.40	0.25
44	50	14 × 9	14	+0.043	+0.120	0	+0.021	−0.018	5.5		3.8		0.40	0.25
50	58	16 × 10	16		+0.050	−0.043	−0.021	−0.061	6		4.3		0.40	0.25
58	65	18 × 11	18						7	+0.2	4.4	+0.2	0.40	0.25
65	75	20 × 12	20						7.5	0	4.9	0	0.60	0.40
75	85	22 × 14	22	+0.052	+0.149	0	+0.026	−0.022	9		5.4		0.60	0.40
85	95	25 × 14	25		+0.065	−0.052	−0.026	−0.074	9		5.4		0.60	0.40
95	110	28 × 16	28						10		6.4		0.60	0.40
110	130	32 × 18	32						11		7.4		0.60	0.40
130	150	36 × 20	36						12		8.4		1.00	0.70
150	170	40 × 22	40	+0.062	+0.180	0	+0.031	−0.026	13		9.4		1.00	0.70
170	200	45 × 25	45		−0.080	−0.062	−0.031	−0.088	15		10.4		1.00	0.70
200	230	50 × 28	50						17		11.4		1.00	0.70
230	260	56 × 32	56						20	+0.3	12.4	+0.3	1.60	1.20
260	290	63 × 32	63	+0.074	+0.220	0	+0.037	−0.032	20	0	12.4	0	1.60	1.20
290	330	70 × 36	70		+0.100	−0.074	−0.037	−0.106	22		14.4		1.60	1.20
330	380	80 × 40	80						25		15.4		2.50	2.00
380	440	90 × 45	90	+0.087	+0.260	0	+0.043	−0.037	28		17.4		2.50	2.00
440	500	100 × 50	100	0	+0.120	−0.087	−0.043	−0.124	31		19.5		2.50	2.00

[a] Tolerance limits J_S9 are quoted from BS 4500, "ISO Limits and Fits," to three significant figures.
All dimensions in millimeters.

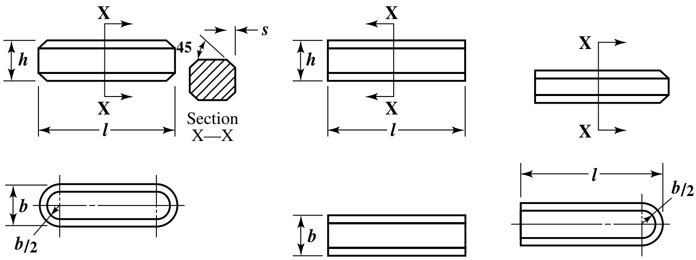
Table 2. British Standard Metric Keyways for Square and Rectangular Taper Keys, *BS 4235:Part 1:1972 (1986)*



All dimensions are in millimeters.

Shaft Nominal Diameter *d* Over	Shaft Nominal Diameter *d* Up to and Incl.	Key Size, $b \times h$	Keyway Width *b*, Shaft and Hub Nom.	Keyway Width *b*, Shaft and Hub Tol. (D10)	Keyway Depth Shaft t_1 Nom.	Keyway Depth Shaft t_1 Tol.	Keyway Depth Hub t_2 Nom.	Keyway Depth Hub t_2 Tol.	Corner Radius of Keyway Max.	Corner Radius of Keyway Min.
Keyways for Square Taper Keys										
6	8	2 × 2	2	+0.060 +0.020	1.2		0.5		0.16	0.08
8	10	3 × 3	3		1.8	+0.1 0	0.9	+0.1 0	0.16	0.08
10	12	4 × 4	4		2.5		1.2		0.16	0.08
12	17	5 × 5	5	+0.078 +0.030	3		1.7		0.25	0.16
17	22	6 × 6	6		3.5	+0.2 0	2.2	+0.2 0	0.25	0.16
Keyways for Rectangular Taper Keys										
22	30	8 × 7	8	+0.098 +0.040	4		2.4		0.25	0.16
30	38	10 × 8	10		5		2.4		0.40	0.25
38	44	12 × 8	12		5		2.4		0.40	0.25
44	50	14 × 9	14	+0.120 +0.050	5.5		2.9		0.40	0.25
50	58	16 × 10	16		6		3.4		0.40	0.25
58	65	18 × 11	18		7	+0.2 0	3.4	+0.2 0	0.40	0.25
65	75	20 × 12	20		7.5		3.9		0.60	0.40
75	85	22 × 14	22	+0.149 +0.065	9		4.4		0.60	0.40
85	95	25 × 14	25		9		4.4		0.60	0.40
95	110	28 × 16	28		10		5.4		0.60	0.40
110	130	32 × 18	32		11		6.4		0.60	0.40
130	150	36 × 20	36		12		7.1		1.00	0.70
150	170	40 × 22	40	+0.180 +0.080	13		8.1		1.00	0.70
170	200	45 × 25	45		15		9.1		1.00	0.70
200	230	50 × 28	50		17		10.1		1.00	0.70
230	260	56 × 32	56		20	+0.3 0	11.1	+0.3 0	1.60	1.20
260	290	63 × 32	63	+0.220 +0.120	20		11.1		1.60	1.20
290	330	70 × 36	70		22		13.1		1.60	1.20
330	380	80 × 40	80		25		14.1		2.50	2.00
380	440	90 × 45	90	+0.260 +0.120	28		16.1		2.50	2.00
440	500	100 × 50	100		31		18.1		2.50	2.00

Table 3. British Standard Metric Square and Rectangular Parallel Keys.
BS 4235:Part 1:1972 (1986)

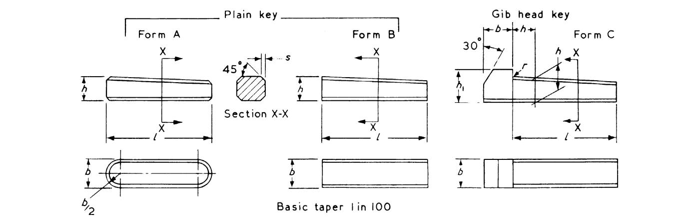


Form A **Form B** **Form C**

Width, *b*		Thickness, *h*		Chamfer, *s*		Length Range, *l*	
Nom.	Tol.[a]	Nom.	Tol.[a]	Min.	Max.	From	To
Square Parallel Keys							
2	0 −0.025	2	0 −0.025	0.16	0.25	6	20
3		3		0.16	0.25	6	36
4	0 −0.030	4	0 −0.030	0.16	0.25	8	45
5		5		0.25	0.40	10	56
6		6		0.25	0.40	14	70
Rectangular Parallel Keys							
8	0 −0.036	7	0 −0.090	0.25	0.40	18	90
10		8		0.40	0.60	22	110
12	0 −0.043	8		0.40	0.60	28	140
14		9		0.40	0.60	36	160
16		10		0.40	0.60	45	180
18		11	0 −0.110	0.40	0.60	50	200
20	0 −0.052	12		0.60	0.80	56	220
22		14		0.60	0.80	63	250
25		14		0.60	0.80	70	280
28		16		0.60	0.80	80	320
32	0 −0.062	18		0.60	0.80	90	360
36		20	0 −0.130	1.00	1.20	100	400
40		22		1.00	1.20	…	…
45		25		1.00	1.20	…	…
50		28		1.00	1.20	…	…
56	0 −0.074	32	0 −0.160	1.60	2.00	…	…
63		32		1.60	2.00	…	…
70		36		1.60	2.00	…	…
80		40		2.50	3.00	…	…
90	0 −0.087	45		2.50	3.00	…	…
100		50		2.50	3.00	…	…

[a] The tolerance on the width and thickness of square taper keys is h9, and on the width and thickness of rectangular keys, h9 and h11, respectively, in accordance with ISO metric limits and fits. All dimensions in millimeters.

Table 4. British Standard Metric Square and Rectangular Taper Keys
BS 4235:Part 1:1972 (1986)



Width *b*		Thickness *h*		Chamfer *s*		Length Range *l*		Gib head h_1	Radius *r*
Nom.	Tol.[a]	Nom.	Tol.[a]	Min.	Max.	From	To	Nom.	Nom.
Square Taper Keys									
2	0 −0.025	2	0 −0.025	0.16	0.25	6	20	…	…
3		3		0.16	0.25	6	36	…	…
4	0 −0.030	4	0 −0.030	0.16	0.25	8	45	7	0.25
5		5		0.25	0.40	10	56	8	0.25
6		6		0.25	0.40	14	70	10	0.25
Rectangular Taper Keys									
8	0 −0.036	7	0 −0.090	0.25	0.40	18	90	11	1.5
10		8		0.40	0.60	22	110	12	1.5
12	0 −0.043	8		0.40	0.60	28	140	12	1.5
14		9		0.40	0.60	36	160	14	1.5
16		10		0.40	0.60	45	180	16	3.2
18		11	0 −0.110	0.40	0.60	50	200	18	3.2
20	0 −0.052	12		0.60	0.80	56	220	20	3.2
22		14		0.60	0.80	63	250	22	3.2
25		14		0.60	0.80	70	280	22	3.2
28		16		0.60	0.80	80	320	25	3.2
32	0 −0.062	18		0.60	0.80	90	360	28	6.4
36		20	0 −0.130	1.00	1.20	100	400	32	6.4
40		22		1.00	1.20	…	…	36	6.4
45		25		1.00	1.20	…	…	40	6.4
50		28		1.00	1.20	…	…	45	6.4
56	0 −0.074	32	0 −0.160	1.60	2.00	…	…	50	9.5
63		32		1.60	2.00	…	…	50	9.5
70		36		1.60	2.00	…	…	56	9.5
80		40		2.50	3.00	…	…	63	9.5
90	0 −0.087	45		2.50	3.00	…	…	70	9.5
100		50		2.50	3.00	…	…	80	9.5

[a] The tolerance on the width and thickness of square taper keys is h9, and on the width and thickness of rectangular taper keys, h9 and h11 respectively, in accordance with ISO metric limits and fits. Does not apply to gib head dimensions.

British Standard Keys and Keyways: Tables 1 through 6 from BS 46:Part 1:1958 (1985) (obsolescent) provide data for rectangular parallel keys and keyways, square parallel keys and keyways, plain and gib head rectangular taper keys and key-ways, plain and gib head square taper keys and keyways, and Woodruff keys and keyways.

Parallel Keys: These keys are used for transmitting unidirectional torques in transmissions not subject to heavy starting loads and where periodic withdrawal or sliding of the hub member may be required. In many instances, particularly couplings, a gib-head cannot be accommodated, and there is insufficient room to drift out the key from behind. It is then necessary to withdraw the component over the key and a parallel key is essential. Parallel square and rectangular keys are normally side fitting with top clearance and are usually retained in the shaft rather more securely than in the hub. The rectangular key is the general-purpose key for shafts greater than 1 inch in diameter; the square key is intended for use with shafts up to and including 1-inch diameter or for shafts up to 6-inch diameter where it is desirable to have a greater key depth than is provided by rectangular keys. In stepped shafts, the larger diameters are usually required by considerations other than torque, e.g., resistance to bending. Where components such as fans, gears, impellers, etc., are attached to the larger shaft diameter, the use of a key smaller than standard for that diameter may be permissible. As this results in unequal disposition of the key in the shaft and its related hub, the dimensions *H* and *h* must be recalculated to maintain the *T*/2 relationship.

British Standard Preferred Lengths of Metric Keys *BS 4235:Part 1:1972 (1986)*

Length	Type of key				Length	Type of key			
	Sq.	Rect.	Sq. Taper	Rect. Taper		Sq.	Rect.	Sq. Taper	Rect. Taper
6	x		x		63	x	x	x	x
8	x		x		70	x	x	x	x
10	x		x		80		x		x
12	x		x		90		x		x
14	x		x		100		x		x
16	x		x		110		x		x
18	x	x	x	x	125		x		x
20	x	x	x	x	140		x		x
22	x	x	x	x	160		x		x
25	x	x	x	x	180		x		x
28	x	x	x	x	200		x		x
32	x	x	x	x	220		x		x
36	x	x	x	x	250		x		x
40	x	x	x	x	280		x		x
45	x	x	x	x	320		x		x
50	x	x	x	x	360		x		x
56	x	x	x	x	400		x		x

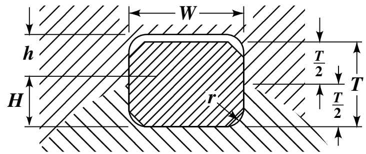
Taper Keys: These keys are used for transmitting heavy unidirectional, reversing, or vibrating torques and in applications where periodic withdrawal of the key may be necessary. Taper keys are usually top fitting, but may be top and side fitting where required, and the keyway in the hub should then have the same width value as the keyway in the shaft. Taper keys of rectangular section are used for general purposes and are of less depth than square keys; square sections are for use with shafts up to and including 1-inch diameter or for shafts up to 6-inch diameter where it is desirable to have greater key depth.

Woodruff Keys: These keys are used for light applications or the angular location of associated parts on tapered shaft ends. They are not recommended for other applications, but if so used, corner radii in the shaft and hub keyways are advisable to reduce stress concentration.

Dimensions and Tolerances for British Parallel and Taper Keys and Keyways: Dimensions and tolerances for key and keyway widths given in Tables 1, 2, 3, and 4 are based on the width of key *W* and provide a fitting allowance. The fitting allowance is designed to permit an interference between the key and the shaft keyway and a slightly easier condition between the key and the hub keyway. In shrink and heavy force fits, it may be found necessary to depart from the width and depth tolerances specified. Any variation in the width of the keyway should be such that the greatest width is at the end from which the key enters and any variation in the depth of the keyway should be such that the greatest depth is at the end from which the key enters.

Keys and keybar normally are not chamfered or radiused as supplied, but this may be done at the time of fitting. Radii and chamfers are given in Tables 1, 2, 3, and 4. Corner radii are recommended for keyways to alleviate stress concentration.

Table 1. British Standard Rectangular Parallel Keys, Keyways, and Keybars *B.S. 46: Part I: 1958*



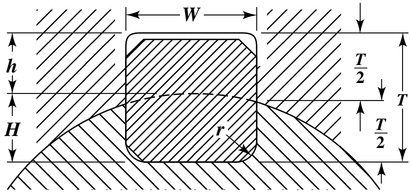
Diameter of Shaft		Key					Keyway in Shaft				Keyway in Hub				Nominal Keyway Radius, r[a]	Keybar			
Over	Up to and Including	Size $W \times T$	Width, W		Thickness, T		Width W_s		Depth H		Width W_h		Depth h			Width W		Thickness T	
			Max.	Min.	Max.	Min.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.		Max.	Min.	Max.	Min.
1	1¼	5/16 × ¼	0.314	0.312	0.253	0.250	0.311	0.312	0.146	0.152	0.312	0.313	0.112	0.118	0.010	0.314	0.312	0.253	0.250
1¼	1½	⅜ × ¼	0.377	0.375	0.253	0.250	0.374	0.375	0.150	0.156	0.375	0.376	0.108	0.114	0.010	0.377	0.375	0.253	0.250
1½	1¾	7/16 × 5/16	0.440	0.438	0.315	0.312	0.437	0.438	0.186	0.192	0.438	0.439	0.135	0.141	0.020	0.440	0.438	0.315	0.312
1¾	2	½ × 5/16	0.502	0.500	0.315	0.312	0.499	0.500	0.190	0.196	0.500	0.501	0.131	0.137	0.020	0.502	0.500	0.315	0.312
2	2½	⅝ × 7/16	0.627	0.625	0.441	0.438	0.624	0.625	0.260	0.266	0.625	0.626	0.185	0.191	0.020	0.627	0.625	0.441	0.438
2½	3	¾ × ½	0.752	0.750	0.503	0.500	0.749	0.750	0.299	0.305	0.750	0.751	0.209	0.215	0.020	0.752	0.750	0.503	0.500
3	3½	⅞ × ⅝	0.877	0.875	0.629	0.625	0.874	0.875	0.370	0.376	0.875	0.876	0.264	0.270	0.062	0.877	0.875	0.629	0.625
3½	4	1 × ¾	1.003	1.000	0.754	0.750	0.999	1.000	0.441	0.447	1.000	1.001	0.318	0.324	0.062	1.003	1.000	0.754	0.750
4	5	1¼ × ⅞	1.253	1.250	0.879	0.875	1.248	1.250	0.518	0.524	1.250	1.252	0.366	0.372	0.062	1.253	1.250	0.879	0.875
5	6	1½ × 1	1.504	1.500	1.006	1.000	1.498	1.500	0.599	0.605	1.500	1.502	0.412	0.418	0.062	1.504	1.500	1.006	1.000
6	7	1½ × 1¼	1.754	1.750	1.256	1.250	1.748	1.750	1.740	0.746	1.750	1.752	0.526	0.532	0.125				
7	8	2 × 1⅜	2.005	2.000	1.381	1.375	1.998	2.000	0.818	0.824	2.000	2.002	0.573	0.579	0.125				
8	9	2¼ × 1½	2.255	2.250	1.506	1.500	2.248	2.250	0.897	0.905	2.250	2.252	0.619	0.627	0.125				
9	10	2½ × 1⅝	2.505	2.500	1.631	1.625	2.498	2.500	0.975	0.983	2.500	2.502	0.666	0.674	0.187				
10	11	2¾ × 1⅞	2.755	2.750	1.881	1.875	2.748	2.750	1.114	1.122	2.750	2.752	0.777	0.785	0.187				
11	12	3 × 2	3.006	3.000	2.008	2.000	2.998	3.000	1.195	1.203	3.000	3.002	0.823	0.831	0.187				
12	13	3¼ × 2⅛	3.256	3.250	2.133	2.125	3.248	3.250	1.273	1.281	3.250	3.252	0.870	0.878	0.187				
13	14	3½ × 2⅜	3.506	3.500	2.383	2.375	3.498	3.500	1.413	1.421	3.500	3.502	0.980	0.988	0.250				
14	15	3¾ × 2½	3.756	3.750	2.508	2.500	3.748	3.750	1.492	1.502	3.750	3.752	1.026	1.036	0.250				
15	16	4 × 2⅝	4.008	4.000	2.633	2.625	3.998	4.000	1.571	1.581	4.000	4.002	1.072	1.082	0.250				
16	17	4¼ × 2⅞	4.258	4.250	2.883	2.875	4.248	4.250	1.711	1.721	4.250	4.252	1.182	1.192	0.312				
17	18	4½ × 3	4.508	4.500	3.010	3.000	4.498	4.500	1.791	1.801	4.500	4.502	1.229	1.239	0.312				
18	19	4¼ × 3⅛	4.758	4.750	3.135	3.125	4.748	4.750	1.868	1.878	4.750	4.752	1.277	1.287	0.312				
19	20	5 × 3⅜	5.008	5.000	3.385	3.375	4.998	5.000	2.010	2.020	5.000	5.002	1.385	1.395	0.312				

Bright keybar is not normally available in sections larger than the above.

All dimensions in inches.

[a] The key chamfer shall be the minimum to clear the keyway radius. Nominal values are given.

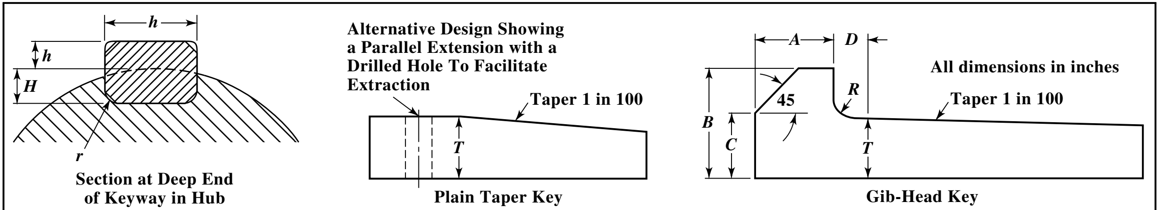
Table 2. British Standard Square Parallel Keys, Keyways, and Keybars *B.S. 46: Part I: 1958*



Diameter of Shaft		Key			Keyway in Shaft				Keyway in Hub				Nominal Keyway Radius, r[a]	Bright Keybar	
Over	Up to and Including	Size, $W \times T$	Width, W and Thickness, T		Width, W_s		Depth, H		Width, W_h		Depth, h			Width, W and Thickness, T	
			Max.	Min.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.		Max.	Min.
1/4	1/2	1/8 × 1/8	0.127	0.125	0.124	0.125	0.072	0.078	0.125	0.126	0.060	0.066	0.010	0.127	0.125
1/2	3/4	3/16 × 3/16	0.190	0.188	0.187	0.188	0.107	0.113	0.188	0.189	0.088	0.094	0.010	0.190	0.188
3/4	1	1/4 × 1/4	0.252	0.250	0.249	0.250	0.142	0.148	0.250	0.251	0.115	0.121	0.010	0.252	0.250
1	1 1/4	5/16 × 5/16	0.314	0.312	0.311	0.312	0.177	0.183	0.312	0.313	0.142	0.148	0.010	0.314	0.312
1 1/4	1 1/2	3/8 × 3/8	0.377	0.375	0.374	0.375	0.213	0.219	0.375	0.376	0.169	0.175	0.010	0.377	0.375
1 1/2	1 3/4	7/16 × 7/16	0.440	0.438	0.437	0.438	0.248	0.254	0.438	0.439	0.197	0.203	0.020	0.440	0.438
1 3/4	2	1/2 × 1/2	0.502	0.500	0.499	0.500	0.283	0.289	0.500	0.501	0.224	0.230	0.020	0.502	0.500
2	2 1/2	5/8 × 5/8	0.627	0.625	0.624	0.625	0.354	0.360	0.625	0.626	0.278	0.284	0.020	0.627	0.625
2 1/2	3	3/4 × 3/4	0.752	0.750	0.749	0.750	0.424	0.430	0.750	0.751	0.333	0.339	0.020	0.752	0.750
3	3 1/2	7/8 × 7/8	0.877	0.875	0.874	0.875	0.495	0.501	0.875	0.876	0.387	0.393	0.062	0.877	0.875
3 1/2	4	1 × 1	1.003	1.000	0.999	1.000	0.566	0.572	1.000	1.001	0.442	0.448	0.062	1.003	1.000
4	5	1 1/4 × 1 1/4	1.253	1.250	1.248	1.250	0.707	0.713	1.250	1.252	0.551	0.557	0.062	1.253	1.250
5	6	1 1/2 × 1 1/2	1.504	1.500	1.498	1.500	0.848	0.854	1.500	1.502	0.661	0.667	0.062	1.504	1.500

[a] The key chamfer shall be the minimum to clear the keyway radius. Nominal values are given. All dimensions in inches.

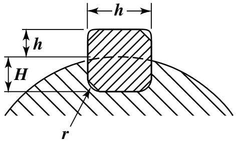
Table 3. British Standard Rectangular Taper Keys and Keyways, Gib-head and Plain *B.S. 46: Part 1: 1958*



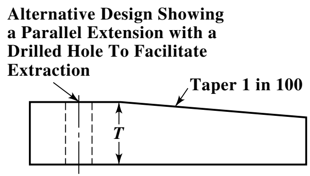
Diameter of Shaft		Key					Keyway in Shaft and Hub								Nominal Keyway Radius, r[a]	Gib-head[b]				Radius, R
Over	Up to and Including	Size, $W \times T$	Width, W		Thickness, T		Keyway in Shaft Width, W_s		Keyway in Hub Width, W_h		Depth in Shaft, H		Depth in Hub at Deep End of Keyway, h			A	B	C	D	
			Max.	Min.	Max.	Min.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.						
1	1 1/4	5/16 × 1/4	0.314	0.312	0.254	0.249	0.311	0.312	0.312	0.313	0.146	0.152	0.090	0.096	0.010	3/8	7/16	1/4	0.3	1/16
1 1/4	1 1/2	3/8 × 1/4	0.377	0.375	0.254	0.249	0.374	0.375	0.375	0.376	0.150	0.156	0.086	0.092	0.010	7/16	7/16	9/32	0.3	1/16
1 1/2	1 3/4	7/16 × 5/16	0.440	0.438	0.316	0.311	0.437	0.438	0.438	0.439	0.186	0.192	0.112	0.118	0.020	1/2	9/16	5/16	0.4	1/16
1 3/4	2	1/2 × 5/16	0.502	0.500	0.316	0.311	0.499	0.500	0.500	0.0.501	0.190	0.196	0.108	0.114	0.020	9/16	5/8	3/8	0.4	1/16
2	2 1/2	5/8 × 7/16	0.627	0.625	0.442	0.437	0.624	0.625	0.625	0.626	0.260	0.266	0.162	0.168	0.020	11/16	3/4	7/16	0.5	1/8
2 1/2	3	3/4 × 1/2	0.752	0.750	0.504	0.499	0.749	0.750	0.750	0.751	0.299	0.305	0.185	0.191	0.020	13/16	7/8	17/32	0.5	1/8
3	3 1/2	7/8 × 5/8	0.877	0.875	0.630	0.624	0.874	0.875	0.875	0.876	0.370	0.376	0.239	0.245	0.062	15/16	1	21/32	0.6	1/8
3 1/2	4	1 × 3/4	1.003	1.000	0.755	0.749	0.999	1.000	1.000	1.001	0.441	0.447	0.293	0.299	0.062	1 1/16	1 1/4	23/32	0.6	1/8
4	5	1 1/4 × 7/8	1.253	1.250	0.880	0.874	1.248	1.250	1.250	1252	0.518	0.524	0.340	0.346	0.062	1 5/16	1 1/2	27/32	0.7	1/4
5	6	1 1/2 × 1	1.504	1.500	1.007	0.999	1.498	1.500	1.500	1.502	0.599	0.605	0.384	0.390	0.062	1 9/16	1 5/8	1 1/32	0.7	1/4
6	7	1 3/4 × 1 1/4	1.754	1.750	1.257	1.249	1.748	1.750	1.750	1.752	0.740	0.746	0.493	0.499	0.125	1 13/16	2	1 7/32	0.8	1/4
7	8	2 × 1 3/8	2.005	2.000	1.382	1.374	1.998	2.000	2.000	2.002	0.818	0.824	0.539	0.545	0.125	2 1/16	2 1/4	1 13/32	0.8	1/4
8	9	2 1/4 × 1 1/2	2.255	2.250	1.509	1.499	2.248	2.250	2.250	2.252	0.897	0.905	0.581	0.589	0.125	2 5/16	2 1/2	1 9/16	0.9	3/8
9	10	2 1/2 × 1 5/8	2.505	2.500	1.634	1.624	2.498	2.500	2.500	2.502	0.975	0.983	0.628	0.636	0.187	2 9/16	2 3/4	1 11/16	0.9	3/8
10	11	2 3/4 × 1 7/8	2.755	2.750	1.884	1.874	2.748	2.750	2.750	2.752	1.114	1.122	0.738	0.746	0.187	2 13/16	3	1 15/16	1.0	3/8
11	12	3 × 2	3.006	3.000	2.014	1.999	2.998	3.000	3.000	3.002	1.195	1.203	0.782	0.790	0.187	3 1/16	3 1/4	2 1/16	1.0	3/8

[a] The key chamfer shall be the minimum to clear the keyway radius. Nominal values shall be given.
[b] Dimensions A, B, C, D, and R pertain to gib-head keys only.
All dimensions in inches.

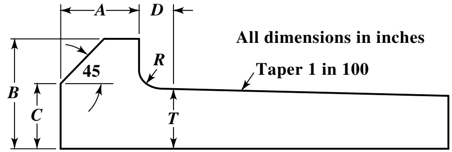
Table 4. British Standard Square Taper Keys and Keyways, Gib-head or Plain *B.S. 46: Part I: 1958*



Section at Deep End of Keyway in Hub



Plain Taper Key



Gib-Head Key

Diameter of Shaft		Key					Keyway in Shaft and Hub										Gib-head[b]				
Over	Up to and Including	Size $W \times T$	Width, W		Thickness, T		Keyway in Shaft Width W_s		Keyway in Hub Width W_h		Depth in Shaft, H		Depth in Hub at Deep End of Keyway, h		Nominal Keyway Radius, r[a]	A	B	C	D	Radius, R
			Max.	Min.	Max.	Min.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.						
1/4	1/2	1/8 × 1/8	0.127	0.125	0.129	0.124	0.124	0.125	0.125	0.126	0.072	0.078	0.039	0.045	0.010	3/16	1/4	5/32	0.1	1/32
1/2	3/4	3/16 × 3/16	0.190	0.188	0.192	0.187	0.187	0.188	0.188	0.189	0.107	0.113	0.067	0.073	0.010	1/4	3/8	7/32	0.2	1/32
3/4	1	1/4 × 1/4	0.252	0.250	0.254	0.249	0.249	0.250	0.250	0.251	0.142	0.148	0.094	0.100	0.010	5/16	7/16	9/32	0.2	1/16
1	1 1/4	5/16 × 5/16	0.314	0.312	0.316	0.311	0.311	0.312	0.312	0.313	0.177	0.183	0.121	0.127	0.010	3/8	9/16	11/32	0.3	1/16
1 1/4	1 1/2	3/8 × 3/8	0.377	0.375	0.379	0.374	0.374	0.375	0.375	0.376	0.213	0.219	0.148	0.154	0.010	7/16	5/8	13/32	0.3	1/16
1 1/2	1 3/4	7/16 × 7/16	0.440	0.438	0.442	0.437	0.437	0.438	0.438	0.439	0.248	0.254	0.175	0.181	0.020	1/2	3/4	15/32	0.4	1/16
1 3/4	2	1/2 × 1/2	0.502	0.500	0.504	0.499	0.499	0.500	0.500	0.501	0.283	0.289	0.202	0.208	0.020	9/16	7/8	17/32	0.4	1/16
2	2 1/2	5/8 × 5/8	0.627	0.625	0.630	0.624	0.624	0.625	0.625	0.626	0.354	0.360	0.256	0.262	0.020	11/16	1	21/32	0.5	1/8
2 1/2	3	3/7 × 3/4	0.752	0.750	0.755	0.749	0.749	0.750	0.750	0.751	0.424	0.430	0.310	0.316	0.020	13/16	1 1/4	25/32	0.5	1/8
3	3 1/2	7/8 × 7/8	0.877	0.875	0.880	0.874	0.874	0.875	0.875	0.876	0.495	0.501	0.364	0.370	0.062	15/16	1 3/8	20/32	0.6	1/8
3 1/2	4	1 × 1	1.003	1.000	1.007	0.999	0.999	1.000	1.000	1.001	0.566	0.572	0.418	0.424	0.062	1 1/16	1 5/8	1 1/32	0.6	1/8
4	5	1 1/4 × 1 1/4	1.253	1.250	1.257	1.249	1.248	1.250	1.250	1.252	0.707	0.713	0.526	0.532	0.062	1 5/16	2	1 9/32	0.7	1/4
5	6	1 1/2 × 1 1/2	1.504	1.500	1.509	1.499	1.498	1.500	1.500	1.502	0.848	0.854	0.635	0.641	0.062	1 9/16	2 1/2	1 17/32	0.7	1/4

[a] The key chamfer shall be the minimum to clear the keyway radius. Nominal values shall be given.

[b] Dimensions A, B, C, D, and R pertain to gib-head keys only.

All dimensions in inches.

Dimensions and Tolerances of British Woodruff Keys and Keyways.—Dimensions and tolerances are shown in Table 5. An optional alternative design of the Woodruff key that differs from the normal form in its depth is given in the illustration accompanying the table. The method of designating British Woodruff Keys is the same as the American method explained in the footnote on page 2348.

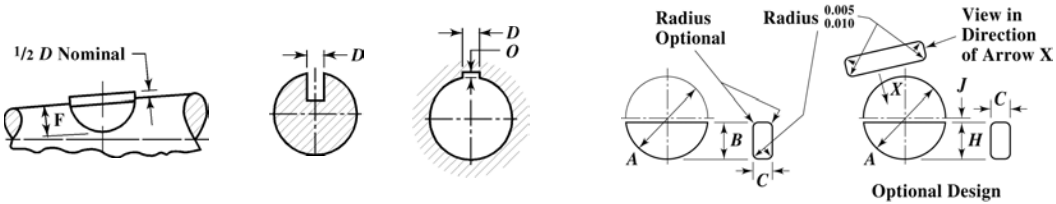


Table 5. British Standard Woodruff Keys and Keyways *BS 6: Part 1: 1958*

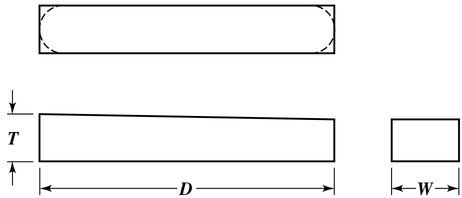
Key and Cutter No.	Key								Keyway								Optional Design		
	Nominal Fractional Size		Diameter *A*		Depth *B*		Thickness *C*		Width in Shaft, *D*		Width in Hub, *E*		Depth in Shaft, *F*		Depth in Hub at Center Line, *G*		Depth of Key, *H*		Dimension, *J*
	Width.	Dia.	Max.	Min.	Max.	Min.	Max.	Min.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Max.	Min.	Nom.
203	1/16	3/8	0.375	0.370	0.171	0.166	0.063	0.062	0.061	0.063	0.063	0.065	0.135	0.140	0.042	0.047	0.162	0.156	1/64
303	3/32	3/8	0.375	0.370	0.171	0.166	0.095	0.094	0.093	0.095	0.095	0.097	0.119	0.124	0.057	0.062	0.162	0.156	1/64
403	1/8	3/8	0.375	0.370	0.171	0.166	0.126	0.125	0.124	0.126	0.126	0.128	0.104	0.109	0.073	0.078	0.162	0.156	1/64
204	1/16	1/2	0.500	0.490	0.203	0.198	0.063	0.062	0.061	0.063	0.063	0.065	0.167	0.172	0.042	0.047	0.194	0.188	3/64
304	3/32	1/2	0.500	0.490	0.203	0.198	0.095	0.094	0.093	0.095	0.095	0.097	0.151	0.156	0.057	0.062	0.194	0.188	3/64
404	1/8	1/2	0.500	0.490	0.203	0.198	0.126	0.125	0.124	0.126	0.126	0.128	0.136	0.141	0.073	0.078	0.194	0.188	3/64
305	3/32	5/8	0.625	0.615	0.250	0.245	0.095	0.094	0.093	0.095	0.095	0.097	0.198	0.203	0.057	0.062	0.240	0.234	1/16
405	1/8	5/8	0.625	0.615	0.250	0.245	0.126	0.125	0.124	0.126	0.126	0.128	0.182	0.187	0.073	0.078	0.240	0.234	1/16
505	5/32	5/8	0.625	0.615	0.250	0.245	0.157	0.156	0.155	0.157	0.157	0.159	0.167	0.172	0.089	0.094	0.240	0.234	1/16
406	1/8	3/4	0.750	0.740	0.313	0.308	0.126	0.125	0.124	0.126	0.126	0.128	0.246	0.251	0.073	0.078	0.303	0.297	1/16

Table 5. *(Continued)* **British Standard Woodruff Keys and Keyways** *BS 6: Part 1: 1958*

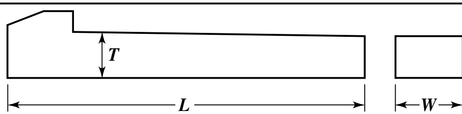
Key and Cutter No.	Key								Keyway								Optional Design		
	Nominal Fractional Size		Diameter *A*		Depth *B*		Thickness *C*		Width in Shaft, *D*		Width in Hub, *E*		Depth in Shaft, *F*		Depth in Hub at Center Line, *G*		Depth of Key, *H*		Dimension, *J*
	Width.	Dia.	Max.	Min.	Max.	Min.	Max.	Min.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Max.	Min.	Nom.
506	5/32	3/4	0.750	0.740	0.313	0.308	0.157	0.156	0.155	0.157	0.157	0.159	0.230	0.235	0.089	0.094	0.303	0.297	1/16
606	3/16	3/4	0.750	0.740	0.313	0.308	0.189	0.188	0.187	0.189	0.189	0.191	0.214	0.219	0.104	0.109	0.303	0.297	1/16
507	5/32	7/8	0.875	0.865	0.375	0.370	0.157	0.156	0.155	0.157	0.157	0.159	0.292	0.297	0.089	0.094	0.365	0.359	1/16
607	3/16	7/8	0.875	0.865	0.375	0.370	0.189	0.188	0.187	0.189	0.189	0.191	0.276	0.281	0.104	0.109	0.365	0.359	1/16
807	1/4	7/8	0.875	0.865	0.375	0.370	0.251	0.250	0.249	0.251	0.251	0.253	0.245	0.250	0.136	0.141	0.365	0.359	1/16
608	3/16	1	1.000	0.990	0.438	0.433	0.189	0.188	0.187	0.189	0.189	0.191	0.339	0.344	0.104	0.109	0.428	0.422	1/16
808	1/4	1	1.000	0.990	0.438	0.433	0.251	0.250	0.249	0.251	0.251	0.253	0.308	0.313	0.136	0.141	0.428	0.422	1/16
1008	5/16	1	1.000	0.990	0.438	0.433	0.313	0.312	0.311	0.313	0.313	0.315	0.277	0.282	0.167	0.172	0.428	0.422	1/16
609	3/16	1 1/8	1.125	1.115	0.484	0.479	0.189	0.188	0.187	0.189	0.189	0.191	0.385	0.390	0.104	0.109	0.475	0.469	5/64
809	1/4	1 1/8	1.125	1.115	0.484	0.479	0.251	0.250	0.249	0.251	0.251	0.253	0.354	0.359	0.136	0.141	0.475	0.469	5/64
1009	5/16	1 1/8	1.125	1.115	0.484	0.479	0.313	0.312	0.311	0.313	0.313	0.315	0.323	0.328	0.167	0.172	0.475	0.469	5/64
810	1/4	1 1/4	1.250	1.240	0.547	0.542	0.251	0.250	0.249	0.251	0.251	0.253	0.417	0.422	0.136	0.141	0.537	0.531	5/64
1010	5/16	1 1/4	1.250	1.240	0.547	0.542	0.313	0.312	0.311	0.313	0.313	0.315	0.386	0.391	0.167	0.172	0.537	0.531	5/64
1210	3/8	1 1/4	1.250	1.240	0.547	0.542	0.376	0.375	0.374	0.376	0.376	0.378	0.354	0.359	0.198	0.203	0.537	0.531	5/64
1011	5/16	1 3/8	1.375	1.365	0.594	0.589	0.313	0.312	0.311	0.313	0.313	0.315	0.433	0.438	0.167	0.172	0.584	0.578	3/32
1211	3/8	1 3/8	1.375	1.365	0.594	0.589	0.376	0.375	0.374	0.376	0.376	0.378	0.402	0.407	0.198	0.203	0.584	0.578	3/32
812	1/4	1 1/2	1.500	1.490	0.641	0.636	0.251	0.250	0.249	0.251	0.251	0.253	0.511	0.516	0.136	0.141	0.631	0.625	7/64
1012	5/16	1 1/2	1.500	1.490	0.641	0.636	0.313	0.312	0.311	0.313	0.313	0.315	0.480	0.485	0.167	0.172	0.631	0.625	7/64
1212	3/8	1 1/2	1.500	1.490	0.641	0.636	0.376	0.375	0.374	0.376	0.376	0.378	0.448	0.453	0.198	0.203	0.631	0.625	7/64

All dimensions are in inches.

Table 6. British Preferred Lengths of Plain (Parallel or Taper) and Gib-head Keys, Rectangular and Square Section *BS 46:Part 1:1958 (1985) Appendix*



Plain Key Size $W \times T$	Overall Length, L														
	3/4	1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 3/4	3	3 1/2	4	4 1/2	5	6
1/8 × 1/8	X	X													
3/16 × 3/16	X	X	X	X	X	X									
1/4 × 1/4	X	X	X	X	X	X	X	X	X	X					
5/16 × 1/4	X	X	X	X	X	X	X	X	X	X					
5/16 × 5/16	X	X	X	X	X	X	X	X	X	X					
3/8 × 1/4		X	X	X	X	X	X	X	X	X	X	X			
3/8 × 3/8		X	X	X	X	X	X	X	X	X	X	X			
7/16 × 5/16				X	X	X	X	X	X	X	X	X			
7/16 × 7/16					X	X	X	X	X	X	X	X			
1/2 × 5/16					X	X	X	X	X	X	X	X	X	X	
1/2 × 1/2						X	X	X	X	X	X	X	X	X	
5/8 × 7/16								X	X	X	X	X	X	X	
5/8 × 5/8								X		X	X	X	X	X	X
3/4 × 1/2										X	X	X	X	X	X
3/4 × 3/4											X	X	X	X	X
7/8 × 5/8											X	X	X	X	X



Gib-head Key Size, $W \times T$	Overall Length, L																
	1 1/2	1 3/4	2	2 1/4	2 1/2	2 3/4	3	3 1/2	4	4 1/2	5	5 1/2	6	6 1/2	7	7 1/2	8
3/16 × 3/16	X	X	X	X	X		X										
1/4 × 1/4	X	X	X	X	X	X	X	X	X								
5/16 × 1/4			X	X	X	X	X	X	X								
5/16 × 5/16			X	X	X	X	X	X	X	X							
3/8 × 1/4			X	X	X	X	X	X	X	X	X						
3/8 × 3/8			X	X	X	X	X	X	X	X	X	X	X				
7/16 × 5/16					X		X	X	X	X	X	X	X				
7/16 × 7/16					X		X	X	X	X	X	X	X				
1/2 × 5/16					X		X	X	X	X	X	X	X	X			
1/2 × 1/2					X		X	X	X	X	X	X	X	X			
5/8 × 7/16							X		X	X	X	X	X	X	X		
5/8 × 5/8									X	X	X	X	X	X	X		X
3/4 × 1/2									X	X	X	X	X	X	X	X	X
3/4 × 3/4									X		X		X		X		X
7/8 × 5/8											X		X	X	X	X	X
7/8 × 7/8													X		X		X
1 × 3/4													X		X	X	X
1 × 1													X		X		X

All dimension are in inches

Flat Belting

Flat belting was originally made from leather because it was the most durable material available and could easily be cut and joined to make a driving belt suitable for use with cylindrical or domed pulleys. This type of belting was popular because it could be used to transmit high torques over long distances and it was employed in factories to drive many small machines from a large common power source such as a steam engine. As electric motors became smaller, more efficient, and more powerful, and new types of belts and chains were made possible by modern materials and manufacturing processes, flat belts fell out of favor. Flat belts are still used for some drive purposes, but leather has been replaced by other natural and synthetic materials such as urethanes, which can be reinforced by high-strength polyamide or steel fabrics to provide properties such as resistance to stretching. The high modulus of elasticity in these flat belts eliminates the need for periodic retensioning that is usually necessary with V-belts.

Driving belts can be given a coating of an elastomer with a high coefficient of friction, to enable belts to grip pulleys without the degree of tension common with earlier materials. Urethanes are commonly used for driving belts where high resistance to abrasion is required, and will also resist attack by chemical solvents of most kinds. Flat belts having good resistance to high temperatures are also available. Typical properties of polyurethane belts include tensile strength up to 40,000 psi, depending on reinforcement type and Shore hardness of 85 to 95. Most polyurethane belts are installed under tension. The amount of the tension varies with the belt cross-section, being greater for belts of small section. Belt tension can be measured by marking lines 10 in. apart on an installed belt, then applying tension until the separation increases by the desired percentage. For 2 per cent tension, the lines on the tensioned belt would be 10.2 in. apart. Mechanical failure may result when belt tensioning is excessive, and 2 to 2.5 per cent elongation should be regarded as the limit.

Flat belts offer high load capacities and are capable of transmitting power over long distances, maintaining relative rotational direction, can operate without lubricants, and are generally inexpensive to maintain or replace when worn. Flat belt systems will operate with little maintenance and only periodic adjustment. Because they transmit motion by friction, flat belts have the ability to slip under excessive loads, providing a fail-safe action to guard against malfunctions. This advantage is offset by the problem that friction drives can both slip and creep so that they do not offer exact, consistent velocity ratios nor precision timing between input and output shafts. Flat belts can be made to any desired length, being joined by reliable chemical bonding processes.

Increasing centrifugal force has less effect on the load-carrying capacity of flat belts at high speeds than it has on V-belts, for instance. The low thickness of a flat belt, compared with a V-belt, places its center of gravity near the pulley surface. Flat belts therefore may be run at surface speeds of up to 16,000 or even 20,000 ft/min (81.28 and 101.6 m/s), although ideal speeds are in the range of 3,000 to 10,000 ft/min (15.25 to 50.8 m/s). Elastomeric drive surfaces on flat belts have eliminated the need for belt dressings that were often needed to keep leather belts in place. These surface coatings can also contain antistatic materials. Belt pulley wear and noise are low with flat belts shock and vibration are damped, and efficiency is generally greater than 98 per cent compared with 96 per cent for V-belts.

Driving belt load capacities can be calculated from torque $T = F(d/2)$ and horsepower $HP = T \times rpm/396{,}000$, where T is the torque in in.-lb, F is the force transmitted in lb, and d is the pulley diameter in in. Pulley width is usually about 10 per cent larger than the belt, and for good tracking, pulleys are often crowned by 0.012 to 0.10 in. for diameters in the range of 1.5 to 80 in.

Before a belt specification is written, the system should be checked for excessive startup and shut-down loads, which sometimes are more than 10 per cent above operating conditions. In overcoming such loads, the belt will transmit considerably more force than during normal operation. Large starting and stopping forces will also shorten belt life unless they are taken into account during the design stage.

Belt speed plays an important role in the amount of load a friction drive system can transmit. Higher speeds will require higher preloads (increased belt tension) to compensate for the higher centrifugal force. In positive drive (toothed belt) systems, higher speeds generate dynamic forces caused by unavoidable tolerance errors that may result in increased tooth or pin stresses and shorter belt life.

Pulley Diameters and Drive Ratios: Minimum pulley diameters determined by belt manufacturers are based on the minimum radius that a belt can wrap around a pulley without stressing the load-carrying members. For positive drive systems, minimum pulley diameters are also determined by the minimum number of teeth that must be engaged with the sprocket to guarantee the operating load.

Diameters of driving and driven pulleys determine the velocity ratio of the input relative to the output shaft and are derived from the following formulas: for all belt systems, velocity ratio $V = D_{pi}/D_{po}$, and for positive (toothed) drive systems, velocity ratio $V = N_i/N_o$, where D_{pi} is the pitch diameter of the driving pulley, D_{po} is the pitch diameter of the driven pulley, N_i is the number of teeth on the driving pulley, and N_o is the number of teeth on the driven pulley. For most drive systems, a velocity ratio of 8:1 is the largest that should be attempted with a single reduction drive, and 6:1 is a reasonable maximum.

Wrap Angles and Center-to-Center Distances: The radial distance for which the belt is in contact with the pulley surface, or the number of teeth in engagement for positive drive belts, is called the wrap angle. Belt and sprocket combinations should be chosen to ensure a wrap angle of about 120° around the smaller pulley. The wrap angle should not be less than 90°, especially with positive drive belts, because if too few teeth are in engagement, the belt may jump a tooth or pin and timing or synchronization may be lost.

For flat belts, the minimum allowable center-to-center distance (CD) for any belt-and-sprocket combination should be chosen to ensure a minimum wrap angle around the smaller pulley. For high-velocity systems, a good rule of thumb is a minimum CD equal to the sum of the pitch diameter of the larger sprocket and one-half the pitch diameter of the smaller sprocket. This formula ensures a minimum wrap angle of approximately 120°, which is generally sufficient for friction drives and will ensure that positive drive belts do not jump teeth.

Pulley Center Distances and Belt Lengths: Maximum center distances of pulleys should be about 15 to 20 times the pitch diameter of the smaller pulley. Greater spacing requires tight control of the belt tension because a small amount of stretch will cause a large drop in tension. Constant belt tension can be obtained by application of an adjustable tensioning pulley applied to the slack side of the belt. Friction drive systems using flat belts require much more tension than positive drive belt systems.

Belt length can be calculated from: $L = 2C + \pi(D_2 + D_1)/2 + (D_2 - D_1)^2/4C$ for friction drives, and length $L = 2C + \pi(D_2 + D_1)/2 + (D_2 + D_1)^2/4C$ for crossed belt friction belt drives, where C is the center distance, D_1 is the pitch diameter of the small pulley, and D_2 is the pitch diameter of the large pulley. For serrated belt drives, the length determined by use of these equations should be divided by the serration pitch. The belt length must then be adjusted to provide a whole number of serrations.

Calculating Diameters and Speeds of Pulleys

Pulley Diameters and Speeds.— If D = diameter of driving pulley, d = diameter of driven pulley, S = speed of driving pulley, and s = speed of driven pulley:

$$D = \frac{d \times s}{S}, \qquad d = \frac{D \times S}{s}, \qquad S = \frac{d \times s}{D}, \qquad \text{and} \qquad s = \frac{D \times S}{d}$$

Example 1: If the diameter of the driving pulley D is 24 inches, its speed is 100 rpm, and the driven pulley is to run at 600 rpm, the diameter of the driven pulley, $d = 24 \times 100/600 = 4$ inches.

Example 2: If the diameter of the driven pulley d is 36 inches, its required speed is to be 150 rpm, and the speed of the driving pulley is to be 600 rpm, the diameter of the driving pulley $D = 36 \times 150/600 = 9$ inches.

Example 3: If the diameter of the driven pulley d is 4 inches, its required speed is 800 rpm, and the diameter of the driving pulley D is 26 inches, the speed of the driving pulley $= 4 \times 800/26 = 123$ rpm.

Example 4: If the diameter of the driving pulley D is 15 inches and its speed is 180 rpm, and the diameter of the driven pulley d is 9 inches, then the speed of the driven pulley = $15 \times 180/9 = 300$ rpm.

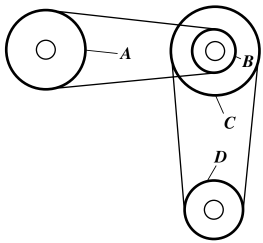
Pulley Diameters in Compound Drive.— If speeds of driving and driven pulleys, A, B, C, and D (see illustration) are known, the first step in finding their diameters is to form a fraction with the driving pulley speed as the numerator and the driven pulley speed as the, denominator, and then reduce this fraction to its lowest terms. Resolve the numerator and the denominator into two pairs of factors (a pair being one factor in the numerator and one in the denominator) and, if necessary, multiply each pair by a trial number that will give pulleys of suitable diameters.

Example: If the speed of pulley A is 260 rpm and the required speed of pulley D is 720 rpm, find the diameters of the four pulleys. Reduced to its lowest terms, the fraction 260/720 = 13/36, which represents the required speed ratio. Resolve this ratio 13/36 into two factors:

$$\frac{13}{36} = \frac{1 \times 13}{2 \times 18}$$

Multiply by trial numbers 12 and 1 to get:

$$\frac{(1 \times 12) \times (13 \times 1)}{(2 \times 12) \times (18 \times 1)} = \frac{12 \times 13}{24 \times 18}$$



Compound Drive with Four Pulleys.

The values 12 and 13 in the numerator represent the diameters of the *driven* pulleys, B and D, and the values 24 and 18 in the denominator represent the diameters of the *driving* pulleys, A and C, as shown in the illustration.

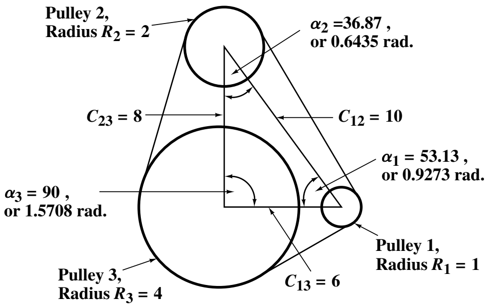
Speed of Driven Pulley in Compound Drive.—If diameters of pulleys *A*, *B*, *C*, and *D* (see illustration above), and speed of pulley *A* are known, the speed of the driven pulley *D* is found from:

$$\frac{\text{driving pulley diameter}}{\text{driven pulley diameter}} \times \frac{\text{driving pulley diameter}}{\text{driven pulley diameter}} \times \text{speed of first driving pulley}$$

Example: If the diameters of driving pulleys *A* and *C* are 18 and 24 inches, diameters of driven pulleys *B* and *D* are 12 and 13 inches, and the speed of driving pulley *A* is 260 rpm, speed of driven pulley

$$D = \frac{18 \times 24}{12 \times 13} \times 260 = 720 \text{ rpm}$$

Length of Belt Traversing Three Pulleys.—The length *L* of a belt traversing three pulleys, as shown in the diagram below, and touching them on one side only, can be found by the following formula.



Flat Belt Traversing Three Pulleys.

Referring to the diagram, R_1, R_2, and R_3 are the radii of the three pulleys; C_{12}, C_{13}, and C_{23} are the center distances; and α_1, α_2, and α_3 are the angles, in radians, of the triangle formed by the center distances. Then:

$$L = C_{12} + C_{13} + C_{23} + \frac{1}{2}\left[\frac{(R_2 - R_1)^2}{C_{12}} + \frac{(R_3 - R_1)^2}{C_{13}} + \frac{(R_3 - R_2)^2}{C_{23}}\right]$$
$$+\pi(R_1 + R_2 + R_3) - (\alpha_1 R_1 + \alpha_2 R_2 + \alpha_3 R_3)$$

For example: assume $R_1 = 1$, $R_2 = 2$, $R_3 = 4$, $C_{12} = 10$, $C_{13} = 6$, $C_{23} = 8$, $\alpha_1 = 53.13$ degrees or 0.9273 radian, $\alpha_2 = 36.87$ degrees or 0.6435 radian, and $\alpha_3 = 90$ degrees or 1.5708 radians. Then:

$$L = 10 + 6 + 8 + \frac{1}{2}\left[\frac{(2-1)^2}{10} + \frac{(4-1)^2}{6} + \frac{(4-2)^2}{8}\right]$$
$$+\pi(1 + 2 + 4) - 0.9273 \times 1 + 0.6435 \times 2 + 1.5708 \times 4$$
$$= 24 + 1.05 + 21.9911 - 8.4975 = 38.5436$$

FLEXIBLE BELTS AND SHEAVES

Flexible belt drives are used in industrial power transmission applications, especially when the speeds of the driver and driven shafts must be different or when shafts must be widely separated. The trend toward higher speed prime movers and the need to achieve a slower, useful driven speed are additional factors favoring the use of belts. Belts have numerous advantages over other means of power transmission; these advantages include overall economy, cleanliness, no need for lubrication, lower maintenance costs, easy installation, dampening of shock loads, and the abilities to be used for clutching and variable speed power transmission between widely spaced shafts.

Power Transmitted By Belts.—With belt drives, the force that produces work acts on the rim of a pulley or sheave and causes it to rotate. Since a belt on a drive must be tight enough to prevent slip, there is a belt pull on both sides of a driven wheel. When a drive is stationary or operating with no power transmitted, the pulls on both sides of the driven wheel are equal. When the drive is transmitting power, however, the pulls are not the same. There is a tight side tension T_T and a slack side tension, T_S. The difference between these two pulls ($T_T - T_S$) is called *effective pull* or *net pull*. This effective pull is applied at the rim of the pulley and is the force that produces work.

Net pull equals horsepower (HP) × 33,000 ÷ belt speed (fpm). Belt speed in fpm can be set by changing the pulley, sprocket, or sheave diameter. The shaft speeds remain the same. Belt speed is directly related to pulley diameter. Double the diameter and the total belt pull is cut in half, reducing the load on the shafts and bearings.

A belt experiences three types of tension as it rotates around a pulley: working tension (tight side – slack side), bending tension, and centrifugal tension.

The *tension ratio* (R) equals tight side divided by slack side tension (measured in pounds). The larger R is, the closer a V-belt is to slipping—the belt is too loose. (Synchronous belts do not slip, because they depend on the tooth grip principle.)

In addition to working tension (tight side – slack side), two other tensions are developed in a belt when it is operating on a drive. *Bending tension* T_B occurs when the belt bends around the pulley. One part of the belt is in tension and the other is in compression, so compressive stresses also occur. The amount of tension depends on the belt's construction and the pulley diameter. *Centrifugal tension* (T_C) occurs as the belt rotates around the drive and is calculated by $T_C = MV^2$, where T_C is centrifugal tension in pounds, M is a constant dependent on the belt's weight, and V is the belt velocity in feet per minute. Neither the bending nor centrifugal tensions are imposed on the pulley, shaft, or bearing—only on the belt.

Combining these three types of tension results in *peak tension* which is important in determining the degree of performance or belt life: $T_{peak} = T_T + T_B + T_C$.

Measuring the Effective Length.—The effective length of a V-belt is determined by placing the belt on a measuring device having two equal diameter sheaves with standard groove dimensions. The shaft of one of the sheaves is fixed. A specified measuring tension is applied to the housing for the shaft of the other sheave, moving it along a graduated scale. The belt is rotated around the sheaves at least two revolutions of the belt to seat it properly in the sheave grooves and to divide the total tension equally between the two strands of the belt.

The effective length of the belt is obtained by adding the effective (outside) circumference of one of the measuring sheaves to twice the center distance. Synchronous belts are measured in a similar manner.

The following sections cover common belts used in industrial applications for power transmission and specified in Rubber Manufacturers Association (RMA), Mechanical Power Transmission Association (MPTA), and The Rubber Association of Canada (RAC) standards. The information presented does not apply to automotive or agricultural drives, for which other standards exist. The belts covered in this section are Narrow, Classical, Double, and Light-Duty V-Belts, V-Ribbed Belts, Variable-Speed Belts, 60 deg V-Belts, and Synchronous (Timing) Belts.

Narrow V-Belts ANSI/RMA IP-22.—Narrow V-belts serve the same applications as multiple, classical V-belts, but allow for a lighter, more compact drive. Three basic cross sections—3V and 3VX, 5V and 5VX, and 8V—are provided, as shown in Fig. 1. The 3VX and 5VX are molded, notched V-belts that have greater power capacity than conventional belts. Narrow V-belts are specified by cross section and effective length and have top widths ranging from $\frac{3}{8}$ to 1 in.

Narrow V-belts usually provide substantial weight and space savings over classical belts. Some narrow belts can transmit up to three times the horsepower of conventional belts in the same drive space, or the same horsepower in one-third to one-half the space. These belts are designed to operate in multiples and are also available in the joined configuration.

Belt Cross Sections: Nominal dimensions of the three cross sections are given in Fig. 1.

Belt Size Designation: Narrow V-belt sizes are identified by a standard belt number. The first figure of this number followed by the letter V denotes the belt cross section. An X following the V indicates a notched cross section. The remaining figures show the effective belt length in tenths of an inch. For example, the number 5VX1400 designates a notched V-belt with a 5V cross section and an effective length of 140.0 in. Standard effective lengths of narrow V-belts are shown in Table 1.

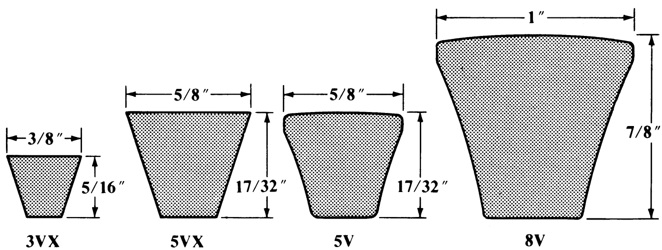


Fig. 1. Nominal Narrow V-Belt Dimensions

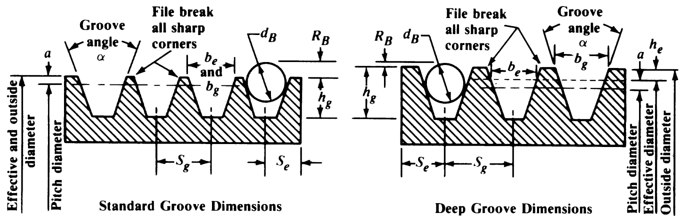
Table 1. Narrow V-Belt Standard Effective Lengths *ANSI/RMA IP-22 (1983)*

Standard Length Designation[a]	Standard Effective Outside Length			Permissible Deviation from Standard Length	Matching Limits for One Set	Standard Length Designation[a]	Standard Effective Outside Length			Permissible Deviation from Standard Length	Matching Limits for One Set
	Cross Section						Cross Section				
	3V	5V	8V				3V	5V	8V		
250	25.0	…	…	±0.3	0.15	1060	106.0	106.0	106.0	±0.6	0.30
265	26.5	…	…	±0.3	0.15	1120	112.0	112.0	112.0	±0.6	0.30
280	28.0	…	…	±0.3	0.15	1180	118.0	118.0	118.0	±0.6	0.30
300	30.0	…	…	±0.3	0.15	1250	125.0	125.0	125.0	±0.6	0.30
315	31.5	…	…	±0.3	0.15	1320	132.0	132.0	132.0	±0.6	0.30
335	33.5	…	…	±0.3	0.15	1400	140.0	140.0	140.0	±0.6	0.30
355	35.5	…	…	±0.3	0.15	1500	…	150.0	150.0	±0.8	0.30
375	37.5	…	…	±0.3	0.15	1600	…	160.0	160.0	±0.8	0.45
400	40.0	…	…	±0.3	0.15	1700	…	170.0	170.0	±0.8	0.45
425	42.5	…	…	±0.3	0.15	1800	…	180.0	180.0	±0.8	0.45
450	45.0	…	…	±0.3	0.15	1900	…	190.0	190.0	±0.8	0.45
475	47.5	…	…	±0.3	0.15	2000	…	200.0	200.0	±0.8	0.45
500	50.0	50.0	…	±0.3	0.15	2120	…	212.0	212.0	±0.8	0.45
530	53.0	53.0	…	±0.4	0.15	2240	…	224.0	224.0	±0.8	0.45
560	56.0	56.0	…	±0.4	0.15	2360	…	236.0	236.0	±0.8	0.45
600	60.0	60.0	…	±0.4	0.15	2500	…	250.0	250.0	±0.8	0.45
630	63.0	63.0	…	±0.4	0.15	2650	…	265.0	265.0	±0.8	0.60
670	67.0	67.0	…	±0.4	0.30	2800	…	280.0	280.0	±0.8	0.60
710	71.0	71.0	…	±0.4	0.30	3000	…	300.0	300.0	±0.8	0.60
750	75.0	75.0	…	±0.4	0.30	3150	…	315.0	315.0	±1.0	0.60
800	80.0	80.0	…	±0.4	0.30	3350	…	335.0	335.0	±1.0	0.60
850	85.0	85.0	…	±0.5	0.30	3550	…	355.0	355.0	±1.0	0.60
900	90.0	90.0	…	±0.5	0.30	3750	…	…	375.0	±1.0	0.60
950	95.0	95.0	…	±0.5	0.30	4000	…	…	400.0	±1.0	0.75
1000	100.0	100.0	100.0	±0.5	0.30	4250	…	…	425.0	±1.2	0.75

[a] To specify belt size, use the Standard Length Designation prefixed by the cross section, for example, 5 V850.

All dimensions in inches.

Table 2. Narrow V-Belt Standard Sheave and Groove Dimensions *ANSI/RMA IP-22 (1983)*



Face Width of Standard and Deep Groove Sheaves $= s_g(N_g - 1) + 2S_e$, where N_g = number of grooves

Cross Section	Standard Groove Outside Diameter	Standard Groove Dimensions								Design Factors	
		Groove Angle, α, ±0.25 deg	b_g ±0.005	b_e (Ref)	h_g (Min)	R_B (Min)	d_B ±0.0005	S_g[a] ±0.015	S_e	Min Recommended OD	$2a$
3V	Up through 3.49	36				0.181					
	Over 6.00 up to and including 12.00	40	0.350	0.350	0.340	0.186	0.3438	0.406	0.344 (+ 0.099, −0.031)	2.65	0.050
	Over 9.99 up to and including 16.00	40	0.600	0.600	0.590	0.332	0.5938	0.688	0.500 +0.125, −0.047)	7.10	0.100
	Over 16.00	42				0.336					
8V	Up through 15.99	38				0.575					
	Over 15.99 up to and including 22.40	40	1.000	1.000	0.990	0.580	1.0000	1.125	0.750 (+0.250, −0.062)	12.50	0.200
	Over 22.40	42				0.585					

[a] Summation of the deviations from S_g for all grooves in any one sheave should not exceed ±0.031 in. The variations in pitch diameter between the grooves in any one sheave must be within the following limits: Up through 19.9 in. outside diameter and up through 6 grooves—0.010 in. (add 0.0005 in. for each additional groove). 20.0 in. and over on outside diameter and up through 10 grooves—0.015 in. (add 0.0005 in. for each additional groove). This variation can be obtained by measuring the distance across two measuring balls or rods placed in the grooves diametrically opposite each other. Comparing this “diameter over balls or rods” measurement between grooves will give the variation in pitch diameter.

Table 2. *(Continued)* **Narrow V-Belt Standard Sheave and Groove Dimensions ANSI/RMA IP-22 (1983)**

Cross Section	Deep Groove Outside Diameter	Deep Groove Dimensions[a]								Design Factors		
		Groove Angle, α, ±0.25 deg	b_g ±0.005	b_e (Ref)	h_g (Min)	R_B (Min)	d_B ±0.0005	S_g[a] ±0.015	S_e	Min Recommended OD	$2a$	$2h_e$
3V	Up through 3.71	36	0.421			0.070						
	Over 3.71 up to and including 6.22	38	0.425			0.073						
	Over 6.22 up to and including 12.22	40	0.429	0.350	0.449	0.076	0.3438	0.500	0.375 (+0.094, −0.031)	2.87	0.050	0.218
	Over 12.22	42	0.434			0.078						
5V	Up through 10.31	38	0.710			0.168						
	Over 10.31 up to and including 16.32	40	0.716	0.600	0.750	0.172	0.5938	0.812	0.562 (+0.125, − 0.047)	7.42	0.100	0.320
	Over 16.32	42	0.723			0.175						
8V	Up through 16.51	38	1.180			0.312						
	Over 16.51 up to and including 22.92	40	1.191	1.000	1.252	0.316	1.0000	1.312	0.844 (+0.250, −0.062)	13.02	0.200	0.524
	Over 22.92	42	1.201			0.321						

[a] Deep groove sheaves are intended for drives with belt offset such as quarter-turn or vertical shaft drives. They may also be necessary where oscillations in the center distance may occur. Joined belts will not operate in deep groove sheaves.

Other Sheave Tolerances		
Outside Diameter	Radial Runout[a]	Axial Runout[a]
Up through 8.0 in. outside diameter ±0.020 in.	Up through 10.0 in. outside diameter 0.010 in.	Up through 5.0 in. outside diameter 0.005 in.
For each additional inch of outside diameter add ±0.0025 in.	For each additional inch of outside diameter add 0.0005 in.	For each additional inch of outside diameter add 0.001 in.

[a] Total indicator reading.
All dimensions in inches.

Sheave Dimensions: Groove angles and dimensions for sheaves and face widths of sheaves for multiple belt drives are given in Table 2, along with various tolerance values. Standard sheave outside diameters are given in Table 3.

Table 3. Standard Sheave Outside Diameters *ANSI/RMA IP-22, 1983*

3V			5V			8V		
Nom	Min	Max	Nom	Min	Max	Nom	Min	Max
2.65	2.638	2.680	7.10	7.087	7.200	12.50	12.402	12.600
2.80	2.795	2.840	7.50	7.480	7.600	13.20	13.189	13.400
3.00	2.953	3.000	8.00	7.874	8.000	14.00	13.976	14.200
3.15	3.150	3.200	8.50	8.346	8.480	15.00	14.764	15.000
3.35	3.346	3.400	9.00	8.819	8.960	16.00	15.748	16.000
3.55	3.543	3.600	9.25	9.291	9.440	17.00	16.732	17.000
3.65	3.642	3.700	9.75	9.567	9.720	18.00	17.717	18.000
4.00	3.937	4.000	10.00	9.843	10.000	19.00	18.701	19.000
4.12	4.055	4.120	10.30	10.157	10.320	20.00	19.685	20.000
4.50	4.409	4.480	10.60	10.433	10.600	21.20	20.866	21.200
4.75	4.646	4.720	10.90	10.709	10.880	22.40	22.047	22.400
5.00	4.921	5.000	11.20	11.024	11.200	23.60	23.622	24.000
5.30	5.197	5.280	11.80	11.811	12.000	24.80	24.803	25.200
5.60	5.512	5.600	12.50	12.402	12.600	30.00	29.528	30.000
6.00	5.906	6.000	13.20	13.189	13.400	31.50	31.496	32.000
6.30	6.299	6.400	14.00	13.976	14.200	35.50	35.433	36.000
6.50	6.496	6.600	15.00	14.764	15.000	40.00	39.370	40.000
6.90	6.890	7.000	16.00	15.748	16.000	44.50	44.094	44.800
8.00	7.874	8.000	18.70	18.701	19.000	50.00	49.213	50.000
10.00	9.843	10.000	20.00	19.685	20.000	52.00	51.969	52.800
10.60	10.433	10.600	21.20	20.866	21.200	63.00	62.992	64.000
12.50	12.402	12.600	23.60	23.622	24.000	71.00	70.866	72.000
14.00	13.976	14.200	25.00	24.803	25.200	79.00	78.740	80.000
16.00	15.748	16.000	28.00	27.953	28.400	99.00	98.425	100.000
19.00	18.701	19.000	31.50	31.496	32.000	…	…	…
20.00	19.685	20.000	37.50	37.402	38.000	…	…	…
25.00	24.803	25.200	40.00	39.370	40.000	…	…	…
31.50	31.496	32.000	44.50	44.094	44.800	…	…	…
33.50	33.465	34.000	50.00	49.213	50.000	…	…	…
…	…	…	63.00	62.992	64.000	…	…	…
…	…	…	71.00	70.866	72.000	…	…	…

All dimensions in inches. The nominal diameters were selected from R40 and R80 preferred numbers (see page 19).

Cross Section Selection: The chart (Fig. 2, on page 2379) is a guide to the V-belt cross section to use for any combination of design horsepower and speed of the faster shaft. When the intersection of the design horsepower and speed of the faster shaft falls near a line between two areas on the chart, it is advisable to investigate the possibilities in both areas. Special circumstances (such as space limitations) may lead to a choice of belt cross section different from that indicated in the chart.

Horsepower Ratings: The horsepower ratings of narrow V-belts can be calculated using the following formula:

$$\mathrm{HP} = d_p r[K_1 - K_2/d_p - K_3(d_p r)^2 - K_4 \log(d_p r)] + K_{SR} r$$

where d_p = the pitch diameter of the small sheave, in.; r = rpm of the faster shaft divided by 1000; K_{SR}, speed ratio correction factor, and K_1, K_2, K_3, and K_4, cross section parameters, are listed in the accompanying tables. This formula gives the basic horsepower rating, corrected for the speed ratio. To obtain the horsepower per belt for an arc of contact other than 180° and for belts shorter or longer than average length, multiply the horsepower obtained from this formula by the length correction factor (Table 4) and the arc of contact correction factor (Table 5).

Table 4. Length Correction Factors

Standard Length Designation	Cross Section 3V	5V	8V	Standard Length Designation	Cross Section 3V	5V	8V
250	0.83			1180	1.12	0.99	0.89
265	0.84			1250	1.13	1.00	0.90
280	0.85			1320	1.14	1.01	0.91
300	0.86			1400	1.15	1.02	0.92
315	0.87			1500		1.03	0.93
335	0.88			1600		1.04	0.94
355	0.89			1700		1.05	0.94
375	0.90			1800		1.06	0.95
400	0.92			1900		1.07	0.96
425	0.93			2000		1.08	0.97
450	0.94			2120		1.09	0.98
475	0.95			2240		1.09	0.98
500	0.96	0.85		2360		1.10	0.99
530	0.97	0.86		2500		1.11	1.00
560	0.98	0.87		2650		1.12	1.01
600	0.99	0.88		2800		1.13	1.02
630	1.00	0.89		3000		1.14	1.03
670	1.01	0.90		3150		1.15	1.03
710	1.02	0.91		3350		1.16	1.04
750	1.03	0.92		3550		1.17	1.05
800	1.04	0.93		3750			1.06
850	1.06	0.94		4000			1.07
900	1.07	0.95		4250			1.08
950	1.08	0.96		4500			1.09
1000	1.09	0.96	0.87	4750			1.09
1060	1.10	0.97	0.88	5000			1.10
1120	1.11	0.98	0.88	…			…

Table 5. Arc of Contact Correction Factors

$\frac{D_e - d_e}{C}$	Arc of Contact, θ, on Small Sheave (deg)	Correction Factor	$\frac{D_e - d_e}{C}$	Arc of Contact, θ, on Small Sheave (deg)	Correction Factor
0.00	180	1.00	0.80	133	0.87
0.10	174	0.99	0.90	127	0.85
0.20	169	0.97	1.00	120	0.82
0.30	163	0.96	1.10	113	0.80
0.40	157	0.94	1.20	106	0.77
0.50	151	0.93	1.30	99	0.73
0.60	145	0.91	1.40	91	0.70
0.70	139	0.89	1.50	83	0.65

Speed Ratio Correction Factors

Speed Ratio[a] Range	K_{SR} Cross Section 3VX	5VX	Speed Ratio[a] Range	K_{SR} Cross Section 5V	8V
1.00–1.01	0.0000	0.0000	1.00–1.01	0.0000	0.0000
1.02–1.03	0.0157	0.0801	1.02–1.05	0.0963	0.4690
1.04–1.06	0.0315	0.1600	1.06–1.11	0.2623	1.2780
1.07–1.09	0.0471	0.2398	1.12–1.18	0.4572	2.2276
1.10–1.13	0.0629	0.3201	1.19–1.26	0.6223	3.0321
1.14–1.18	0.0786	0.4001	1.27–1.38	0.7542	3.6747
1.19–1.25	0.0944	0.4804	1.39–1.57	0.8833	4.3038
1.26–1.35	0.1101	0.5603	1.58–1.94	0.9941	4.8438
1.36–1.57	0.1259	0.6405	1.95–3.38	1.0830	5.2767
Over 1.57	0.1416	0.7202	Over 3.38	1.1471	5.5892

[a] D_p/d_p, where D_p (d_p) is the pitch diameter of the large (small) sheave.

Cross Section Correction Factors

Cross Section	K_1	K_2	K_3	K_4
3VX	1.1691	1.5295	1.5229×10^{-4}	0.15960
5VX	3.3038	7.7810	3.6432×10^{-4}	0.43343
5V	3.3140	10.123	5.8758×10^{-4}	0.46527
8V	8.6628	49.323	1.5804×10^{-3}	1.1669

Number of Belts: The number of belts required for an application is obtained by dividing the design horsepower by the corrected horsepower rating for one belt.

Minimum Sheave Size: The recommended minimum sheave size depends on the rpm of the faster shaft. Minimum sheave diameters for each belt cross-section are listed in Table 3.

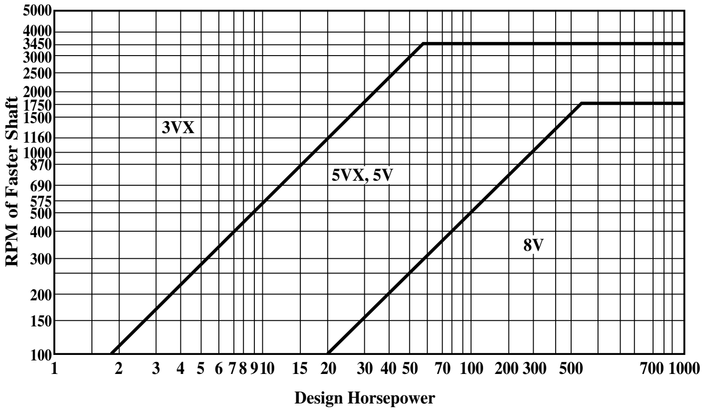


Fig. 2. Selection of Narrow V-Belt Cross Section

Arc of contact on the small sheave may be determined by the formulas.

Exact formula:

$$\text{Arc of Contact (deg)} = 2\cos^{-1}\left(\frac{D_e - d_e}{2C}\right)$$

Approximate formula:

$$\text{Arc of Contact (deg)} = 180 - \frac{(D_e - d_e)60}{C}$$

where: D_e = Effective diameter of large sheave, inch

d_e = Effective diameter of small sheave, inch

C = Center distance, inch

Classical V-Belts ANSI/RMA IP-20.—Classical V-belts are most commonly used in heavy-duty applications and include these standard cross sections: A, AX, B, BX, C, CX, D, and DX (Fig. 3, page 2383). Top widths range from ½ to 1¼ in. and are specified by cross section and nominal length. Classical belts can be teamed in multiples of two or more. These multiple drives can transmit up to several hundred horsepower continuously and absorb reasonable shock loads.

Belt Cross Sections: Nominal dimensions of the four cross sections are given in Fig. 3.

Belt Size Designation: Classical V-belt sizes are identified by a standard belt number consisting of a letter-numeral combination. The letter identifies the cross section; the numeral identifies the length as shown in Table 6. For example, A60 indicates an A cross section and a standard length designation of 60. An X following the section letter designation indicates a molded notch cross section, for example, AX60.

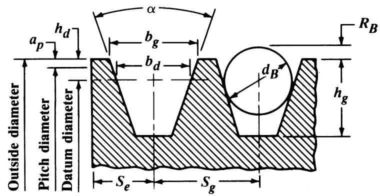
Table 6. Classical V-Belt Standard Datum Length *ANSI/RMA IP-20, 1988*

Standard Length Designation[a]	Standard Datum lengths				Permissible Deviations from Std. Datum Length	Matching Limits for One Set
	Cross Section					
	A, AX	B, BX	C, CX	D		
26	27.3	…	…	…	+0.6, −0.6	0.15
31	32.3	…	…	…	+0.6, −0.6	0.15
35	36.3	36.8	…	…	+0.6, −0.6	0.15
38	39.3	39.8	…	…	+0.7, −0.7	0.15
42	43.3	43.8	…	…	+0.7, −0.7	0.15
46	47.3	47.8	…	…	+0.7, −0.7	0.15
51	52.3	52.8	53.9	…	+0.7, −0.7	0.15
55	56.3	56.8	…	…	+0.7, −0.7	0.15
60	61.3	61.8	62.9	…	+0.7, −0.7	0.15
68	69.3	69.8	70.9	…	+0.7, −0.7	0.30
75	75.3	76.8	77.9	…	+0.7, −0.7	0.30
80	81.3	…	…	…	+0.7, −0.7	0.30
81	…	82.8	83.9	…	+0.7, −0.7	0.30
85	86.3	86.8	87.9	…	+0.7, −0.7	0.30
90	91.3	91.8	92.9	…	+0.8, −0.8	0.30
96	97.3	…	98.9	…	+0.8, −0.8	0.30
97	…	98.8	…	…	+0.8, −0.8	0.30
105	106.3	106.8	107.9	…	+0.8, −0.8	0.30
112	113.3	113.8	114.9	…	+0.8, −0.8	0.30
120	121.3	121.8	122.9	123.3	+0.8, −0.8	0.30
128	129.3	129.8	130.9	131.3	+0.8, −0.8	0.30
144	…	145.8	146.9	147.3	+0.8, −0.8	0.30
158	…	159.8	160.9	161.3	+1.0, −1.0	0.45
173	…	174.8	175.9	176.3	+1.0, −1.0	0.45
180	…	181.8	182.9	183.3	+1.0, −1.0	0.45
195	…	196.8	197.9	198.3	+1.1, −1.1	0.45
210	…	211.8	212.9	213.3	+1.1, −1.1	0.45
240	…	240.3	240.9	240.8	+1.3, −1.3	0.45
270	…	270.3	270.9	270.8	+1.6, −1.6	0.60
300	…	300.3	300.0	300.8	+1.6, −1.6	0.60
330	…	…	330.9	330.8	+2.0, −2.0	0.60
360	…	…	380.9	360.8	+2.0, −2.0	0.60
540	…	…	…	540.8	+3.3, −3.3	0.90
390	…	…	390.9	390.8	+2.0, −2.0	0.75
420	…	…	420.9	420.8	+3.3, −3.3	0.75
480	…	…	…	480.8	+3.3, −3.3	0.75
600	…	…	…	600.8	+3.3, −3.3	0.90
660	…	…	…	660.8	+3.3, −3.3	0.90

[a] To specify belt size use the Standard Length Designation prefixed by the letter indicating the cross section, e.g., B90.

All dimensions in inches.

Table 7. Classical V-Belt Sheave and Groove Dimensions *ANSI/RMA IP-20, 1988*



Face Width of Standard and Deep Groove Sheaves $= S_g(N_g - 1) + 2S_e$, where N_g = number of grooves

Standard Groove Dimensions													Design Factors	
Cross Section	Datum[a] Diameter Range	α Groove Angle ±0.33°	b_d Ref	b_g		h_g Min	$2h_d$	R_B Min	d_B ±0.0005	S_g[b] ±0.025	S_e		Min Recom. Datum Diameter	$2a_p$
A, AX	Through 5.4	34	0.418	0.494	± 0.005	0.460	0.250	0.148	0.4375	0.625	0.375	+ 0.090	A 3.0	0
	Over 5.4	38		0.504				0.149	($\frac{7}{16}$)			− 0.062	AX 2.2	
B, BX	Through 7.0	34	0.530	0.637	± 0.006	0.550	0.350	0.189	0.5625	0.750	0.500	+ 0.120	B 5.4	0
	Over 7.0	38		0.650				0.190	($\frac{9}{16}$)			− 0.065	BX 4.0	
Combination A, AX Belt	Through 7.4[c]	34	0.508[d]	0.612	±0.006	0.612	0.634[e]	0.230	0.5625 ($\frac{9}{16}$)	0.750	0.500	+0.120 −0.065	A 3.6[c]	0.37
	Over 7.4	38		0.625			0.602[e]	0.226					AX 2.8	
Combination B, BX Belt	Through 7.4[c]	34		0.612	±0.006		0.333[e]	0.230					B 5.7[c]	−0.01
	Over 7.4	38		0.625			0.334[e]	0.226					BX 4.3	
C, CX	Through 7.99	34	0.757	0.879	±0.007	0.750	0.400	0.274	0.7812 ($\frac{25}{32}$)	1.000	0.688	+0.160 −0.070	C 9.0 CX 6.8	0
	Over 7.99 to and incl. 12.0	36		0.887				0.276						
	Over 12.0	38		0.895				0.277						
D	Through 12.99	34	1.076	1.259	± 0.008	1.020	0.600	0.410	1.1250 ($1\frac{1}{8}$)	1.438	0.875	+ 0.220 − 0.080	13.0	0
	Over 12.99 to and incl. 17.0	36		1.271				0.410						
	Over 17.0	38		1.283				0.411						

Table 7. *(Continued)* **Classical V-Belt Sheave and Groove Dimensions** *ANSI/RMA IP-20, 1988*

Deep Groove Dimensions[f]													Design Factors	
Cross Section	Datum[a] Dia. Range	α Groove Angle ± 0.33°	b_g Ref	b_g		h_g Min	$2h_d$ Ref	R_B Min	d_B ± 0.0005	S_g[b] ± 0.025	S_e		Min Rec. Datum Diameter	$2a_p$
B, BX	Through 7.0	34	0.530	0.747	± 0.006	0.730	0.710	0.007	0.5625 (9⁄16)	0.875	0.562	+ 0.120 / − 0.065	B 5.4	0.36
	Over 7.0	38		0.774				0.008					BX 4.0	
C, CX	Through 7.99	34	0.757	1.066	± 0.007	1.055	1.010	− 0.035	0.7812 (25⁄32)	1.250	0.812	+ 0.160 / − 0.070	C 9.0 / CX 6.8	0.61
	Over 7.99 to and incl. 12.0	36		1.085				− 0.032						
	Over 12.0	38		1.105				−0.031						
D	Through 12.99	34	1.076	1.513	±0.008	1.435	1.430	−0.010	1.1250 (1⅛)	1.750	1.062	+0.220 / −0.080	13.0	0.83
	Over 12.99 to and incl. 17.0	36		1.514				−0.009						
	Over 17.0	38		1.569				−0.008						

[a] The A/AX, B/BX combination groove should be used when deep grooves are required for A or AX belts.

[b] Summation of the deviations from S_g for all grooves in any one sheave should not exceed ±0.050 in. The variation in datum diameter between the grooves in any one sheave must be within the following limits: Through 19.9 in. outside diameter and through 6 grooves: 0.010 in. (add 0.0005 in. for each additional groove). 20.0 in. and over on outside diameter and through 10 grooves: 0.015 in. (add 0.0005 in. for each additional groove). This variation can be obtained by measuring the distance across two measuring balls or rods placed diametrically opposite each other in a groove. Comparing this "diameter over balls or rods" measurement between grooves will give the variation in datum diameter.

[c] Diameters shown for combination grooves are outside diameters. A specific datum diameter does not exist for either A or B belts in combination grooves.

[d] The b_d value shown for combination grooves is the "constant width" point, but does not represent a datum width for either A or B belts ($2h_d = 0.340$ ref).

[e] $2h_d$ values for combination grooves are calculated based on b_d for A and B grooves.

[f] Deep groove sheaves are intended for drives with belt offset such as quarter-turn or vertical shaft drives. Joined belts will not operate in deep groove sheaves. Also, A and AX joined belts will not operate in A/AX and B/BX combination grooves.

Other Sheave Tolerances		
Outside Diameter	Radial Runout[a]	Axial Runout[a]
Through 8.0 in. outside diameter ±0.020 in. For each additional inch of outside diameter add ±0.005 in.	Through 10.0 in. outside diameter 0.010 in. For each additional inch of outside diameter add 0.0005 in.	Through 5.0 in. outside diameter 0.005 in. For each additional inch of outside diameter add 0.001 in.

[a] Total indicator readings.

A, AX & B, BX Combin. All dimensions in inches.

Sheave Dimensions: Groove angles and dimensions for sheaves and the face widths of sheaves for multiple belt drives are given in Table 7, along with various tolerance values.

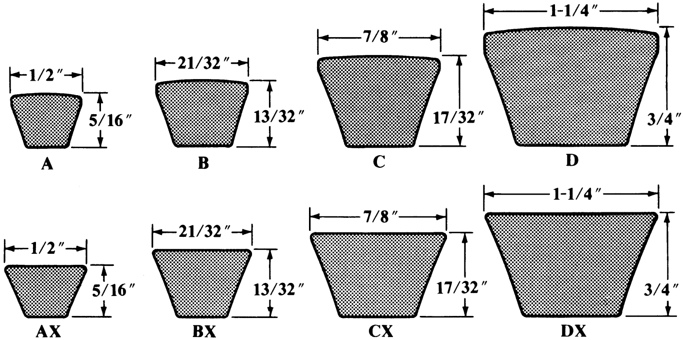


Fig. 3. Classical V-Belt Cross Sections

Cross Section Selection: Use the chart (Fig. 4) as a guide to the Classical V-belt cross section for any combination of design horsepower and speed of the faster shaft. When the intersection of the design horsepower and speed of the faster shaft falls near a line between two areas on the chart, the possibilities in both areas should be investigated. Special circumstances (such as space limitations) may lead to a choice of belt cross section different from that indicated in the chart.

Horsepower Ratings: The horsepower rating formulas for classical V-belts are:

$$\mathbf{A:}\mathrm{HP} = d_p r\left[1.004 - \frac{1.652}{d_p} - 1.547\times10^{-4}(d_p r)^2 - 0.2126\log(d_p r)\right] + 1.652r\left(1 - \frac{1}{K_{SR}}\right)$$

$$\mathbf{AX:}\mathrm{HP} = d_p r\left[1.462 - \frac{2.239}{d_p} - 2.198\times10^{-4}(d_p r)^2 - 0.4238\log(d_p r)\right] + 2.239r\left(1 - \frac{1}{K_{SR}}\right)$$

$$\mathbf{B:}\mathrm{HP} = d_p r\left[1.769 - \frac{4.372}{d_p} - 3.081\times10^{-4}(d_p r)^2 - 0.3658\log(d_p r)\right] + 4.372r\left(1 - \frac{1}{K_{SR}}\right)$$

$$\mathbf{BX:}\text{HP} = d_p r\left[2.051 - \frac{3.532}{d_p} - 3.097\times10^{-4}(d_p r)^2 - 0.5735\log(d_p r)\right]$$
$$+3.532r\left(1 - \frac{1}{K_{SR}}\right)$$

$$\mathbf{C:}\text{HP} = d_p r\left[3.325 - \frac{12.07}{d_p} - 5.828\times10^{-4}(d_p r)^2 - 0.6886\log(d_p r)\right]$$
$$+12.07r\left(1 - \frac{1}{K_{SR}}\right)$$

$$\mathbf{CX:}\text{HP} = d_p r\left[3.272 - \frac{6.655}{d_p} - 5.298\times10^{-4}(d_p r)^2 - 0.8637\log(d_p r)\right]$$
$$+6.655r\left(1 - \frac{1}{K_{SR}}\right)$$

$$\mathbf{D:}\text{HP} = d_p r\left[7.160 - \frac{43.21}{d_p} - 1.384\times10^{-3}(d_p r)^2 - 1.454\log(d_p r)\right]$$
$$+43.21r\left(1 - \frac{1}{K_{SR}}\right)$$

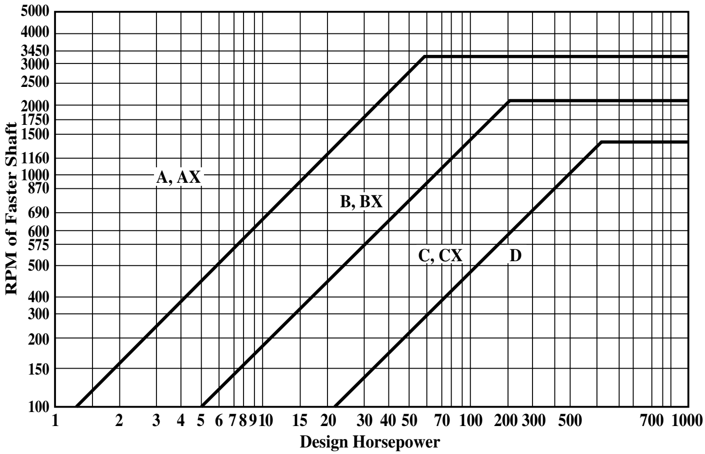


Fig. 4. Selection of Classic V-Belt Cross Sections

In these equations, d_p = pitch diameter of small sheave, in.; r = rpm of the faster shaft divided by 1000; K_{SR} = speed ratio factor given in the accompanying table. These formulas give the basic horsepower rating, corrected for the speed ratio. To obtain the horsepower per belt for an arc of contact other than 180 degrees and for belts shorter or longer than average length, multiply the horsepower obtained from these formulas by the length correction factor (Table 8) and the arc of contact correction factor (Table 9).

Table 8. Length Correction Factors

Std. Length Designation	Cross Section			
	A, AX	B, BX	C, CX	D
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
195	…	1.17	1.08	0.96
210	…	1.18	1.07	0.98
240	…	1.22	1.10	1.00
270	…	1.24	1.13	1.02
300	…	1.27	1.15	1.04
330	…	…	1.17	1.06
360	…	…	1.18	1.07
390	…	…	1.20	1.09
420	…	…	1.21	1.10
480	…	…	…	1.13
540	…	…	…	1.15
600	…	…	…	1.17
660	…	…	…	1.18

Table 9. Arc of Contact Correction Factors

$\frac{D_d - d_d}{C}$	Arc of Contact, θ, Small Sheave (deg)	Correction Factor		$\frac{D_d - d_d}{C}$	Arc of Contact, θ Small Sheave (deg)	Correction Factor	
		V-V	V-Flat[a]			V-V	V-Flat[a]
0.00	180	1.00	0.75	0.80	133	0.87	0.85
0.10	174	0.99	0.76	0.90	127	0.85	0.85
0.20	169	0.97	0.78	1.00	120	0.82	0.82
0.30	163	0.96	0.79	1.10	113	0.80	0.80
0.40	157	0.94	0.80	1.20	106	0.77	0.77
0.50	151	0.93	0.81	1.30	99	0.73	0.73
0.60	145	0.91	0.83	1.40	91	0.70	0.70
0.70	139	0.89	0.84	1.50	83	0.65	0.65

[a] A V-flat drive is one using a small sheave and a large diameter flat pulley.

Speed Ratio Correction Factors

Speed Ratio[a] Range	K_{SR}	Speed Ratio[a] Range	K_{SR}
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

[a] D_p/d_p, where D_p (d_p) is the pitch diameter of the large (small) sheave.

Arc of contact on the small sheave may be determined by the formulas.

Exact formula: $\text{Arc of Contact (deg)} = 2\cos^{-1}\left(\frac{D_d - d_d}{2C}\right)$

Approximate formula: $\text{Arc of Contact (deg)} = 180 - \left(\frac{(D_d - d_d)60}{C}\right)$

where D_d = Datum diameter of large sheave or flat pulley, inch; d_d = Datum diameter of small sheave, inch; and, C = Center distance, inch.

Number of Belts: The number of belts required for an application is obtained by dividing the design horsepower by the corrected horsepower rating for one belt.

Minimum Sheave Size: The recommended minimum sheave size depends on the rpm of the faster shaft. Minimum groove diameters for each belt cross section are listed in Table 11.

Double V-Belts ANSI/RMA IP-21.—Double V-belts or hexagonal belts are used when power input or takeoff is required on both sides of the belt. Designed for use on "serpentine" drives, which consist of sheaves rotating in opposite directions, the belts are available in AA, BB, CC, and DD cross sections and operate in standard classical sheaves. They are specified by cross section and nominal length.

Belt Cross Sections: Nominal dimensions of the four cross sections are given in Fig. 5.

Belt Size Designation: Double V-belt sizes are identified by a standard belt number, consisting of a letter-numeral combination. The letters identify the cross section; the numbers identify length as shown in Column 1 of Table 10. For example, AA51 indicates an AA cross section and a standard length designation of 51.

Table 10. Double V-Belt Standard Effective Lengths *ANSI/RMA IP-21, 1984*

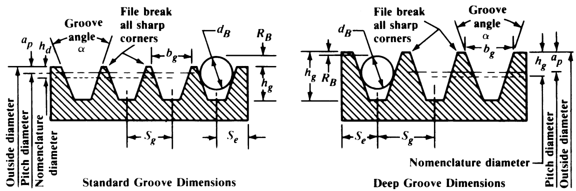
Standard Length Designation[a]	Standard Effetive Length				Permissible Deviation from Standard Effective Length	Matching Limits for One Set
	Cross Section					
	AA	BB	CC	DD		
51	53.1	53.9	…	…	±0.7	0.15
55	…	57.9	…	…	±0.7	0.15
60	62.1	62.9	…	…	±0.7	0.15
68	70.1	70.9	…	…	±0.7	0.30
75	77.1	77.9	…	…	±0.7	0.30
80	82.1	…	…	…	±0.7	0.30
81	…	83.9	85.2	…	±0.7	0.30
85	87.1	87.9	89.2	…	±0.7	0.30
90	92.1	92.9	94.2	…	±0.8	0.30
96	98.1	…	100.2	…	±0.8	0.30
97	…	99.9	…	…	±0.8	0.30
105	107.1	107.9	109.2	…	±0.8	0.30
112	114.1	114.9	116.2	…	±0.8	0.30
120	122.1	122.9	124.2	125.2	±0.8	0.30
128	130.1	130.9	132.2	133.2	±0.8	0.30
144	…	146.9	148.2	149.2	±0.8	0.30
158	…	160.9	162.2	163.2	±1.0	0.45
173	…	175.9	177.2	178.2	±1.0	0.45
180	…	182.9	184.2	185.2	±1.0	0.45
195	…	197.9	199.2	200.2	±1.1	0.45
210	…	212.9	214.2	215.2	±1.1	0.45
240	…	241.4	242.2	242.7	±1.3	0.45
270	…	271.4	272.2	272.7	±1.6	0.60
300	…	301.4	302.2	302.7	±1.6	0.60
330	…	…	332.2	332.7	±2.0	0.60
360	…	…	362.2	362.7	±2.0	0.60

[a] To specify belt size use the Standard Length Designation prefixed by the letters indicating cross section; for example, BB90.

All dimensions in inches.

Sheave Dimensions: Groove angles and dimensions for sheaves and face widths of sheaves for multiple belt drives are given in Table 11, along with various tolerance values.

Table 11. Double V-Belt Sheave and Groove Dimensions *ANSI/RMP IP-21, 1984*



Face Width of Standard and Deep Groove Sheaves $= S_g(N_g - 1) + 2S_e$, where N_g = number of grooves

Standard Groove Dimensions												Drive Design Factors	
Cross Section	Outside Diameter Range	Groove Angle, α ±0.33°	b_g		h_g (Min.)	$2h_d$	R_B (Min.)	d_B ±0.0005	S_g[a] ±0.025	S_e		Min. Recomm. Outside Diam.	$2a_p$[b]
AA	Up through 5.65 Over 5.65	34 38	0.494 0.504	±0.005	0.460	0.250	0.148 0.149	0.4375 (7⁄16)	0.625	0.375	+0.090 −0.062	3.25	0.0
BB	Up through 7.35 Over 7.35	34 38	0.637 0.650	±0.006	0.550	0.350	0.189 0.190	0.5625 (9⁄16)	0.750	0.500	+0.120 −0.065	5.75	0.0
AA-BB	Up through 7.35 Over 7.35	34 38	0.612 0.625	±0.006	0.612	A = 0.750 B = 0.350	0.230 0.226	0.5625 (9⁄16)	0.750	0.500	+0.120 −0.065	A = 3.620 B = 5.680	A = 0.370 B = −0.070
CC	Up through 8.39 Over 8.39 up to and including 12.40 Over 12.40	34 36 38	0.879 0.887 0.895	} ±0.007	0.750	0.400	0.274 0.276 0.277	0.7812 (25⁄32)	1.000	0.688	+0.160 −0.070	9.4	0.0
DD	Up through 13.59 Over 13.59 up to and including 17.60 Over 17.60	34 36 38	1.259 1.271 1.283	} ±0.008	1.020	0.600	0.410 0.410 0.411	1.1250 (1 1⁄8)	1.438	0.875	+0.220 −0.080	13.6	0.0

Table 11. *(Continued)* **Double V-Belt Sheave and Groove Dimensions** *ANSI/RMP IP-21, 1984*

Deep Groove Dimensions[c]												Drive Design Factors	
Cross Section	Outside Diameter Range	Groove Angle, α ±0.33°	b_g		h_g (Min.)	$2h_d$	R_B (Min.)	d_B ±0.0005	S_g[a] ±0.025	S_e		Minimum Recommended Outside Diameter	$2a_p$
AA	Up through 5.96	34	0.589	±0.005	0.615	0.560	−0.009	0.4375	0.750	0.438	+0.090	3.56	0.310
	Over 5.96	38	0.611				−0.008	(7⁄16)			−0.062		
BB	Up through 7.71	34	0.747	±0.006	0.730	0.710	+0.007	0.5625	0.875	0.562	+0.120	6.11	0.360
	Over 7.71	38	0.774				+0.008	(9⁄16)			−0.065		
CC	Up through 9.00	34	1.066	±0.007	1.055	1.010	−0.035	0.7812	1.250	0.812	+0.160	10.01	0.610
	Over 9.00 up to and including 13.01	36	1.085				−0.032	(25⁄32)			−0.070		
	Over 13.01	38	1.105				−0.031						
DD	Up through 14.42	34	1.513	±0.008	1.435	1.430	−0.010	1.1250	1.750	1.062	+0.220	14.43	0.830
	Over 14.42 up to and including 18.43	36	1.541				−0.009	(1⅛)			−0.080		
	Over 18.43	38	1.569				−0.008						

[a] Summation of the deviations from S_g for all grooves in any one sheave shall not exceed ±0.050 in. The variation in pitch diameter between the grooves in any one sheave must be within the following limits: Up through 19.9 in. outside diameter and up through 6 grooves: 0.010 in. (add 0.005 in. for each additional groove). 20.0 in. and over on outside diameter and up through 10 grooves: 0.015 in. (add 0.0005 in. for each additional groove). This variation can be obtained easily by measuring the distance across two measuring balls or rods placed diametrically opposite each other in a groove. Comparing this "diameter over balls or rods" measurement between grooves will give the variation in pitch diameter.

[b] The a_p values shown for the A/B combination sheaves are the geometrically derived values. These values may be different from those shown in manufacturer's catalogs.

[c] Deep groove sheaves are intended for drives with belt offset such as quarter-turn or vertical shaft drives.

Other Sheave Tolerances		
Outside Diameter	Radial Runout[a]	Axial Runout[a]
Up through 4.0 in. outside diameter ±0.020 in. For each additional inch of outside diameter add ±0.005 in.	Up through 10.0 in. outside diameter ±0.010 in. For each additional inch of outside diameter add 0.0005 in.	Up through 5.0 in. outside diameter 0.005 in. For each additional inch of outside diameter add 0.001 in.

[a] Total indicator reading.

All dimensions in inches.

Cross Section Selection: Use the chart (Fig. 5) as a guide to the double V-belt cross section for any combination of design horsepower and speed of the faster shaft. When the intersection of the design horsepower and speed of the faster shaft falls near a line between two areas on the chart, it is best to investigate the possibilities in both areas. Special circumstances (such as space limitations) may lead to a choice of belt cross section different from that indicated in the chart.

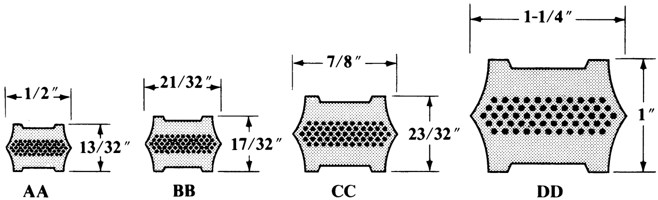


Fig. 5. Double-V Belt Cross Section

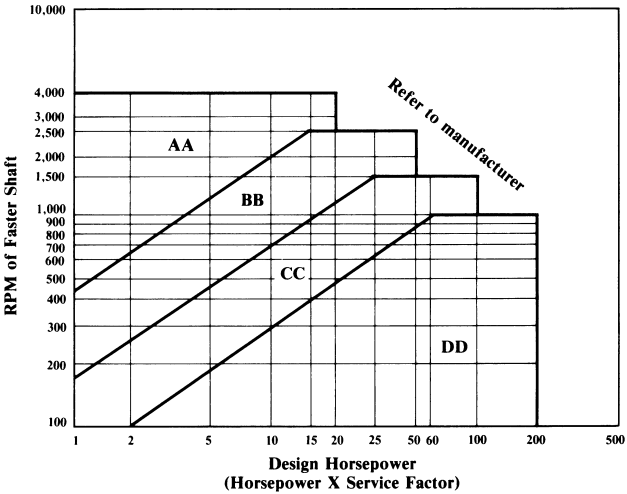


Fig. 6. Selection of Double V-Belt Cross Section

Effective Diameter Determination: Fig. 6 shows the relationship of effective diameter, outside diameter, and nomenclature diameter. Nomenclature diameter is used when ordering sheaves for double V-belt drives. The effective diameter is determined as follows:

$$\text{Effective diameter} = \text{Nomenclature diameter} + 2h_d - 2a_p$$

The values of $2h_d$ and $2a_p$ are given in Table 11.

Double V-belt Length Determination: The effective belt length of a specific drive may be determined by making a scaled layout of the drive. Draw the sheaves in terms of their effective diameters and in the position when a new belt is applied and first brought to driving tension. Next, measure the tangents and calculate the effective arc length (AL_e) of each sheave (see Table 12 for a glossary of terms):

$$AL_e = \frac{d_e\theta}{115}$$

The effective length of the belt will then be the sum of the tangents and the connecting arc lengths. Manufacturers may be consulted for mathematical calculation of effective belt length for specific drive applications.

Table 12. Glossary of Terms for Double V-belt Calculations

AL_e	=	Length, arc, effective, in.
$2a_p$	=	Diameter, differential, pitch to outside, in.
d	=	Diameter, pitch, in. (same as effective diameter)
d_e	=	Diameter, effective, in.
$2h_d$	=	Diameter differential, nomenclature to outside, in.
K_f	=	Factor, length – flex correction
L_e	=	Length, effective, in.
n	=	Sheaves, number on drive
P_d	=	Power, design, horsepower (transmitted horsepower × service factor)
R	=	Ratio, tight side to slack side tension
$R/(R-1)$	=	Factor, tension ratio
r	=	Angular velocity, faster shaft, rpm/1000
S	=	Speed, belt, fpm/1000
T_e	=	Tension, effective pull, lbf
T_r	=	Tension, allowable tight side, lbf
T_S	=	Tension, slack side, lbf
T_T	=	Tension, tight side, lbf
θ	=	Angle, arc of belt contact, deg

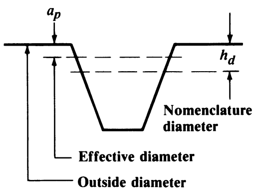


Fig. 7. Effective, Outside, and Nomenclature Sheave Diameters

Number of Belts Determination: The number of belts required may be determined on the basis of allowable tight side tension rating (T_r) at the most severe sheave. The allowable tight side tensions per belt are given in Tables 13 through 16, and must be multiplied by the length-flex correction factors (K_f) listed in Table 17. To select the allowable tight side tension from the tables for a given sheave, the belt speed and effective diameter of the sheave in question are required.

Double V-Belt Drive Design Method: The fourteen drive design steps are as follows:

1) Number the sheaves starting from the driver in the opposite direction to belt rotation; include the idlers.

2) Select the proper service factor for each loaded driven unit.

3) Multiply the horsepower requirement for each loaded driven sheave by the corresponding service factor. This is the design horsepower at each sheave.

4) Calculate driver design horsepower. This hp is equal to the sum of all the driven design horsepowers.

5) Calculate belt speed (S) in thousands of feet per minute: $S = rd/3.820$.

6) Calculate effective tension (T_e) for each loaded sheave: $T_e = 33P_d/S$.

7) Determine minimum $R/(R-1)$ for each loaded sheave from Table 18 using the arc of contact determined from the drive layout.

8) In most drives, slippage will occur first at the driver sheave. Assume this to be true and calculate T_T and T_S for the driver: $T_T = T_e\,[R/(R-1)]$ and $T_s = T_T - T_e$. Use $R/(R-1)$ from Step 7 and T_e from Step 6 for the driver sheave.

9) Starting with the first driven sheave, determine T_T and T_S for each segment of the drive. The T_T for the driver becomes T_S for that sheave and is equal to $T_T - T_e$. Proceed around the drive in like manner.

10) Calculate actual $R/(R-1)$ for each sheave using: $R/(R-1) = T_T/T_e = T_T/(T_T - T_S)$. The T_T and T_S values are for those determined in Step 9. If these values are equal to or greater than those determined in Step 7, the assumption that slippage will first occur at the driver is correct and the next two steps are not necessary. If the value is less, the assumption was not correct, so proceed with Step 11.

11) Take the sheave where the actual value $R/(R-1)$ (Step 10) is less than the minimum, as determined in Step 7, and calculate a new T_T and T_S for this sheave using the minimum $R/(R-1)$ as determined in Step 7: $T_T = T_e\,[R/(R-1)]$ and $T_S = T_T - T_e$.

12) Start with this sheave and recalculate the tension in each segment of the drive as in Step 9.

13) The length-flex factor (K_f) is taken from Table 17. Before using this table, calculate the value of L_e/n. Be sure to use the appropriate belt cross-section column when selecting the correction factor.

14) Beginning with the driver sheave, determine the number of belts (N_b) needed to satisfy the conditions at each loaded sheave using: $N_b = T_T/T_r K_f$. Note: T_T is tight side tension as determined in Step 9 or 11 and 12. T_r is allowable tight side tension as shown in Tables 15-18. K_f is the length-flex correction factor from Table 17. The sheave that requires the largest number of belts is the number of belts required for the drive. Any fraction of a belt should be treated as a whole belt.

Table 13. Allowable Tight Side Tension for an AA Section

Belt Speed (fpm)	Sheave Effective Diameter (in.)							
	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5
200	30	46	57	66	73	79	83	88
400	23	38	49	58	65	71	76	80
600	18	33	44	53	60	66	71	75
800	14	30	41	50	57	63	67	72
1000	12	27	38	47	54	60	65	69
1200	9	24	36	45	52	57	62	66
1400	7	22	34	42	49	55	60	64
1600	5	20	32	40	47	53	58	62
1800	3	18	30	38	46	51	56	60
2000	1	16	28	37	44	50	54	58
2200	…	15	26	35	42	48	53	57
2400	…	13	24	33	40	46	51	55
2600	…	11	23	31	39	44	49	53
2800	…	9	21	30	37	43	47	51
3000	…	8	19	28	35	41	46	50
3200	…	6	17	26	33	39	44	48
3400	…	4	16	24	31	37	42	46
3600	…	2	14	23	30	35	40	44
3800	…	1	12	21	28	34	38	43
4000	…	…	10	19	26	32	37	41
4200	…	…	8	17	24	30	35	39
4400	…	…	6	15	22	28	33	37
4600	…	…	4	13	20	26	31	35
4800	…	…	2	11	18	24	29	33
5000	…	…	…	9	16	22	27	31
5200	…	…	…	7	14	20	24	28
5400	…	…	…	4	12	17	22	26
5600	…	…	…	2	9	15	20	24
5800	…	…	…	…	7	13	18	22

The allowable tight side tension must be evaluated for each sheave in the system (see Step 14). Values must be corrected by K_f from Table 17.

Table 14. Allowable Tight Side Tension for a BB Section

Belt Speed (fpm)	Sheave Effective Diameter (in.)								
	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0
200	81	93	103	111	119	125	130	135	140
400	69	81	91	99	107	113	118	123	128
600	61	74	84	92	99	106	111	116	121
800	56	68	78	87	94	101	106	111	115
1000	52	64	74	83	90	96	102	107	111
1200	48	60	71	79	86	93	98	103	107
1400	45	57	67	76	83	89	95	100	104
1600	42	54	64	73	80	86	92	97	101
1800	39	51	61	70	77	84	89	94	98
2000	36	49	59	67	74	81	86	91	96
2200	34	46	56	64	72	78	84	89	93
2400	31	43	53	62	69	75	81	86	90
2600	29	41	51	59	67	73	78	83	88
2800	26	38	48	57	64	70	76	81	85
3000	23	35	45	54	61	68	73	78	82
3200	21	33	43	51	59	65	70	75	80
3400	18	30	40	49	56	62	68	73	77
3600	15	27	37	46	53	59	65	70	74
3800	12	24	35	43	50	57	62	67	71
4000	9	22	32	40	47	54	59	64	69
4200	7	19	29	37	45	51	56	61	66
4400	4	16	26	34	42	48	53	58	63
4600	1	13	23	31	39	45	50	55	60
4800	…	10	20	28	35	42	47	52	57
5000	…	6	16	25	32	39	44	49	53
5200	…	3	13	22	29	35	41	46	50
5400	…	…	10	18	26	32	38	42	47
5600	…	…	6	15	22	29	34	39	43
5800	…	…	3	11	19	25	31	36	40

The allowable tight side tension must be evaluated for each sheave in the system (see Step 14). Values must be corrected by K_f from Table 17.

Table 15. Allowable Tight Side Tension for a CC Section

Belt Speed (fpm)	Sheave Effective Diameter (in.)								
	7.0	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0
200	121	158	186	207	228	244	257	268	278
400	99	135	164	187	206	221	234	246	256
600	85	122	151	173	192	208	221	232	242
800	75	112	141	164	182	198	211	222	232
1000	67	104	133	155	174	190	203	214	224
1200	60	97	126	149	167	183	196	207	217
1400	54	91	120	142	161	177	190	201	211
1600	48	85	114	137	155	171	184	196	205
1800	43	80	108	131	150	166	179	190	200
2000	38	75	103	126	145	160	174	185	195
2200	33	70	98	121	140	155	169	180	190
2400	28	65	93	116	135	150	164	175	185
2600	23	60	88	111	130	145	159	170	180
2800	18	55	83	106	125	140	154	165	175
3000	13	50	78	101	120	135	149	160	170
3200	8	45	73	96	115	130	144	155	165
3400	3	39	68	91	110	125	138	150	160
3600	…	34	63	86	104	120	133	145	154
3800	…	29	58	80	99	115	128	139	149
4000	…	24	52	75	94	109	123	134	144
4200	…	18	47	70	88	104	117	128	138
4400	…	12	41	64	83	98	112	123	133
4600	…	7	35	58	77	93	106	117	127
4800	…	1	29	52	71	87	100	111	121
5000	…	…	23	46	65	81	94	105	115
5200	…	…	17	40	59	75	88	99	109
5400	…	…	11	34	53	68	81	93	103
5600	…	…	5	27	46	62	75	86	96
5800	…	…	…	21	40	55	68	80	90

The allowable tight side tension must be evaluated for each sheave in the system (see Step 14). Values must be corrected by K_f from Table 17.

Table 16. Allowable Tight Side Tension for a DD Section

Belt Speed (fpm)	Sheave Effective Diameter (in.)								
	12,0	13.0	14.0	15.0	16.0	17.0	18.0	19.0	20.0
200	243	293	336	373	405	434	459	482	503
400	195	245	288	325	358	386	412	434	455
600	167	217	259	297	329	358	383	406	426
800	146	196	239	276	308	337	362	385	405
1000	129	179	222	259	291	320	345	368	389
1200	114	164	207	244	277	305	331	353	374
1400	101	151	194	231	263	292	318	340	361
1600	89	139	182	219	251	280	305	328	349
1800	78	128	170	207	240	269	294	317	337
2000	67	117	159	196	229	258	283	306	326
2200	56	106	149	186	218	247	272	295	316
2400	45	95	138	175	208	236	262	284	305
2600	35	85	128	165	197	226	251	274	294
2800	24	74	117	154	187	215	241	263	284
3000	14	64	106	144	176	205	230	253	273
3200	3	53	96	133	165	194	219	242	263
3400	…	42	85	122	155	183	209	231	252
3600	…	31	74	111	144	172	198	220	241
3800	…	20	63	100	132	161	186	209	230
4000	…	9	51	89	121	150	175	198	218
4200	…	…	40	77	109	138	163	186	207
4400	…	…	28	65	97	126	152	174	195
4600	…	…	16	53	85	114	139	162	183
4800	…	…	3	40	73	102	127	150	170
5000	…	…	…	28	60	89	114	137	158
5200	…	…	…	15	47	76	101	124	145
5400	…	…	…	1	34	62	88	111	131
5600	…	…	…	…	20	49	74	97	118
5800	…	…	…	…	6	35	60	83	104

The allowable tight side tension must be evaluated for each sheave in the system (see Step 14). Values must be corrected by K_f from Table 17.

Table 17. Length-Flex Correction Factors K_f

$\frac{L_e}{n}$	Belt Cross Section				$\frac{L_e}{n}$	Belt Cross Section			
	AA	BB	CC	DD		AA	BB	CC	DD
10	0.64	0.58	…	…	70	…	1.03	0.95	0.91
15	0.74	0.68	…	…	80	…	1.06	0.98	0.94
20	0.82	0.74	0.68	…	90	…	1.09	1.00	0.96
25	0.87	0.79	0.73	0.70	100	…	1.11	1.03	0.99
30	0.92	0.84	0.77	0.74	110	…	…	1.05	1.00
35	0.96	0.87	0.80	0.77	120	…	…	1.06	1.02
40	0.99	0.90	0.83	0.80	130	…	…	1.08	1.04
45	1.02	0.93	0.86	0.82	140	…	…	1.10	1.05
50	1.05	0.95	0.88	0.84	150	…	…	1.11	1.07
60	…	0.99	0.92	0.88	…	…	…	…	…

Minimum Sheave Size: The recommended minimum sheave size depends on the rpm of the faster shaft. Minimum sheave diameters for each cross-section belt are listed in Table 11.

Tension Ratings: The tension rating formulas are:

$$\text{AA} \quad T_r = 118.5 - \frac{318.2}{d} - 0.8380S^2 - 25.76\log S$$

$$\text{BB} \quad T_r = 186.3 - \frac{665.1}{d} - 1.269S^2 - 39.02\log S$$

$$\text{CC}\quad T_r = 363.9 - \frac{2060}{d} - 2.400S^2 - 73.77\log S$$

$$\text{DD}\quad T_r = 783.1 - \frac{7790}{d} - 5.078S^2 - 156.1\log S$$

where T_r = The allowable tight side tension for a double-V belt drive, lbf (not corrected for tension ratio or length-flex correction factor)

d = Pitch diameter of small sheave, inch

S = Belt speed, fpm/1000

Table 18. Tension Ratio/Arc of Contact Factors

Arc of Contact, θ (deg.)	Design $\frac{R}{R-1}$	Arc of Contact, θ(deg.)	Design $\frac{R}{R-1}$
300	1.07	170	1.28
290	1.08	160	1.31
280	1.09	150	1.35
270	1.10	140	1.40
260	1.11	130	1.46
250	1.12	120	1.52
240	1.13	110	1.60
230	1.15	100	1.69
220	1.16	90	1.81
210	1.18	80	1.96
200	1.20	70	2.15
190	1.22	60	2.41
180	1.25	50	2.77

Light Duty V-Belts ANSI/RMA IP-23.—Light duty V-belts are typically used with fractional horsepower motors or small engines, and are designed primarily for fractional horsepower service. These belts are intended and specifically designed for use with small diameter sheaves and drives of loads and service requirements that are within the capacity of a single belt.

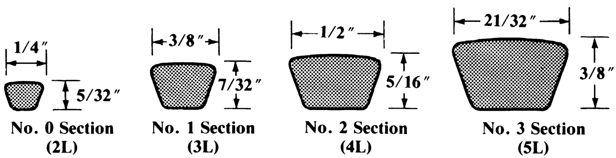


Fig. 8. Light Duty V-Belt Cross Sections

The four belt cross sections and sheave groove sizes are 2L, 3L, 4L, and 5L. The 2L is generally used only by OEMs and is not covered in the RMA standards.

Belt Cross Sections.—Nominal dimensions of the four cross sections are given in Fig. 8.

Belt Size Designation.—V-belt sizes are identified by a standard belt number, consisting of a letter-numeral combination. The first number and letter identify the cross section; the remaining numbers identify length as shown in Table 19. For example, a 3L520 belt has a 3L cross section and a length of 52.0 in.

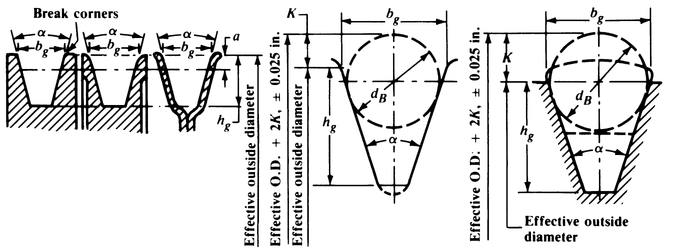
Table 19. Light Duty V-Belt Standard Dimensions *ANSI/RMA IP-23, 1968*

Standard Effective Outside Length (in.)				Permissible Deviation From Standard Effective Length (in.)	Standard Effective Outside Length (in.)				Permissible Deviation From Standard Effective Length (in.)
Cross Section					Cross Section				
2L	3L	4L	5L		2L	3L	4L	5L	
8	…	…	…	+0.12, −0.38	…	…	53	53	+0.25, −0.62
9	…	…	…	+0.12, −0.38	…	54	54	54	+0.25, −0.62
10	…	…	…	+0.12, −0.38	…	…	55	55	+0.25, −0.62
11	…	…	…	+0.12, −0.38	…	56	56	56	+0.25, −0.62
12	…	…	…	+0.12, −0.38	…	…	57	57	+0.25, −0.62
13	…	…	…	+0.12, −0.38	…	58	58	58	+0.25, −0.62
14	14	…	…	+0.12, −0.38	…	…	59	59	+0.25, −0.62
15	15	…	…	+0.12, −0.38	…	60	60	60	+0.25, −0.62
16	16	…	…	+0.12, −0.38	…	…	61	61	+0.31, −0.69
17	17	…	…	+0.12, −0.38	…	…	62	62	+0.31, −0.69
18	18	18	…	+0.12, −0.38	…	…	63	63	+0.31, −0.69
19	19	19	…	+0.12, −0.38	…	…	64	64	+0.31, −0.69
20	20	20	…	+0.12, −0.38	…	…	65	65	+0.31, −0.69
…	21	21	…	+0.25, −0.62	…	…	66	66	+0.31, −0.69
…	22	22	…	+0.25, −0.62	…	…	67	67	+0.31, −0.69
…	23	23	…	+0.25, −0.62	…	…	68	68	+0.31, −0.69
…	24	24	…	+0.25, −0.62	…	…	69	69	+0.31, −0.69
…	25	25	25	+0.25, −0.62	…	…	70	70	+0.31, −0.69
…	26	26	26	+0.25, −0.62	…	…	71	71	+0.31, −0.69
…	27	27	27	+0.25, −0.62	…	…	72	72	+0.31, −0.69
…	28	28	28	+0.25, −0.62	…	…	73	73	+0.31, −0.69
…	29	29	29	+0.25, −0.62	…	…	74	74	+0.31, −0.69
…	30	30	30	+0.25, −0.62	…	…	75	75	+0.31, −0.69
…	31	31	31	+0.25, −0.62	…	…	76	76	+0.31, −0.69
…	32	32	32	+0.25, −0.62	…	…	77	77	+0.31, −0.69
…	33	33	33	+0.25, −0.62	…	…	78	78	+0.31, −0.69
…	34	34	34	+0.25, −0.62	…	…	79	79	+0.31, −0.69
…	35	35	35	+0.25, −0.62	…	…	80	80	+0.62, −0.88
…	36	36	36	+0.25, −0.62	…	…	82	82	+0.62, −0.88
…	37	37	37	+0.25, −0.62	…	…	84	84	+0.62, −0.88
…	38	38	38	+0.25, −0.62	…	…	86	86	+0.62, −0.88
…	39	39	39	+0.25, −0.62	…	…	88	88	+0.62, −0.88
…	40	40	40	+0.25, −0.62	…	…	90	90	+0.62, −0.88
…	41	41	41	+0.25, −0.62	…	…	92	92	+0.62, −0.88
…	42	42	42	+0.25, −0.62	…	…	94	94	+0.62, −0.88
…	43	43	43	+0.25, −0.62	…	…	96	96	+0.62, −0.88
…	44	44	44	+0.25, −0.62	…	…	98	98	+0.62, −0.88
…	45	45	45	+0.25, −0.62	…	…	100	100	+0.62, −0.88
…	46	46	46	+0.25, −0.62	…	…	…	…	…
…	47	47	47	+0.25, −0.62	…	…	…	…	…
…	48	48	48	+0.25, −0.62					
…	49	49	49	+0.25, −0.62	…	…	…	…	…
…	50	50	50	+0.25, −0.62	…	…	…	…	…
…	…	51	51	+0.25, −0.62	…	…	…	…	…
…	52	52	52	+0.25, −0.62	…	…	…	…	…

All dimensions in inches.

Sheave Dimensions: Groove angles and dimensions for sheaves and various sheave tolerances are given in Table 20.

Table 20. Light Duty V-Belt Sheave and Groove Dimensions
ANSI/RMA IP-23, 1968



Belt Section	Effective Outside Diameter Min. Recomm.	Effective Outside Diameter Range	α Groove Angle ±0°20′ (deg)	d_B Ball Diameter ±0.0005	2K	b_g (Ref)	h_g (min)	$2a$[a]
2L	0.8	Less Than 1.50	32	0.2188	0.176	0.240	0.250	0.04
		1.50 to 1.99	34		0.182			
		2.00 to 2.50	36		0.188			
		Over 2.50	38		0.194			
3L	1.5	Less Than 2.20	32	0.3125	0.177	0.364	0.406	0.06
		2.20 to 3.19	34		0.191			
		3.20 to 4.20	36		0.203			
		Over 4.20	38		0.215			
4L	2.5	Less Than 2.65	30	0.4375	0.299	0.490	0.490	0.10
		2.65 to 3.24	32		0.316			
		3.25 to 5.65	34		0.331			
		Over 5.65	38		0.358			
5L	3.5	Less Than 3.95	30	0.5625	0.385	0.630	0.580	0.16
		3.95 to 4.94	32		0.406			
		4.95 to 7.35	34		0.426			
		Over 7.35	38		0.461			

[a] The diameter used in calculating speed ratio and belt speed is obtained by subtracting the $2a$ value from the Effective Outside Diameter of the sheave.

Other Sheave Tolerances		
Outside Diameters	Outside Diameter Eccentricity[a]	Groove Side Wobble & Runout[a]
For outside diameters under 6.0 in. ±0.015 in. For outside diameters 6.0 to 12.0 in. ±0.020 in. For outside diameters over 12.0 in. ±0.040 in.	For outside diameters 10.0 in. and under 0.010 in. For each additional inch of outside diameter, add 0.0005 in.	For outside diameters 20.0 in. and under 0.0015 in. per inch of outside diameter. For each additional inch of outside diameter, add 0.0005 in.

[a] Total indicator reading.

All dimensions in inches.

Horsepower Ratings: The horsepower ratings for light duty V-belts can be calculated from the following formulas:

$$3L \quad HP = r\left(\frac{0.2164d^{0.91}}{r^{0.09}} - 0.2324 - 0.0001396r^2d^3\right)$$

$$4L \quad HP = r\left(\frac{0.4666d^{0.91}}{r^{0.09}} - 0.7231 - 0.0002286r^2d^3\right)$$

$$5L \quad HP = r\left(\frac{0.7748d^{0.91}}{r^{0.09}} - 1.727 - 0.0003641r^2d^3\right)$$

where $d = d_0 - 2a$; d_0 = effective outside diameter of small sheave, in.; r = rpm of the faster shaft divided by 1000. The corrected horsepower rating is obtained by dividing the horsepower rating by the combined correction factor (Table 21), which accounts for drive geometry and service factor requirements.

Table 21. Combined Correction Factors

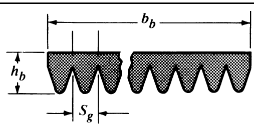
Type of Driven Unit	Speed Ratio: Less than 1.5	Speed Ratio: 1.5 and Over
Fans and blowers	1.0	0.9
Domestic laundry machines	1.1	1.0
Centrifugal pumps	1.1	1.0
Generators	1.2	1.1
Rotary compressors	1.2	1.1
Machine tools	1.3	1.2
Reciprocating pumps	1.4	1.3
Reciprocating compressors	1.4	1.3
Wood working machines	1.4	1.3

V-Ribbed Belts ANSI/RMA IP-26.—V-ribbed belts are a cross between flat belts and V-belts. The belt is basically flat with V-shaped ribs projecting from the bottom, which guide the belt and provide greater stability than that found in a flat belt. The ribs operate in grooved sheaves.

V-ribbed belts do not have the wedging action of a V-belt and thus operate at higher tensions. This design provides excellent performance in high-speed and serpentine applications, and in drives that utilize small diameter sheaves. The V-ribbed belt comes in five cross sections: H, J, K, L, and M, specified by effective length, cross section and number of ribs.

Belt Cross Sections: Nominal dimensions of the five cross sections are given in Table 22.

Table 22. Nominal Dimensions of V-Ribbed Belt Cross Sections
ANSI/RMA IP-26, 1977

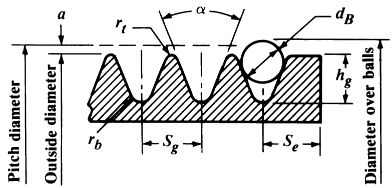


$b_b = N_r \times S_g$, where N_r = number of ribs and S_g is sheave groove spacing

Cross Section	h_b	S_g	Standard Number of Ribs
H	0.12	0.063	…
J	0.16	0.092	4, 6, 10, 16, 20
K	0.24	0.140	…
L	0.38	0.185	6, 8, 10, 12, 14, 16, 18, 20
M	0.66	0.370	6, 8, 10, 12, 14, 16, 18, 20

All dimensions in inches.

Table 23. V-Ribbed Belt Sheave and Groove Dimensions *ANSI/RMA IP-26, 1977*



Face width $= S_e(N_g - 1) + 2S_e$, where N_g is number of grooves

Cross Section	Minimum Recommended Outside Diameter	α Groove Angle ±0.25 (deg)	S_g[a]	r_t +0.005, −0.000	$2a$	r_b	h_g (min)	d_B ±0.0005	S_e
H	0.50	40	0.063 ±0.001	0.005	0.020	0.013 +0.000 −0.005	0.041	0.0469	0.080 +0.020 −0.010
J	0.80	40	0.092 ±0.001	0.008	0.030	0.015 +0.000 −0.005	0.071	0.0625	0.125 +0.030 −0.015
K	1.50	40	0.140 ±0.002	0.010	0.038	0.020 +0.000 −0.005	0.122	0.1093	0.125 +0.050 −0.000
L	3.00	40	0.185 ±0.002	0.015	0.058	0.015 +0.000 −0.005	0.183	0.1406	0.375 +0.075 −0.030
M	7.00	40	0.370 ±0.003	0.030	0.116	0.030 +0.000 −0.010	0.377	0.2812	0.500 +0.100 −0.040

[a] Summation of the deviations from S_g for all grooves in any one sheave shall not exceed ±0.010 in.

Other Sheave Tolerances[a]		
Outside Diameter	Radial Runout[b]	Axial Runout[b]
Up through 2.9 in. outside diameter ±0.010 in.	Up through 2.9 in. outside diameter 0.005 in.	0.001 in. per inch of outside diameter
Over 2.9 in. to and including 8.0 in. outside diameter ±0.020 in.	Over 2.9 in. to and including 10.0 in. outside diameter 0.010 in.	
For each additional inch of outside diameter over 8.0 in., add ±0.0025 in.	For each additional inch of outside diameter over 10.0 in., add 0.0005 in.	

[a] Variations in pitch diameter between the grooves in any one sheave must be within the following limits: Up through 2.9 in. outside diameter and up through 6 grooves, 0.002 in. (add 0.001 in. for each additional groove); over 2.9 in. to and including 19.9 in. and up through 10 grooves, 0.005 in. (add 0.0002 in. for each additional groove); over 19.9 in. and up through 10 grooves, 0.010 in. (add 0.0005 in. for each additional groove). This variation can be obtained by measuring the distance across two measuring balls or rods placed in the grooves diametrically opposite each other. Comparing this “diameter-over-balls or -rods” measurement between grooves will give the variation in pitch diameter.

[b] Total indicator reading.

All dimensions in inches

Belt Size Designation: Belt sizes are identified by a standard belt number, which consists of belt effective length to the nearest tenth of an inch, a letter designating cross section, and the number of ribs. For example, 540L6 signifies a 54.0 in. effective length, L belt, six ribs wide.

Sheave Dimensions.: Groove angles and dimensions for sheaves and face widths of sheaves for multiple belt drives are given in Table 23, along with various tolerance values.

Cross Section Selection.: Use the chart (Fig. 9) as a guide to the V-ribbed belt cross section for any combination of design horsepower and speed of the faster shaft. When the intersection of the design horsepower and speed of the faster shaft falls near a line between two areas on the chart, the possibilities in both areas should be explored. Special circumstances (such as space limitations) may lead to a choice of belt cross section different from that indicated in the chart. H and K cross sections are not included because of their specialized use. Belt manufacturers should be contacted for specific data.

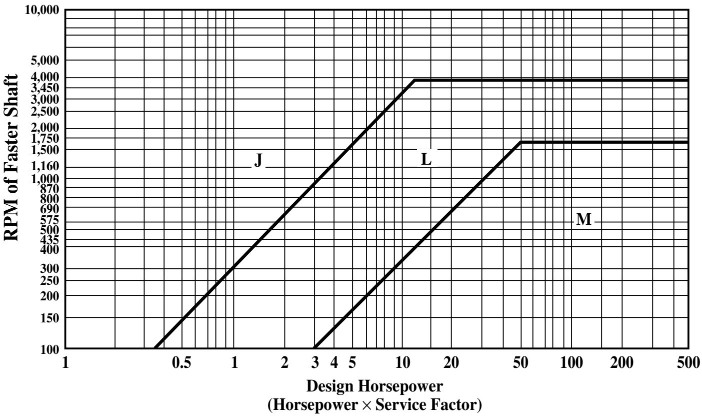


Fig. 9. Selection of V-Ribbed Belt Cross Section

Horsepower Ratings.: The horsepower rating formulas are:

$$\textbf{J:}\mathrm{HP} = d_p r\left[\frac{0.1240}{(d_p r)^{0.09}} - \frac{0.08663}{d_p} - 0.2318\times10^{-4}(d_p r)^2\right] + 0.08663 r\left[1 - \frac{1}{K_{SR}}\right]$$

$$\textbf{L:}\mathrm{HP} = d_p r\left[\frac{0.5761}{(d_p r)^{0.09}} - \frac{0.8987}{d_p} - 1.018\times10^{-4}(d_p r)^2\right] + 0.8987 r\left[1 - \frac{1}{K_{SR}}\right]$$

$$\textbf{M:}\mathrm{HP} = d_p r\left[\frac{1.975}{(d_p r)^{0.09}} - \frac{6.597}{d_p} - 3.922\times10^{-4}(d_p r)^2\right] + 6.597 r\left[1 - \frac{1}{K_{SR}}\right]$$

In these equations, d_p = pitch diameter of the small sheave, in.; r = rpm of the faster shaft divided by 1000; K_{SR} = speed ratio factor given in the accompanying table. These formulas give the maximum horsepower per rib recommended, corrected for the speed ratio. To obtain the horsepower per rib for an arc of contact other than 180 degrees, and for belts longer or shorter than the average length, multiply the horsepower obtained from these formulas by the length correction factor (Table 25) and the arc of contact correction factor (Table 26).

Table 24. V-Ribbed Belt Standard Effective Lengths *ANSI/RMA IP-26, 1977*

J Cross Section			L Cross Section			M Cross Section		
Standard Length Designation[a]	Standard Effective Length	Permissible Deviation From Standard Length	Standard Length Designation[a]	Standard Effective Length	Permissible Deviation From Standard Length	Standard Length Designation[a]	Standard Effective Length	Permissible Deviation From Standard Length
180	18.0	+0.2, –0.2	500	50.0	+0.2, –0.4	900	90.0	+0.4, –0.7
190	19.0	+0.2, –0.2	540	54.0	+0.2, –0.4	940	94.0	+0.4, –0.8
200	20.0	+0.2, –0.2	560	56.0	+0.2, –0.4	990	99.0	+0.4, –0.8
220	22.0	+0.2, –0.2	615	61.5	+0.2, –0.5	1060	106.0	+0.4, –0.8
240	24.0	+0.2, –0.2	635	63.5	+0.2, –0.5	1115	111.5	+0.4, –0.9
260	26.0	+0.2, –0.2	655	65.5	+0.2, –0.5	1150	115.0	+0.4, –0.9
280	28.0	+0.2, –0.2	675	67.5	+0.3, –0.6	1185	118.5	+0.4, –0.9
300	30.0	+0.2, –0.3	695	69.5	+0.3, –0.6	1230	123.0	+0.4, –1.0
320	32.0	+0.2, –0.3	725	72.5	+0.3, –0.6	1310	131.0	+0.5, –1.1
340	34.0	+0.2, –0.3	765	76.5	+0.3, –0.6	1390	139.0	+0.5, –1.1
360	36.0	+0.2, –0.3	780	78.0	+0.3, –0.6	1470	147.0	+0.6, –1.2
380	38.0	+0.2, –0.3	795	79.5	+0.3, –0.6	1610	161.0	+0.6, –1.2
400	40.0	+0.2, –0.4	815	81.5	+0.3, –0.7	1650	165.0	+0.6, –1.3
430	43.0	+0.2, –0.4	840	84.0	+0.3, –0.7	1760	176.0	+0.7, –1.4
460	46.0	+0.2, –0.4	865	86.5	+0.3, –0.7	1830	183.0	+0.7, –1.4
490	49.0	+0.2, –0.4	915	91.5	+0.4, –0.7	1980	198.0	+0.8, –1.6
520	52.0	+0.2, –0.4	975	97.5	+0.4, –0.8	2130	213.0	+0.8, –1.6
550	55.0	+0.2, –0.4	990	99.0	+0.4, –0.8	2410	241.0	+0.9, –1.6
580	58.0	+0.2, –0.5	1065	106.5	+0.4, –0.8	2560	256.0	+1.0, –1.8
610	61.0	+0.2, –0.5	1120	112.0	+0.4, –0.9	2710	271.0	+1.1, –2.2
650	65.0	+0.2, –0.5	1150	115.0	+0.4, –0.9	3010	301.0	+1.2, –2.4

[a] To specify belt size, use the standard length designation, followed by the letter indicating belt cross section and the number of ribs desired. For example: 865L10.
All dimensions in inches.

Table 25. Length Correction Factors

Std. Length Designation	Cross Section J	L	M	Std. Length Designation	Cross Section J	L	M
180	0.83	…	…	1230	…	1.08	0.94
200	0.85	…	…	1310	…	1.10	0.96
240	0.89	…	…	1470	…	1.12	0.098
280	0.92	…	…	1610	…	1.14	1.00
320	0.95	…	…	1830	…	1.17	1.03
360	0.98	…	…	1980	…	1.19	1.05
400	1.00	…	…	2130	…	1.21	1.06
440	1.02	…	…	2410	…	1.24	1.09
500	1.05	0.89	…	2710	…	…	1.12
550	1.07	0.91	…	3010	…	…	1.14
610	1.09	0.93	…	3310	…	…	1.16
690	1.12	0.96	…	3610	…	…	1.18
780	1.16	0.98	…	3910	…	…	1.20
910	1.18	1.02	0.88	4210	…	…	1.22
940	1.19	1.02	0.89	4810	…	…	1.25
990	1.20	1.04	0.90	5410	…	…	1.28
1060	…	1.05	0.91	6000	…	…	1.30
1150	…	1.07	0.93	…	…	…	…

Table 26. Arc of Contact Correction Factors

$\frac{D_o - d_o}{C}$	Arc of Contact, θ, on Small Sheave, (deg)	Correction Factor
0.00	180	1.00
0.10	174	0.98
0.20	169	0.97
0.30	163	0.95
0.40	157	0.94
0.50	151	0.92
0.60	145	0.90
0.70	139	0.88
0.80	133	0.85
0.90	127	0.83
1.00	120	0.80
1.10	113	0.77
1.20	106	0.74
1.30	99	0.71
1.40	91	0.67
1.50	83	0.63

Number of Ribs: The number of ribs required for an application is obtained by dividing the design horsepower by the corrected horsepower rating for one rib.

Arc of contact on the small sheave may be determined by the following formulas:

Exact Formula:

$$\text{Arc of Contact (deg)} = 2\cos^{-1}\left(\frac{D_o - d_o}{2C}\right)$$

Approximate Formula:

$$\text{Arc of Contact (deg)} = 180 - \frac{(D_o - d_o)60}{C}$$

where D_o = Effective outside diameter of large sheave, in; d_o = Effective outside diameter of small sheave, in; and, C = Center distance, inch.

Speed Ratio Correction Factors

Speed Ratio[a]	K_{SR}
1.00 to and incl. 1.10	1.0000
Over 1.01 to and incl. 1.04	1.0136
Over 1.04 to and incl. 1.08	1.0276
Over 1.08 to and incl. 1.12	1.0419
Over 1.12 to and incl. 1.18	1.0567
Over 1.18 to and incl. 1.24	1.0719
Over 1.24 to and incl. 1.34	1.0875
Over 1.34 to and incl. 1.51	1.1036
Over 1.51 to and incl. 1.99	1.1202
Over 1.99	1.1373

[a] D_p/d_p, where D_p (d_p) is the pitch diameter of the large (small) sheave.

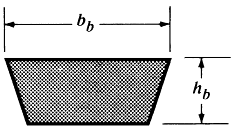
Variable Speed Belts ANSI, RMA IP-25.—For drives that require more speed variation than can be obtained with conventional industrial V-belts, standard-line variable-speed drives are available. These drives use special wide, thin belts. Package units of standard-line variable-speed belts and sheaves, combined with the motor and output gearbox are available in ranges from approximately ½ through 100 horsepower.

The speed ranges of variable-speed drives can be much greater than those drives using classical V-belts. Speed ranges up to 10:1 can be obtained on lower horsepower units.

This section covers 12 variable speed belt cross sections and sheave groove sizes designed 1422V, 1922V, 2322V 1926V, 2926V, 3226V, 2530V, 3230V, 4430V, 4036V, 4436V, and 4836V. The industry supplies many other sizes that are not listed in this section.

Belt Cross Sections and Lengths: Nominal dimensions of the 12 cross sections are given in Table 27, and lengths in Table 28.

Table 27. Normal Variable-Speed Belt Dimensions *ANSI/RMA IP-25, 1982*



Cross Section	b_b	h_b	h_b/b_b	Cross Section	b_b	h_b	h_b/b_b
1422V	0.88	0.31	0.35	2530V	1.56	0.59	0.38
1922V	1.19	0.38	0.32	3230V	2.00	0.62	0.31
2322V	1.44	0.44	0.31	4430V	2.75	0.69	0.25
1926V	1.19	0.44	0.37	4036V	2.50	0.69	0.28
2926V	1.81	0.50	0.28	4436V	2.75	0.72	0.26
3226V	2.0	0.53	0.27	4836V	3.00	0.75	0.25

All dimensions in inches.

Table 28. Variable-Speed V-Belt Standard Belt Lengths *ANSI/RMA IP-25, 1982*

Standard Pitch Length Designation	Standard Effective Lengths												Permissible Deviations From Standard Length
	Cross Section												
	1422V	1922V	2322V	1926V	2926V	3226V	2530V	3230V	4430V	4036V	4436V	4836V	
315	32.1	…	…	…	…	…	…	…	…	…	…	…	±0.7
335	34.1	…	…	…	…	…	…	…	…	…	…	…	±0.7
355	36.1	36.2	…	36.3	…	…	…	…	…	…	…	…	±0.7
375	38.1	38.2	…	38.3	…	…	…	…	…	…	…	…	±0.7
400	40.6	40.7	40.8	40.8	…	…	…	…	…	…	…	…	±0.7
425	43.1	43.2	43.3	43.3	…	…	…	…	…	…	…	…	±0.8
450	45.6	45.7	45.8	45.8	…	…	…	…	…	…	…	…	±0.8
475	48.1	48.2	48.3	48.3	…	…	…	…	…	…	…	…	±0.8
500	50.6	50.7	50.8	50.8	…	…	50.9	…	…	…	…	…	±0.8
530	53.6	53.7	53.8	53.8	53.9	…	53.9	…	…	…	…	…	±0.8
560	56.6	56.7	56.8	56.8	56.9	56.9	56.9	57.1	57.3	57.3	57.3	57.4	±0.9
600	60.6	60.7	60.8	60.8	60.9	60.9	60.9	61.1	61.3	61.3	61.3	61.4	±0.9
630	63.6	63.7	63.8	63.8	63.9	63.9	63.9	64.1	64.3	64.3	64.3	64.4	±0.9
670	67.6	67.7	67.8	67.8	67.9	67.9	67.9	68.1	68.3	68.3	68.3	68.4	±0.9
710	71.6	71.7	71.8	71.8	71.9	71.9	71.9	72.1	72.3	72.3	72.3	72.4	±0.9
750	75.6	75.7	75.8	75.8	75.9	75.9	75.9	76.1	76.3	76.3	76.3	76.4	±1.0
800	…	80.7	80.8	80.8	80.9	80.9	80.9	81.1	81.3	81.3	81.3	81.4	±1.0
850	…	85.7	85.8	85.8	85.9	85.9	85.9	86.1	86.3	86.3	86.3	86.4	±1.1
900	…	90.7	90.8	90.8	90.9	90.9	90.9	91.1	91.3	91.3	91.3	91.4	±1.1
950	…	95.7	95.8	95.8	95.9	95.9	95.9	96.1	96.3	96.3	96.3	96.4	±1.1
1000	…	100.7	100.8	100.8	100.9	100.9	100.9	101.1	101.3	101.3	101.3	101.4	±1.2
1060	…	106.7	106.8	106.8	106.9	106.9	106.9	107.1	107.3	107.3	107.3	107.4	±1.2
1120	…	112.7	112.8	112.8	112.9	112.9	112.9	113.1	113.3	113.3	113.3	113.4	±1.2
1180	…	118.7	118.8	118.8	118.9	118.9	118.9	119.1	119.3	119.3	119.3	119.4	±1.3
1250	…	…	…	…	125.9	125.9	125.9	126.1	126.3	126.3	126.3	126.4	±1.3
1320	…	…	…	…	…	132.9	…	133.1	133.3	133.3	133.3	133.4	±1.3

All dimensions in inches.

The lengths given in this table are not necessarily available from all manufacturers. Availability should be investigated prior to design commitment.

Table 29. Variable-Speed Sheave and Groove Dimensions

Cross Section	Standard Groove Dimensions: Variable: α Groove Angle ±0.67 (deg)	b_g[a] Closed +0.000 −0.030	b_{go} Open Max	h_{gv} Min	S_g ±0.03	Companion: α Groove Angle ±0.33 (deg)	b_g ±0.010	h_g Min	S_g ±0.03	Drive Design Factors: Min. Recomm. Pitch Diameter	$2a$	$2av$ Max	CL Min
1422V	22	0.875	1.63	2.33	1.82	22	0.875	0.500	1.82	2.0	0.20	3.88	0.08
1922V	22	1.188	2.23	3.14	2.42	22	1.188	0.562	2.42	3.0	0.22	5.36	0.08
2322V	22	1.438	2.71	3.78	2.89	22	1.438	0.625	2.89	3.5	0.25	6.52	0.08
1926V	26	1.188	2.17	2.65	2.36	26	1.188	0.625	2.36	3.0	0.25	4.26	0.08
2926V	26	1.812	3.39	4.00	3.58	26	1.812	0.750	3.58	3.5	0.30	6.84	0.08
3226V	26	2.000	3.75	4.41	3.96	26	2.000	0.781	3.96	4.0	0.30	7.60	0.08
2530V	30	1.562	2.81	3.01	2.98	30	1.562	0.844	2.98	4.0	0.30	4.64	0.10
3230V	30	2.000	3.67	3.83	3.85	30	2.000	0.875	3.85	4.5	0.35	6.22	0.10
4430V	30	2.750	5.13	5.23	5.38	30	2.750	0.938	5.38	5.0	0.40	8.88	0.10
4036V	36	2.500	4.55	3.95	4.80	36	2.500	0.938	4.80	4.5	0.40	6.32	0.10
4436V	36	2.750	5.03	4.33	5.30	36	2.750	0.969	5.30	5.0	0.40	7.02	0.10
4836V	36	3.000	5.51	4.72	5.76	36	3.000	1.000	5.76	6.0	0.45	7.74	0.10

[a] The effective width (b_e), a reference dimension, is the same as the ideal top width of closed variable-speed sheave (b_g) and the ideal top width of the companion sheave (b_g).

Other Sheave Tolerances: Outside Diameter		Radial Runout[a]		Axial Runout[a]	
Up through 4.0 in. outside diameter	±0.020 in.	Up through 10.0 in. outside diameter	0.010 in.	Up through 5.0 in. outside diameter	0.005 in.
For each additional inch of outside diameter add ±0.005 in.		For each additional inch of outside diameter add 0.0005 in.		For each additional inch of outside diameter add 0.001 in.	

[a] Total indicator reading.

Surface Finish: Machined Surface Area	Max Surface Roughness Height, R_a (AA) (μ in.)	Machined Surface Area	Max Surface Roughness Height, R_a (AA) (μ in.)
V-Sheave groove sidewalls	125	Straight bores with 0.002 in. or less total tolerance	125
Rim edges and ID, Hub ends and OD	500	Taper and straight bores with total tolerance over 0.002 in.	250

All dimensions in inches, except where noted.

Belt Size Designation: Variable-speed belt sizes are identified by a standard belt number. The first two digits denote the belt top width in sixteenths of an inch; the third and fourth digits indicate the angle of the groove in which the belt is designed to operate. The letter V (for variable) follows the first four digits. The digits after the V indicate the pitch length to the nearest 0.1 in. For example, 1422V450 is a belt of ⅞ in. ($^{14}/_{16}$ in.) nominal top width designed to operate in a sheave of 22 degree groove angle and having a pitch length of 45.0 in.

Sheave Groove Data: A variable speed sheave is an assembly of movable parts, designed to permit one or both flanges of the sheave to be moved axially causing a radial movement of the variable speed belt in the sheave groove. This radial movement permits stepless speed variation within the physical limits of the sheave and the belt. A companion sheave may be a solid sheave having a constant diameter and groove profile or another variable sheave. Variable speed sheave designs should conform to the dimensions in Table 29 and Fig. 10. The included angle of the sheaves, top width, and clearance are boundary dimensions. Groove angles and dimensions of companion sheaves should conform to Table 29 and Fig. 11. Various tolerance values are also given in Table 29.

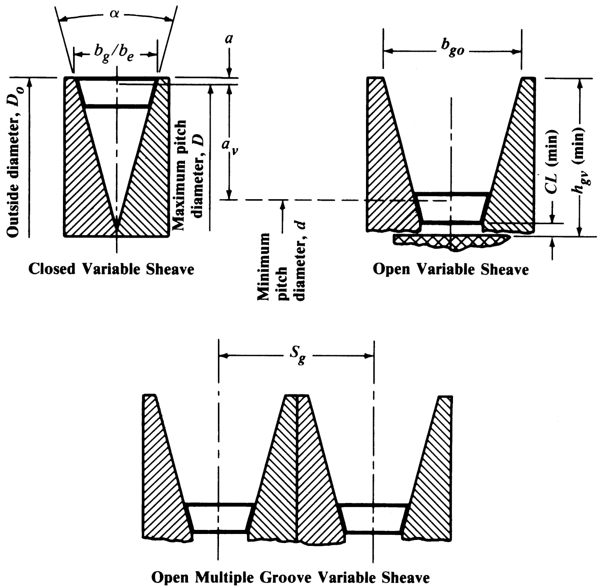


Fig. 10. Variable Sheaves

Variable-Speed Drive Design: Variable-speed belts are designed to operate in sheaves that are an assembly of movable parts. The sheave design permits one or both flanges of the sheave to be moved axially, causing a radial movement of the variable-speed belt in the sheave groove. The result is a stepless speed variation within the physical limits of the sheave and the variable-speed belt. Therefore, besides transmitting power, variable-speed belt drives provide speed variation.

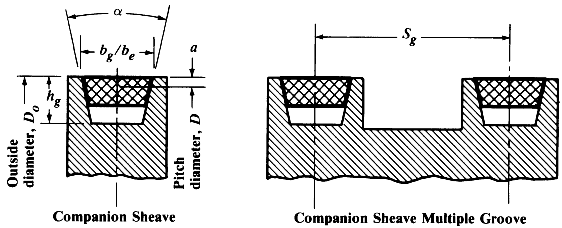


Fig. 11. Companion Sheaves

The factors that determine the amount of pitch diameter change on variable-speed sheaves are belt top width, belt thickness, and sheave angle. This pitch diameter change, combined with the selected operating pitch diameters for a sheave, determines the possible speed variation.

The range of output speeds from a variable-speed sheave drive is established by the companion sheave and is a function of the ratio of the pitch diameter of the companion sheave to the maximum and minimum pitch diameters of the variable sheave. Speed variation is usually obtained by varying the center distance between the two sheaves. This type of drive seldom exceeds a speed variation of 3:1.

For a single variable-speed sheave drive, the speed variation

$$\text{Speed variation} = \frac{\text{PD Max}}{\text{PD Min}} \text{ (of variable sheave)}$$

For a dual variable-speed sheave drive, which is frequently referred to as a compound drive because both sheaves are variable, the speed variation is

$$\text{Speed variation} = \frac{DR(DN)}{dr(dn)}$$

where DR = Max driver PD

DN = Max driven PD

dr = Min driver PD

dn = Min driven PD

With this design, the center distance is generally fixed and speed variation is usually accomplished by mechanically altering the pitch diameter of one sheave. In this type of drive, the other sheave is spring loaded to make an opposite change in the pitch diameter and to provide the correct belt tension. Speed variations of up to 10: 1 are common on this type of drive.

Speed Ratio Adjustment: All speed ratio changes must be made while the drives are running. Attempting to make adjustments while the unit is stopped creates unnecessary and possibly destructive forces on both the belt and sheaves. In stationary control drives, the belt tension should be released to allow the flanges to adjust without belt force interference.

Cross Section Selection: Selection of a variable speed belt cross section is based on the drive design horsepower and speed variation. Table 29 shows the maximum pitch diameter variation ($2av$) that each cross section can attain.

Horsepower Ratings: The general horsepower formulas for variable-speed belts are:

$$1422\text{V HP} = d_p r\left[0.4907(d_p r)^{-0.09} - \frac{0.8378}{d_p} - 0.000337(d_p r)^2\right] + 0.8378 r\left(1 - \frac{1}{K_{SR}}\right)$$

$$1922\,VHP = d_p r\left[0.8502(d_p r)^{-0.09} - \frac{1.453}{d_p} - 0.000538(d_p r)^2\right] + 1.453r\left(1 - \frac{1}{K_{SR}}\right)$$

$$2322\,VHP = d_p r\left[1.189(d_p r)^{-0.09} - \frac{2.356}{d_p} - 0.000777(d_p r)^2\right] + 2.356r\left(1 - \frac{1}{K_{SR}}\right)$$

$$1926\,VHP = d_p r\left[1.046(d_p r)^{-0.09} - \frac{1.833}{d_p} - 0.000589(d_p r)^2\right] + 1.833r\left(1 - \frac{1}{K_{SR}}\right)$$

$$2926\,VHP = d_p r\left[1.769(d_p r)^{-0.09} - \frac{4.189}{d_p} - 0.001059(d_p r)^2\right] + 4.189r\left(1 - \frac{1}{K_{SR}}\right)$$

$$3226\,VHP = d_p r\left[2.073(d_p r)^{-0.09} - \frac{5.236}{d_p} - 0.001217(d_p r)^2\right] + 5.236r\left(1 - \frac{1}{K_{SR}}\right)$$

$$2530\,VHP = d_p r\left[2.395(d_p r)^{-0.09} - \frac{6.912}{d_p} - 0.001148(d_p r)^2\right] + 6.912r\left(1 - \frac{1}{K_{SR}}\right)$$

$$3230\,VHP = d_p r\left[2.806(d_p r)^{-0.09} - \frac{7.854}{d_p} - 0.001520(d_p r)^2\right] + 7.854r\left(1 - \frac{1}{K_{SR}}\right)$$

$$4430\,VHP = d_p r\left[3.454(d_p r)^{-0.09} - \frac{7.854}{d_p} - 0.002196(d_p r)^2\right] + 9.818r\left(1 - \frac{1}{K_{SR}}\right)$$

$$4036\,VHP = d_p r\left[3.566(d_p r)^{-0.09} - \frac{9.687}{d_p} - 0.002060(d_p r)^2\right] + 9.687r\left(1 - \frac{1}{K_{SR}}\right)$$

$$4436\,VHP = d_p r\left[4.041(d_p r)^{-0.09} - \frac{11.519}{d_p} - 0.002297(d_p r)^2\right] + 11.519r\left(1 - \frac{1}{K_{SR}}\right)$$

$$4836\,VHP = d_p r\left[4.564(d_p r)^{-0.09} - \frac{13.614}{d_p} - 0.002634(d_p r)^2\right] + 13.614r\left(1 - \frac{1}{K_{SR}}\right)$$

In these equations, d_p = pitch diameter of small sheave, in.; r = rpm of faster shaft divided by 1000; K_{SR} = speed ratio factor given in the accompanying table. These formulas give the basic horsepower rating, corrected for the speed ratio. To obtain the horsepower for arcs of contact other than 180 degrees and for belts longer or shorter than average length, multiply the horsepower obtained from these formulas by the arc of contact correction factor (Table 30) and the length correction factor (Table 31).

Table 30. Arc of Contact Correction Factors

$\frac{D-d}{C}$	Arc of Contact, θ, on Small Sheave, (deg)	Correction Factor	$\frac{D-d}{C}$	Arc of Contact, θ, on Small Sheave, (deg)	Correction Factor
0.00	180	1.00	0.80	0.80	0.87
0.10	174	0.99	0.90	0.90	0.85
0.20	169	0.97	1.00	1.00	0.82
0.30	163	0.96	1.10	1.10	0.80
0.40	157	0.94	1.20	1.20	0.77
0.50	151	0.93	1.30	1.30	0.73
0.60	145	0.91	1.40	1.40	0.70
0.70	139	0.89	1.50	1.50	0.65

Table 31. Length Correction Factors

Standard Pitch Length Designation	Cross Section											
	1422V	1922V	2322V	1926V	2926V	3226V	2530V	3230V	4430V	4036V	4436V	4836V
315	0.93	…	…	…	…	…	…	…	…	…	…	…
335	0.94	…	…	…	…	…	…	…	…	…	…	…
355	0.95	0.90	…	0.90	…	…	…	…	…	…	…	…
375	0.96	0.91	…	0.91	…	…	…	…	…	…	…	…
400	0.97	0.92	0.90	0.92	…	…	…	…	…	…	…	…
425	0.98	0.93	0.91	0.93	…	…	…	…	…	…	…	…
450	0.99	0.94	0.92	0.94	…	…	…	…	…	…	…	…
475	1.00	0.95	0.93	0.95	…	…	…	…	…	…	…	…
500	1.01	0.95	0.94	0.95	…	…	0.90	…	…	…	…	…
530	1.02	0.96	0.95	0.96	0.92	…	0.92	…	…	…	…	…
560	1.03	0.97	0.96	0.97	0.93	0.92	0.93	0.91	0.90	0.91	0.91	0.92
600	1.04	0.98	0.97	0.98	0.94	0.93	0.94	0.93	0.92	0.93	0.92	0.93
630	1.05	0.99	0.98	0.99	0.95	0.94	0.95	0.94	0.93	0.94	0.93	0.94
670	1.06	1.00	0.99	1.00	0.97	0.95	0.96	0.95	0.94	0.95	0.95	0.95
710	1.07	1.01	1.00	1.01	0.98	0.96	0.98	0.96	0.96	0.96	0.96	0.96
750	1.08	1.02	1.01	1.02	0.99	0.98	0.99	0.97	0.97	0.97	0.97	0.98
800	…	1.03	1.02	1.03	1.00	0.99	1.00	0.99	0.99	0.99	0.99	0.99
850	…	1.04	1.03	1.04	1.01	1.00	1.01	1.00	1.00	1.00	1.00	1.00
900	…	1.05	1.04	1.05	1.02	1.01	1.02	1.01	1.01	1.01	1.01	1.01
950	…	1.06	1.05	1.06	1.03	1.02	1.04	1.02	1.03	1.02	1.02	1.02
1000	…	1.07	1.06	1.07	1.04	1.03	1.05	1.03	1.04	1.03	1.04	1.03
1060	…	1.08	1.07	1.07	1.06	1.04	1.06	1.05	1.06	1.05	1.05	1.04
1120	…	1.09	1.08	1.08	1.07	1.06	1.07	1.06	1.07	1.06	1.06	1.06
1180	…	1.09	1.09	1.09	1.08	1.07	1.08	1.07	1.08	1.07	1.07	1.07
1250	…	…	…	…	1.09	1.08	1.10	1.08	1.10	1.08	1.09	1.08
1320	…	…	…	…	…	1.09	…	1.09	1.11	1.09	1.10	1.09

Rim Speed: The material and design selected for sheaves must be capable of withstanding the high rim speeds that may occur in variable-speed drives. The rim speed is calculated as follows: Rim speed (fpm) = $(\pi/12)(D_o)$ (rpm).

60 Degree V-Belts.—60 degree V-belts are ideal for compact drives. Their 60 degree angle and ribbed top are specifically designed for long life on small diameter sheaves. These belts offer extremely smooth operation at high speeds (in excess of 10,000 rpm) and can be used on drives with high speed ratios. They are available in 3M, 5M, 7M, and 11M (3, 5, 7, 11 mm) cross sections (top widths) and are commonly found in the joined configuration, which provides extra stability and improved performance. They are specified by cross section and nominal length; for example, a 5M315 designation indicates a belt having a 5 mm cross section and an effective length of 315 mm.

Speed Ratio Correction Factors

Speed Ratio[a]	K_{SR}	Speed Ratio[a]	K_{SR}
1.00–1.01	1.0000	1.19–1.24	1.0719
1.02–1.04	1.0136	1.25–1.34	1.0875
1.05–1.08	1.0276	1.35–1.51	1.1036
1.09–1.12	1.0419	1.52–1.99	1.1202
1.13–1.18	1.0567	2.0 and over	1.1373

[a] D_p/d_p, where D_p (d_p) is the pitch diameter of the large (small) sheave.

Arc of contact on the small sheave may be determined by the formulas:

Exact Formula: $$\text{Arc of Contact (deg)} = 2\cos^{-1}\left(\frac{D-d}{2C}\right)$$

Approximate Formula:

$$\text{Arc of Contact (deg)} = 180 - \frac{(D-d)60}{C}$$

where D = Pitch diameter of large sheave or flat pulley, inch

d = Pitch diameter of small sheave, inch

C = Center distance, inch

Industry standards have not yet been published for 60 degree V-belts. Therefore, belt manufacturers should be contacted for specific applications, specifications, and additional information.

Synchronous Belts ANSI/RMA IP-24.—Synchronous belts are also known as timing or positive-drive belts. These belts have evenly spaced teeth on their surfaces, which mesh with teeth on pulleys or sprockets to produce a positive, no-slip transmission of power. Such designs should not be confused with molded notched V-belts, which transmit power by means of the wedging action of the V-shape.

Synchronous belts are used where driven shaft speeds must be synchronized to the rotation of the driver shaft and to eliminate the noise and maintenance problems of chain drives.

Standard Timing Belts: Conventional trapezoidal, or rectangular tooth, timing belts come in six cross sections, which relate to the pitch of the belt. Pitch is the distance from center to center of the teeth. The six basic cross sections or pitches are MXL (mini extra light), XL (extra light), L (light), H (heavy), XH (extra heavy), and XXH (double extra heavy) (Fig. 12). Belts are specified by pitch length, cross section (pitch), and width.

Double-sided timing belts have identical teeth on both sides of the belt and are used where synchronization is required from each belt face. They are available in XL, L, and H cross sections.

Size Designations: Synchronous belt sizes are identified by a standard number. The first digits specify the belt length to 0.1 in. followed by the belt section (pitch) designation. The digits following the belt section designation represent the nominal belt width times 100. For example, an L section belt 30.000 in. pitch length and 0.75 in. in width would be specified as a 300L075 synchronous belt.

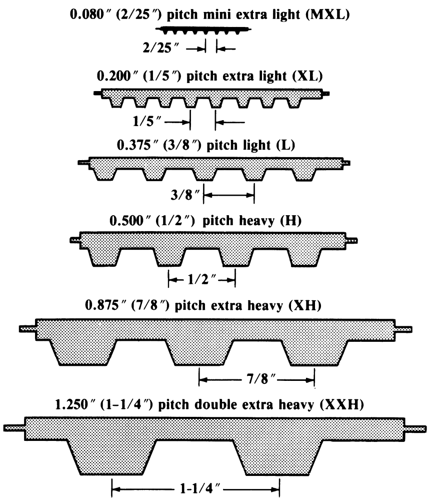


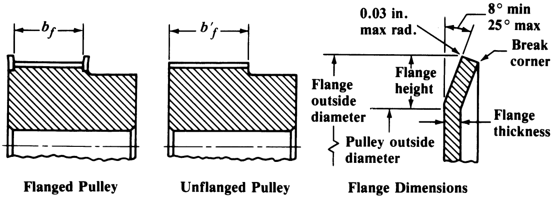
Fig. 12. Standard Synchronous Belt Sections

The RMA nomenclature for double-sided belts is the same as for single-sided belts with the addition of the prefix "D" in front of the belt section. However, some manufacturers use their own designation system for double-sided belts.

Standard Sections: Belt sections are specified in terms of pitch. Table 33 gives the Standard Belt Sections and their corresponding pitches.

Pitch Lengths: Standard belt pitch lengths, belt length designations, and numbers of teeth are shown in Table 34. Belt length tolerances are also given in this table; these tolerances apply to all belt sections and represent the total manufacturing tolerance on belt length.

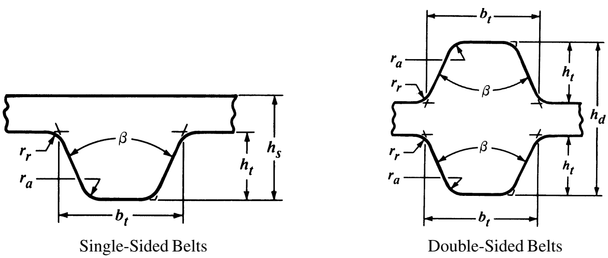
Table 32. Synchronous Belt Standard Pulley and Flange Dimensions
ANSI/RMA IP-24, 1983



Belt Section	Standard Nominal Pulley Width	Standard Pulley Width Designation	Minimum Pulley Width: Flanged b_f	Minimum Pulley Width: Unflanged b'_f	Flange: Thickness (min)	Flange: Height[a] (min)
MXL	0.25	025	0.28	0.35	0.023	0.020
XL	0.38	037	0.41	0.48	0.029	0.040
L	0.50	050	0.55	0.67	0.050	0.065
	0.75	075	0.80	0.92		
	1.00	100	1.05	1.17		
H	1.00	100	1.05	1.23	0.050	0.080
	1.50	150	1.55	1.73		
	2.00	200	2.08	2.26		
	3.00	300	3.11	3.29		
XH	2.00	200	2.23	2.46	0.098	0.190
	3.00	300	3.30	3.50		
	4.00	400	4.36	4.59		
XXH	2.00	200	2.23	2.52	0.127	0.245
	3.00	300	3.30	3.59		
	4.00	400	4.36	4.65		
	5.00	500	5.42	5.72		

Nominal Tooth Dimensions: Table 33 shows the nominal tooth dimensions for each of the standard belt sections. Tooth dimensions for single- and double-sided belts are identical.

Table 33. Synchronous Belt Nominal Tooth and Section Dimensions
ANSI/RMP IP-24, 1983



Belt Section (Pitch)	β Tooth Angle(deg)	h_t	b_t	r_a	r_r	h_s	h_d
MXL (0.080)	40	0.020	0.045	0.005	0.005	0.045	…
XL (0.200)	50	0.050	0.101	0.015	0.015	0.090	…
L (0.375)	40	0.075	0.183	0.020	0.020	0.14	…
H (0.500)	40	0.090	0.241	0.040	0.040	0.16	…
XH (0.875)	40	0.250	0.495	0.047	0.062	0.44	…
XXH (1.250)	40	0.375	0.750	0.060	0.090	0.62	…
DXL (0.200)	50	0.050	0.101	0.015	0.015	…	0.120
DL (0.375)	40	0.075	0.183	0.020	0.020	…	0.180
DH (0.500)	40	0.090	0.241	0.040	0.040	…	0.234

All dimensions in inches.

Table 34. Synchronous Belt Standard Pitch Lengths and Tolerances *ANSI/RMA IP-24, 1983*

Belt Length Designation	Pitch Length	Permissible Deviation From Standard Length	Number of Teeth for Standard Lengths MXL (0.080)	XL (0.200)	L (0.375)	H (0.500)	XH (0.875)	XXH (1.250)	Belt Length Designation	Pitch Length	Permissible Deviation From Standard Length	Number of Teeth for Standard Lengths MXL (0.080)	XL (0.200)	L (0.375)	H (0.500)	XH (0.875)	XXH (1.250)
36	3.600	±0.016	45						230	23.000	±0.024	…	115	…	…		
40	4.000	±0.016	50						240	24.000	±0.024	…	120	64	48		
44	4.400	±0.016	55						250	25.000	±0.024	…	125	…	…		
48	4.800	±0.016	60						255	25.500	±0.024	…	…	68	…		
56	5.600	±0.016	70						260	26.000	±0.024	…	130	…	…		
60	6.000	±0.016	75	30					270	27.000	±0.024	…	…	72	54		
64	6.400	±0.016	80	…					285	28.500	±0.024	…	…	76	…		
70	7.000	±0.016	…	35					300	30.000	±0.024	…	…	80	60		
72	7.200	±0.016	90	…					322	32.250	±0.026	…	…	86	…		
80	8.000	±0.016	100	40					330	33.000	±0.026	…	…	…	66		
88	8.800	±0.016	110	…					345	34.500	±0.026			92	…		
90	9.000	±0.016	…	45					360	36.000	±0.026			…	72		
100	10.000	±0.016	125	50					367	36.750	±0.026			98	…		
110	11.000	±0.018	…	55					390	39.000	±0.026			104	78		
112	11.200	±0.018	140	…					420	42.000	±0.030			112	84		
120	12.000	±0.018	…	60	…				450	45.000	±0.030			120	90	…	
124	12.375	±0.018	…	…	33				480	48.000	±0.030			128	96	…	
124	12.400	±0.018	155	…	…				507	50.750	±0.032			…	…	58	
130	13.000	±0.018	…	65	…				510	51.000	±0.032			136	102	…	
140	14.000	±0.018	175	70	…				540	54.000	±0.032			144	108	…	
150	15.000	±0.018	…	75	40				560	56.000	±0.032			…	…	64	
160	16.000	±0.020	200	80	…				570	57.000	±0.032			…	114	…	
170	17.000	±0.020	…	85	…				600	60.000	±0.032			160	120	…	
180	18.000	±0.020	225	90	…				630	63.000	±0.034			…	126	72	
187	18.750	±0.020	…	…	50				660	66.000	±0.034			…	132	…	
190	19.000	±0.020	…	95	…				700	70.000	±0.034				140	80	56
200	20.000	±0.020	250	100	…				750	75.000	±0.036				150	…	…
210	21.000	±0.024	…	105	56				770	77.000	±0.036				…	88	…
220	22.000	±0.024	…	110	…				800	80.000	±0.036				160	…	64
225	22.500	±0.024	…	…	60				840	84.000	±0.038				…	96	…

All dimensions in inches.

Widths.: Standard belt widths, width designations, and width tolerances are shown in Table 35.

Table 35. Synchronous Belt Standard Widths and Tolerances
ANSI/RMA IP-24, 1983

Belt Section	Standard Belt Widths		Tolerances on Width for Belt Pitch Lengths		
	Designation	Dimensions	Up to and including 33 in.	Over 33 in. up to and including 66 in.	Over 66 in.
MXL (0.080)	012 019 025	0.12 0.19 0.25	+0.02 −0.03	…	…
XL (0.200)	025 037	0.25 0.38	+0.02 −0.03	…	…
L (0.375)	050 075 100	0.50 0.75 1.00	+0.03 −0.03	+0.03 −0.05	…
H (0.500)	075 100 150	0.75 1.00 1.50	+0.03 −0.03	+0.03 −0.05	+0.03 −0.05
	200	2.00	+0.03 −0.05	+0.05 −0.05	+0.05 −0.06
	300	3.00	+0.05 −0.06	+0.06 −0.06	+0.06 −0.08
XH (0.875)	200 300 400	2.00 3.00 4.00	…	+0.19 −0.19	+0.19 −0.19
XXH (1.250)	200 300 400 500	2.00 3.00 4.00 5.00	…	…	+0.19 −0.19

Length Determination.: The pitch length of a synchronous belt is determined by placing the belt on a measuring fixture having two pulleys of equal diameter, a method of applying force, and a means of measuring the center distance between the two pulleys. The position of one of the two pulleys is fixed and the other is movable along a graduated scale.

Synchronous Belt Pulley Diameters: Table 36 lists the standard pulley diameters by belt section (pitch). Fig. 13 defines the pitch, pitch diameter, outside diameter and pitch line differential.

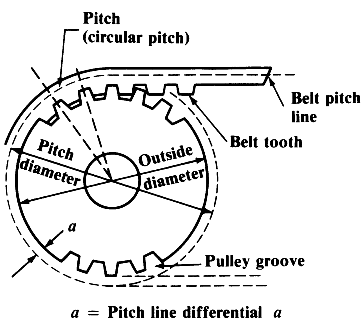


Fig. 13. Synchronous Belt Pulley Dimensions

Table 36. Synchronous Belt Standard Pulley Diameters *ANSI/RMA IP-24, 1983*

Number of Grooves	Belt Section											
	MXL (0.080)		XL (0.200)		L (0.375)		H (0.500)		XH (0.875)		XXH (1.250)	
	Diameters		Diameters		Diameters		Diameters		Diameters		Diameters	
	Pitch	Outside	Pitch	Outside	Pitch	Outside	Pitch	Outside	Pitch	Outside	Pitch	Outside
10	0.255	0.235	0.637	0.617	1.194[a]	1.164	…	…	…	…	…	…
12	0.306	0.286	0.764	0.744	1.432[a]	1.402	…	…	…	…	…	…
14	0.357	0.337	0.891	0.871	1.671	1.641	2.228[a]	2.174	…	…	…	…
16	0.407	0.387	1.019	0.999	1.910	1.880	2.546	2.492	…	…	…	…
18	0.458	0.438	1.146	1.126	2.149	2.119	2.865	2.811	5.013	4.903	7.162	7.042
20	0.509	0.489	1.273	1.253	2.387	2.357	3.183	3.129	5.570	5.460	7.958	7.838
22	0.560	0.540	1.401	1.381	2.626	2.596	3.501	3.447	6.127	6.017	8.754	8.634
24	0.611	0.591	1.528	1.508	2.865	2.835	3.820	3.766	6.685	6.575	9.549	9.429
26	0.662	0.642	…	…	3.104	3.074	4.138	4.084	7.242	7.132	10.345	10.225
28	0.713	0.693	1.783	1.763	3.342	3.312	4.456	4.402	7.799	7.689	…	…
30	0.764	0.744	1.910	1.890	3.581	3.551	4.775	4.721	8.356	8.246	11.937	11.817
32	0.815	0.795	2.037	2.017	3.820	3.790	5.093	5.039	8.913	8.803	…	…
34	0.866	0.846	…	…	…	…	…	…	…	…	13.528	13.408
36	0.917	0.897	2.292	2.272	4.297	4.267	5.730	5.676	…	…	…	…
40	1.019	0.999	2.546	2.526	4.775	4.745	6.366	6.312	11.141	11.031	15.915	15.795
42	1.070	1.050	2.674	2.654	…	…	…	…	…	…	…	…
44	1.120	1.100	2.801	2.781	5.252	5.222	7.003	6.949	…	…	…	…
48	1.222	1.202	3.056	3.036	5.730	5.700	7.639	7.585	13.369	13.259	19.099	18.979
60	1.528	1.508	3.820	3.800	7.162	7.132	9.549	9.495	16.711	16.601	23.873	23.753
72	1.833	1.813	4.584	4.564	8.594	8.564	11.459	11.405	20.054	19.944	28.648	28.528
84	…	…	…	…	10.027	9.997	13.369	13.315	23.396	23.286	…	…
90	…	…	…	…	…	…	…	…	…	…	35.810	35.690
96	…	…	…	…	…	…	15.279	15.225	26.738	26.628	…	…
120	…	…	…	…	…	…	19.099	19.045	33.423	33.313	…	…

All dimensions in inches.

* Usually not available in all widths — consult supplier.

Widths: Standard pulley widths for each belt section are shown in Table 32. The nominal pulley width is specified in terms of the maximum standard belt width the pulley will accommodate. The minimum pulley width, whether flanged or unflanged, is also shown in Table 32, along with flange dimensions and various pulley tolerances.

Pulley Size Designation: Synchronous belt pulleys are designated by the number of grooves, the belt section, and a number representing 100 times the nominal width. For example, a 30 groove L section pulley with a nominal width of 0.75 in. would be designated by 30L075. Pulley tolerances are shown in Table 37.

Table 37. Pulley Tolerances (All Sections)

Outside Diameter Range	Outside Diameter Tolerance	Pitch to Pitch Tolerance: Adjacent Grooves	Pitch to Pitch Tolerance: Accumulative Over 90 Degrees
Up thru 1.000	+0.002 −0.000	±0.001	±0.003
Over 1.000 to and including 2.000	+0.003 −0.000	±0.001	±0.004
Over 2.000 to and including 4.000	+0.004 −0.000	±0.001	±0.005
Over 4.000 to and including 7.000	+0.005 −0.000	±0.001	±0.005
Over 7.000 to and including 12.000	+0.006 −0.000	±0.001	±0.006
Over 12.000 to and including 20.000	+0.007 −0.000	±0.001	±0.007
Over 20.000	+0.008 −0.000	±0.001	±0.008

Radial Runout[a]	Axial Runout[b]
For outside diameters 8.0 in. and under 0.005 in. For each additional inch of outside diameter add 0.0005 in.	For outside diameters 1.0 in. and under 0.001 in. For each additional inch of outside diameter up through 10.0 in., add 0.001 in. For each additional inch of outside diameter over 10.0 in., add 0.0005 in.

[a] Flange outside diameter equals pulley outside diameter plus twice flange height.
[b] Total indicator reading.
All dimensions in inches.

Cross Section Selection: The chart (Fig. 14) may be used as a guide to the selection of a synchronous belt for any combination of design horsepower and speed of the faster shaft. When the intersection of the design horsepower and speed of the faster shaft falls near a line between two areas on the chart, the possibilities in both areas should be explored. Special circumstances (such as space limitations) may result in selection of a belt cross section different from that indicated in the chart. Belt manufacturers should be contacted for specific data.

Torque Ratings: It is customary to use torque load requirements rather than horsepower load when designing drives using the small pitch MXL section belts. These belts operate on small diameters resulting in relatively low belt speeds, so torque is essentially constant for all rpm. The torque rating formulas for MXL sections are:

$$Q_r = d[1.13 - 1.38\times10^{-3}d^2] \text{ for belt width} = 0.12 \text{ in.}$$

$$Q_r = d[1.88 - 2.30\times10^{-3}d^2] \text{ for belt width} = 0.19 \text{ in.}$$

$$Q_r = d[2.63 - 3.21\times10^{-3}d^2] \text{ for belt width} = 0.25 \text{ in.}$$

where Q_r = the maximum torque rating (lbf-in.) for a belt of specified width having six or more teeth in mesh and a pulley surface speed of 6500 fpm or less. Torque ratings for drives with less than six teeth in mesh must be corrected as shown in Table 38. d = pitch diameter of smaller pulley, inch.

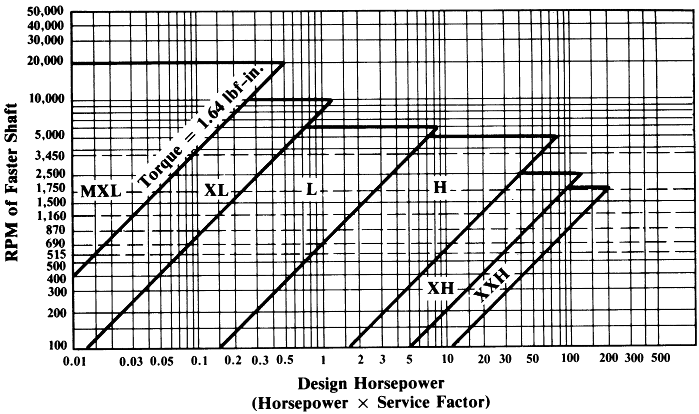


Fig. 14. Selection of Synchronous Belt Cross Section

Table 38. Teeth in Mesh Factor

Teeth in Mesh	Factor K_z	Teeth in Mesh	Factor K_z
6 or more	1.00	3	0.40
5	0.80	2	0.20
4	0.60		

Horsepower Rating Formulas: The horsepower rating formulas for synchronous belts, other than the MLX section, are determined from the following formulas, where the number in parentheses is the belt width in inches.

$$\mathrm{XL}(0.38)\mathrm{HP} = dr[0.0916 - 7.07\times10^{-5}(dr)^2]$$

$$\mathrm{L}(1.00)\mathrm{HP} = dr[0.436 - 3.01\times10^{-4}(dr)^2]$$

$$\mathrm{H}(3.00)\mathrm{HP} = dr[3.73 - 1.41\times10^{-3}(dr)^2]$$

$$\mathrm{XH}(4.00)\mathrm{HP} = dr[7.21 - 4.68\times10^{-3}(dr)^2]$$

$$\mathrm{XXH}(5.00)\mathrm{HP} = dr[11.4 - 7.81\times10^{-3}(dr)^2]$$

where HP = the maximum horsepower rating recommended for the specified standard belt width having six or more teeth in mesh and a pulley surface speed of 6500 fpm or less. Horsepower ratings for drives with less than six teeth in mesh must be corrected as shown in Table 38. d = pitch diameter of smaller pulley, in. r = rpm of faster shaft divided by 1000. Total horsepower ratings are the same for double-sided as for single-sided belts. Contact manufacturers for percentage of horsepower available for each side of the belt.

Finding the Required Belt Width: The belt width should not exceed the small pulley diameter or excessive side thrust will result.

Torque Rating Method (MXL Section): Divide the design torque by the teeth in mesh factor to obtain the corrected design torque. Compare the corrected design torque with the torque rating given in Table 39 for the pulley diameter being considered. Select the narrowest belt width that has a torque rating equal to or greater than the corrected design torque.

Table 39. Torque Rating for MXL Section (0.080 in. Pitch)

Belt Width, (in.)	Rated Torque (lbf-in.) for Small Pulley (Number of Grooves and Pitch Diameter, in.)									
	10MXL 0.255	12MXL 0.306	14MXL 0.357	16MXL 0.407	18MXL 0.458	20MXL 0.509	22MXL 0.560	24MXL 0.611	28MXL 0.713	30MXL 0.764
0.12	0.29	0.35	0.40	0.46	0.52	0.57	0.63	0.69	0.81	0.86
0.19	0.48	0.58	0.67	0.77	0.86	0.96	1.05	1.15	1.34	1.44
0.25	0.67	0.80	0.94	1.07	1.20	1.34	1.47	1.61	1.87	2.01

Horsepower Rating Method (XL, L, H, XH, and XXH Sections): Multiply the horsepower rating for the widest standard belt of the selected section by the teeth in mesh factor to obtain the corrected horsepower rating. Divide the design horsepower by the corrected horsepower rating to obtain the required belt width factor. Compare the required belt width factor with those shown in Table 40. Select the narrowest belt width that has a width factor equal to or greater than the required belt width factor.

Table 40. Belt Width Factor

Belt Section	Belt Width (in.)											
	0.12	0.19	0.25	0.38	0.50	0.75	1.00	1.50	2.00	3.00	4.00	5.00
MXL (0.080)	0.43	0.73	1.00									
XL (0.200)			0.62	1.00								
L (0.375)					0.45	0.72	1.00					
H (0.500)						0.21	0.29	0.45	0.63	1.00		
XH (0.875)									0.45	0.72	1.00	
XXH (1.250)									0.35	0.56	0.78	1.00

Drive Selection: Information on design and selection of synchronous belt drives is available in engineering manuals published by belt manufacturers. Manufacturers should be consulted on such matters as preferred stock sizes, desirable speeds, center distances, etc.

Minimum Pulley Size: The recommended minimum pulley size depends on the rpm of the faster shaft. Minimum sheave diameters for each cross-section belt are listed in Table 36.

Selection of Flanged Pulleys: To determine when to use flanged pulleys, consider the following conditions:

1) On all two-pulley drives, the minimum flanging requirements are two flanges on one pulley, or one flange on each pulley on opposite sides.

2) On drives where the center distance is more than eight times the diameter of the small pulley, both pulleys should be flanged on both sides.

3) On vertical shaft drives, one pulley should be flanged on both sides and other pulleys in the system should be flanged on the bottom side only.

4) On drives with more than two pulleys, the minimum flanging requirements are two flanges on every other pulley, or one flange on every pulley, alternating sides around the system.

Service Factors: Service factors for V-belts are listed in Table 41 and for synchronous belts in Table 42.

Belt Storage and Handling.—To achieve maximum belt performance, proper belt storage procedures should always be practiced. If belts are not stored properly, their performance can be adversely affected. Four key rules are:

1) Do not store belts on floors unless they are protected by appropriate packaging.

2) Do not store belts near windows where the belts may be exposed to direct sunlight or moisture.

3) Do not store belts near electrical devices that may generate ozone (transformers, electric motors, etc.).

4) Do not store belts in areas where solvents or chemicals may be present in the atmosphere.

Table 41. Service Factors for V-Belts

Driving Unit	*AC Motors:* Normal Torque, Squirrel Cage, Synchronous and Split Phase. *DC Motors:* Shunt Wound. *Engines:* Multiple Cylinder Internal Combustion.		
Types of Driven Machines	Intermittent Service (3–5 hours daily or seasonal)	Normal Service (8–10 hours daily)	Continuous Service (16–24 hours daily)
Agitators for liquids; Blowers and exhausters; Centrifugal pumps & compressors; Fans up to 10 horsepower; Light duty conveyors	1.1	1.2	1.3
Belt conveyors for sand, grain, etc.; Dough mixers; Fans over 10 horsepower; Generators; Line shafts; Laundry machinery; Machine tools; Punches, presses, shears; Printing machinery; Positive displacement rotary pumps; Revolving and vibrating screens	1.2	1.3	1.4
Brick machinery; Bucket elevators; Exciters; Piston compressors; Conveyors (drag, pan, screw); Hammer mills; Paper mill beaters; Piston pumps; Positive displacement blowers; Pulverizers; Saw mill and woodworking machinery; Textile machinery	1.4	1.5	1.6
Crushers (gyratory, jaw, roll); Mills (ball, rod, tube); Hoists; Rubber calendars, extruders, mills	1.5	1.6	1.8

Driving Unit	*AC Motors:* High Torque, High Slip, Repulsion-Induction, Single Phase, Series Wound, Slip Ring. *DC Motors:* Series Wound, Compound Wound. *Engines:* Single Cylinder Internal Combustion. *Line Shafts*, *Clutches*		
Types of Driven Machines	Intermittent Service (3–5 hours daily or seasonal)	Normal Service (8–10 hours daily)	Continuous Service (16–24 hours daily)
Agitators for liquids; Blowers and exhausters; Centrifugal pumps & compressors; Fans up to 10 horsepower; Light duty conveyors	1.1	1.2	1.3
Belt conveyors for sand, grain, etc.; Dough mixers; Fans over 10 horsepower; Generators; Line shafts; Laundry machinery; Machine tools; Punches, presses, shears; Printing machinery; Positive displacement rotary pumps; Revolving and vibrating screens	1.2	1.3	1.4
Brick machinery; Bucket elevators; Exciters; Piston compressors; Conveyors (drag, pan, screw); Hammer mills; Paper mill beaters; Piston pumps; Positive displacement blowers; Pulverizers; Saw mill and woodworking machinery; Textile machinery	1.4	1.5	1.6
Crushers (gyratory, jaw, roll); Mills (ball, rod, tube); Hoists; Rubber calendars, extruders, mills	1.5	1.6	1.8

The machines listed above are representative samples only. Select the group listed above whose load characteristics most closely approximate those of the machine being considered.

Belts should be stored in a cool, dry environment. When stacked on shelves, the stacks should be short enough to avoid excess weight on the bottom belts, which may cause distortion. When stored in containers, the container size and contents should be sufficiently limited to avoid distortion.

V-Belts: A common method is to hang the belts on pegs or pin racks. Very long belts stored this way should use sufficiently large pins or crescent shaped "saddles" to prevent their weight from causing distortion.

Joined V-belts, Synchronous Belts, V-Ribbed Belts: Like V-belts, these belts may be stored on pins or saddles with precautions taken to avoid distortion. However, belts of this type up to approximately 120 in. are normally shipped in a "nested" configuration and should be stored in the same manner. Nests are formed by laying a belt on its side on a flat surface and placing as many belts inside the first belt as possible without undue force. When the nests are tight and are stacked with each rotated 180° from the one below, they may be stacked without damage.

Belts of this type over 120 in. may be "rolled up" and tied for shipment. These rolls may be stacked for easy storage. Care should be taken to avoid small bend radii which could damage the belts.

Table 42. Service Factors for Synchronous Belt Drives

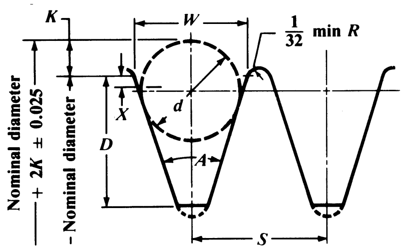
Driving Units	*AC Motors:* Normal Torque, Squirrel Cage, Synchronous and Split Phase. *DC Motors:* Shunt Wound. *Engines:* Multiple Cylinder Internal Combustion.		
Types of Driven Machines	Intermittent Service (3–5 hours daily or seasonal)	Normal Service (8–10 hours daily)	Continuous Service (16–24 hours daily)
Display, Dispensing, Projection, Medical equipment; Instrumentation; Measuring devices	1.0	1.2	1.4
Appliances, sweepers, sewing machines; Office equipment; Wood lathes, band saws	1.2	1.4	1.6
Conveyors: belt, light package, oven, screens, drums, conical	1.3	1.5	1.7
Agitators for liquids; Dough mixers; Drill presses, lathes; Screw machines, jointers; Circular saws, planes; Laundry, Paper, Printing machinery	1.4	1.6	1.8
Agitators for semiliquids; Brick machinery (except pug mills); Conveyor belt: ore, coal, sand; Line shafts; Machine tools: grinder, shaper, boring mill, milling machines; Pumps: centrifugal, gear, rotary	1.5	1.7	1.9
Conveyor: apron, pan, bucket, elevator; Extractors, washers; Fans, blowers; centifugal, induced draft exhausters; Generators & exciters; Hoists, elevators; Rubber calenders, mills, extruders; Saw mill, Textile machinery inc. looms, spinning frames, twisters	1.6	1.8	2.0
Centrifuges; Conveyors: flight, screw; Hammer mills; Paper pulpers	1.7	1.9	2.1
Brick & clay pug mills; Fans, blowers, propeller mine fans, positive blowers	1.8	2.0	2.2
Driving Units	*AC Motors:* High Torque, High Slip, Repulsion-Induction, Single Phase Series Wound and Slip Ring. *DC Motors:* Series Wound and Compound Wound. *Engines:* Single Cylinder Internal Combustion. *Line Shafts*. *Clutches*.		
Types of Driven Machines	Intermittent Service (3–5 hours daily or seasonal)	Normal Service (8–10 hours daily)	Continuous Service (16–24 hours daily)
Display, Dispensing, Projection, Medical equipment; Instrumentation; Measuring devices	1.2	1.4	1.6
Appliances, sweepers, sewing machines; Office equipment; Wood lathes, band saws	1.4	1.6	1.8
Conveyors: belt, light package, oven, screens, drums, conical	1.5	1.7	1.9
Agitators for liquids; Dough mixers; Drill presses, lathes; Screw machines, jointers; Circular saws, planes; Laundry, Paper, Printing machinery	1.6	1.8	2.0
Agitators for semiliquids; Brick machinery (except pug mills); Conveyor belt: ore, coal, sand; Line shafts; Machine tools:grinder, shaper, boring mill, milling machines; Pumps: centrifugal, gear, rotary	1.7	1.9	2.1
Conveyor: apron, pan, bucket, elevator; Extractors, washers; Fans, blowers; centifugal, induced draft exhausters; Generators & exciters; Hoists, elevators; Rubber calenders, mills, extruders; Saw mill, Textile machinery inc. looms, spinning frames, twisters	1.8	2.0	2.2
Centrifuges; Conveyors: flight, screw; Hammer mills; Paper pulpers	1.9	2.1	2.3
Brick & clay pug mills; Fans, blowers, propeller mine fans, positive blowers	2.0	2.2	2.4

Synchronous belts will not slip, and therefore must be belted for the highest loadings anticipated in the system. A minimum service factor of 2.0 is recommended for equipment subject to chocking.

Variable Speed Belts: Variable speed belts are more sensitive to distortion than most other belts, and should not be hung from pins or racks but stored on shelves in the sleeves in which they are shipped.

SAE Standard V-Belts.—The data for V-belts and pulleys shown in Table 43 cover nine sizes, three of which — 0.250, 0.315, and 0.440 — were added in 1977 to conform to existing practice. This standard was reaffirmed in 1987.

Table 43. SAE V-Belt and Pulley Dimensions



SAE Size	Recommended Min. Eff Dia[a]	*A* Groove Angle (deg) ±0.5	*W* Eff. Groove Width	*D* Groove Depth Min	*d* Ball or Rod Dia (±0.0005)	2*K* 2 × Ball Extension	2*X*[b]	*S* Groove[c] Spacing (±0.015)
0.250	2.25	36	0.248	0.276	0.2188	0.164	0.04	0.315
0.315	2.25	36	0.315	0.354	0.2812	0.222	0.05	0.413
0.380	2.40	36	0.380	0.433	0.3125	0.154	0.06	0.541
0.440	2.75	36	0.441	0.512	0.3750	0.231	0.07	0.591
0.500	3.00	36	0.500	0.551	0.4375	0.314	0.08	0.661
11⁄16	3.00	34	0.597	0.551	0.500	0.258	0.00	0.778
	Over 4.00	36				0.280		
	Over 6.00	38				0.302		
¾	3.00	34	0.660	0.630	0.5625	0.328	0.02	0.841
	Over 4.00	36				0.352		
	Over 6.00	38				0.374		
⅞	3.50	34	0.785	0.709	0.6875	0.472	0.04	0.966
	Over 4.50	36				0.496		
	Over 6.00	38				0.520		
1	4.00	34	0.910	0.827	0.8125	0.616	0.06	1.091
	Over 6.00	36				0.642		
	Over 8.00	38				0.666		

[a] Pulley effective diameters below those recommended should be used with caution, because power transmission and belt life may be reduced.

[b] The *X* dimension is radial; 2*X* is to be subtracted from the effective diameter to obtain "pitch diameter" for speed ratio calculations.

[c] These values are intended for adjacent grooves of the same effective width (*W*). Choice of pulley manufacture or belt design parameter may justify variance from these values. The *S* dimension should be the same on all multiple groove pulleys in a drive using matched belts.

All dimensions in inches.

V-belts are produced in a variety of constructions in a basic trapezoidal shape and are to be dimensioned in such a way that they are functional in pulleys dimensioned as described in the standard. Standard belt lengths are in increments of ½ inch up to and including 80 inches. Standard lengths above 80 inches up to and including 100 inches are in increments of 1 inch, without fractions. Standard belt length tolerances are based on the center distance and are as follows: For belt lengths of 50 inches or less, ± 0.12 inch; over 50 to 60 inches, inclusive, ± 0.16 inch; over 60 to 80 inches, inclusive, ± 0.19; and over 80 to 100 inches, inclusive, ± 0.22.

TRANSMISSION CHAINS

Types of Chains

In addition to the standard roller and inverted tooth types, a wide variety of drive chains of different construction is available. Such chains are manufactured to various degrees of precision ranging from unfinished castings or forgings to chains having certain machined parts. Practically all of these chains as well as standard roller chains can be equipped with attachments to fit them for conveyor use. A few such types are briefly described in the following paragraphs. Detailed information about them can be obtained from the manufacturers.

Types of chains.—*Detachable Chains:* The links of this type of chain, which are identical, are easily detachable. Each has a hook-shaped end in which the bar of the adjacent link articulates. These chains are available in malleable iron or pressed steel. The chief advantage is the ease with which any link can be removed.

Cast Roller Chains: Cast roller chains are constructed, wholly or partly, of cast metal parts and are available in various styles. In general the rollers and side bars are accurately made castings without machine finish. The links are usually connected by means of forged pins secured by nuts or cotters. Such chains are used for slow speeds and moderate loads, or where the precision of standard roller chains is not required.

Pintle Chains: Unlike the roller chain, the pintle chain is composed of hollow-cored cylinders cast or forged integrally with two offset side bars and each link identical. The links are joined by pins inserted in holes in the ends of the side bars and through the cored holes in the adjacent links. Lugs prevent turning of the pins in the side bars ensuring articulation of the chain between the pin and the cored cylinder.

Standard Roller Transmission Chains

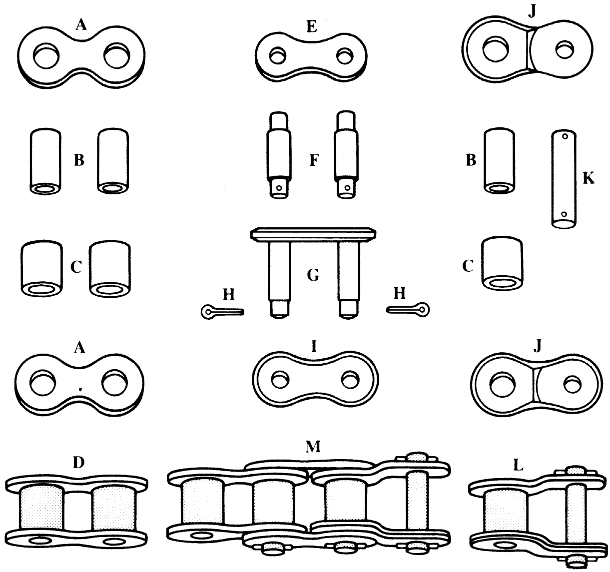
A roller chain is made up of two kinds of links: roller links and pin links alternately spaced throughout the length of the chain as shown in Table 1.

Roller chains are manufactured in several types, each designed for the particular service required. All roller chains are so constructed that the rollers are evenly spaced throughout the chain. The outstanding advantage of this type of chain is the ability of the rollers to rotate when contacting the teeth of the sprocket. Two arrangements of roller chains are in common use: the single-strand type and the multiple-strand type. In the latter type, two or more chains are joined side by side by means of common pins which maintain the alignment of the rollers in the different strands.

Types of roller chains.—*Standard roller chains* are manufactured to the specifications in the American National Standard for precision power transmission roller chains, attachments, and sprockets ANSI/ASME B29.1M-1993 and, where indicated, the data in the subsequent tables have been taken from this standard. These roller chains and sprockets are commonly used for the transmission of power in industrial machinery, machine tools, motor trucks, motorcycles, tractors, and similar applications. In tabulating the dimensional information in ANSI/ASME B29.1M, customary inch-pound units were used. Metric (SI) units are given in separate tabulations in the Standard.

Nonstandard roller chains, developed individually by various manufacturers prior to the adoption of the ANSI standard, are similar in form and construction to standard roller chains but do not conform dimensionally to standard chains. Some sizes are still available from the originating manufacturers for replacement on existing equipment. They are not recommended for new installations, since their manufacture is being discontinued as rapidly as possible.

Table 1. ANSI Nomenclature for Roller Chain Parts *ANSI/ASME B29.1M-1993*



Roller Link D. — An inside link consisting of two inside plates, two bushings, and two rollers.

Pin Link G and E. — An outside link consisting of two pin-link plates assembled with two pins.

Inside Plate A. — One of the plates forming the tension members of a roller link.

Pin Link Plate E. — One of the plates forming the tension members of a pin link.

Pin F. — A stud articulating within a bushing of an inside link and secured at its ends by the pin-link plates.

Bushing B. — A cylindrical bearing in which the pin turns.

Roller C. — A ring or thimble which turns over a bushing.

Assembled Pins G. — Two pins assembled with one pin-link plate.

Connecting-Link G and I. — A pin link having one side plate detachable.

Connecting-Link Plate I. — The detachable pin-link plate belonging to a connecting link. It is retained by cotter pins or by a one-piece spring clip (not shown).

Connecting Link Assembly M. — A unit designed to connect two roller links.

Offset Link L. — A link consisting of two offset plates assembled with a bushing and roller at one end and an offset link pin at the other.

Offset Plate J. — One of the plates forming the tension members of the offset link.

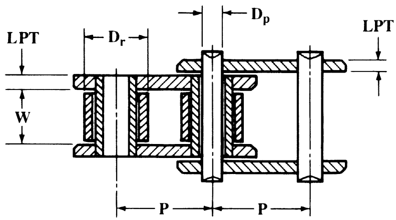
Offset Link Pin K. — A pin used in offset links.

Standard double-pitch roller chains are like standard roller chains, except that their link plates have twice the pitch of the corresponding standard-pitch chain. Their design conforms to specifications in the ANSI Standard for double-pitch power transmission roller chains and sprockets ANSI/ASME B29.3M-1994. They are especially useful for low speeds, moderate loads, or long center distances.

Transmission Roller Chain

Standard Roller Chain Nomenclature, Dimensions and Loads.—Standard nomenclature for roller chain parts are given in Table 1. Dimensions for Standard Series roller chain are given in Table 2.

Table 2. ANSI Roller Chain Dimensions *ASME/ANSI B29.1M-1986*



Pitch P	Max. Roller Diameter D_r	Standard Series					Heavy Series
		Standard Chain No.	Width W	Pin Diameter D_p	Thickness of Link Plates *LPT*	Measuring Load,† Lb.	Thickness of Link Plates *LPT*
0.250	[a]0.130	25	0.125	0.0905	0.030	18	…
0.375	[a]0.200	35	0.188	0.141	0.050	18	…
0.500	0.306	41	0.250	0.141	0.050	18	…
0.500	0.312	40	0.312	0.156	0.060	31	…
0.625	0.400	50	0.375	0.200	0.080	49	…
0.750	0.469	60	0.500	0.234	0.094	70	0.125
1.000	0.625	80	0.625	0.312	0.125	125	0.156
1.250	0.750	100	0.750	0.375	0.156	195	0.187
1.500	0.875	120	1.000	0.437	0.187	281	0.219
1.750	1.000	140	1.000	0.500	0.219	383	0.250
2.000	1.125	160	1.250	0.562	0.250	500	0.281
2.250	1.406	180	1.406	0.687	0.281	633	0.312
2.500	1.562	200	1.500	0.781	0.312	781	0.375
3.000	1.875	240	1.875	0.937	0.375	1000	0.500

[a] Bushing diameter. This size chain has no rollers.

All dimensions are in inches.

Roller Diameters D_r are approximately $\frac{5}{8}P$.

The *width W* is defined as the distance between the link plates. It is approximately $\frac{5}{8}$ of the chain pitch.

Pin Diameters D_p are approximately $\frac{5}{16}P$ or $\frac{1}{2}$ of the roller diameter.

Thickness LPT of Inside and Outside Link Plates for the standard series is approximately $\frac{1}{8}P$.

Thickness of Link Plates for the heavy series of any pitch is approximately that of the next larger pitch Standard Series chain.

Maximum Height of Roller Link Plates = 0.95 *P*.
Maximum Height of Pin Link Plates = 0.82 *P*.
Maximum Pin Diameter = nominal pin diameter + 0.0005 inch.
Minimum Hole in Bushing = nominal pin diameter + 0.0015 inch.
Maximum Width of Roller Link = nominal width of chain + (2.12 × nominal link plate thickness.)
Minimum Distance between Pin Link Plates = maximum width of roller link + 0.002 inch.

Chain Pitch: Distance in inches between centers of adjacent joint members. Other dimensions are proportional to the pitch.

Tolerances for Chain Length: New chains, under standard measuring load, must not be underlength. Overlength tolerance is $0.001/(\text{pitch in inches})^2 + 0.015$ inch per foot. Length measurements are to be taken over a length of at least 12 inches.

Measuring Load: The load in pounds under which a chain should be measured for length. It is equal to one per cent of the ultimate tensile strength, with a minimum of 18 pounds and a maximum of 1000 pounds for both single and multiple-strand chain.

Minimum Ultimate Tensile Strength: For single-strand chain, equal to or greater than $12{,}500 \times (\text{pitch in inches})^2$ pounds. The minimum tensile strength or breaking strength of a multiple-strand chain is equal to that of a single-strand chain multiplied by the number of strands. Minimum ultimate tensile strength is indicative only of the tensile strength quality of the chain, not the maximum load that can be applied.

Standard Roller Chain Numbers.—The right-hand figure in the chain number is zero for roller chains of the usual proportions, 1 for a lightweight chain, and 5 for a rollerless bushing chain. The numbers to the left of the right-hand figure denote the number of ⅛ inches in the pitch. The letter *H* following the chain number denotes the heavy series; thus the number 80 *H* denotes a 1-inch pitch heavy chain. The hyphenated number 2 suffixed to the chain number denotes a double strand, 3 a triple strand, 4 a quadruple strand chain and so on.

Heavy Series: These chains, made in ¾-inch and larger pitches, have thicker link plates than those of the regular standard. Their value is only in the acceptance of higher loads at lower speeds.

Light-weight Machinery Chain: This chain is designated as No. 41. It is ½ inch pitch; ¼ inch wide; has 0.306-inch diameter rollers and a 0.141-inch pin diameter. The minimum ultimate tensile strength is 1500 pounds.

Multiple-strand Chain: This is essentially an assembly of two or more single-strand chains placed side by side with pins that extend through the entire width to maintain alignment of the different strands.

Types of Sprockets.—Four different designs or types of roller-chain sprockets are shown by the sectional views, Fig. 1. Type *A* is a plain plate; type *B* has a hub on one side only; type *C*, a hub on both sides; and type *D*, a detachable hub. Also used are shear pin and slip clutch sprockets designed to prevent damage to the drive or to other equipment caused by overloads or stalling.

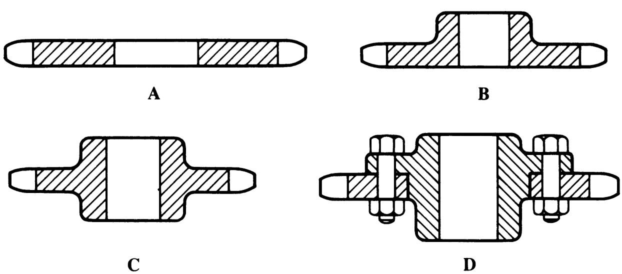
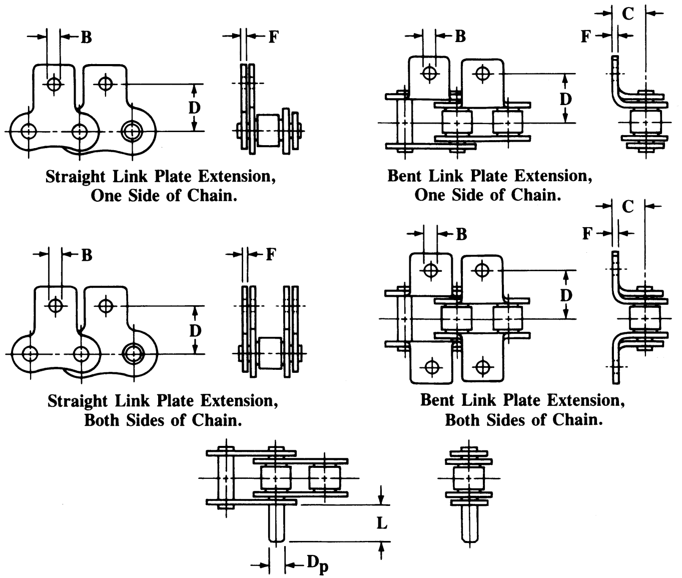


Fig. 1. Types of Sprockets

Attachments.—Modifications to standard chain components to adapt the chain for use in conveying, elevating, and timing operations are known as "attachments." The components commonly modified are: 1) the link plates, which are provided with extended lugs which may be straight or bent ; and 2) the chain pins, which are extended in length so as to project substantially beyond the outer surface of the pin link plates.

Hole diameters, thicknesses, hole locations and offset dimensions for straight link and bent link plate extensions and lengths and diameters of extended pins are given in Table 3.

Table 3. Straight and Bent Link Plate Extensions and Extended Pin Dimensions
ANSI/ASME B29.1M-1993



Straight Link Plate Extension, One Side of Chain.

Bent Link Plate Extension, One Side of Chain.

Straight Link Plate Extension, Both Sides of Chain.

Bent Link Plate Extension, Both Sides of Chain.

Extended Pin, One Side of Chain.

Chain No.	Straight Link Plate Extension			Bent Link Plate Extension				Extended Pin	
	B min.	*D*	*F*	*B* min.	*C*	*D*	*F*	D_p Nominal	*L*
35	0.102	0.375	0.050	0.102	0.250	0.375	0.050	0.141	0.375
40	0.131	0.500	0.060	0.131	0.312	0.500	0.060	0.156	0.375
50	0.200	0.625	0.080	0.200	0.406	0.625	0.080	0.200	0.469
60	0.200	0.719	0.094	0.200	0.469	0.750	0.094	0.234	0.562
80	0.261	0.969	0.125	0.261	0.625	1.000	0.125	0.312	0.750
100	0.323	1.250	0.156	0.323	0.781	1.250	0.156	0.375	0.938
120	0.386	1.438	0.188	0.386	0.906	1.500	0.188	0.437	1.125
140	0.448	1.750	0.219	0.448	1.125	1.750	0.219	0.500	1.312
160	0.516	2.000	0.250	0.516	1.250	2.000	0.250	0.562	1.500
200	0.641	2.500	0.312	0.641	1.688	2.500	0.312	0.781	1.875

All dimensions are in inches.

Sprocket Classes.—The American National Standard ANSI/ASME B29.1M-1993 provides for two classes of sprockets designated as Commercial and Precision. The selection of either is a matter of drive application judgment. The usual moderate to slow speed commercial drive is adequately served by Commercial sprockets. Where extreme high speed in combination with high load is involved, or where the drive involves fixed centers, critical timing, or register problems, or close clearance with outside interference, then the use of Precision sprockets may be more appropriate.

As a general guide, drives requiring Type A or Type B lubrication (see page 2443) would be served by Commercial sprockets. Drives requiring Type C lubrication may require Precision sprockets; the manufacturer should be consulted.

Keys, Keyways, and Set Screws.—To secure sprockets to the shaft, both keys and set screws should be used. The key is used to prevent rotation of the sprocket on the shaft. Keys should be fitted carefully in the shaft and sprocket keyways to eliminate all backlash, especially on the fluctuating loads. A set screw should be located over a flat key to secure it against longitudinal displacement.

Where a set screw is to be used with a parallel key, the following sizes are recommended by the American Chain Association. For a sprocket bore and shaft diameter in the range of

$\frac{1}{2}$ through $\frac{7}{8}$ inch, a $\frac{1}{4}$-inch set screw

$\frac{15}{16}$ through $1\frac{3}{4}$ inches, a $\frac{3}{8}$-inch set screw

$1\frac{13}{16}$ through $2\frac{1}{4}$ inches, a $\frac{1}{2}$-inch set screw

$2\frac{5}{16}$ through $3\frac{1}{4}$ inches, a $\frac{5}{8}$-inch set screw

$3\frac{3}{8}$ through $4\frac{1}{2}$ inches, a $\frac{3}{4}$-inch set screw

$4\frac{3}{4}$ through $5\frac{1}{2}$ inches, a $\frac{7}{8}$-inch set screw

$5\frac{3}{4}$ through $7\frac{3}{8}$ inches, a 1-inch set screw

$7\frac{1}{2}$ through $12\frac{1}{2}$ inches, a $1\frac{1}{4}$-inch set screw

Sprocket Diameters.—The various diameters of roller chain sprockets are shown in Fig. 2. These are defined as follows.

Pitch Diameter: The pitch diameter is the diameter of the pitch circle that passes through the centers of the link pins as the chain is wrapped on the sprocket.

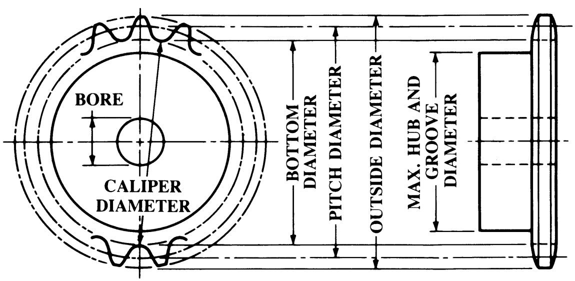


Fig. 2. Sprocket Diameters

Because the chain pitch is measured on a straight line between the centers of adjacent pins, the chain pitch lines form a series of chords of the sprocket pitch circle. Sprocket pitch diameters for one-inch pitch and for 9 to 108 teeth are given in Table 4. For lower (5 to 8) or higher (109 to 200) numbers of teeth use the following formula in which P = pitch, N = number of teeth: Pitch Diameter = $P \div \sin(180° \div N)$.

Table 4. ANSI Roller Chain Sprocket Diameters *ANSI/ASME B29.1M-1993*

These diameters and caliper factors apply only to chain of 1-inch pitch. For any other pitch, multiply the values given below by the pitch.

Caliper Dia. (even teeth) = Pitch Diameter − Roller Dia.

Caliper Dia. (odd teeth) = Caliper factor × Pitch − Roller Dia.

See Table 5 for tolerances on Caliper Diameters.

No. Teeth[a]	Pitch Diameter	Outside Diameter		Caliper Factor	No. Teeth[a]	Pitch Diameter	Outside Diameter		Caliper Factor
		Turned	Topping Hob Cut				Turned	Topping Hob Cut	
9	2.9238	3.348	3.364	2.8794	59	18.7892	19.363	19.361	18.7825
10	3.2361	3.678	3.676		60	19.1073	19.681	19.680	
11	3.5495	4.006	3.990	3.5133	61	19.4255	20.000	19.998	19.4190
12	3.8637	4.332	4.352		62	19.7437	20.318	20.316	
13	4.1786	4.657	4.666	4.1481	63	20.0618	20.637	20.634	20.0556
14	4.4940	4.981	4.982		64	20.3800	20.956	20.952	
15	4.8097	5.304	5.298	4.7834	65	20.6982	21.274	21.270	20.6921
16	5.1258	5.627	5.614		66	21.0164	21.593	21.588	
17	5.4422	5.949	5.930	5.4190	67	21.3346	21.911	21.907	21.3287
18	5.7588	6.271	6.292		68	21.6528	22.230	22.225	
19	6.0755	6.593	6.609	6.0548	69	21.9710	22.548	22.543	21.9653
20	6.3924	6.914	6.926		70	22.2892	22.867	22.861	
21	6.7095	7.235	7.243	6.6907	71	22.6074	23.185	23.179	22.6018
22	7.0267	7.555	7.560		72	22.9256	23.504	23.498	
23	7.3439	7.876	7.877	7.3268	73	23.2438	23.822	23.816	23.2384
24	7.6613	8.196	8.195		74	23.5620	24.141	24.134	
25	7.9787	8.516	8.512	7.9630	75	23.8802	24.459	24.452	23.8750
26	8.2962	8.836	8.829		76	24.1984	24.778	24.770	
27	8.6138	9.156	9.147	8.5992	77	24.5166	25.096	25.089	24.5116
28	8.9314	9.475	9.465		78	24.8349	25.415	25.407	
29	9.2491	9.795	9.782	9.2355	79	25.1531	25.733	25.725	25.1481
30	9.5668	10.114	10.100		80	25.4713	26.052	26.043	
31	9.8845	10.434	10.418	9.8718	81	25.7896	26.370	26.362	25.7847
32	10.2023	10.753	10.736		82	26.1078	26.689	26.680	
33	10.5201	11.073	11.053	10.5082	83	26.4260	27.007	26.998	26.4213
34	10.8379	11.392	11.371		84	26.7443	27.326	27.316	
35	11.1558	11.711	11.728	11.1446	85	27.0625	27.644	27.635	27.0579
36	11.4737	12.030	12.046		86	27.3807	27.962	27.953	
37	11.7916	12.349	12.364	11.7810	87	27.6990	28.281	28.271	27.6945
38	12.1095	12.668	12.682		88	28.0172	28.599	28.589	
39	12.4275	12.987	13.000	12.4174	89	28.3354	28.918	28.907	28.3310
40	12.7455	13.306	13.318		90	28.6537	29.236	29.226	
41	13.0635	13.625	13.636	13.0539	91	28.9719	29.555	29.544	28.9676
42	13.3815	13.944	13.954		92	29.2902	29.873	29.862	
43	13.6995	14.263	14.272	13.6904	93	29.6084	30.192	30.180	29.6042
44	14.0175	14.582	14.590		94	29.9267	30.510	30.499	
45	14.3355	14.901	14.908	14.3269	95	30.2449	30.828	30.817	30.2408
46	14.6535	15.219	15.226		96	30.5632	31.147	31.135	
47	14.9717	15.538	15.544	14.9634	97	30.8815	31.465	31.454	30.8774
48	15.2898	15.857	15.862		98	31.1997	31.784	31.772	
49	15.6079	16.176	16.180	15.5999	99	31.5180	32.102	32.090	31.5140
50	15.9260	16.495	16.498		100	31.8362	32.421	32.408	
51	16.2441	16.813	16.816	16.2364	101	32.1545	32.739	32.727	32.1506
52	16.5622	17.132	17.134		102	32.4727	33.057	33.045	
53	16.8803	17.451	17.452	16.8729	103	32.7910	33.376	33.363	32.7872
54	17.1984	17.769	17.770		104	33.1093	33.694	33.681	
55	17.5165	18.088	18.089	17.5094	105	33.4275	34.013	34.000	33.4238
56	17.8347	18.407	18.407		106	33.7458	34.331	34.318	
57	18.1528	18.725	18.725	18.1459	107	34.0641	34.649	34.636	34.0604
58	18.4710	19.044	19.043		108	34.3823	34.968	34.954	

[a] For 5 – 8 and 109–200 teeth see text, pages 2426, 2428.

Bottom Diameter: The bottom diameter is the diameter of a circle tangent to the curve (called the seating curve) at the bottom of the tooth gap. It equals the pitch diameter minus the diameter of the roller.

Caliper Diameter: The caliper diameter is the same as the bottom diameter for a sprocket with an even number of teeth. For a sprocket with an odd number of teeth, it is defined as the distance from the bottom of one tooth gap to that of the nearest opposite tooth gap. The caliper diameter for an even tooth sprocket is equal to pitch diameter–roller diameter. The caliper diameter for an odd tooth sprocket is equal to caliper factor–roller diameter. Here, the caliper factor = $PD[\cos(90° \div N)]$, where PD = pitch diameter and N = number of teeth. Caliper factors for 1-in. pitch and sprockets having 9–108 teeth are given in Table 4. For other tooth numbers use above formula. Caliper diameter tolerances are minus only and are equal to $0.002P\sqrt{N} + 0.006$ inch for the Commercial sprockets and $0.001P\sqrt{N} + 0.003$ inch for Precision sprockets. Tolerances are given in Table 5.

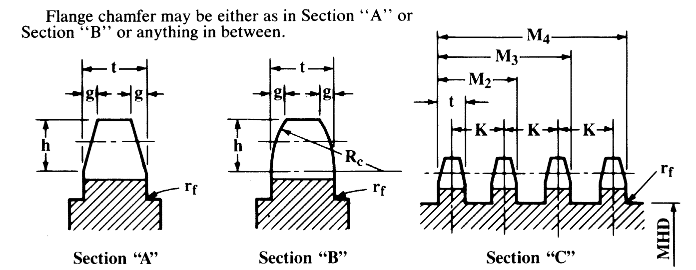
Table 5. Minus Tolerances on the Caliper Diameters of Precision Sprockets
ANSI/ASME B29.1M-1993

Pitch	Number of Teeth				
	Up to 15	16–24	25–35	36–48	49–63
0.250	0.004	0.004	0.004	0.005	0.005
0.375	0.004	0.004	0.004	0.005	0.005
0.500	0.004	0.005	0.0055	0.006	0.0065
0.625	0.005	0.0055	0.006	0.007	0.008
0.750	0.005	0.006	0.007	0.008	0.009
1.000	0.006	0.007	0.008	0.009	0.010
1.250	0.007	0.008	0.009	0.010	0.012
1.500	0.007	0.009	0.0105	0.012	0.013
1.750	0.008	0.010	0.012	0.013	0.015
2.000	0.009	0.011	0.013	0.015	0.017
2.250	0.010	0.012	0.014	0.016	0.018
2.500	0.010	0.013	0.015	0.018	0.020
3.000	0.012	0.015	0.018	0.021	0.024
Pitch	Number of Teeth				
	64–80	81–99	100–120	121–143	144 up
0.250	0.005	0.005	0.006	0.006	0.006
0.375	0.006	0.006	0.006	0.007	0.007
0.500	0.007	0.0075	0.008	0.0085	0.009
0.625	0.009	0.009	0.009	0.010	0.011
0.750	0.010	0.010	0.011	0.012	0.013
1.000	0.011	0.012	0.013	0.014	0.015
1.250	0.013	0.014	0.016	0.017	0.018
1.500	0.015	0.016	0.018	0.019	0.021
1.750	0.017	0.019	0.020	0.022	0.024
2.000	0.019	0.021	0.023	0.025	0.027
2.250	0.021	0.023	0.025	0.028	0.030
2.500	0.023	0.025	0.028	0.030	0.033
3.000	0.027	0.030	0.033	0.036	0.039

Minus tolerances for Commercial sprockets are twice those shown in this table.

Outside Diameter: OD is the diameter over the tips of teeth. Sprocket ODs for 1-in. pitch and 9–108 teeth are given in Table 4. For other tooth numbers the OD may be determined by the following formulas in which O = approximate OD; P = pitch of chain; N = number of sprocket teeth: $O = P[0.6 + \cot(180° \div N)]$, for turned sprocket; O = pitch diameter − roller diameter + 2 × whole depth of topping hob cut, for topping hob cut sprocket.*

Table 6. American National Standard Roller Chain Sprocket Flange Thickness and Tooth Section Profile Dimension *ANSI/ASME B29.1M-1993*



Sprocket Flange Thickness										
		Maximum Sprocket Flange Thickness, *t*			Minus Tolerance on *t*		Tolerance on *M*		Max. Variation of *t* on Each Flange	
Std. Chain No.	Width of Chain, *W*	Single	Double & Triple	Quad. & Over	Commercial	Precision	Commercial Plus or Minus	Precision Minus Only	Commercial	Precision
25	0.125	0.110	0.106	0.096	0.021	0.007	0.007	0.007	0.021	0.004
35	0.188	0.169	0.163	0.150	0.027	0.008	0.008	0.008	0.027	0.004
41	0.250	0.226	…	…	0.032	0.009	…	…	0.032	0.004
40	0.312	0.284	0.275	0.256	0.035	0.009	0.009	0.009	0.035	0.004
50	0.375	0.343	0.332	0.310	0.036	0.010	0.010	0.010	0.036	0.005
60	0.500	0.459	0.444	0.418	0.036	0.011	0.011	0.011	0.036	0.006
80	0.625	0.575	0.556	0.526	0.040	0.012	0.012	0.012	0.040	0.006
100	0.750	0.0692	0.669	0.633	0.046	0.014	0.014	0.014	0.046	0.007
120	1.000	0.924	0.894	0.848	0.057	0.016	0.016	0.016	0.057	0.008
140	1.000	0.924	0.894	0.848	0.057	0.016	0.016	0.016	0.057	0.008
160	1.250	1.156	1.119	1.063	0.062	0.018	0.018	0.018	0.062	0.009
180	1.406	1.302	1.259	1.198	0.068	0.020	0.020	0.020	0.068	0.010
200	1.500	1.389	1.344	1.278	0.072	0.021	0.021	0.021	0.072	0.010
240	1.875	1.738	1.682	1.602	0.087	0.025	0.025	0.025	0.087	0.012

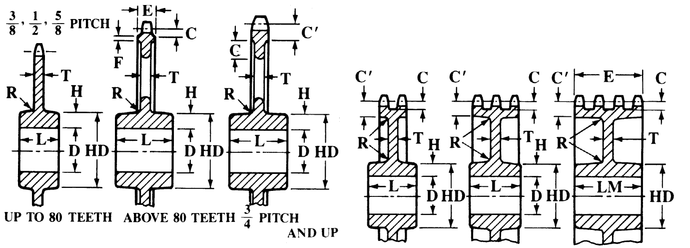
Sprocket Tooth Section Profile Dimensions						
					Transverse Pitch *K*	
Std. Chain No.	Chain Pitch *P*	Depth of Chamfer *h*	Width of Chamfer *g*	Minimum Radius R_c	Standard Series	Heavy Series
25	0.250	0.125	0.031	0.265	0.252	…
35	0.375	0.188	0.047	0.398	0.399	…
41	0.500	0.250	0.062	0.531	…	…
40	0.500	0.250	0.062	0.531	0.566	…
50	0.625	0.312	0.078	0.664	0.713	…
60	0.750	0.375	0.094	0.796	0.897	1.028
80	1.000	0.500	0.125	1.062	1.153	1.283
100	1.250	0.625	0.156	1.327	1.408	1.539
120	1.500	0.750	0.188	1.593	1.789	1.924
140	1.750	0.875	0.219	1.858	1.924	2.055
160	2.000	1.000	0.250	2.124	2.305	2.437
180	2.250	1.125	0.281	2.392	2.592	2.723
200	2.500	1.250	0.312	2.654	2.817	3.083
240	3.000	1.500	0.375	3.187	3.458	3.985

All dimensions are in inches. r_f max = 0.04 *P* for max. hub diameter.

* This dimension was added in 1984 as a desirable goal for the future. It should in no way obsolete existing tools or sprockets. The whole depth *WD* is found from the formula: $WD = \frac{1}{2}D_r + P[0.3 - \frac{1}{2}\tan(90 \text{ deg} \div N_a)]$, where N_a is the intermediate number of teeth for the topping hob. For teeth range 5, $N_a = 5$; 6, 6; 7–8, 7.47; 9–11, 9.9; 12–17, 14.07; 18–34, 23.54; 35 and over, 56.

Proportions of Sprockets.—Typical proportions of single-strand and multiple-strand cast roller chain sprockets, as provided by the American Chain Association, are shown in Table 7. Typical proportions of roller chain bar-steel sprockets, also provided by this association, are shown in Table 8.

Table 7. Typical Proportions of Single-Strand and Multiple-Strand Cast Roller Chain Sprockets



Single-Strand Multiple-Strand

Sprocket Web Thickness, T, for Various Pitches P															
Single-Strand								Multiple-Strand							
P	T	P	T	P	T	P	T	P	T	P	T	P	T	P	T
3⁄8	.312	3⁄4	.437	1½	.625	2¼	1.000	3⁄8	.375	3⁄4	.500	1½	.750	2¼	1.125
½	.375	1	.500	1¾	.750	2½	1.125	½	.406	1	.562	1 ¾	.875	2½	1.250
5⁄8	.406	1¼	.562	2	.875	3	1.250	5⁄8	.437	1¼	.625	2	1.000	3	1.500

Formulas for Dimensions of Single and Multiple Sprockets

$H = 0.375 + \frac{D}{6} + 0.01\ PD$

$E = 0.625P + 0.93W$

$F = 0.150 + 0.25P$

$L = 4H$ for semi-steel castings

$G = 2T$

$C = 0.5P$

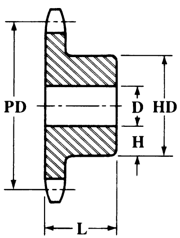
$R = 0.4P$ for single-strand sprockets

$C' = 0.9P$

$R = 0.5T$ for multiple-strand sprockets

All dimensions in inches. Where: P = chain pitch and W = nominal chain width.

Table 8. Typical Proportions of Roller Chain Bar-steel Sprockets



$H = Z + D/6 + 0.01\ PD$

For PD up to 2 inches, Z = 0.125 inch; for 2–4 inches, Z = 0.187 inch; for 4–6 inches, 0.25 inch; and for over 6 inches, 0.375 inch.

Hub length $L = 3.3\ H$, normally, with a minimum of $2.6H$.

Hub diameter $HD = D + 2H$, but not more than the maximum hub diameter MHD given by the formula:

$$MHD = P\left(\cot\frac{180°}{N} - 1\right) - 0.030$$

where: P = Chain pitch, in inches

N = Number of sprocket teeth

When sprocket wheels are designed with spokes, the usual assumptions made in order to determine suitable proportions are as follows: 1) That the maximum torque load acting on a sprocket is the chain tensile strength times the sprocket pitch radius; 2) That the torque load is equally divided between the arms by the rim; and 3) That each arm acts as a cantilever beam.

The arms are generally elliptical in cross section, with the major axis twice the minor axis.

Selection of Chain and Sprockets.—The smallest applicable pitch of roller chain is desirable for quiet operation and high speed. The horsepower capacity varies with the chain pitch as shown in Table . However, short pitch with high working load can often be obtained by the use of multiple-strand chain.

The small sprocket selected must be large enough to accommodate the shaft. Table 9 gives maximum bore and hub diameters consistent with commercial practice for sprockets with up to 25 teeth.

After selecting the small sprocket, the number of teeth in the larger sprocket is determined by the desired ratio of the shaft speed. Overemphasis on the exactness in the speed ratio may result in a cumbersome and expensive installation. In most cases, satisfactory operation can be obtained with a minor change in speed of one or both shafts.

To properly use this table the following factors must be taken into consideration:

1) Service factors
2) Multiple Strand Factors
3) Lubrication

Service Factors: See Table 14.

Multiple Strand Factors: For two strands, the multiple strand factor is 1.7; for three strands, it is 2.5; and for four strands, it is 3.3.

Lubrication:

Required type of lubrication is indicated at the bottom of each roller chain size section of the table. For a description of each type of lubrication, see page 2443.

Type A — Manual or Drip Lubrication
Type B — Bath or Disc Lubrication
Type C — Oil Stream Lubrication

To find the required horsepower table rating, use the following formula:

$$\text{Required hp Table Rating} = \frac{\text{hp to be Transmitted} \times \text{Service Factor}}{\text{Multiple-Strand Factor}}$$

¼-inch Pitch Standard Single-Strand Roller Chain — No. 25

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	50	100	300	500	700	900	1200	1500	1800	2100	2500	3000	3500
	Horsepower Rating												
11	0.03	0.05	0.14	0.23	0.31	0.39	0.50	0.62	0.73	0.83	0.98	1.15	1.32
12	0.03	0.06	0.16	0.25	0.34	0.43	0.55	0.68	0.80	0.92	1.07	1.26	1.45
13	0.04	0.06	0.17	0.27	0.37	0.47	0.60	0.74	0.87	1.00	1.17	1.38	1.58
14	0.04	0.07	0.19	0.30	0.40	0.50	0.65	0.80	0.94	1.08	1.27	1.49	1.71
15	0.04	0.08	0.20	0.32	0.43	0.54	0.70	0.86	1.01	1.17	1.36	1.61	1.85
16	0.04	0.08	0.22	0.34	0.47	0.58	0.76	0.92	1.09	1.25	1.46	1.72	1.98
17	0.05	0.09	0.23	0.37	0.50	0.62	0.81	0.99	1.16	1.33	1.56	1.84	2.11
18	0.05	0.09	0.25	0.39	0.53	0.66	0.86	1.05	1.24	1.42	1.66	1.96	2.25
19	0.05	0.10	0.26	0.41	0.56	0.70	0.91	1.11	1.31	1.50	1.76	2.07	2.38
20	0.06	0.10	0.28	0.44	0.59	0.74	0.96	1.17	1.38	1.59	1.86	2.19	2.52
21	0.06	0.11	0.29	0.46	0.62	0.78	1.01	1.24	1.46	1.68	1.96	2.31	2.66
22	0.06	0.11	0.31	0.48	0.66	0.82	1.07	1.30	1.53	1.76	2.06	2.43	2.79
23	0.06	0.12	0.32	0.51	0.69	0.86	1.12	1.37	1.61	1.85	2.16	2.55	2.93
24	0.07	0.13	0.34	0.53	0.72	0.90	1.17	1.43	1.69	1.94	2.27	2.67	3.07
25	0.07	0.13	0.35	0.56	0.75	0.94	1.22	1.50	1.76	2.02	2.37	2.79	3.21
26	0.07	0.14	0.37	0.58	0.79	0.98	1.28	1.56	1.84	2.11	2.47	2.91	3.34
28	0.08	0.15	0.40	0.63	0.85	1.07	1.38	1.69	1.99	2.29	2.68	3.15	3.62
30	0.08	0.16	0.43	0.68	0.92	1.15	1.49	1.82	2.15	2.46	2.88	3.40	3.90
32	0.09	0.17	0.46	0.73	0.98	1.23	1.60	1.95	2.30	2.64	3.09	3.64	4.18
35	0.10	0.19	0.51	0.80	1.08	1.36	1.76	2.15	2.53	2.91	3.41	4.01	4.61
40	0.12	0.22	0.58	0.92	1.25	1.57	2.03	2.48	2.93	3.36	3.93	4.64	5.32
45	0.13	0.25	0.66	1.05	1.42	1.78	2.31	2.82	3.32	3.82	4.47	5.26	6.05
	Type A				Type B								

3/8-inch Pitch Standard Single-Strand Roller Chain — No. 35

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	50	100	300	500	700	900	1200	1500	1800	2100	2500	3000	3500
	Horsepower Rating												
11	0.10	0.18	0.49	0.77	1.05	1.31	1.70	2.08	2.45	2.82	3.30	2.94	2.33
12	0.11	0.20	0.54	0.85	1.15	1.44	1.87	2.29	2.70	3.10	3.62	3.35	2.66
13	0.12	0.22	0.59	0.93	1.26	1.57	2.04	2.49	2.94	3.38	3.95	3.77	3.00
14	0.13	0.24	0.63	1.01	1.36	1.71	2.21	2.70	3.18	3.66	4.28	4.22	3.35
15	0.14	0.25	0.68	1.08	1.47	1.84	2.38	2.91	3.43	3.94	4.61	4.68	3.71
16	0.15	0.27	0.73	1.16	1.57	1.97	2.55	3.12	3.68	4.22	4.94	5.15	4.09
17	0.16	0.29	0.78	1.24	1.68	2.10	2.73	3.33	3.93	4.51	5.28	5.64	4.48
18	0.17	0.31	0.83	1.32	1.78	2.24	2.90	3.54	4.18	4.80	5.61	6.15	4.88
19	0.18	0.33	0.88	1.40	1.89	2.37	3.07	3.76	4.43	5.09	5.95	6.67	5.29
20	0.19	0.35	0.93	1.48	2.00	2.51	3.25	3.97	4.68	5.38	6.29	7.20	5.72
21	0.20	0.37	0.98	1.56	2.11	2.64	3.42	4.19	4.93	5.67	6.63	7.75	6.15
22	0.21	0.38	1.03	1.64	2.22	2.78	3.60	4.40	5.19	5.96	6.97	8.21	6.59
23	0.22	0.40	1.08	1.72	2.33	2.92	3.78	4.62	5.44	6.25	7.31	8.62	7.05
24	0.23	0.42	1.14	1.80	2.44	3.05	3.96	4.84	5.70	6.55	7.66	9.02	7.51
25	0.24	0.44	1.19	1.88	2.55	3.19	4.13	5.05	5.95	6.84	8.00	9.43	7.99
26	0.25	0.46	1.24	1.96	2.66	3.33	4.31	5.27	6.21	7.14	8.35	9.84	8.47
28	0.27	0.50	1.34	2.12	2.88	3.61	4.67	5.71	6.73	7.73	9.05	10.7	9.47
30	0.29	0.54	1.45	2.29	3.10	3.89	5.03	6.15	7.25	8.33	9.74	11.5	10.5
32	0.31	0.58	1.55	2.45	3.32	4.17	5.40	6.60	7.77	8.93	10.4	12.3	11.6
35	0.34	0.64	1.71	2.70	3.66	4.59	5.95	7.27	8.56	9.84	11.5	13.6	13.2
40	0.39	0.73	1.97	3.12	4.23	5.30	6.87	8.40	9.89	11.4	13.3	15.7	16.2
45	0.45	0.83	2.24	3.55	4.80	6.02	7.80	9.53	11.2	12.9	15.1	17.8	19.3
	Type A		Type B							Type C			

1/2-inch Pitch Standard Single-Strand Roller Chain — No. 40

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	50	100	200	300	400	500	700	900	1000	1200	1400	1600	1800
	Horsepower Rating												
11	0.23	0.43	0.80	1.16	1.50	1.83	2.48	3.11	3.42	4.03	4.63	5.22	4.66
12	0.25	0.47	0.88	1.27	1.65	2.01	2.73	3.42	3.76	4.43	5.09	5.74	5.31
13	0.28	0.52	0.96	1.39	1.80	2.20	2.97	3.73	4.10	4.83	5.55	6.26	5.99
14	0.30	0.56	1.04	1.50	1.95	2.38	3.22	4.04	4.44	5.23	6.01	6.78	6.70
15	0.32	0.60	1.12	1.62	2.10	2.56	3.47	4.35	4.78	5.64	6.47	7.30	7.43
16	0.35	0.65	1.20	1.74	2.25	2.75	3.72	4.66	5.13	6.04	6.94	7.83	8.18
17	0.37	0.69	1.29	1.85	2.40	2.93	3.97	4.98	5.48	6.45	7.41	8.36	8.96
18	0.39	0.73	1.37	1.97	2.55	3.12	4.22	5.30	5.82	6.86	7.88	8.89	9.76
19	0.42	0.78	1.45	2.09	2.71	3.31	4.48	5.62	6.17	7.27	8.36	9.42	10.5
20	0.44	0.82	1.53	2.21	2.86	3.50	4.73	5.94	6.53	7.69	8.83	9.96	11.1
21	0.46	0.87	1.62	2.33	3.02	3.69	4.99	6.26	6.88	8.11	9.31	10.5	11.7
22	0.49	0.91	1.70	2.45	3.17	3.88	5.25	6.58	7.23	8.52	9.79	11.0	12.3
23	0.51	0.96	1.78	2.57	3.33	4.07	5.51	6.90	7.59	8.94	10.3	11.6	12.9
24	0.54	1.00	1.87	2.69	3.48	4.26	5.76	7.23	7.95	9.36	10.8	12.1	13.5
25	0.56	1.05	1.95	2.81	3.64	4.45	6.02	7.55	8.30	9.78	11.2	12.7	14.1
26	0.58	1.09	2.04	2.93	3.80	4.64	6.28	7.88	8.66	10.2	11.7	13.2	14.7
28	0.63	1.18	2.20	3.18	4.11	5.03	6.81	8.54	9.39	11.1	12.7	14.3	15.9
30	0.68	1.27	2.38	3.42	4.43	5.42	7.33	9.20	10.1	11.9	13.7	15.4	17.2
32	0.73	1.36	2.55	3.67	4.75	5.81	7.86	9.86	10.8	12.8	14.7	16.5	18.4
35	0.81	1.50	2.81	4.04	5.24	6.40	8.66	10.9	11.9	14.1	16.2	18.2	20.3
40	0.93	1.74	3.24	4.67	6.05	7.39	10.0	12.5	13.8	16.3	18.7	21.1	23.4
45	1.06	1.97	3.68	5.30	6.87	8.40	11.4	14.2	15.7	18.5	21.2	23.9	26.6
	Type A		Type B								Type C		

½-inch Pitch Light Weight Machinery Roller Chain — No. 41

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	10	25	50	100	200	300	400	500	700	900	1000	1200	1400
	Horsepower Rating												
11	0.03	0.07	0.13	0.24	0.44	0.64	0.82	1.01	1.37	1.71	1.88	1.71	1.36
12	0.03	0.07	0.14	0.26	0.49	0.70	0.91	1.11	1.50	1.88	2.07	1.95	1.55
13	0.04	0.08	0.15	0.28	0.53	0.76	0.99	1.21	1.63	2.05	2.25	2.20	1.75
14	0.04	0.09	0.16	0.31	0.57	0.83	1.07	1.31	1.77	2.22	2.44	2.46	1.95
15	0.04	0.09	0.18	0.33	0.62	0.89	1.15	1.41	1.91	2.39	2.63	2.73	2.17
16	0.04	0.10	0.19	0.36	0.66	0.95	1.24	1.51	2.05	2.57	2.82	3.01	2.39
17	0.05	0.11	0.20	0.38	0.71	1.02	1.32	1.61	2.18	2.74	3.01	3.29	2.61
18	0.05	0.12	0.22	0.40	0.75	1.08	1.40	1.72	2.32	2.91	3.20	3.59	2.85
19	0.05	0.12	0.23	0.43	0.80	1.15	1.49	1.82	2.46	3.09	3.40	3.89	3.09
20	0.06	0.13	0.24	0.45	0.84	1.21	1.57	1.92	2.60	3.26	3.59	4.20	3.33
21	0.06	0.14	0.26	0.48	0.89	1.28	1.66	2.03	2.74	3.44	3.78	4.46	3.59
22	0.06	0.14	0.27	0.50	0.93	1.35	1.74	2.13	2.89	3.62	3.98	4.69	3.85
23	0.06	0.15	0.28	0.53	0.98	1.41	1.83	2.24	3.03	3.80	4.17	4.92	4.11
24	0.07	0.16	0.29	0.55	1.03	1.48	1.92	2.34	3.17	3.97	4.37	5.15	4.38
25	0.07	0.17	0.31	0.57	1.07	1.55	2.00	2.45	3.31	4.15	4.57	5.38	4.66
26	0.07	0.17	0.32	0.60	1.12	1.61	2.09	2.55	3.46	4.33	4.76	5.61	4.94
28	0.08	0.19	0.35	0.65	1.21	1.75	2.26	2.77	3.74	4.69	5.16	6.08	5.52
30	0.08	0.20	0.38	0.70	1.31	1.88	2.44	2.98	4.03	5.06	5.56	6.55	6.13
32	0.09	0.22	0.40	0.75	1.40	2.02	2.61	3.20	4.33	5.42	5.96	7.03	6.75
35	0.10	0.24	0.44	0.83	1.54	2.22	2.88	3.52	4.76	5.97	6.57	7.74	7.72
40	0.12	0.27	0.51	0.96	1.78	2.57	3.33	4.07	5.50	6.90	7.59	8.94	9.43
45	0.14	0.31	0.58	1.08	2.02	2.92	3.78	4.62	6.25	7.84	8.62	10.2	11.3
	Type A				Type B							Type C	

⅝-inch Pitch Standard Single-Strand Roller Chain — No. 50

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	25	50	100	200	300	400	500	700	900	1000	1200	1400	1600
	Horsepower Rating												
11	0.24	0.45	0.84	1.56	2.25	2.92	3.57	4.83	6.06	6.66	7.85	8.13	6.65
12	0.26	0.49	0.92	1.72	2.47	3.21	3.92	5.31	6.65	7.31	8.62	9.26	7.58
13	0.29	0.54	1.00	1.87	2.70	3.50	4.27	5.78	7.25	7.97	9.40	10.4	8.55
14	0.31	0.58	1.09	2.03	2.92	3.79	4.63	6.27	7.86	8.64	10.2	11.7	9.55
15	0.34	0.63	1.17	2.19	3.15	4.08	4.99	6.75	8.47	9.31	11.0	12.6	10.6
16	0.36	0.67	1.26	2.34	3.38	4.37	5.35	7.24	9.08	9.98	11.8	13.5	11.7
17	0.39	0.72	1.34	2.50	3.61	4.67	5.71	7.73	9.69	10.7	12.6	14.4	12.8
18	0.41	0.76	1.43	2.66	3.83	4.97	6.07	8.22	10.3	11.3	13.4	15.3	13.9
19	0.43	0.81	1.51	2.82	4.07	5.27	6.44	8.72	10.9	12.0	14.2	16.3	15.1
20	0.46	0.86	1.60	2.98	4.30	5.57	6.80	9.21	11.5	12.7	15.0	17.2	16.3
21	0.48	0.90	1.69	3.14	4.53	5.87	7.17	9.71	12.2	13.4	15.8	18.1	17.6
22	0.51	0.95	1.77	3.31	4.76	6.17	7.54	10.2	12.8	14.1	16.6	19.1	18.8
23	0.53	1.00	1.86	3.47	5.00	6.47	7.91	10.7	13.4	14.8	17.4	20.0	20.1
24	0.56	1.04	1.95	3.63	5.23	6.78	8.29	11.2	14.1	15.5	18.2	20.9	21.4
25	0.58	1.09	2.03	3.80	5.47	7.08	8.66	11.7	14.7	16.2	19.0	21.9	22.8
26	0.61	1.14	2.12	3.96	5.70	7.39	9.03	12.2	15.3	16.9	19.9	22.8	24.2
28	0.66	1.23	2.30	4.29	6.18	8.01	9.79	13.2	16.6	18.3	21.5	24.7	27.0
30	0.71	1.33	2.48	4.62	6.66	8.63	10.5	14.3	17.9	19.7	23.2	26.6	30.0
32	0.76	1.42	2.66	4.96	7.14	9.25	11.3	15.3	19.2	21.1	24.9	28.6	32.2
35	0.84	1.57	2.93	5.46	7.86	10.2	12.5	16.9	21.1	23.2	27.4	31.5	35.5
40	0.97	1.81	3.38	6.31	9.08	11.8	14.4	19.5	24.4	26.8	31.6	36.3	41.0
45	1.10	2.06	3.84	7.16	10.3	13.4	16.3	22.1	27.7	30.5	35.9	41.3	46.5
	Type A			Type B					Type C				

¾-inch Pitch Standard Single-Strand Roller Chain — No. 60

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	25	50	100	150	200	300	400	500	600	700	800	900	1000
	Horsepower Rating												
11	0.41	0.77	1.44	2.07	2.69	3.87	5.02	6.13	7.23	8.30	9.36	10.4	11.4
12	0.45	0.85	1.58	2.28	2.95	4.25	5.51	6.74	7.94	9.12	10.3	11.4	12.6
13	0.50	0.92	1.73	2.49	3.22	4.64	6.01	7.34	8.65	9.94	11.2	12.5	13.7
14	0.54	1.00	1.87	2.69	3.49	5.02	6.51	7.96	9.37	10.8	12.1	13.5	14.8
15	0.58	1.08	2.01	2.90	3.76	5.41	7.01	8.57	10.1	11.6	13.1	14.5	16.0
16	0.62	1.16	2.16	3.11	4.03	5.80	7.52	9.19	10.8	12.4	14.0	15.6	17.1
17	0.66	1.24	2.31	3.32	4.30	6.20	8.03	9.81	11.6	13.3	15.0	16.7	18.3
18	0.70	1.31	2.45	3.53	4.58	6.59	8.54	10.4	12.3	14.1	15.9	17.7	19.5
19	0.75	1.39	2.60	3.74	4.85	6.99	9.05	11.1	13.0	15.0	16.9	18.8	20.6
20	0.79	1.47	2.75	3.96	5.13	7.38	9.57	11.7	13.8	15.8	17.9	19.8	21.8
21	0.83	1.55	2.90	4.17	5.40	7.78	10.1	12.3	14.5	16.7	18.8	20.9	23.0
22	0.87	1.63	3.05	4.39	5.68	8.19	10.6	13.0	15.3	17.5	19.8	22.0	24.2
23	0.92	1.71	3.19	4.60	5.96	8.59	11.1	13.6	16.0	18.4	20.8	23.1	25.4
24	0.96	1.79	3.35	4.82	6.24	8.99	11.6	14.2	16.8	19.3	21.7	24.2	26.6
25	1.00	1.87	3.50	5.04	6.52	9.40	12.2	14.9	17.5	20.1	22.7	25.3	27.8
26	1.05	1.95	3.65	5.25	6.81	9.80	12.7	15.5	18.3	21.0	23.7	26.4	29.0
28	1.13	2.12	3.95	5.69	7.37	10.6	13.8	16.8	19.8	22.8	25.7	28.5	31.4
30	1.22	2.28	4.26	6.13	7.94	11.4	14.8	18.1	21.4	24.5	27.7	30.8	33.8
32	1.31	2.45	4.56	6.57	8.52	12.3	15.9	19.4	22.9	26.3	29.7	33.0	36.3
35	1.44	2.69	5.03	7.24	9.38	13.5	17.5	21.4	25.2	29.0	32.7	36.3	39.9
40	1.67	3.11	5.81	8.37	10.8	15.6	20.2	24.7	29.1	33.5	37.7	42.0	46.1
45	1.89	3.53	6.60	9.50	12.3	17.7	23.0	28.1	33.1	38.0	42.9	47.7	52.4
	Type A		Type B							Type C			

1-inch Pitch Standard Single-Strand Roller Chain — No. 80

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	25	50	100	150	200	300	400	500	600	700	800	900	1000
	Horsepower Ratings												
11	0.97	1.80	3.36	4.84	6.28	9.04	11.7	14.3	16.9	19.4	21.9	23.0	19.6
12	1.06	1.98	3.69	5.32	6.89	9.93	12.9	15.7	18.5	21.3	24.0	26.2	22.3
13	1.16	2.16	4.03	5.80	7.52	10.8	14.0	17.1	20.2	23.2	26.2	29.1	25.2
14	1.25	2.34	4.36	6.29	8.14	11.7	15.2	18.6	21.9	25.1	28.4	31.5	28.2
15	1.35	2.52	4.70	6.77	8.77	12.6	16.4	20.0	23.6	27.1	30.6	34.0	31.2
16	1.45	2.70	5.04	7.26	9.41	13.5	17.6	21.5	25.3	29.0	32.8	36.4	34.4
17	1.55	2.88	5.38	7.75	10.0	14.5	18.7	22.9	27.0	31.0	35.0	38.9	37.7
18	1.64	3.07	5.72	8.25	10.7	15.4	19.9	24.4	28.7	33.0	37.2	41.4	41.1
19	1.74	3.25	6.07	8.74	11.3	16.3	21.1	25.8	30.4	35.0	39.4	43.8	44.5
20	1.84	3.44	6.41	9.24	12.0	17.2	22.3	27.3	32.2	37.0	41.7	46.3	48.1
21	1.94	3.62	6.76	9.74	12.6	18.2	23.5	28.8	33.9	39.0	43.9	48.9	51.7
22	2.04	3.81	7.11	10.2	13.3	19.1	24.8	30.3	35.7	41.0	46.2	51.4	55.5
23	2.14	4.00	7.46	10.7	13.9	20.1	26.0	31.8	37.4	43.0	48.5	53.9	59.3
24	2.24	4.19	7.81	11.3	14.6	21.0	27.2	33.2	39.2	45.0	50.8	56.4	62.0
25	2.34	4.37	8.16	11.8	15.2	21.9	28.4	34.7	40.9	47.0	53.0	59.0	64.8
26	2.45	4.56	8.52	12.3	15.9	22.9	29.7	36.2	42.7	49.1	55.3	61.5	67.6
28	2.65	4.94	9.23	13.3	17.2	24.8	32.1	39.3	46.3	53.2	59.9	66.7	73.3
30	2.85	5.33	9.94	14.3	18.5	26.7	34.6	42.3	49.9	57.3	64.6	71.8	78.9
32	3.06	5.71	10.7	15.3	19.9	28.6	37.1	45.4	53.5	61.4	69.2	77.0	84.6
35	3.37	6.29	11.7	16.9	21.9	31.6	40.9	50.0	58.9	67.6	76.3	84.8	93.3
40	3.89	7.27	13.6	19.5	25.3	36.4	47.2	57.7	68.0	78.1	88.1	99.0	108
45	4.42	8.25	15.4	22.2	28.7	41.4	53.6	65.6	77.2	88.7	100	111	122
	Type A	Type B						Type C					

1¼-inch Pitch Standard Single-Strand Roller Chain — No. 100

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	10	25	50	100	150	200	300	400	500	600	700	800	900
	Horsepower Rating												
11	0.81	1.85	3.45	6.44	9.28	12.0	17.3	22.4	27.4	32.3	37.1	32.8	27.5
12	0.89	2.03	3.79	7.08	10.2	13.2	19.0	24.6	30.1	35.5	40.8	37.3	31.3
13	0.97	2.22	4.13	7.72	11.1	14.4	20.7	26.9	32.8	38.7	44.5	42.1	35.3
14	1.05	2.40	4.48	8.36	12.0	15.6	22.5	29.1	35.6	41.9	48.2	47.0	39.4
15	1.13	2.59	4.83	9.01	13.0	16.8	24.2	31.4	38.3	45.2	51.9	52.2	43.7
16	1.22	2.77	5.17	9.66	13.9	18.0	26.0	33.6	41.1	48.4	55.6	57.5	48.2
17	1.30	2.96	5.52	10.3	14.8	19.2	27.7	35.9	43.9	51.7	59.4	63.0	52.8
18	1.38	3.15	5.88	11.0	15.8	20.5	29.5	38.2	46.7	55.0	63.2	68.6	57.5
19	1.46	3.34	6.23	11.6	16.7	21.7	31.2	40.5	49.5	58.3	67.0	74.4	62.3
20	1.55	3.53	6.58	12.3	17.7	22.9	33.0	42.8	52.3	61.6	70.8	79.8	67.3
21	1.63	3.72	6.94	13.0	18.7	24.2	34.8	45.1	55.1	65.0	74.6	84.2	72.4
22	1.71	3.91	7.30	13.6	19.6	25.4	36.6	47.4	58.0	68.3	78.5	88.5	77.7
23	1.80	4.10	7.66	14.3	20.6	26.7	38.4	49.8	60.8	71.7	82.3	92.8	83.0
24	1.88	4.30	8.02	15.0	21.5	27.9	40.2	52.1	63.7	75.0	86.2	97.2	88.5
25	1.97	4.49	8.38	15.6	22.5	29.2	42.0	54.4	66.6	78.4	90.1	102	94.1
26	2.05	4.68	8.74	16.3	23.5	30.4	43.8	56.8	69.4	81.8	94.0	106	99.8
28	2.22	5.07	9.47	17.7	25.5	33.0	47.5	61.5	75.2	88.6	102	115	112
30	2.40	5.47	10.2	19.0	27.4	35.5	51.2	66.3	81.0	95.5	110	124	124
32	2.57	5.86	10.9	20.4	29.4	38.1	54.9	71.1	86.9	102	118	133	136
35	2.83	6.46	12.0	22.5	32.4	42.0	60.4	78.3	95.7	113	130	146	156
40	3.27	7.46	13.9	26.0	37.4	48.5	69.8	90.4	111	130	150	169	188
45	3.71	8.47	15.8	29.5	42.5	55.0	79.3	103	126	148	170	192	213
	Type A		Type B				Type C						

1½-inch Pitch Standard Single-Strand Roller Chain — No. 120

No. of Teeth Small Spkt.	Revolutions per Minute — Small Sprocket[a]												
	10	25	50	100	150	200	300	400	500	600	700	800	900
	Horsepower Rating												
11	1.37	3.12	5.83	10.9	15.7	20.3	29.2	37.9	46.3	54.6	46.3	37.9	31.8
12	1.50	3.43	6.40	11.9	17.2	22.3	32.1	41.6	50.9	59.9	52.8	43.2	36.2
13	1.64	3.74	6.98	13.0	18.8	24.3	35.0	45.4	55.5	65.3	59.5	48.7	40.8
14	1.78	4.05	7.56	14.1	20.3	26.3	37.9	49.1	60.1	70.8	66.5	54.4	45.6
15	1.91	4.37	8.15	15.2	21.9	28.4	40.9	53.0	64.7	76.3	73.8	60.4	50.6
16	2.05	4.68	8.74	16.3	23.5	30.4	43.8	56.8	69.4	81.8	81.3	66.5	55.7
17	2.19	5.00	9.33	17.4	25.1	32.5	46.8	60.6	74.1	87.3	89.0	72.8	61.0
18	2.33	5.32	9.92	18.5	26.7	34.6	49.8	64.5	78.8	92.9	97.0	79.4	66.5
19	2.47	5.64	10.5	19.6	28.3	36.6	52.8	68.4	83.6	98.5	105	86.1	72.1
20	2.61	5.96	11.1	20.7	29.9	38.7	55.8	72.2	88.3	104	114	92.9	77.9
21	2.75	6.28	11.7	21.9	31.5	40.8	58.8	76.2	93.1	110	122	100	83.8
22	2.90	6.60	12.3	23.0	33.1	42.9	61.8	80.1	97.9	115	131	107	89.9
23	3.04	6.93	12.9	24.1	34.8	45.0	64.9	84.0	103	121	139	115	96.1
24	3.18	7.25	13.5	25.3	36.4	47.1	67.9	88.0	108	127	146	122	102
25	3.32	7.58	14.1	26.4	38.0	49.3	71.0	91.9	112	132	152	130	109
26	3.47	7.91	14.8	27.5	39.7	51.4	74.0	95.9	117	138	159	138	115
28	3.76	8.57	16.0	29.8	43.0	55.7	80.2	104	127	150	172	154	129
30	4.05	9.23	17.2	32.1	46.3	60.0	86.4	112	137	161	185	171	143
32	4.34	9.90	18.5	34.5	49.6	64.3	92.6	120	147	173	199	188	158
35	4.78	10.9	20.3	38.0	54.7	70.9	102	132	162	190	219	215	180
40	5.52	12.6	23.5	43.9	63.2	81.8	118	153	187	220	253	…	…
45	6.27	14.3	26.7	49.8	71.7	92.9	134	173	212	250	287	…	…
	Type A	Type B					Type C						

[a] For lower or higher rpm, larger chain sizes, and rpm above 3500, see B29.1M-1993.
For use of table see page 2431.

Table 9. Recommended Roller Chain Sprocket Maximum Bore and Hub Diameters

	Roller Chain Pitch									
	3/8		1/2		5/8		3/4		1	
No. of Teeth	Max. Bore	Max. Hub Dia.	Max. Bore	Max. Hub Dia.	Max. Bore	Max. Hub Dia.	Max. Bore	Max. Hub Dia.	Max. Bore	Max. Hub Dia.
11	19/32	55/64	25/32	1 11/64	31/32	1 15/32	1 1/4	1 49/64	1 5/8	2 3/8
12	5/8	63/64	7/8	1 21/64	1 5/32	1 43/64	1 9/32	2 1/64	1 25/32	2 45/64
13	3/4	1 7/64	1	1 1/2	1 9/32	1 7/8	1 1/2	2 1/4	2	3 1/64
14	27/32	1 15/64	1 5/32	1 21/32	1 5/16	2 5/64	1 3/4	2 1/2	2 9/32	3 11/32
15	7/8	1 23/64	1 1/4	1 13/16	1 17/32	2 9/32	1 25/32	2 3/4	2 13/32	3 43/64
16	31/32	1 15/32	1 9/32	1 63/64	1 11/16	2 31/64	1 31/32	2 63/64	2 23/32	3 63/64
17	1 3/32	1 19/32	1 3/8	2 9/64	1 25/32	2 11/16	2 7/32	3 7/32	2 13/16	4 5/16
18	1 7/32	1 23/32	1 17/32	2 19/64	1 7/8	2 57/64	2 9/32	3 15/32	3 1/8	4 41/64
19	1 1/4	1 27/32	1 11/16	2 29/64	2 1/16	3 5/64	2 7/16	3 45/64	3 5/16	4 61/64
20	1 9/32	1 61/64	1 25/32	2 5/8	2 1/4	3 9/32	2 11/16	3 61/64	3 1/2	5 9/32
21	1 5/16	2 5/64	1 25/32	2 25/32	2 9/32	3 31/64	2 13/16	4 3/16	3 3/4	5 19/32
22	1 7/16	2 13/16	1 15/16	2 15/16	2 7/16	3 11/16	2 15/16	4 7/16	3 7/8	5 59/64
23	1 9/16	2 5/16	2 3/32	3 3/32	2 5/8	3 57/64	3 1/8	4 43/64	4 3/16	6 15/64
24	1 11/16	2 7/16	2 1/4	3 17/64	2 13/16	4 5/64	3 1/4	4 29/32	4 9/16	6 9/16
25	1 3/4	2 9/16	2 9/32	3 27/64	2 27/32	4 9/32	3 3/8	5 5/32	4 11/16	6 7/8
	Roller Chain Pitch									
	1 1/4		1 1/2		1 3/4		2		2 1/2	
No. of Teeth	Max. Bore	Max. Hub Dia.	Max. Bore	Max. Hub Dia.	Max. Bore	Max. Hub Dia.	Max. Bore	Max. Hub Dia.	Max. Bore	Max. Hub Dia.
11	1 31/32	2 31/32	2 5/16	3 37/64	2 13/16	4 11/64	3 9/32	4 25/32	3 15/16	5 63/64
12	2 9/32	3 3/8	2 3/4	4 1/16	3 1/4	4 3/4	3 5/8	5 27/64	4 23/32	6 51/64
13	2 17/32	3 25/32	3 1/16	4 35/64	3 9/16	5 5/16	4 1/16	6 5/64	5 3/32	7 39/64
14	2 11/16	4 3/16	3 5/16	5 1/32	3 7/8	5 7/8	4 11/16	6 23/32	5 23/32	8 27/64
15	3 3/32	4 19/32	3 3/4	5 33/64	4 7/16	6 29/64	4 7/8	7 3/8	6 1/4	9 7/32
16	3 9/32	5	4	6	4 11/16	7 1/64	5 1/2	8 1/64	7	10 1/32
17	3 21/32	5 13/32	4 15/32	6 31/64	5 1/16	7 37/64	5 11/16	8 21/32	7 7/16	10 27/32
18	3 25/32	5 51/64	4 21/32	6 31/32	5 5/8	8 9/64	6 1/4	9 5/16	8 1/8	11 41/64
19	4 3/16	6 13/64	4 15/16	7 29/64	5 11/16	8 45/64	6 7/8	9 61/64	9	12 7/16
20	4 19/32	6 39/64	5 7/16	7 15/16	6 1/4	9 17/64	7	10 19/32	9 3/4	13 1/4
21	4 11/16	7	5 11/16	8 27/64	6 13/16	9 53/64	7 3/4	11 15/64	10	14 3/64
22	4 7/8	7 13/32	5 7/8	8 57/64	7 1/4	10 25/64	8 3/8	11 7/8	10 7/8	14 27/32
23	5 5/16	7 13/16	6 3/8	9 3/8	7 7/16	10 15/16	9	12 33/64	11 5/8	15 21/32
24	5 11/16	8 13/64	6 13/16	9 55/64	8	11 1/2	9 5/8	13 5/32	13	16 29/64
25	5 23/32	8 39/64	7 1/4	10 11/32	8 9/16	12 1/16	10 1/4	13 51/64	13 1/2	17 1/4

All dimensions in inches.

For standard key dimensions see pages 2342 through 2343.

Source: American Chain Association.

Center Distance between Sprockets.—The center-to-center distance between sprockets, as a general rule, should not be less than 1 1/2 times the diameter of the larger sprocket and not less than thirty times the pitch nor more than about 50 times the pitch, although much depends upon the speed and other conditions. A center distance equivalent to 80 pitches may be considered an approved maximum. Very long center distances result in catenary tension in the chain. If roller-chain drives are designed correctly, the center-to-center distance for some transmissions may be so short that the sprocket teeth nearly touch each other, assuming that the load is not too great and the number of teeth is not too small. To avoid interference of the sprocket teeth, the center distance must, of course, be somewhat greater than one-half the sum of the outside diameters of the sprockets. The chain should extend around at least 120 degrees of the pinion circumference, and this minimum amount of contact is obtained for all center distances provided the ratio is less than 3½ to 1. Other things being equal, a fairly long chain is recommended in preference to the shortest one allowed by the sprocket diameters, because the rate of chain elongation due to natural wear is inversely proportional to the length, and also because the greater elasticity of the longer strand tends to absorb irregularities of motion and to decrease the effect of shocks.

If possible, the center distance should be adjustable in order to take care of slack due to elongation from wear and this range of adjustment should be at least one and one-half pitches. A little slack is desirable as it allows the chain links to take the best position on the sprocket teeth and reduces the wear on the bearings. Too much sag or an excessive distance between the sprockets may cause the chain to whip up and down — a condition detrimental to smooth running and very destructive to the chain. The sprockets should run in a vertical plane, the sprocket axes being approximately horizontal, unless an idler is used on the slack side to keep the chain in position. The most satisfactory results are obtained when the slack side of the chain is on the bottom.

Center Distance for a Given Chain Length.—When the distance between the driving and driven sprockets can be varied to suit the length of the chain, this center distance for a tight chain may be determined by the following formula, in which c = center-to-center distance in inches; L = chain length in pitches; P = pitch of chain; N = number of teeth in large sprocket; n = number of teeth in small sprocket.

$$c = \frac{P}{8}\left(2L - N - n + \sqrt{(2L - N - n)^2 - 0.810(N - n)^2}\right)$$

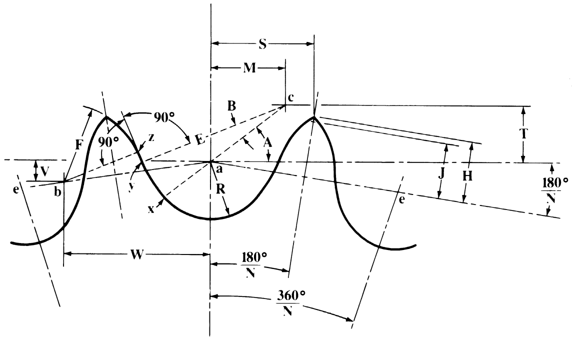
This formula is approximate, but the error is less than the variation in the length of the best chains. The length L in pitches should be an even number for a roller chain, so that the use of an offset connecting link will not be necessary.

Idler Sprockets.—When sprockets have a fixed center distance or are non-adjustable, it may be advisable to use an idler sprocket for taking up the slack. The idler should preferably be placed against the slack side between the two strands of the chain. When a sprocket is applied to the tight side of the chain to reduce vibration, it should be on the lower side and so located that the chain will run in a straight line between the two main sprockets. A sprocket will wear excessively if the number of teeth is too small and the speed too high, because there is impact between the teeth and rollers even though the idler carries practically no load.

Length of Driving Chain.—The total length of a block chain should be given in multiples of the pitch, whereas for a roller chain, the length should be in multiples of twice the pitch, because the ends must be connected with an outside and inside link. The length of a chain can be calculated accurately enough for ordinary practice by the use of the following formula, in which L = chain length in pitches; C = center distance in pitches; N = number of teeth in large sprocket; n = number of teeth in small sprocket:

$$L = 2C + \frac{N}{2} + \frac{n}{2} + \left(\frac{N - n}{2\pi}\right)^2 \times \frac{1}{C}$$

Table 10. ANSI Sprocket Tooth Form for Roller Chain *ANSI/ASME B29.1M-1993*



Seating Curve Data—Inches

P	D_r	Min. R	Min. D_s	D_s Tol.[a]	P	D_r	Min. R	Min. D_s	D_s Tol.[a]
0.250	0.130	0.0670	0.134	0.0055	1.250	0.750	0.3785	0.757	0.0070
0.375	0.200	0.1020	0.204	0.0055	1.500	0.875	0.4410	0.882	0.0075
0.500	0.306	0.1585	0.317	0.0060	1.750	1.000	0.5040	1.008	0.0080
0.500	0.312	0.1585	0.317	0.0060	2.000	1.125	0.5670	1.134	0.0085
0.625	0.400	0.2025	0.405	0.0060	2.250	1.406	0.7080	1.416	0.0090
0.750	0.469	0.2370	0.474	0.0065	2.500	1.562	0.7870	1.573	0.0095
1.000	0.625	0.3155	0.631	0.0070	3.000	1.875	0.9435	1.887	0.0105

[a] Plus tolerance only.

P = pitch (ae)

N = number of teeth D_r = nominal roller diameter

D_s = seating curve diameter = $1.005\,D_r + 0.003$ (in inches)

$R = \frac{1}{2} D_s$ (D_s has only plus tolerance)

$A = 35° + (60° \div N)$ $B = 18° - (56° \div N)$ $ac = 0.8\,D_r$

$M = 0.8\,D_r \cos(35° + (60° \div N))$

$T = 0.8\,D_r \sin(35° + (60° \div N))$

$E = 1.3025\,D_r + 0.0015$ (in inches)

Chord $xy = (2.605\,D_r + 0.003)\sin 9° - (28° \div N))$ (in inches)

$yz = D_r\,[1.4 \sin(17° - (64° \div N)) - 0.8 \sin(18° - (56° \div N))]$

Length of a line between a and $b = 1.4\,D_r$

$W = 1.4\,D_r \cos(180° \div N)$; $V = 1.4\,D_r \sin(180° \div N)$

$F = D_r\,[0.8 \cos(18° - (56° \div N)) + 1.4 \cos(17° - (64° \div N)) - 1.3025] - 0.0015$ inch

$H = \sqrt{F^2 - (1.4D_r - 0.5P)^2}$

$S = 0.5\,P \cos(180° \div N) + H \sin(180° \div N)$

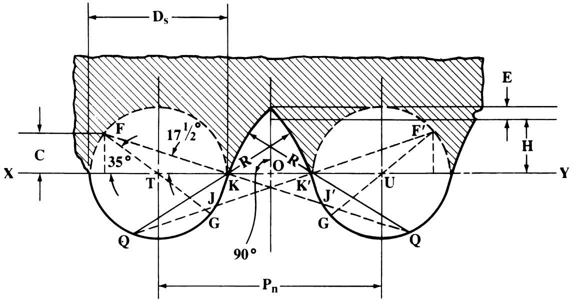
Approximate O.D. of sprocket when J is $0.3\,P = P\,[0.6 + \cot(180° \div N)]$

O.D. of sprocket when tooth is pointed $+ P \cot(180° \div N) + \cos(180° \div N)(D_s - D_r) + 2H$

Pressure angle for new chain $= xab = 35° - (120° \div N)$

Minimum pressure angle $= xab - B = 17° - (64° \div N)$;
Average pressure angle $= 26° - (92° \div N)$

Table 11. Standard Hob Design for Roller Chain Sprockets



Section Normal to Hob Teeth

Data for Laying Out Hob Outlines — Inches								
P	P_n	H	E	O.D.	W	Bore	Keyway	No. Gashes
1/4	0.2527	0.0675	0.0075	2 5/8	2 1/2	1.250	1/4 × 1/8	13
3/8	0.379	0.101	0.012	3 1/8	2 1/2	1.250	1/4 × 1/8	13
1/2	0.506	0.135	0.015	3 3/8	2 1/2	1.250	1/4 × 1/8	12
5/8	0.632	0.170	0.018	3 5/8	2 1/2	1.250	11/4 × 1/8	12
3/4	0.759	0.202	0.023	3 3/4	2 7/8	1.250	1/4 × 1/8	11
1	1.011	0.270	0.030	4 3/8	3 3/4	1.250	1/4 × 1/8	11
1 1/4	1.264	0.337	0.038	4 3/4	4 1/2	1.250	1/4 × 1/8	10
1 1/2	1.517	0.405	0.045	5 3/8	5 1/4	1.250	1/4 × 1/8	10
1 3/4	1.770	0.472	0.053	6 3/8	6	1.500	3/8 × 3/16	9
2	2.022	0.540	0.060	6 7/8	6 3/4	1.500	3/8 × 3/16	9
2 1/4	2.275	0.607	0.068	8	8 1/2	1.750	3/8 × 3/16	8
2 1/2	2.528	0.675	0.075	8 5/8	9 3/8	1.750	3/8 × 3/16	8
3	3.033	0.810	0.090	9 3/4	11 1/4	2.000	1/2 × 3/6	8

Hobs designed for a given roller diameter (D_r) and chain pitch (P) will cut any number of teeth.

P = Pitch of Chain

P_n = Normal Pitch of Hob = 1.011 P inches

D_s = Minimum Diameter of Seating Curve = 1.005 D_r + 0.003 inches

F = Radius Center for Arc GK; $TO = OU = P_n \div 2$

$H = 0.27\,P$; $E = 0.03\,P$ = Radius of Fillet Circle

Q is located on line passing through F and J. Point J is intersection of line XY with circle of diameter D_s. R is found by trial and the arc of this radius is tangent to arc KG at K and to fillet radius.

OD = Outside Diameter = 1.7 (Bore + D_r + 0.7 P) approx.

D_h = Pitch Diameter = $OD - D_s$; M = Helix Angle; $\sin M = P_n \div \pi D_h$

L = Lead = $P_n \div \cos M$; W = Width = Not less than 2 × Bore, or 6 D_r, or 3.2 P

To the length obtained by this formula, add enough to make a whole number (and for a roller chain, an even number) of pitches. If a roller chain has an odd number of pitches, it will be necessary to use an offset connecting link.

Another formula for obtaining chain length in which D = distance between centers of shafts; R = pitch radius of large sprocket; r = pitch radius of small sprocket: N = number of teeth in large sprocket; n = number of teeth in small sprocket; P = pitch of chain and sprockets; and l = required chain length in inches, is:

$$l = \frac{180° + 2\alpha}{360°}NP + \frac{180° - 2\alpha}{360°}nP + 2D\cos\alpha \qquad \text{where } \sin\alpha = \frac{R - r}{D}$$

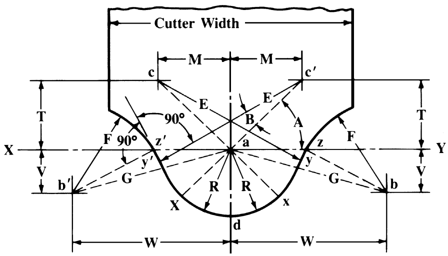
Cutting Standard Sprocket Tooth Form.—The proportions and seating curve data for the standard sprocket tooth form for roller chain are given in Table 10. Either formed or generating types of sprocket cutters may be employed.

Hobs: Only one hob will be required to cut any number of teeth for a given pitch and roller diameter. All hobs should be marked with pitch and roller diameter to be cut. Formulas and data for standard hob design are given in Table 11.

Space Cutters: Five cutters of this type will be required to cut from 7 teeth up for any given roller diameter. The ranges are, respectively, 7–8, 9–11, 12–17, 18–34, and 35 teeth and over. If less than 7 teeth is necessary, special cutters conforming to the required number of teeth should be used.

The regular cutters are based upon an intermediate number of teeth N_a, equal to $2N_1N_2 \div (N_1 + N_2)$ in which N_1 = minimum number of teeth and N_2 = maximum number of teeth for which cutter is intended; but the topping curve radius F (see diagram in Table 12) is designed to produce adequate tooth height on a sprocket of N_2 teeth. The values of N_a for the several cutters are, respectively, 7.47, 9.9, 14.07, 23.54, and 56. Formulas and construction data for space cutter layout are given in Table 12 and recommended cutter sizes are given in Table 13.

Table 12. Standard Space Cutters for Roller-Chain Sprockets



Data for Laying Out Space Cutter

Range of Teeth	*M*	*T*	*W*	*V*
7–8	0.5848 D_r	0.5459 D_r	1.2790 D_r	0.5694 D_r
9–11	0.6032 D_r	0.5255 D_r	1.3302 D_r	0.4365 D_r
12–17	0.6194 D_r	0.5063 D_r	1.3694 D_r	0.2911 D_r
18–34	0.6343 D_r	0.4875 D_r	1.3947 D_r	0.1220 D_r
35 up	0.6466 D_r	0.4710 D_r	1.4000 D_r	0
Range of Teeth	*F*	Chord *xy*	*yz*	Angle *Yab*
7–8	0.8686 D_r – 0.0015	0.2384 D_r + 0.0003	0.0618 D_r	24°
9–11	0.8554 D_r – 0.0015	0.2800 D_r + 0.0003	0.0853 D_r	18° 10′
12–17	0.8364 D_r – 0.0015	0.3181 D_r + 0.0004	0.1269 D_r	12°
18–34	0.8073 D_r – 0.0015	0.3540 D_r + 0.0004	0.1922 D_r	5°
35 up	0.7857 D_r – 0.0015	0.3850 D_r + 0.0004	0.2235 D_r	0°

E (same for all ranges) = 1.3025 D_r + 0.0015; *G* (same for all ranges) = 1.4 D_r

See Table 13 for recommended cutter sizes.

Angle *Yab* is equal to 180° ÷ *N* when the cutter is made for a specific number of teeth. For the design of cutters covering a range of teeth, angle *Yab* was determined by layout to ensure chain roller clearance and to avoid pointed teeth on the larger sprockets of each range. It has values as given below for cutters covering the range of teeth shown. The following formulas are for cutters covering the standard ranges of teeth where N_a equals intermediate values given on page 2440.

$$W = 1.4D_r \cos Yab \qquad V = 1.4D_r \sin Yab$$

$$yz = D_r\left[1.4\sin\left(17° + \frac{116°}{N_a} - Yab\right) - 0.8\sin\left(18° - \frac{56°}{N_a}\right)\right]$$

$$F = D_r\left[0.8\cos\left(18° - \frac{56°}{N_a}\right) + 1.4\cos\left(17° + \frac{116°}{N_a} - Yab\right) - 1.3025\right] - 0.0015 \text{ in.}$$

For other points, use the value of N_a for N in the standard formulas in Table 10.

Table 13. Recommended Space Cutter Sizes for Roller-Chain Sprockets

		Number of Teeth					
		6	7–8	9–11	12–17	18–34	35 up
Pitch	Roller Dia.	Cutter Diameter (Minimum)					
0.250	0.130	2.75	2.75	2.75	2.75	2.75	2.75
0.375	0.200	2.75	2.75	2.75	2.75	2.75	2.75
0.500	0.312	3.00	3.00	3.12	3.12	3.12	3.12
0.625	0.400	3.12	3.12	3.25	3.25	3.25	3.25
0.725	0.469	3.25	3.25	3.38	3.38	3.38	3.38
1.000	0.625	3.88	4.00	4.12	4.12	4.25	4.25
1.250	0.750	4.25	4.38	4.50	4.50	4.62	4.62
1.500	0.875	4.38	4.50	4.62	4.62	4.75	4.75
1.750	1.000	5.00	5.12	5.25	5.38	5.50	5.50
2.000	1.125	5.38	5.50	5.62	5.75	5.88	5.88
2.250	1.406	5.88	6.00	6.25	6.38	6.50	6.50
2.500	1.563	6.38	6.62	6.75	6.88	7.00	7.12
3.000	1.875	7.50	7.75	7.88	8.00	8.00	8.25
Pitch	Roller Dia.	Cutter Width (Minimum)					
0.250	0.130	0.31	0.31	0.31	0.31	0.28	0.28
0.375	0.200	0.47	0.47	0.47	0.44	0.44	0.41
0.500	0.312	0.75	0.75	0.75	0.75	0.72	0.69
0.625	0.400	0.75	0.75	0.75	0.75	0.72	0.69
0.750	0.469	0.91	0.91	0.91	0.88	0.84	0.81
1.000	0.625	1.50	1.50	1.47	1.47	1.41	1.34
1.250	0.750	1.81	1.81	1.78	1.75	1.69	1.62
1.500	0.875	1.81	1.81	1.78	1.75	1.69	1.62
1.750	1.000	2.09	2.09	2.06	2.03	1.97	1.88
2.000	1.125	2.41	2.41	2.38	2.31	2.25	2.16
2.250	1.406	2.69	2.69	2.66	2.59	2.47	2.41
2.500	1.563	3.00	3.00	2.94	2.91	2.75	2.69
3.000	1.875	3.59	3.59	3.53	3.47	3.34	3.22

Where the same roller diameter is commonly used with chains of two different pitches it is recommended that stock cutters be made wide enough to cut sprockets for both chains.

Marking of Cutters.— All cutters are to be marked, giving pitch, roller diameter and range of teeth to be cut.

Bores for Sprocket Cutters (recommended practice) are approximately as calculated from the formula:

$$\text{Bore} = 0.7\sqrt{(\text{Width of Cutter} + \text{Roller Diameter} + 0.7\text{ Pitch})}$$

and are equal to 1 inch for ¼- through ¾-inch pitches; 1¼ inches for 1- through 1½-inch for 1¾- through 2¼-inch pitches; 1¾ inches for 2½-inch pitch; and 2 inches for 3-inch pitch.

Minimum Outside Diameters of Space Cutters for 35 teeth and over (recommended practice) are approximately as calculated from the formula:

$$\text{Outside Diameter} = 1.2(\text{Bore} + \text{Roller Diameter} + 0.7\text{ Pitch}) + 1\text{ in.}$$

Shaper Cutters: Only one will be required to cut any number of teeth for a given pitch and roller diameter. The manufacturer should be referred to for information concerning the cutter form design to be used.

Sprocket Manufacture.—Cast sprockets have cut teeth, and the rim, hub face, and bore are machined. The smaller sprockets are generally cut from steel bar stock, and are finished all over. Sprockets are often made from forgings or forged bars. The extent of finishing depends on the particular specifications that are applicable. Many sprockets are made by welding a steel hub to a steel plate. This process produces a one-piece sprocket of desired proportions and one that can be heat-treated.

Sprocket Materials.—For large sprockets, cast iron is commonly used, especially in drives with large speed ratios, since the teeth of the larger sprocket are subjected to fewer chain engagements in a given time. For severe service, cast steel or steel plate is preferred.

The smaller sprockets of a drive are usually made of steel. With this material the body of the sprocket can be heat-treated to produce toughness for shock resistance, and the tooth surfaces can be hardened to resist wear.

Stainless steel or bronze may be used for corrosion resistance, and Formica, nylon or other suitable plastic materials for special applications.

Roller Chain Drive Ratings.—In 1961, under auspices of The American Sprocket Chain Manufacturers Association (now called American Chain Association), a joint research program was begun to study pin-bushing interaction at high speeds and to gain further data on the phenomenon of chain joint galling among other research areas. These studies have shown that a separating film of lubricant is formed in chain joints in a manner similar to that found in journal bearings. These developments appear in ANSI/ASME B29.1M-1993, and are contained in Table . The ratings shown in Table are below the galling range.

The horsepower ratings in Table 14 apply to lubricated, single-pitch, single-strand roller chains, both ANSI Standard and Heavy series. To obtain ratings of multiple-strand chains, a multiple-strand factor is applied.

The ratings in Table 14 are based upon: 1) A service factor of 1.; 2) A chain length of approximately 100 pitches.; 3) Use of recommended lubrication methods.; and 4) A drive arrangement where two aligned sprockets are mounted on parallel shafts in a horizontal plane..

Under these conditions, approximately 15,000 hours of service life at full load operation may be expected.

Table 14. Roller Chain Drive Service Factors

Type of Driven Load	Type of Input Power		
	Internal Combustion Engine with Hydraulic Drive	Electric Motor or Turbine	Internal Combustion Engine with Mechanical Drive
Smooth	1.0	1.0	1.2
Moderate Shock	1.2	1.3	1.4
Heavy Shock	1.4	1.5	1.7

Substantial increases in rated speed loads can be utilized, as when a service life of less than 15,000 hours is satisfactory, or when full load operation is encountered only during a portion of the required service life. Chain manufacturers should be consulted for assistance with any special application requirements.

The horsepower ratings shown in Table relate to the speed of the smaller sprocket and drive selections are made on this basis, whether the drive is speed reducing or speed increasing. Drives with more than two sprockets, idlers, composite duty cycles, or other unusual conditions often require special consideration. Where quietness or extra smooth operation are of special importance, small-pitch chain operating over large diameter sprockets will minimize noise and vibration.

When making drive selection, consideration is given to the loads imposed on the chain by the type of input power and the type of equipment to be driven. Service factors are used to compensate for these loads and the *required* horsepower rating of the chain is determined by the following formula:

$$\text{Required hp Table Rating} = \frac{\text{hp to be Transmitted} \times \text{Service Factor}}{\text{Multiple-Strand Factor}}$$

Service Factors: The service factors in Table 14 are for normal chain loading. For unusual or extremely severe operating conditions not shown in this table, it is desirable to use larger service factors.

Multiple-Strand Factors: The horsepower ratings for multiple-strand chains equal single-strand ratings multiplied by these factors: for two strands, a factor of 1.7; for three strands, 2.5; and and for four strands, 3.3.

Lubrication.—It has been shown that a separating wedge of fluid lubricant is formed in operating chain joints much like that formed in journal bearings. Therefore, fluid lubricant must be applied to ensure an oil supply to the joints and minimize metal-to-metal contact. If supplied in sufficient volume, lubrication also provides effective cooling and impact damping at higher speeds. For this reason, it is important that lubrication recommendations be followed. *The ratings in* Table *apply only to drives lubricated in the manner specified in this table*.

Chain drives should be protected against dirt and moisture and the oil supply kept free of contamination. Periodic oil change is desirable. A good grade of non-detergent petroleum base oil is recommended. Heavy oils and greases are generally too stiff to enter and fill the chain joints. The following lubricant viscosities are recommended: For temperatures of 20° to 40°F, use SAE 20 lubricant; for 40° to 100°, use SAE 30; for 100° to 120°, use SAE 40; and for 120° to 140°, use SAE 50.

There are three basic types of lubrication for roller chain drives. The recommended type shown in Table as Type A, Type B, or Type C is influenced by the chain speed and the amount of power transmitted. These are *minimum* lubrication requirements and the use of a better type (for example, Type C instead of Type B) is acceptable and may be beneficial. Chain life can vary appreciably depending upon the way the drive is lubricated. The better the chain lubrication, the longer the chain life. For this reason, it is important that the lubrication recommendations be followed when using the ratings given in Table . The types of lubrication are as follows:

Type A — Manual or Drip Lubrication: In manual lubrication, oil is applied copiously with a brush or spout can at least once every eight hours of operation. Volume and frequency should be sufficient to prevent overheating of the chain or discoloration of the chain joints. In drip lubrication, oil drops from a drip lubricator are directed between the link plate edges. The volume and frequency should be sufficient to prevent discoloration of the lubricant in the chain joints. Precautions must be taken against misdirection of the drops by windage.

Type B — Bath or Disc Lubrication: In bath lubrication, the lower strand of the chain runs through a sump of oil in the drive housing. The oil level should reach the pitch line of the chain at its lowest point while operating. In disc lubrication, the chain operates above the oil level. The disc picks up oil from the sump and deposits it onto the chain, usually by means of a trough. The diameter of the disc should be such as to produce rim speeds of between 600 and 8000 feet per minute.

Type C — Oil Stream Lubrication: The lubricant is usually supplied by a circulating pump capable of supplying each chain drive with a continuous stream of oil. The oil should be applied inside the chain loop evenly across the chain width, and directed at the slack strand.

The chain manufacturer should be consulted when it appears desirable to use a type of lubricant other than that recommended.

Installation and Alignment.—Sprockets should have the tooth form, thickness, profile, and diameters conforming to ASME/ANSI B29.1M. For maximum service life small sprockets operating at moderate to high speeds, or near the rated horsepower, should have hardened teeth. Normally, large sprockets should not exceed 120 teeth.

In general a center distance of 30 to 50 chain pitches is most desirable. The distance between sprocket centers should provide at least a 120 degree chain wrap on the smaller sprocket. Drives may be installed with either adjustable or fixed center distances. Adjustable centers simplify the control of chain slack. Sufficient housing clearance must always be provided for the chain slack to obtain full chain life.

Accurate alignment of shafts and sprocket tooth faces provides uniform distribution of the load across the entire chain width and contributes substantially to optimum drive life.

Shafting, bearings, and foundations should be suitable to maintain the initial alignment. Periodic maintenance should include an inspection of alignment.

Example of Roller Chain Drive Design Procedure.—The selection of a roller chain and sprockets for a specific design requirement is best accomplished by a systematic step-by-step procedure such as is used in the following example.

Example: Select a roller chain drive to transmit 10 horsepower from a countershaft to the main shaft of a wire drawing machine. The countershaft is $1\frac{15}{16}$-inches diameter and operates at 1000 rpm. The main shaft is also $1\frac{15}{16}$-inches diameter and must operate between 378 and 382 rpm. Shaft centers, once established, are fixed and by initial calculations must be approximately $22\frac{1}{2}$ inches. The load on the main shaft is uneven and presents "peaks," which place it in the heavy shock load category. The input power is supplied by an electric motor. The driving head is fully enclosed and all parts are lubricated from a central system.

Step 1. Service Factor: From Table 14 the service factor for heavy shock load and an electric motor drive is 1.5.

Step 2. Design Horsepower: The horsepower upon which the chain selection is based (design horsepower) is equal to the specified horsepower multiplied by the service factor, $10 \times 1.5 = 15$ hp.

Step 3. Chain Pitch and Small Sprocket Size for Single-Strand Drive: In Table under 1000 rpm, a $\frac{5}{8}$-inch pitch chain with a 24-tooth sprocket or a $\frac{3}{4}$-inch pitch chain with a 15-tooth sprocket are possible choices.

Step 4. Check of Chain Pitch and Sprocket Selection: From Table 9 it is seen that only the 24-tooth sprocket in Step 3 can be bored to fit the $1\frac{15}{16}$-inch diameter main shaft. In Table a $\frac{5}{8}$-pitch chain at a small sprocket speed of 1000 rpm is rated at 15.5 hp for a 24-tooth sprocket.

Step 5. Selection of Large Sprocket: Since the driver is to operate at 1000 rpm and the driven at a minimum of 378 rpm, the speed ratio 1000⁄378 = 2.646. Therefore the large sprocket should have $24 \times 2.646 = 63.5$ (use 63) teeth.

This combination of 24 and 63 teeth will produce a main drive shaft speed of 381 rpm which is within the limitation of 378 to 382 rpm established in the original specification.

Step 6. Computation of Chain Length: Since the 24- and 63-tooth sprockets are to be placed on $22\frac{1}{2}$-inch centers, the chain length is determined from the formula:

$$L = 2C + \frac{N}{2} + \frac{n}{2} + \left(\frac{N-n}{2\pi}\right)^2 \times \frac{1}{C}$$

where L = chain length in pitches; C = shaft center distance in pitches; N = number of teeth in large sprocket; and n = number of teeth in small sprocket.

$$L = 2 \times 36 + \frac{63 + 24}{2} + \left(\frac{63 - 24}{6.28}\right)^2 \times \frac{1}{36} = 116.57 \text{ pitches}$$

Step 7. Correction of Center Distance: Since the chain is to couple at a whole number of pitches, 116 pitches will be used and the center distance recomputed based on this figure using the formula on page 2437 where c is the center distance in inches and P is the pitch.

$$c = \frac{P}{8}(2L - N - n + \sqrt{(2L - N - n)^2 - 0.810(N - n)^2})$$

$$c = \frac{5}{64}(2 \times 116 - 63 - 24 + \sqrt{(2 \times 116 - 63 - 24)^2 - 0.810(63 - 24)^2})$$

$$c = \frac{5}{64}(145 + 140.69) = 22.32 \text{ inches, say } 22\frac{3}{8} \text{ inches}$$

STANDARDS FOR ELECTRIC MOTORS

Classes of NEMA Standards.—National Electrical Manufacturers Association Standards, available from the Association at 2101 L Street, NW, Washington, DC 20037, are of two classes: 1) *NEMA Standard,* which relates to a product commercially standardized and subject to repetitive manufacture, which standard has been approved by at least 90 per cent of the members of the Subdivision eligible to vote thereon; and 2) *Suggested Standard for Future Design,* which may not have been regularly applied to a commercial product, but which suggests a sound engineering approach to future development and has been approved by at least two-thirds of the members of the Subdivision eligible to vote thereon.

Authorized Engineering Information consists of explanatory data and other engineering information of an informative character not falling within the classification of NEMA Standard or Suggested Standard for Future Design.

Mounting Dimensions and Frame Sizes for Electric Motors.—Dimensions for foot-mounted electric motors as standardized in the United States by the National Electrical Manufacturers Association (NEMA) include the spacing of bolt holes in the feet of the motor, the distance from the bottom of the feet to the center-line of the motor shaft, the size of the conduit, the length and diameter of shaft, and other dimensions likely to be required by designers or manufacturers of motor-driven equipment. The Standard provides dimensions for face-mounted and flange-mounted motors by means of standard motor frame numbers.

Standard dimensions also are given where the motor is to be mounted upon a belt-tightening base or upon rails.

The NEMA standards also prescribe lettering for dimension drawings, mounting and terminal housing locations and dimensions, symbols and terminal connections, and provision for grounding of field wiring. In addition, the standards give recommended knock-out and clearance hole dimensions; tolerances on shaft extension diameters and keyseats; methods of measuring shaft run-out and eccentricity, also face runout of mounting surfaces; and tolerances of face-mounted and flanged-mounted motors.

Design Letters of Polyphase Integral-horsepower Motors.—Designs A, B, C, and D motors are squirrel-cage motors designed to withstand full voltage starting and developing locked-rotor torque and breakdown torque, drawing locked-rotor current, and having a slip as specified below:

Design A: Locked-rotor torque as shown in Table 2, breakdown torque as shown in Table 3, locked-rotor current higher than the values shown in Table 1, and a slip at rated load of less than 5 per cent. Motors with 10 or more poles may have a slightly greater slip.

Table 1. NEMA Standard Locked-rotor Current of 3-phase 60-hertz Integral-horsepower Squirrel-cage Induction Motors Rated at 230 Volts

Horsepower	Locked-rotor Current, Amps.	Design Letters	Horsepower	Locked-rotor Current, Amps.	Design Letters	Horsepower	Locked-rotor Current, Amps.	Design Letters
½	20	B, D	7½	127	B, C, D	50	725	B, C, D
¾	25	B, D	10	162	B, C, D	60	870	B, C, D
1	30	B, D	15	232	B, C, D	75	1085	B, C, D
1½	40	B, D	20	290	B, C, D	100	1450	B, C, D
2	50	B, D	25	365	B, C, D	125	1815	B, C, D
3	64	B, C, D	30	435	B, C, D	150	2170	B, C, D
5	92	B, C, D	40	580	B, C, D	200	2900	B, C

Note: The locked-rotor current of a motor is the steady-state current taken from the line with the rotor locked and with rated voltage and frequency applied to the motor.

For motors designed for voltages other than 230 volts, the locked-rotor current is inversely proportional to the voltages. For motors larger than 200 hp, see NEMA Standard MG 1-12.34.

Table 2. NEMA Standard Locked-rotor Torque of Single-speed Polyphase 60- and 50-hertz Squirrel-cage Integral-horsepower Motors with Continuous Ratings

Hp	Designs A and B							Design C		
	Synchronous Speed, rpm									
60 hertz	3600	1800	1200	900	720	600	514	1800	1200	900
50 hertz	3000	1500	1000	750	…	…	…	1500	1000	750
	Percent of Full-load Torque[a]									
½	…	…	…	140	140	115	110	…	…	…
¾	…	…	175	135	135	115	110	…	…	…
1	…	275	170	135	135	115	110	…	…	…
1½	175	250	165	130	130	115	110	…	…	…
2	170	235	160	130	125	115	110	…	…	…
3	160	215	155	130	125	115	110	…	250	225
5	150	185	150	130	125	115	110	250	250	225
7½	140	175	150	125	120	115	110	250	225	200
10	135	165	150	125	120	115	110	250	225	200
15	130	160	140	125	120	115	110	225	200	200
20	130	150	135	125	120	115	110	200 for all sizes above 15 hp.		
25	130	150	135	125	120	115	110			
30	130	150	135	125	120	115	110			
40	125	140	135	125	120	115	110			
50	120	140	135	125	120	115	110			
60	120	140	135	125	120	115	110			
75	105	140	135	125	120	115	110	For Design D motors, see footnote.		
100	105	125	125	125	120	115	110			
125	100	110	125	120	115	115	110			
150	100	110	120	120	115	115	…			
200	100	100	120	120	115	…	…			

[a] These values represent the upper limit of application for these motors.

Note: The locked-rotor torque of a motor is the minimum torque which it will develop at rest for all angular positions of the rotor, with rated voltage applied at rated frequency.

The locked-rotor torque of Design D, 60- and 50-hertz 4-, 6-, and 8-pole single-speed, polyphase squirrel-cage motors rated 150 hp and smaller, with rated voltage and frequency applied is 275 per cent of full-load torque, which represents the upper limit of application for these motors.

For motors larger than 200 hp, see NEMA Standard MG 1-12.37.

Table 3. NEMA Standard Breakdown Torque of Single-speed Polyphase Squirrel-cage, Integral-horsepower Motors with Continuous Ratings

Horsepower	Synchronous Speed, rpm						
60 hertz	3600	1800	1200	900	720	600	514
50 hertz	3000	1500	1000	750	…	…	…
	Per Cent of Full Load Torque						
	Designs A and B[a]						
½	…	…	…	225	200	200	200
¾	…	…	275	220	200	200	200
1	…	300	265	215	200	200	200
1½	250	280	250	210	200	200	200
2	240	270	240	210	200	200	200
3	230	250	230	205	200	200	200
5	215	225	215	205	200	200	200
7½	200	215	205	200	200	200	200
10–125, incl.	200	200	200	200	200	200	200
150	200	200	200	200	200	200	…
200	200	200	200	200	200	…	…
	Design C						
3	…	…	225	200	…	…	…
5	…	200	200	200	…	…	…
7½–200, incl.	…	190	190	190	…	…	…

[a] Design A values are in excess of those shown.

These values represent the upper limit of the range of application for these motors. For above 200 hp, see NEMA Standard MG1-12.38.

Design B: Locked-rotor torque as shown in Table 2, breakdown torque as shown in Table 3, locked-rotor current not exceeding that in Table 1, and a slip at rated load of less than 5 per cent. Motors with 10 or more poles may have a slightly greater slip.

Design C: Locked-rotor torque for special high-torque applications up to values shown in Table 2, breakdown torque up to values shown in Table 3, locked-rotor current not exceeding values shown in Table 1 and a slip at rated load of less than 5 per cent.

Design D: Locked-rotor torque as indicated in Table 2, locked-rotor current not greater than that shown in Table 1 and a slip at rated load of 5 per cent or more.

Torque and Current Definitions.—The definitions which follow have been adopted as standard by the National Electrical Manufacturers Association.

Locked-Rotor or Static Torque: The locked-rotor torque of a motor is the minimum torque which it will develop at rest for all angular positions of the rotor, with rated voltage applied at rated frequency.

Breakdown Torque: The breakdown torque of a motor is the maximum torque which the motor will develop, with rated voltage applied at rated frequency, without an abrupt drop in speed (see Table 4).

Full-Load Torque: The full-load torque of a motor is the torque necessary to produce its rated horsepower at full load speed. In pounds at 1-foot radius, it is equal to the horsepower times 5252 divided by the full-load speed.

Pull-Out Torque: The pull-out torque of a synchronous motor is the maximum sustained torque which the motor will develop at synchronous speed with rated voltage applied at rated frequency and with normal excitation.

Pull-In Torque: The pull-in torque of a synchronous motor is the maximum constant torque under which the motor will pull its connected inertia load into synchronism at rated voltage and frequency, when its field excitation is applied.

Pull-Up Torque: The pull-up torque of an alternating current motor is the minimum torque developed by the motor during the period of acceleration from rest to the speed at which breakdown torque occurs. For motors which do not have a definite breakdown torque, the pull-up torque is the minimum torque developed up to rated speed.

Locked Rotor Current: The locked rotor current of a motor is the steady-state current taken from the line with the rotor locked and with rated voltage (and rated frequency in the case of alternating-current motors) applied to the motor.

Table 4. NEMA Standard Breakdown Torque of Polyphase Wound-rotor Motors with Continuous Ratings — 60- and 50-hertz

Horsepower	Speed, rpm			Horsepower	Speed, rpm		
	1800	1200	900		1800	1200	900
	Per cent of Full-load Torque				Per cent of Full-load Torque		
1	…	…	250	7½	275	250	225
1½	…	…	250	10	275	250	225
2	275	275	250	15	250	225	225
3	275	275	250	20–200, incl.	225	225	225
5	275	275	250	…	…	…	…

These values represent the upper limit of the range of application for these motors.

Standard Direction of Motor Rotation.—The standard direction of rotation for all non-reversing direct-current motors, all alternating-current single-phase motors, all synchronous motors, and all universal motors, is *counterclockwise* when facing that end of the motor opposite the drive.

This rule does not apply to two- and three-phase induction motors, as in most applications the phase sequence of the power lines is rarely known.

Motor Types According to Variability of Speed.—Five types of motors classified according to variability of speed are:

Constant-speed Motors: In this type of motor the normal operating speed is constant or practically constant; for example, a synchronous motor, an induction motor with small slip, or a direct-current shunt-wound motor.

Varying-speed Motor: In this type of motor, the speed varies with the load, ordinarily decreasing when the load increases; such as a series-wound or repulsion motor.

Adjustable-speed Motor: In this type of motor, the speed can be varied gradually over a considerable range, but when once adjusted remains practically unaffected by the load; such as a direct-current shunt-wound motor with field resistance control designed for a considerable range of speed adjustment.

The base speed of an adjustable-speed motor is the lowest rated speed obtained at rated load and rated voltage at the temperature rise specified in the rating.

Adjustable Varying-speed Motor: This type of motor is one in which the speed can be adjusted gradually, but when once adjusted for a given load will vary in considerable degree with the change in load; such as a direct-current compound-wound motor adjusted by field control or a wound-rotor induction motor with rheostatic speed control.

Multispeed Motor: This type of motor is one which can be operated at any one of two or more definite speeds, each being practically independent of the load; such as a direct-current motor with two armature windings or an induction motor with windings capable of various pole groupings. In the case of multispeed permanent-split capacitor and shaded pole motors, the speeds are dependent upon the load.

Pull-up Torque.—NEMA Standard pull up torques for single-speed, polyphase, squirrel-cage integral-horsepower motors, Designs A and B, with continuous ratings and with rated voltage and frequency applied are as follows: When the locked-rotor torque given in Table 2 is 110 per cent or less, the pull-up torque is 90 per cent of the locked-rotor torque;

when the locked-rotor torque is greater than 110 per cent but less than 145 per cent, the pull-up torque is 100 per cent of full-load torque; and when the locked-rotor torque is 145 per cent or more, the pull-up torque is 70 per cent of the locked-rotor torque.

For Design C motors, with rated voltage and frequency applied, the pull-up torque is not less than 70 per cent of the locked-rotor torque as given in Table 2.

Types and Characteristics of Electric Motors

Types of Direct-Current Motors.—Direct-current motors may be grouped into three general classes: series-wound; shunt-wound; and compound-wound.

In the *series-wound motor* the field windings, which are fixed in the stator frame, and the armature windings, which are placed around the rotor, are connected in series so that all current passing through the armature also passes through the field. In the *shunt-wound motor,* both armature and field are connected across the main power supply so that the armature and field currents are separate. In the *compound-wound motor,* both series and shunt field windings are provided and these may be connected so that the currents in both are flowing in the same direction, called *cumulative compounding,* or so that the currents in each are flowing in opposite directions, called *differential compounding.*

Characteristics of Series-wound Direct-Current Motors.—In the series-wound motor, any increase in load results in more current passing through the armature and the field windings. As the field is strengthened by this increased current, the motor speed decreases. Conversely, as the load is decreased the field is weakened and the speed increases and at very light loads may become excessive. For this reason, series-wound direct-current motors are usually directly connected or geared to the load to prevent "run-away." (A series-wound motor, designated as series-shunt wound, is sometimes provided with a light shunt field winding to prevent dangerously high speeds at light loads.) The increase in armature current with increasing load produces increased torque, so that the series-wound motor is particularly suited to heavy starting duty and where severe overloads may be expected. Its speed may be adjusted by means of a variable resistance placed in series with the motor, but due to variation with load, the speed cannot be held at any constant value. This variation of speed with load becomes greater as the speed is reduced. Series-wound motors are used where the load is practically constant and can easily be controlled by hand. They are usually limited to traction and lifting service.

Shunt-wound Direct-Current Motors.—In the shunt-wound motor, the strength of the field is not affected appreciably by change in the load, so that a fairly constant speed (about 10 to 12 per cent drop from no load to full load speed) is obtainable. This type of motor may be used for the operation of machines requiring an approximately constant speed and imposing low starting torque and light overload on the motor.

The shunt-wound motor becomes an adjustable-speed motor by means of field control or by armature control. If a variable resistance is placed in the field circuit, the amount of current in the field windings and hence the speed of rotation can be controlled. As the speed increases, the torque decreases proportionately, resulting in nearly constant horsepower. A speed range of 6 to 1 is possible using field control, but 4 to 1 is more common. Speed regulation is somewhat greater than in the constant-speed shunt-wound motors, ranging from about 15 to 22 per cent. If a variable resistance is placed in the armature circuit, the voltage applied to the armature can be reduced and hence the speed of rotation can be reduced over a range of about 2 to 1. With armature control, speed regulation becomes poorer as speed is decreased, and is about 100 per cent for a 2 to 1 speed range. Since the current through the field remains unchanged, the torque remains constant.

Machine Tool Applications: The adjustable-speed shunt-wound motors are useful on larger machines of the boring mill, lathe, and planer type and are particularly adapted to spindle drives because constant horsepower characteristics permit heavy cuts at low speed and light or finishing cuts at high speed. They have long been used for planer drives because they can provide an adjustable low speed for the cutting stroke and a high speed for the return stroke. Their application has been limited, however, to plants in which direct-current power is available.

Adjustable-voltage Shunt-wound Motor Drive.—More extensive use of the shunt-wound motor has been made possible by a combination drive that includes a means of converting alternating current to direct current. This conversion may be effected by a self-contained unit consisting of a separately excited direct-current generator driven by a constant speed alternating-current motor connected to the regular alternating-current line, or by an electronic rectifier with suitable controls connected to the regular alternating-current supply lines. The latter has the advantage of causing no vibration when mounted directly on the machine tool, an important factor in certain types of grinders.

In this type of adjustable-speed, shunt-wound motor drive, speed control is effected by varying the voltage applied to the armature while supplying constant voltage to the field. In addition to providing for the adjustment of the voltage supplied by the conversion unit to the armature of the shunt-wound motor, the amount of current passing through the motor field may also be controlled. In fact, a single control may be provided to vary the motor speed from minimum to base speed (speed of the motor at full load with rated voltage on armature and field) by varying the voltage applied to the armature and from base speed to maximum speed by varying the current flowing through the field. When so controlled, the motor operates at constant torque up to base speed and at constant horsepower above base speed.

Speed Range: Speed ranges of at least 20 to 1 below base speed and 4 or 5 to 1 above base speed (a total range of 100 to 1, or more) are obtainable as compared with about 2 to 1 below normal speed and 3 or 4 to 1 above normal speed for the conventional type of control. Speed regulation may be as great as 25 per cent at high speeds. Special electronic controls, when used with this type shunt motor drive, make possible maintenance of motor speeds with as little variation as ½ to 1 per cent of full load speed from full load to no load over a line voltage variation of ± 10 per cent and over any normal variation in motor temperature and ambient temperature.

Applications: These direct-current, adjustable-voltage drives, as they are sometimes called, have been applied successfully to such machine tools as planers, milling machines, boring mills and lathes, as well as to other industrial machines where wide, stepless speed control, uniform speed under all operating conditions, constant torque acceleration and adaptability to automatic operation are required.

Compound-wound Motors.—In the compound-wound motor, the speed variation due to load changes is much less than in the series-wound motor, but greater than in the shunt-wound motor (ranging up to 25 per cent from full load to no load). It has a greater starting torque than the shunt-wound motor, is able to withstand heavier overloads, but has a narrower adjustable speed range. Standard motors of this type have a cumulative-compound winding, the differential-compound winding being limited to special applications. They are used where the starting load is very heavy or where the load changes suddenly and violently as with reciprocating pumps, printing presses and punch presses.

Types of Polyphase Alternating-Current Motors.—The most widely used polyphase motors are of the induction type. The *"squirrel cage" induction motor* consists of a wound stator which is connected to an external source of alternating-current power and a laminated steel core rotor with a number of heavy aluminum or copper conductors set into the core around its periphery and parallel to its axis. These conductors are connected together at each end of the rotor by a heavy ring, which provides closed paths for the currents induced in the rotor to circulate. The rotor bars form, in effect, a "squirrel cage" from which the motor takes its name.

Wound-rotor type of *Induction motor:* This type has in addition to a squirrel cage, a series of coils set into the rotor which are connected through slip-rings to external variable resistors. By varying the resistance of the wound-rotor circuits, the amount of current flowing in these circuits and hence the speed of the motor can be controlled. Since the rotor of an induction motor is not connected to the power supply, the motor is said to operate by transfer action and is analogous to a transformer with a short-circuited secondary that is free to rotate. Induction motors are built with a wide range of speed and torque characteristics which are discussed under "Operating Characteristics of Squirrel-cage Induction Motors."

Synchronous Motor: The other type of polyphase alternating-current motor used industrially is the *synchronous motor.* In contrast to the induction motor, the rotor of the synchronous motor is connected to a direct-current supply which provides a field that rotates in step with the alternating-current field in the stator. After having been brought up to synchronous speed, which is governed by the frequency of the power supply and the number of poles in the rotor, the synchronous motor operates at this constant speed throughout its entire load range.

Operating Characteristics of Squirrel-cage Induction Motors.—In general, squirrel-cage induction motors are simple in design and construction and offer rugged service. They are essentially constant-speed motors, their speed changing very little with load and not being subject to adjustment. They are used for a wide range of industrial applications calling for integral horsepower ratings. According to the NEMA (National Electrical Manufacturers Association) Standards, there are four classes of squirrel-cage induction motors designated respectively as *A*, *B*, *C*, and *D*.

Design A motors are not commonly used since Design *B* has similar characteristics with the advantage of lower starting current.

Design B: motors may be designated as a general purpose type suitable for the majority of polyphase alternating-current applications such as blowers, compressors, drill presses, grinders, hammer mills, lathes, planers, polishers, saws, screw machines, shakers, stokers, etc. The starting torque at 1800 R.P.M. is 250 to 275 per cent of full load torque for 3 H.P. and below; for 5 H.P. to 75 H. P. ratings the starting torque ranges from 185 to 150 per cent of full load torque. They have low starting current requirements, usually no more than 5 to 6 times full load current and can be started at full voltage. Their slip (difference between synchronous speed and actual speed at rated load) is relatively low.

Design C: motors have high starting torque (up to 250 per cent of full load torque) but low starting current. They can be started at full voltage. Slip at rated load is relatively low. They are used for compressors requiring a loaded start, heavy conveyors, reciprocating pumps and other applications requiring high starting torque.

Design D: motors have high slip at rated load, that is, the motor speed drops off appreciably as the load increases, permitting use of the stored energy of a flywheel. They provide heavy starting torque, up to 275 per cent of full load torque, are quiet in operation and have relatively low starting current. Applications are for impact, shock and other high peak loads or flywheel drives such as trains, elevators, hoists, punch and drawing presses, shears, etc.

Design F: motors are no longer standard. They had low starting torque, about 125 per cent of full-load torque, and low starting current. They were used to drive machines which required infrequent starting at no load or at very light load.

Multiple-Speed Induction Motors.—This type has a number of windings in the stator so arranged and connected that the number of effective poles and hence the speed can be changed. These motors are for the same types of starting conditions as the conventional squirrel-cage induction motors and are available in designs that provide constant horsepower at all rated speeds and in designs that provide constant torque at all rated speeds.

Typical speed combinations obtainable in these motors are 600, 900, 1200 and 1800 R.P.M.; 450, 600, 900 and 1200 R.P.M.; and 600, 720, 900 and 1200 R.P.M.

Where a gradual change in speed is called for, a wound rotor may be provided in addition to the multiple stator windings.

Wound-Rotor Induction Motors.—These motors are designed for applications where extremely low starting current with high starting torque are called for, such as in blowers, conveyors, compressors, fans and pumps. They may be employed for adjustable-varying speed service where the speed range does not extend below 50 per cent of synchronous speed, as for steel plate-forming rolls, printing presses, cranes, blowers, stokers, lathes and milling machines of certain types. The speed regulation of a wound rotor induction motor ranges from 5 to 10 per cent at maximum speed and from 18 to 30 per cent at low speed. They are also employed for reversing service as in cranes, gates, hoists and elevators.

High-Frequency Induction Motors.—This type is used in conjunction with frequency changers when very high speeds are desired, as on grinders, drills, routers, portable tools or woodworking machinery. These motors have an advantage over the series-wound or universal type of high speed motor in that they operate at a relatively constant speed over the entire load range. A motor-generator set, a two-unit frequency converter or a single unit inductor frequency converter may be used to supply three-phase power at the frequency required. The single unit frequency converter may be obtained for delivering any one of a number of frequencies ranging from 360 to 2160 cycles and it is self-driven and self-excited from the general polyphase power supply.

Synchronous Motors.—These are widely used in electric timing devices; to drive machines that must operate in synchronism; and also to operate compressors, rolling mills, crushers which are started without load, paper mill screens, shredders, vacuum pumps and motor-generator sets. Synchronous motors have an inherently high power factor and are often employed to make corrections for the low power factor of other types of motors on the same system.

Types of Single-Phase Alternating-Current Motors.—Most of the single-phase alternating-current motors are basically induction motors distinguished by different arrangements for starting. (A single-phase induction motor with only a squirrel-cage rotor has no starting torque.) In the *capacitor-start* single-phase motor, an auxiliary winding in the stator is connected in series with a capacitor and a centrifugal switch. During the starting and accelerating period the motor operates as a two-phase induction motor. At about two-thirds full-load speed, the auxiliary circuit is disconnected by the switch and the motor then runs as a single-phase induction motor. In the *capacitor-start, capacitor-run* motor, the auxiliary circuit is arranged to provide high effective capacity for high starting torque and to remain connected to the line but with reduced capacity during the running period. In the *single-value capacitor* or *capacitor split-phase* motor, a relatively small continuously-rated capacitor is permanently connected in one of the two stator windings and the motor both starts and runs like a two-phase motor.

In the *repulsion-start* single-phase motor, a drum-wound rotor circuit is connected to a commutator with a pair of short-circuited brushes set so that the magnetic axis of the rotor winding is inclined to the magnetic axis of the stator winding. The current flowing in this rotor circuit reacts with the field to produce starting and accelerating torques. At about two-thirds full load speed the brushes are lifted, the commutator is short circuited and the motor runs as a single-phase squirrel-cage motor. The *repulsion* motor employs a repulsion winding on the rotor for both starting and running. The *repulsion-induction* motor has an outer winding on the rotor acting as a repulsion winding and an inner squirrel-cage winding. As the motor comes up to speed, the induced rotor current partially shifts from the repulsion winding to the squirrel-cage winding and the motor runs partly as an induction motor.

In the *split-phase* motor, an auxiliary winding in the stator is used for starting with either a resistance connected in series with the auxiliary winding (*resistance-start*) or a reactor in series with the main winding (*reactor-start*).

The *series-wound* single-phase motor has a rotor winding in series with the stator winding as in the series-wound direct-current motor. Since this motor may also be operated on direct current, it is called a *universal* motor.

Characteristics of Single-Phase Alternating-Current Motors.—Single-phase motors are used in sizes up to about 7½ horsepower for heavy starting duty chiefly in home and commercial appliances for which polyphase power is not available. The *capacitor-start* motor is available in normal starting torque designs for such applications as centrifugal pumps, fans, and blowers and in high-starting torque designs for reciprocating compressors, pumps, loaded conveyors, or belts. The *capacitor-start, capacitor-run* motor is exceptionally quiet in operation when loaded to at least 50 per cent of capacity. It is available in low-torque designs for fans and centrifugal pumps and in high-torque designs for applications similar to those of the capacitor-start motor.

The *capacitor split-phase* motor requires the least maintenance of all single-phase motors, but has very low starting torque. Its high maximum torque makes it potentially useful in floor sanders or in grinders where momentary overloads due to excessive cutting pressure are experienced. It is also used for slow-speed direct connected fans.

The *repulsion-start, induction-run* motor has higher starting torque than the capacitor motors, although for the same current, the capacitor motors have equivalent pull-up and maximum torque. Electrical and mechanical noise and the extra maintenance sometimes required are disadvantages. These motors are used for compressors, conveyors and stokers starting under full load. The *repulsion-induction* motor has relatively high starting torque and low starting current. It also has a smooth speed-torque curve with no break and a greater ability to withstand long accelerating periods than capacitor type motors. It is particularly suitable for severe starting and accelerating duty and for high inertia loads such as laundry extractors. Brush noise is, however, continuous.

The *repulsion* motor has no limiting synchronous speed and the speed changes with the load. At certain loads, slight changes in load cause wide changes in speed. A brush shifting arrangement may be provided to adjust the speed which may have a range of 4 to 1 if full rated constant torque is applied but a decreasing range as the torque falls below this value. This type of motor may be reversed by shifting the brushes beyond the neutral point. These motors are suitable for machines requiring constant-torque and adjustable speed.

The *split-phase* and *universal* motors are limited to about ⅓ H.P. ratings and are used chiefly for small appliance and office machine applications.

Motors with Built-in Speed Reducers.—Electric motors having built-in speed-changing units are compact and the design of these motorized speed reducers tends to improve the appearance of the machines which they drive. There are several types of these speed reducers; they may be classified according to whether they are equipped with worm gearing, a regular gear train with parallel shafts, or planetary gearing.

The claims made for the worm gearing type of reduction unit are that the drive is quiet in operation and well adapted for use where the slow-speed shaft must be at right angles to the motor shaft and where a high speed ratio is essential.

For very low speeds, the double reduction worm gearing units are suitable. In these units two sets of worm gearing form the gear train, and both the slow-speed shaft and the armature shaft are parallel. The intermediate worm gear shaft can be built to extend from the housing, if required, so as to make two countershaft speeds available on the same unit.

In the parallel-shaft type of speed reducer, the slow-speed shaft is parallel with the armature shaft. The slow-speed shaft is rotated by a pinion on the armature shaft, this pinion meshing with a larger gear on the slow-speed shaft.

Geared motors having built-in speed-changing units are available with constant-mesh change gears for varying the speed ratio.

Planetary gearing permits a large speed reduction with few parts; hence, it is well adapted for geared-head motor units where economy and compactness are essential. The slow-speed shaft is in line with the armature shaft.

Factors Governing Motor Selection

Speed, Horsepower, Torque and Inertia Requirements.—Where more than one speed or a range of speeds are called for, one of the following types of motors may be selected, depending upon other requirements: For direct-current, the standard shunt-wound motor with field control has a 2 to 1 range in some designs; the adjustable speed motor may have a range of from 3 to 1 up to 6 to 1; the shunt motor with adjustable voltage supply has a range up to 20 to 1 or more below base speed and 4 or 5 to 1 above base speed, making a total range of up to 100 to 1 or more. For polyphase alternating current, multi-speed squirrel-cage induction motors have 2, 3 or 4 fixed speeds; the wound-rotor motor has a 2 to 1 range. The two-speed wound-rotor motor has a 4 to 1 range. The brush-shifting shunt motor has a 4 to 1 range. The brush-shifting series motor has a 3 to 1 range; and the squirrel-cage motor with a variable-frequency supply has a very wide range. For single-phase alternating current, the brush-shifting repulsion motor has a 2½ to 1 range; the capacitor motor with tapped winding has a 2 to 1 range and the multi-speed capacitor motor has 2 or 3 fixed speeds. Speed regulation (variation in speed from no load to full load) is greatest with motors having series field windings and entirely absent with synchronous motors.

Horsepower: Where the load to be carried by the motor is not constant but follows a definite cycle, a horsepower-time curve enables the peak horsepower to be determined as well as the root-mean-square-average horsepower, which indicates the proper motor rating from a heating standpoint. Where the load is maintained at a constant value for a period of from 15 minutes to 2 hours depending on the size, the horsepower rating required will usually not be less than this constant value. When selecting the size of an induction motor, it should be kept in mind that this type of motor operates at maximum efficiency when it is loaded to full capacity. Where operation is to be at several speeds, the horsepower requirement for each speed should be considered.

Torque: Starting torque requirements may vary from 10 per cent of full load to 250 per cent of full load torque depending upon the type of machine being driven. Starting torque may vary for a given machine because of frequency of start, temperature, type and amount of lubricant, etc., and such variables should be taken into account. The motor torque supplied to the machine must be well above that required by the driven machine at all points up to full speed. The greater the excess torque, the more rapid the acceleration. The approximate time required for acceleration from rest to full speed is given by the formula:

$$\text{Time} = \frac{N \times WR^2}{T_a \times 308} \text{ seconds}$$

where N = Full load speed in R.P.M.

T_a = Torque = average foot-pounds available for acceleration.

WR^2 = Inertia of rotating part in pounds feet squared (W = weight and R = radius of gyration of rotating part).

308 = Combined constant converting minutes into seconds, weight into mass and radius into circumference.

If the time required for acceleration is greater than 20 seconds, special motors or starters may be required to avoid overheating.

The running torque T_r is found by the formula:

$$T_r = \frac{5250 \times \text{HP}}{N} \text{ foot pounds}$$

where *H.P.* = Horsepower being supplied to the driven machine

N = Running speed in R.P.M.

5250 = Combined constant converting horsepower to foot-pounds per minute and work per revolution into torque.

The peak horsepower determines the maximum torque required by the driven machine and the motor must have a maximum running torque in excess of this value.

Inertia: The inertia or flywheel effect of the rotating parts of a driven machine will, if large, appreciably affect the accelerating time and, hence, the amount of heating in the motor. If synchronous motors are used, the inertia (WR^2) of both the motor rotor and the rotating parts of the machine must be known since the pull-in torque (torque required to bring the driven machine up to synchronous speed) varies approximately as the square root of the total inertia of motor and load.

Space Limitations in Motor Selection.—If the motor is to become an integral part of the machine which it drives and space is at a premium, a partial motor may be called for. A complete motor is one made up of a stator, a rotor, a shaft, and two end shields with bearings. A *partial motor* is without one or more of these elements. One common type is furnished without drive-end end shield and bearing and is directly connected to the end or side of the machine which it drives, such as the headstock of a lathe. A so-called *shaftless type of motor* is supplied without shaft, end shields or bearings and is intended for built-in application in such units as multiple drilling machines, precision grinders, deep well pumps, compressors and hoists where the rotor is actually made a part of the driven machine. Where a partial motor is used, however, proper ventilation, mounting, alignment and bearings must be arranged for by the designer of the machine to which it is applied.

Sometimes it is possible to use a motor having a smaller frame size and wound with Class *B* insulation, permitting it to be subjected to a higher temperature rise than the larger-frame Class *A* insulated motor having the same horsepower rating.

Temperatures.—The applicability of a given motor is limited not only by its load starting and carrying ability, but also by the temperature which it reaches under load. Motors are given temperature ratings which are based upon the type of insulation (Class A or Class B are the most common) used in their construction and their type of frame (open, semienclosed, or enclosed).

Insulating Materials: Class A materials are: cotton, silk, paper, and similar organic materials when either impregnated or immersed in a liquid dielectric; molded and laminated materials with cellulose filler, phenolic resins, and other resins of similar properties;

films and sheets of cellulose acetate and other cellulose derivatives of similar properties;

and varnishes (enamel) as applied to conductors.

Class B insulating materials are: materials or combinations of materials such as mica, glass fiber, asbestos, etc., with suitable bonding substances. Other materials shown capable of operation at Class B temperatures may be included.

Ambient Temperature and Allowable Temperature Rise: Normal ambient temperature is taken to be 40°C (104°F). For open general-purpose motors with Class A insulation, the normal temperature rise on which the performance guarantees are based is 40°C (104°F).

Motors with Class A insulation having protected, semiprotected, drip-proof, or splash-proof, or drip-proof protected enclosures have a 50°C (122°F) rise rating.

Motors with Class A insulation and having totally enclosed, fan-cooled, explosion-proof, waterproof, dust-tight, submersible, or dust-explosion-proof enclosures have a 55°C (131°F) rise rating.

Motors with Class B insulation are permissible for total temperatures up to 110 degrees C (230°F) for open motors and 115°C (239°F) for enclosed motors.

Motors Exposed to Injurious Conditions.—Where motors are to be used in locations imposing unusual operating conditions, the manufacturer should be consulted, especially where any of the following conditions apply: exposure to chemical fumes; operation in damp places; operation at speeds in excess of specified overspeed; exposure to combustible or explosive dust; exposure to gritty or conducting dust; exposure to lint; exposure to steam; operation in poorly ventilated rooms; operation in pits, or where entirely enclosed in boxes; exposure to inflammable or explosive gases; exposure to temperatures below 10°C (50°F); exposure to oil vapor; exposure to salt air; exposure to abnormal shock or vibration from external sources; where the departure from rated voltage is excessive; and or where the alternating-current supply voltage is unbalanced.

Improved insulating materials and processes and greater mechanical protection against falling materials and liquids make it possible to use general-purpose motors in many locations where special-purpose motors were previously considered necessary. *Splash-proof motors* having well-protected ventilated openings and specially treated windings are used where they are to be subjected to falling and splashing water or are to be washed down as with a hose. Where climatic conditions are not severe, this type of motor is also successfully used in unprotected outdoor installations.

If the surrounding atmosphere carries abnormal quantities of metallic, abrasive, or non-explosive dust or acid or alkali fumes, a *totally enclosed fan-cooled motor* may be called for. In this type, the motor proper is completely enclosed but air is blown through an outer shell that completely or partially surrounds the inner case. If the dust in the atmosphere tends to pack or solidify and close the air passages of open splash-proof or totally enclosed fan-cooled motors, *totally enclosed (nonventilated) motors* are used. This type, which is limited to low horsepower ratings, is also used for outdoor service in mild or severe climates.

Table 1. Characteristics and Applications of D.C. Motors, 1–300 hp

<table>
<tr><th>Type</th><th>Starting Duty</th><th>Maximum Momentary Running Torque</th><th>Speed Regulation</th><th>Speed Control[a]</th><th>Applications</th></tr>
<tr><td>Shunt-wound, constant-speed</td><td rowspan="3">Medium starting torque. Varies with voltage supplied to armature, and is limited by starting resistor to 125 to 200% full-load torque</td><td rowspan="3">125 to 200%. Limited by commutation</td><td>8 to 12%</td><td>Basic speed to 200% basic speed by field control</td><td rowspan="2">Drives where starting requirements are not severe. Use constant-speed or adjustable-speed, depending on speed required. Centrifugal pumps, fans, blowers, conveyors, elevators, wood- and metalworking machines</td></tr>
<tr><td>Shunt-wound, adjustable speed</td><td>10 to 20%, increases with weak fields</td><td>Basic speed to 60% basic speed (lower for some ratings) by field control</td></tr>
<tr><td>Shunt-wound, adjustable voltage control</td><td>Up to 25%. Less than 5% obtainable with special rotating regulator</td><td>Basic speed to 2% basic speed and basic speed to 200% basic speed</td><td>Drives where wide, stepless speed control, uniform speed, constant-torque acceleration and adaptability to automatic operation are required. Planers, milling machines, boring machines, lathes, etc.</td></tr>
<tr><td>Compound wound, constant-speed</td><td>Heavy starting torque, Limited by starting resistor to 130 to 260% of full-load torque</td><td>130 to 260%. Limited by commutation</td><td>Standard compounding 25%. Depends on amount of series winding</td><td>Basic speed to 125% basic speed by field control</td><td>Drives requiring high starting torque and fairly constant speed. Pulsating loads. Shears, bending rolls, pumps, conveyors, crushers, etc.</td></tr>
<tr><td>Series-wound, varying-speed</td><td>Very heavy starting torque. Limited to 300 to 350% full-load torque</td><td>300 to 350%. Limited by commutation</td><td>Very high. Infinite no-load speed</td><td>From zero to maximum speed, depending on control and load</td><td>Drives where very high starting torque is required and speed can be regulated. Cranes, hoists, gates, bridges, car dumpers, etc.</td></tr>
</table>

[a] Minimum speed below basic speed by armature control limited by heating.

Table 2. Characteristics and Applications of Polyphase AC Motors

Polyphase Type	Ratings hp	Speed Regulation	Speed Control	Starting Torque	Breakdown Torque	Applications
General-purpose squirrel cage, normal stg current, normal stg torque. Design B	0.5 to 200	Less than 5%	None, except multi-speed types, designed for two to four fixed speeds	100 to 250% of full-load	200 to 300% of full-load	Constant-speed service where starting torque is not excessive. Fans, blowers, rotary compressors, centrifugal pumps, woodworking machines, machine tools, line shafts
Full-voltage starting, high stg torque, normal stg current, squirrel-cage, Design C	3 to 150	Less than 5%	None except multi-speed types, designed for two to four fixed speeds	200 to 250% of full-load	190 to 225% of full-load	Constant-speed service where fairly high starting torque is required at infrequent intervals with starting current of about 500% full-load. Reciprocating pumps and compressors, conveyors, crushers, pulverizers, agitators, etc.
Full-voltage starting, high stg-torque, high-slip squirrel cage, Design D	0.5 to 150	Drops about 7 to 12% from no load to full load	None, except multi-speed types, designed for two to four fixed speeds	275% of full-load depending on speed and rotor resistance	275% of full-load Will usually not stall until loaded to its maximum torque, which occurs at standstill	Constant-speed service and high-starting torque if starting not too frequent, and for taking high-peak loads with or without flywheels. Punch presses, die stamping, shears, bulldozers, bailers, hoists, cranes, elevators, etc.
Wound-rotor, external-resistance starting	0.5 to several thousand	With rotor rings short-circuited drops about 3% for large to 5% for small sizes	Speed can be reduced to 50% of normal by rotor resistance. Speed varies inversely as the load	Up to 300% depending on external resistance in rotor circuit and how distributed	200% when rotor slip rings are short circulated	Where high-starting torque with low-starting current or where limited speed control is required. Fans, centrifugal and plunger pumps, compressors, conveyors, hoists, cranes, ball mills, gate hoists, etc.
Synchronous	25 to several thousand	Constant	None, except special motors designed for two fixed speeds	40% for slow speed to 160% for medium speed 80% p-f designs. Special high-torque designs	Pull-out torque of unity-p-f motors 170%; 80%-p-f motors 225%. Special designs up to 300%	For constant-speed service, direct connection to slow-speed machines and where power-factor correction is required.

In addition to these special-purpose motors, there are two types of *explosion-proof motors* designed for hazardous locations. One type is for operation in hazardous dust locations (Class II, Group *G* of the National Electrical Code) and the other is for atmospheres containing explosive vapors and fumes classified as Class I, Group *D* (gasoline, naphtha, alcohols, acetone, lacquer-solvent vapors, natural gas).

Electric Motor Maintenance

Electric Motor Inspection Schedule.—Frequency and thoroughness of inspection depend upon such factors as 1) importance of the motor in the production scheme; 2) percentage of days the motor operates; 3) nature of service; and 4) winding conditions.

The following schedules, recommended by the General Electric Company, and covering both AC and DC motors are based on average conditions in so far as duty and dirt are concerned.

Weekly Inspection.—1) *Surroundings.* Check to see if the windings are exposed to any dripping water, acid or alcoholic fumes; also, check for any unusual amount of dust, chips, or lint on or about the motor. See if any boards, covers, canvas, etc., have been misplaced that might interfere with the motor ventilation or jam moving parts.

2) *Lubrication of sleeve-bearing motors.* In sleeve-bearing motors check oil level, if a gage is used, and fill to the specified line. If the journal diameter is less than 2 inches, the motor should be stopped before checking the oil level. For special lubricating systems, such as wool-packed, forced lubrication, flood and disk lubrication, follow instruction book. Oil should be added to bearing housing only when motor is at rest. A check should be made to see if oil is creeping along the shaft toward windings where it may harm the insulation.

3) *Mechanical condition.* Note any unusual noise that may be caused by metal-to-metal contact or any odor as from scorching insulation varnish.

4) *Ball or roller bearings.* Feel ball- or roller-bearing housings for evidence of vibration, and listen for any unusual noise. Inspect for creepage of grease on inside of motor.

5) *Commutators and brushes.* Check brushes and commutator for sparking. If the motor is on cyclic duty it should be observed through several cycles. Note color and surface condition of the commutator. A stable copper oxide-carbon film (as distinguished from a pure copper surface) on the commutator is an essential requirement for good commutation. Such a film may vary in color all the way from copper to straw, chocolate to black. It should be clean and smooth and have a high polish. All brushes should be checked for wear and pigtail connections for looseness. The commutator surface may be cleaned by using a piece of dry canvas or other hard, nonlinting material that is wound around and securely fastened to a wooden stick, and held against the rotating commutator.

6) *Rotors and armatures.* The air gap on sleeve-bearing motors should be checked, especially if they have been recently overhauled. After installing new bearings, make sure that the average reading is within 10 per cent, provided reading should be less than 0.020 inch. Check air passages through punchings and make sure they are free of foreign matter.

7) *Windings.* If necessary clean windings by suction or mild blowing. After making sure that the motor is dead, wipe off windings with dry cloth, note evidence of moisture, and see if any water has accumulated in the bottom of frame. Check if any oil or grease has worked its way up to the rotor or armature windings. Clean with carbon tetrachloride in a well-ventilated room.

8) *General.* This is a good time to check the belt, gears, flexible couplings, chain, and sprockets for excessive wear or improper location. The motor starting should be checked to make sure that it comes up to proper speed each time power is applied.

Monthly or Bimonthly Inspection.—1) *Windings.* Check shunt, series, and commutating field windings for tightness. Try to move field spools on the poles, as drying out may have caused some play. If this condition exists, a service shop should be consulted. Check motor cable connections for tightness.

2) *Brushes.* Check brushes in holders for fit and free play. Check the brush-spring pressure. Tighten brush studs in holders to take up slack from drying out of washers, making sure that studs are not displaced, particularly on DC motors. Replace brushes that are worn down almost to the brush rivet, examine brush faces for chipped toes or heels, and for heat cracks. Damaged brushes should be replaced immediately.

3) *Commutators.* Examine commutator surface for high bars and high mica, or evidence of scratches or roughness. See that the risers are clean and have not been damaged.

4) *Ball or roller bearings.* On hard-driven, 24-hour service ball- or roller-bearing motors, purge out old grease through drain hole and apply new grease. Check to make sure grease or oil is not leaking out of the bearing housing. If any leakage is present, correct the condition before continuing to operate.

5) *Sleeve bearings.* Check sleeve bearings for wear, including end-play bearing surfaces. Clean out oil wells if there is evidence of dirt or sludge. Flush with lighter oil before refilling.

6) *Enclosed gears.* For motors with enclosed gears, open drain plug and check oil flow for presence of metal scale, sand, or water. If condition of oil is bad, drain, flush, and refill as directed. Rock rotor to see if slack or backlash is increasing.

7) *Loads.* Check loads for changed conditions, bad adjustment, poor handling, or control.

8) *Couplings and other drive details.* Note if belt-tightening adjustment is all used up. Shorten belt if this condition exists. See if belt runs steadily and close to inside (motor edge) of pulley. Chain should be checked for evidence of wear and stretch. Clean inside of chain housing. Check chain-lubricating system. Note inclination of slanting base to make sure it does not cause oil rings to rub on housing.

Annual or Biannual Inspection.— 1) *Windings.* Check insulation resistance by using either a megohmmeter or a voltmeter having a resistance of about 100 ohms per volt. Check insulation surfaces for dry cracks and other evidence of need for coatings of insulating material. Clean surfaces and ventilating passages thoroughly if inspection shows accumulation of dust. Check for mold or water standing in frame to determine if windings need to be dried out, varnished, and baked.

2) *Air gap and bearings.* Check air gap to make sure that average reading is within 10 per cent, provided reading should be less than 0.020 inch. All bearings, ball, roller, and sleeve should be thoroughly checked and defective ones replaced. Waste-packed and wick-oiled bearings should have waste or wicks renewed, if they have become glazed or filled with metal or dirt, making sure that new waste bears well against shaft.

3) *Rotors (squirrel-cage).* Check squirrel-cage rotors for broken or loose bars and evidence of local heating. If fan blades are not cast in place, check for loose blades. Look for marks on rotor surface indicating foreign matter in air gap or a worn bearing.

4) *Rotors (wound).* Clean wound rotors thoroughly around collector rings, washers, and connections. Tighten connections if necessary. If rings are rough, spotted, or eccentric, refer to service shop for refinishing. See that all top sticks or wedges are tight. If any are loose, refer to service shop.

5) *Armatures.* Clean all armature air passages thoroughly if any are obstructed. Look for oil or grease creeping along shaft, checking back to bearing. Check commutator for surface condition, high bars, high mica, or eccentricity. If necessary, remachine the commutator to secure a smooth fresh surface.

6) *Loads.* Read load on motor with instruments at no load, full load, or through an entire cycle, as a check on the mechanical condition of the driven machine.

ADHESIVES AND SEALANTS

By strict definition, an adhesive is any substance that fastens or bonds materials to be joined (adherends) by means of surface attachment. The bond durability depends on the strength of the adhesive to the substrate (adhesion) and the strength within the adhesive (cohesion). Besides bonding a joint, an adhesive may serve as a seal against foreign matter. When an adhesive performs both bonding and sealing functions, it is usually referred to as an adhesive sealant. Joining materials with adhesives offers significant benefits compared with mechanical methods of uniting two materials.

Among these benefits are that an adhesive distributes a load over an area rather than concentrating it at a point, resulting in a more even distribution of stresses. The adhesive bonded joint is therefore more resistant to flexural and vibrational stresses than, for example, a bolted, riveted, or welded joint. Another benefit is that an adhesive forms a seal as well as a bond. This seal prevents the corrosion that may occur with dissimilar metals, such as aluminum and magnesium, or mechanically fastened joints, by providing a dielectric insulation between the substrates. An adhesive also joins irregularly shaped surfaces more easily than does a mechanical fastener. Other benefits include negligible weight addition and virtually no change to part dimensions or geometry.

Most adhesives are available in liquids, gels, pastes, and tape forms. The growing variety of adhesives available can make the selection of the proper adhesive or sealant a challenging experience. In addition to the technical requirements of the adhesive, time and costs are also important considerations. Proper choice of an adhesive is based on knowledge of the suitability of the adhesive or sealant for the particular substrates. Appropriate surface preparation, curing parameters, and matching the strength and durability characteristics of the adhesive to its intended use are essential. The performance of an adhesive-bonded joint depends on a wide range of these factors, many of them quite complex. Adhesive suppliers can usually offer essential expertise in the area of appropriate selection.

Adhesives can be classified as structural or nonstructural. In general, an adhesive can be considered structural when it is capable of supporting heavy loads; nonstructural when it cannot support such loads. Many adhesives and sealants, under various brand names, may be available for a particular bonding application. It is always advisable to check the adhesive manufacturers' information before making an adhesive sealant selection. Also, testing under end-use conditions is always suggested to help ensure bonded or sealed joints meet or exceed expected performance requirements.

Though not meant to be all-inclusive, the following information correlates the features of some successful adhesive compositions available in the marketplace.

Bonding Adhesives

Reactive-type bonding adhesives are applied as liquids and react (cure) to solids under appropriate conditions. The cured adhesive is either a thermosetting or thermoplastic polymer. These adhesives are supplied as two-component no-mix, two-component mix, and one-component no-mix types, which are discussed in the following paragraphs.

Two-Component No-Mix Adhesives

Types of Adhesives.—*Anaerobic (Urethane Methacrylate Ester) Structural Adhesives:* Anaerobic structural adhesives are mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between metal substrates. These adhesives can be used for large numbers of industrial purposes where high reliability of bond joints is required. Benefits include: no mixing is required (no pot-life or waste problems), flexible/durable bonds are made that withstand thermal cycling, have excellent resistance to solvents and severe environments, and rapid cure at room temperatures (eliminating expensive ovens). The adhesives are easily dispensed with automatic equipment. An activator is usually required to be present on one surface to initiate the cure for these adhesives. Applications for these adhesives include bonding of metals, magnets (ferrites), glass, thermosetting plastics, ceramics, and stone.

Acrylic Adhesives: Acrylic adhesives are composed of a polyurethane polymer backbone with acrylate end groups. They can be formulated to cure through heat or the use of an activator applied to the substrate surface, but many industrial acrylic adhesives are cured by light. Light-cured adhesives are used in applications where the bond geometry allows light to reach the adhesive and the production rate is high enough to justify the capital expense of a light source. Benefits include: no mixing is required (no pot-life or waste problems); formulations cure (solidify) with activator, heat, or light; the adhesive will bond to a variety of substrates, including metal and most thermoplastics; and tough and durable bonds are produced with a typical resistance to the effects of temperatures up to 180°C. Typical applications include automobile body parts (steel stiffeners), assemblies subjected to paint-baking cycles, speaker magnets to pole plates, and bonding of motor magnets, sheet steel, and many other structural applications. Other applications include bonding glass, sheet metal, magnets (ferrite), thermosetting and thermoplastic plastics, wood, ceramics, and stone.

Two-Component Mix Adhesives

Types of Adhesives.—*Epoxy Adhesives:* Two-component epoxy adhesives are well-established adhesives that offer many benefits in manufacturing. The reactive components of these adhesives are separated prior to use, so they usually have a good shelf life without refrigeration. Polymerization begins upon mixing, and a thermoset polymer is formed. Epoxy adhesives cure to form thermosetting polymers made up of a base side with the polymer resin and a second part containing the catalyst. The main benefit of these systems is that the depth of cure is unlimited. As a result, large volume can be filled for work such as potting, without the cure being limited by the need for access to an external influence such as moisture or light to activate the curing process.

For consistent adhesive performance, it is important that the mix ratio remain constant to eliminate variations in adhesive performance. Epoxies can be handled automatically, but the equipment involves initial and maintenance costs. Alternatively, adhesive components can be mixed by hand. However, this approach involves labor costs and the potential for human error. The major disadvantage of epoxies is that they tend to be very rigid and consequently have low peel strength. This lack of peel strength is less of a problem when bonding metal to metal than it is when bonding flexible substrates such as plastics.

Applications of epoxy adhesives include bonding, potting, and coating of metals, bonding of glass, rigid plastics, ceramics, wood, and stone.

Polyurethane Adhesives: Like epoxies, polyurethane adhesives are available as two-part systems or as one-component frozen premixes. They are also available as one-part moisture-cured systems. Polyurethane adhesives can provide a wide variety of physical properties. Their flexibility is greater than that of most epoxies. Coupled with the high cohesive strength, this flexibility provides a tough polymer able to achieve better peel strength and lower flexural modulus than most epoxy systems. This superior peel resistance allows use of polyurethanes in applications that require high flexibility. Polyurethanes bond very well to a variety of substrates, though a primer may be needed to prepare the substrate surface. These primers are moisture-reactive and require several hours to react sufficiently for the parts to be used. Such a time requirement may cause a production bottleneck if the bond-strength requirements are such that a primer is needed.

Applications for polyurethane adhesives include bonding of metals, glass, rubber, thermosetting and thermoplastic plastics, and wood.

One-Component No-Mix Adhesives

Types of Adhesives.—*Light-Curable Adhesives:* Light-curing systems use a unique curing mechanism. The adhesives contain photoinitiators that absorb light energy and dissociate to form radicals. These radicals then initiate the polymerization of the polymers, oligomers, and monomers in the adhesive. The photoinitiator acts as a chemical solar cell, converting the light energy into chemical energy for the curing process. Typically, these systems are formulated for use with ultraviolet light sources. However, newer products have been formulated for use with visible light sources.

One of the biggest benefits that light-curing adhesives offer to the manufacturer is the elimination of the work time to work-in-progress trade-off, which is embodied in most adhesive systems. With light-curing systems, the user can take as much time as needed to position the part without fear of the adhesive curing. Upon exposure to the appropriate light source, the adhesive then can be fully cured in less than 1 minute, minimizing the costs associated with work in progress. Adhesives that utilize light as the curing mechanism are often one-part systems with good shelf life, which makes them even more attractive for manufacturing use.

Applications for light-curable adhesives include bonding of glass, and glass to metal, tacking of wires, surface coating, thin-film encapsulation, clear substrate bonding, and potting of components,

Cyanoacrylate Adhesives (Instant Adhesives): Cyanoacrylates or instant adhesives are often called Superglue™. Cyanoacrylates are one-part adhesives that cure rapidly, as a result of the presence of surface moisture, to form high-strength bonds, when confined between two substrates. Cyanoacrylates have excellent adhesion to many substrates, including most plastics and they achieve fixture strength in seconds and full strength within 24 hours. These qualities make cyanoacrylates suitable for use in automated production environments. They are available in viscosities ranging from water-thin liquids to thixotropic gels.

Because cyanoacrylates are a relatively mature adhesive family, a wide variety of specialty formulations is now available to help the user address difficult assembly problems. One of the best examples is the availability of polyolefin primers, which allow users to obtain high bond strengths on difficult-to-bond plastics such as polyethylene and polypropylene. One common drawback of cyanoacrylates is that they form a very rigid polymer matrix, resulting in very low peel strengths. To address this problem, formulations have been developed that are rubber-toughened. Although the rubber toughening improves the peel strength of the system to some extent, peel strength remains a weak point for this system, and, therefore, cyanoacrylates are poor candidates for joint designs that require high peel resistance. In manufacturing environments with low relative humidity, the cure of the cyanoacrylate can be significantly retarded.

This problem can be addressed in one of two ways. One approach is to use accelerators that deposit active species on the surface to initiate the cure of the product. The other approach is to use specialty cyanoacrylate formulations that have been engineered to be surface-insensitive. These formulations can cure rapidly even on dry or slightly acidic surfaces.

Applications for cyanoacrylate adhesives include bonding of thermoplastic and thermosetting plastics, rubber, metals, wood, and leather, also strain relief of wires.

Hot-Melt Adhesives: Hot-melt adhesives are widely used in assembly applications. In general, hot-melt adhesives permit fixturing speeds that are much faster than can be achieved with water- or solvent-based adhesives. Usually supplied in solid form, hot-melt adhesives liquify when exposed to elevated temperatures. After application, they cool quickly, solidifying and forming a bond between two mating substrates. Hot-melt adhesives have been used successfully for a wide variety of adherends and can greatly reduce both the need for clamping and the length of time for curing. Some drawbacks with hot-melt adhesives are their tendency to string during dispensing and relatively low-temperature resistance.

Applications for hot-melt adhesives are bonding of fabrics, wood, paper, plastics, and cardboard.

Rubber-Based Solvent Cements: Rubber-based solvent cements are adhesives made by combining one or more rubbers or elastomers in a solvent. These solutions are further modified with additives to improve the tack or stickiness, the degree of peel strength, flexibility, and the viscosity or body. Rubber-based adhesives are used in a wide variety of applications such as contact adhesive for plastics laminates like counter tops, cabinets, desks, and tables. Solvent-based rubber cements have also been the mainstay of the shoe and leather industry for many years.

Applications for rubber-based solvent cements include bonding of plastics laminates, wood, paper, carpeting, fabrics, and leather.

Moisture-Cured Polyurethane Adhesives: Like heat-curing systems, moisture-cured polyurethanes have the advantage of a very simple curing process. These adhesives start to cure when moisture from the atmosphere diffuses into the adhesive and initiates the polymerization process. In general, these systems will cure when the relative humidity is above 25 per cent, and the rate of cure will increase as the relative humidity increases.

The dependence of these systems on the permeation of moisture through the polymer is the source of their most significant process limitations. As a result of this dependence, depth of cure is limited to between 0.25 and 0.5 in. (6.35 and 12.7 mm). Typical cure times are in the range of 12 to 72 hours. The biggest use for these systems is for windshield bonding in automobile bodies.

Applications for moisture-cured polyurethane adhesives include bonding of metals, glass, rubber, thermosetting and thermoplastic plastics, and wood.

Retaining Compounds

The term *retaining compounds* is used to describe adhesives used in circumferential assemblies joined by inserting one part into the other. In general, retaining compounds are anaerobic adhesives composed of mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between cylindrical machine components. A typical example is a bearing held in an electric motor housing with a retaining compound. The first retaining compounds were launched in 1963, and the reaction among users of bearings was very strong because these retaining compounds enabled buyers of new bearings to salvage worn housings and minimize their scrap rate.

The use of retaining compounds has many benefits, including elimination of bulk needed for high friction forces, ability to produce more accurate assemblies and to augment or replace press fits, increased strength in heavy press fits, and reduction of machining costs. Use of these compounds also helps in dissipating heat through assembly, and eliminating distortion when installing drill bushings, fretting corrosion and backlash in keys and splines, and bearing seizure during operation.

The major advantages of retaining compounds for structural assemblies are that they require less severe machining tolerances and no securing of parts. Components are assembled quickly and cleanly, and they transmit high forces and torques, including dynamic forces. Retaining compounds also seal, insulate, and prevent micromovements so that neither fretting corrosion nor stress corrosion occurs. The adhesive joint can be taken apart easily after heating above 450°F (230°C) for a specified time.

Applications for retaining compounds include mounting of bearings in housings or on shafts, avoiding distortion of precision tooling and machines, mounting of rotors on shafts, inserting drill jig bushings, retaining cylinder linings, holding oil filter tubes in castings, retaining engine-core plugs, restoring accuracy to worn machine tools, and eliminating keys and set screws.

Threadlocking

The term *threadlocker* is used to describe adhesives used in threaded assemblies for locking the threaded fasteners by filling the spaces between the nut and bolt threads with a hard, dense material that prevents loosening. In general, thread-lockers are anaerobic adhesives comprising mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between threaded components. A typical example is a mounting bolt on a motor or a pump. Threadlocker strengths range from very low strength (removable) to high strength (permanent).

It is important that the total length of the thread is coated and that there is no restriction to the curing of the threadlocker material. (Certain oils or cleaning systems can impede or even completely prevent the adhesive from curing by anaerobic reaction.) The liquid threadlocker may be applied by hand or with special dispensing devices. Proper coating (wetting) of a thread is dependent on the size of the thread, the viscosity of the adhesive, and the geometry of the parts. With blind-hole threads, it is essential that the adhesive be applied all the way to the bottom of the threaded hole. The quantity must be such that after assembly, the displaced adhesive fills the whole length of the thread.

Some threadlocking products cured by anaerobic reaction have a positive influence on the coefficient of friction in the thread. The values are comparable with those of oiled bolts. Prestress and installation torque therefore can be defined exactly. This property allows threadlocking products cured by anaerobic reaction to be integrated into automated production lines using existing assembly equipment. The use of thread-lockers has many benefits including ability to lock and seal all popular bolt and nut sizes with all industrial finishes, and to replace mechanical locking devices. The adhesive can seal against most industrial fluids and will lubricate threads so that the proper clamp load is obtained. The materials also provide vibration-resistant joints that require handtool dismantling for servicing, prevent rusting of threads, and cure (solidify) without cracking or shrinking.

The range of applications includes such uses as locking and sealing nuts on hydraulic pistons, screws on vacuum cleaner bell housings, track bolts on bulldozers, hydraulic-line fittings, screws on typewriters, oil-pressure switch assembly, screws on carburetors, rocker nuts, machinery driving keys, and on construction equipment.

Sealants

The primary role of a sealant composition is the prevention of leakage from or access by dust, fluids, and other materials to assembly structures. Acceptable leak rates can range from a slight drip to bubbletight to molecular diffusion through the base materials. Equipment users in the industrial market want trouble-free operation, but it is not always practical to specify zero leak rates. Factors influencing acceptable leak rates are toxicity, product or environmental contamination, combustibility, economics, and personnel considerations. All types of fluid seals perform the same basic function: they seal the process fluid (gas, liquid, or vapor) and keep it where it belongs. A general term for these assembly approaches is gasketing. Many products are being manufactured that are capable of sealing a variety of substrates.

Types of Sealants.—*Anaerobic Formed-in-Place Gasketing Materials:* Mechanical assemblies that require the joining of metal-to-metal flange surfaces have long been designed with prefabricated, precut materials required to seal the imperfect surfaces of the assembly. Numerous gasket materials that have been used to seal these assemblies include paper, cork, asbestos, wood, metals, dressings, and even plastics. Fluid seals are divided into static and dynamic systems, depending on whether or not the parts move in relationship to each other. Flanges are classed as static systems, although they may be moved relative to each other by vibration, temperature, and/or pressure changes, shocks, and impacts.

The term *anaerobic formed-in-place gasketing* is used to describe sealants that are used in flanged assemblies to compensate for surface imperfections of metal-to-metal components by filling the space between the substrates with a flexible, nonrunning material. In general, anaerobic formed-in-place gaskets are sealants made up of mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between components. A typical example is sealing two halves of a split crankcase.

The use of anaerobic formed-in-place gaskets has many benefits, including the ability to seal all surface imperfections, allow true metal-to-metal contact, eliminate compression set and fastener loosening, and add structural strength to assemblies. These gaskets also help improve torque transmission between bolted flange joints, eliminate bolt retorquing needed with conventional gaskets, permit use of smaller fasteners and lighter flanges, and provide for easy disassembly and cleaning.

Applications in which formed-in-place gasketing can be used to produce leakproof joints include pipe flanges, split crankcases, pumps, compressors, power takeoff covers, and axle covers. These types of gaskets may also be used for repairing damaged conventional gaskets and for coating soft gaskets.

Silicone Rubber Formed-in-Place Gasketing: Another type of formed-in-place gasket uses room-temperature vulcanizing (RTV) silicone rubbers. These materials are one-component sealants that cure on exposure to atmospheric moisture. They have excellent properties for vehicle use such as flexibility, low volatility, good adhesion, and high resistance to most automotive fluids. The materials will also withstand temperatures up to 600°F (320°C) for intermittent operation.

RTV silicones are best suited for fairly thick section (gap) gasketing applications where flange flexing is greatest. In the form of a very thin film, for a rigid metal-to-metal seal, the cured elastomer may abrade and eventually fail under continual flange movement. The RTV silicone rubber does not unitize the assembly, and it requires relatively clean, oil-free surfaces for sufficient adhesion and leakproof seals.

Because of the silicone's basic polymeric structure, RTV silicone elastomers have several inherent characteristics that make them useful in a wide variety of applications. These properties include outstanding thermal stability at temperatures from 400 to 600°F (204 to 320°C), and good low-temperature flexibility at −85 to −165°F (–65 to −115°C). The material forms an instant seal, as is required of all liquid gaskets, and will fill large gaps up to 0.250 in. (6.35 mm) for stamped metal parts and flanges. The rubber also has good stability in ultraviolet light and excellent weathering resistance.

Applications for formed-in-place RTV silicones in the automotive field are valve, camshaft and rocker covers, manual transmission (gearbox) flanges, oil pans, sealing panels, rear axle housings, timing chain covers, and window plates. The materials are also used on oven doors and flues.

Tapered Pipe-thread Sealing

Thread sealants are used to prevent leakage of gases and liquids from pipe joints. All joints of this type are considered to be dynamic because of vibration, changing pressures, or changing temperatures.

Several types of sealants are used on pipe threads including noncuring pipe dopes, which are one of the oldest methods of sealing the spiral leak paths of threaded joints. In general, pipe dopes are pastes made from oils and various fillers. They lubricate joints and jam threads but provide no locking advantage. They also squeeze out under pressure, and have poor solvent resistance. Noncuring pipe dopes are not suitable for use on straight threads.

Another alternative is solvent-drying pipe dopes, which are an older method of sealing tapered threaded joints. These types of sealant offer the advantages of providing lubrication and orifice jamming and they also extrude less easily than noncuring pipe dopes. One disadvantage is that they shrink during cure as the solvents evaporate and fittings must be retorqued to minimize voids. These materials generally lock the threaded joint together by friction. A third type of sealer is the trapped elastomer supplied in the form of a thin tape incorporating polytetrafluorethylene (PTFE). This tape gives a good initial seal and resists chemical attack, and is one of the only materials used for sealing systems that will seal against oxygen gas.

Some other advantages of PTFE are that it acts as a lubricant, allows for high torquing, and has a good resistance to various solvents. Some disadvantages are that it may not provide a true seal between the two threaded surfaces, and it lubricates in the off direction, so it may allow fittings to loosen. In dynamic joints, tape may allow creep, resulting in leakage over time. The lubrication effect may allow overtightening, which can add stress or lead to breakage. Tape also may be banned in some hydraulic systems due to shredding, which may cause clogging of key orifices.

Anaerobic Pipe Sealants.—*Anaerobic Pipe Sealants:* The term *anaerobic pipe sealants* is used to describe anaerobic sealants used in tapered threaded assemblies for sealing and locking threaded joints. Sealing and locking are accomplished by filling the space between the threads with the sealant. In general, these pipe sealants are anaerobic adhesives consisting of mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between threaded components to form an insoluble tough plastics. The strength of anaerobic pipe sealants is between that of elastomers and yielding metal.

Clamp loads need be only tight enough to prevent separation in use. Because they develop strength by curing after they are in place, these sealants are generally forgiving of tolerances, tool marks, and slight misalignment. These sealants are formulated for use on metal substrates. If the materials are used on plastics, an activator or primer should be used to prepare the surfaces.

Among the advantages of these anaerobic sealers are that they lubricate during assembly, they seal regardless of assembly torque, and they make seals that correspond with the burst rating of the pipe. They also provide controlled disassembly torque, do not cure outside the joint, and are easily dispensed on the production line. These sealants also have the lowest cost per sealed fitting. Among the disadvantages are that the materials are not suitable for oxygen service, for use with strong oxidizing agents, or for use at temperatures above 200°C. The sealants also are typically not suitable for diameters over M80 (approximately 3 inches).

The many influences faced by pipe joints during service should be known and understood at the design stage, when sealants are selected. Sealants must be chosen for reliability and long-term quality. Tapered pipe threads must remain leak-free under the severest vibration and chemical attack, also under heat and pressure surges.

Applications of aerobic sealants are found in industrial plant fluid power systems, the textile industry, chemical processing, utilities and power generation facilities, petroleum refining, and in marine, automotive, and industrial equipment. The materials are also used in the pulp and paper industries, in gas compression and distribution, and in waste-treatment facilities.

MOTION CONTROL

The most important factor in the manufacture of accurately machined components is the control of motion, whatever power source is used. For all practical purposes, motion control is accomplished by electrical or electronic circuits, energizing or deenergizing actuators such as electric motors or solenoid valves connected to hydraulic or pneumatic cylinders or motors. The accuracy with which a machine tool slide, for example, may be brought to a required position, time after time, controls the dimensions of the part being machined. This accuracy is governed by the design of the motion control system in use.

There is a large variety of control systems, with power outputs from milliwatts to megawatts, and they are used for many purposes besides motion control. Such a system may control a mechanical positioning unit, which may be linear or rotary, its velocity, acceleration, or combinations of these motion parameters. A control system may also be used to set voltage, tension, and other manufacturing process variables and to actuate various types of solenoid-operated valves. The main factors governing design of control systems are whether they are to be open- or closed-loop; what kinds and amounts of power are available; and the function requirements.

Factors governing selection of control systems are listed in Table 1.

Table 1. Control System Application Factors

Type of System	Nature of required control motion, i.e., position, velocity, acceleration
Accuracy	Controlled output versus input
Mechanical Load	Viscous friction, coulomb friction, starting friction, load inertia
Impact Loads	Hitting mechanical stops and load disturbances
Ratings	Torque or force, and speed
Torque	Peak instantaneous torque
Duty Cycle	Load response, torque level, and duration and effect on thermal response
Ambient Temperature	Relation to duty cycle and internal temperature rise, and to the effect of temperature on the sensor
Speed of Response	Time to reach commanded condition. Usually defined by a response to a stepped command
Frequency Response	Output to input ratio versus frequency, for varying frequency and specified constant input amplitude. Usually expressed in decibels
No-Load Speed	Frequently applies to maximum kinetic energy and to impact on stops; avoiding overspeeding
Backdriving	With power off, can the load drive the motor? Is a fail-safe brake required? Can the load backdrive with power on without damage to the control electronics? (Electric motor acting as a generator)
Power Source	Range of voltage and frequency within which the system must work. Effect of line transients
Environmental Conditions	Range of nonoperating and operating conditions, reliability and serviceability, scheduled maintenance

Open-Loop Systems.—The term open-loop typically describes use of a rheostat or variable resistance to vary the input voltage and thereby adjust the speed of an electric motor, a low-accuracy control method because there is no output sensor to measure the performance. However, use of stepper motors (see Table 2, and page 2473) in open-loop systems can make them very accurate. Shafts of stepper motors are turned through a fixed angle for every electrical pulse transmitted to them. The maximum pulse rate can be high, and the shaft can be coupled with step-down gear drives to form inexpensive, precise drive units with wide speed ranges. Although average speed with stepper motors is exact, speed modulation can occur at low pulse rates and drives can incur serious resonance problems.

Table 2. Control Motor Types

AC Motors	Induction motors, simplest, lowest cost, most rugged, can work directly off the ac line or through an inexpensive, efficient, and compact thyristor controller. Useful in fan and other drives where power increases rapidly with speed as well as in simple speed regulation. Ac motors are larger than comparable permanent-magnet motors
Two-Phase Induction Motors	Often used as control motors in small electromechanical control systems. Power outputs range from a few milliwatts to tens of watts
Split-Field Series Motors	Work on both ac and dc. Feature high starting torque, low cost, uniform power output over a wide speed range, and are easily reversed with a single-pole three-position switch. Very easy to use with electric limit switches for controlling angle of travel
Permanent-Magnet Motors	Operate on dc, with high power output and high efficiency. The most powerful units use rare-earth magnets and are more expensive than conventional types. Lower-cost ferrite magnets are much less expensive and require higher gear-reduction ratios, but at their higher rated speeds are very efficient
Brushless DC Motors	Use electrical commutation and may be applied as simple drive motors or as four-quadrant control motors. The absence of brushes for commutation ensures high reliability and low electromagnetic interference
Stepper Motors	Index through a fixed angle for each input pulse so that speed is in exact proportion to pulse rate and the travel angle increases uniformly with the number of pulses. Proper application in systems with backlash and load inertia requires special care
Wound-Field DC Motors	For subfractional to integral horsepower applications where size is not significant. Cost is moderate because permanent magnets are not required. Depending on the windings, output characteristics can be adjusted for specific applications

Open-loop systems are only as accurate as the input versus output requirement can be calibrated, including the effects of changes in line voltage, temperature, and other operating conditions.

Closed-Loop Systems.—Table 3 shows some parameters and characteristics of closed-loop systems, and a simple example of such a system is shown below. A command may be input by a human operator, it may be derived from another piece of system equipment, or it may be generated by a computer. Generally, the command is in the form of an electrical signal. The system response is converted by the output sensor to a compatible, scaled electrical signal that may be compared with the input command, the difference constituting an error signal. It is usually required that the error be small, so it is amplified and applied to an appropriate driving unit. The driver may take many forms, but for motion control it is usually a motor.

The amplified error voltage drives the motor to correct the error. If the input command is constant, the system is a closed-loop regulator.

Closed-loop systems use feedback sensors that measure system output and give instructions to the power drive components, based on the measured values. A typical closed-loop speed control, for instance, uses a tachometer as a feedback sensor and will correct automatically for differences between the tachometer output and the commanded speed. All motion control systems require careful design to achieve good practical performance. Closed-loop systems generally cost more than open-loop systems because of the extra cost of the tachometer or transducer used for output measurement. Faster response components also increase cost.

Table 3. Closed-Loop System Parameters and Characteristics

Step Response	The response of the system to a step change in the input command. The response to a large step, which can saturate the system amplifier, is different from the response to a small nonsaturating step. Initial overshoots may not be permissible in some types of equipment
Frequency Response	System response to a specified small-amplitude sinusoidal command where frequency is varied over the range of interest. The response is in decibels (dB), where dB = 20 $\log_{10}$(output/input). This characteristic determines whether the system is responsive enough to meet requirements
Bandwidth	The effective range of input frequencies within which the control system responds well. The bandwidth is often described by the point where the frequency response is down by three decibels. Bandwidth is usually defined in Hz (cycles/sec) or $\omega = 2\pi \times$ Hz (radians/sec)
Loading	The torque required to drive the load and the load inertia. The amplifier must supply enough power to meet acceleration as well as output power requirements. If the load is nonlinear, its effect on error must be within specifications. Behavior may vary considerably, depending on whether the load aids or opposes motor torque, as in a hoist
Output Stiffness	A measure of the system's response to load disturbances. Dynamic stiffness measures the system's response to a rapidly varying load
Resonant Peaks	Can show up in frequency-response testing as sharp (undamped) resonances. These resonances cannot be tolerated in the normal frequency range of the control system because they can lead to oscillation and vibration
No load, or maximum speed and maximum torque	Can be controlled by voltage or current limiting in the electronic amplifier. A slip clutch can also be used for torque limiting, particularly to avoid impact damage

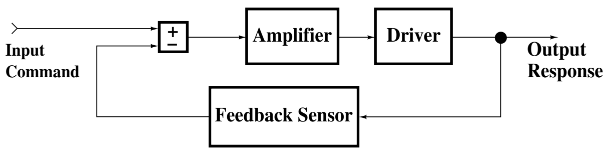


Fig. 1. General Arrangement of a Closed-Loop Control System

Accuracy of closed-loop systems is directly related to the accuracy of the sensor, so that choosing between open-loop and closed-loop controls may mean choosing between low price and consistent, accurate repeatability. In the closed-loop arrangement in Fig. 1, the sensor output is compared with the input command and the difference is amplified and applied to the motor to produce a correction. When the amplifier gain is high (the difference is greatly enlarged), even a small error will generate a correction. However, a high gain can lead to an unstable system due to inherent delays between the electrical inputs and outputs, especially with the motor.

Response accuracy depends not only on the precision of the feedback sensor and the gain of the amplifier, but also on the rate at which the command signal changes. The ability of the control system to follow rapidly changing inputs is naturally limited by the maximum motor speed and acceleration.

Amplified corrections cannot be applied to the motor instantaneously, and the motor does not respond immediately. Overshoots and oscillations can occur and the system must be adjusted or tuned to obtain acceptable performance. This adjustment is called damping the system response. Table 4 lists a variety of methods of damping, some of which require specialized knowledge.

Table 4. Means of Damping System Response

Network Damping	Included in the electrical portion of the closed loop. The networks adjust amplitude and phase to minimize control system feedback oscillations. Notch networks are used to reduce gain at specific frequencies to avoid mechanical resonance oscillations
Tachometer Damping	Feedback proportional to output velocity is added to the error signal for system stabilization
Magnetic Damping	Viscous or inertial dampers on the motor rear shaft extension for closed-loop stabilization. Similar dampers use silicone fluid instead of magnetic means to provide damping
Nonlinear Damping	Used for special characteristics. Inverse error damping provides low damping for large errors, permitting fast slewing toward zero and very stable operation at zero. Other nonlinearities meet specific needs, for example, coulomb friction damping works well in canceling backlash oscillations
Damping Algorithms	With information on output position or velocity, or both, sampled data may be used with appropriate algorithms to set motor voltage for an optimum system response

The best damping methods permit high error amplification and accuracy, combined with the desired degree of stability. Whatever form the output takes, it is converted by the output sensor to an electrical signal of compatible form that can be compared with the input command. The error signal thus generated is amplified before being applied to the driving unit.

Drive Power.—Power for the control system often depends on what is available and may vary from single- and three-phase ac 60 or 400 Hz, through dc and other types. Portable or mobile equipment is usually battery-powered dc or an engine-driven electrical generator. Hydraulic and pneumatic power may also be available. Cost is often the deciding factor in the choice.

Table 5. Special Features of Controllers

Linear or Pulse-Width Modulated	Linear is simpler, PWM is more complex and can generate electromagnetic interference, but is more efficient
Current Limiting	Sets limits to maximum line or motor current. Limits the torque output of permanent magnet motors. Can reduce starting transients and current surges
Voltage Limiting	Sets limits to maximum motor speed. Permits more uniform motor performance over a wide range of line voltages
Energy Absorption	Ability of the controller to absorb energy from a dc motor drive, back-driven by the load
EMI Filtering	Especially important when high electrical gain is required, as in thermocouple circuits, for example
Isolation	Of input and output, sometimes using optoisolators, or transformers, when input and output circuits require a high degree of isolation

Control Function.—The function of the control is usually set by the designer of the equipment and needs careful definition because it is the basis for the overall design. For instance, in positioning a machine tool table, such aspects as speed of movement and permissible variations in speed, accuracy of positioning, repeatability, and overshoot are among dozens of factors that must be considered. Some special features of controllers are listed in Table 5. Complex electromechanical systems require more knowledge of design and debugging than are needed for strictly mechanical systems.

Electromechanical Control Systems.—Wiring is the simplest way to connect components, so electromechanical controls are more versatile than pure hydraulic or pneumatic controls. The key to this versatility is often in the controller, the fundamental characteristic of which is its power output. The power output must be compatible with motor and load requirements. Changes to computer chips or software can usually change system performance to suit the application.

When driving a dc motor, for instance, the controller must supply sufficient power to match load requirements as well as motor operating losses, at minimum line voltage and maximum ambient temperature. The system's wiring must not be greatly sensitive to transient or steady-state electrical interference, and power lines must be separated from control signal lines, or appropriately shielded and isolated to avoid cross-coupling. Main lines to the controller must often include electrical interference filters so that the control system does not affect the power source, which may influence other equipment connected to the same source. For instance, an abruptly applied step command can be smoothed out so that heavy motor inrush currents are avoided. The penalty is a corresponding delay in response.

Use of current limiting units in a controller will not only set limits to line currents, but will also limit motor torque. Electronic torque limiting can frequently avoid the need for mechanical torque limiting. An example of the latter is using a slip clutch to avoid damage due to overtravel, the impact of which usually includes the kinetic energy of the moving machine elements. In many geared systems, most of the kinetic energy is in the motor. Voltage limiting is less useful than current limiting but may be needed to isolate the motor from voltage transients on the power line, to prevent overspeeding, as well as to protect electronic components.

Mechanical Stiffness.—When output motion must respond to a rapidly changing input command, the control system must have a wide bandwidth. Where the load mass (in linear motion systems) or the polar moment of inertia (in rotary systems) is high, there is a possibility of resonant oscillations. For the most stable and reliable systems, with a defined load, a high system mechanical stiffness is preferred. To attain this stiffness requires strengthening shafts, preloading bearings, and minimizing free play or backlash. In the best-performing systems, motor and load are coupled without intervening compliant members. Even tightly bolted couplings can introduce compliant oscillations resulting from extremely minute slippages caused by the load motions.

Backlash is a factor in the effective compliance of any coupling but has little effect on the resonant frequency because little energy is exchanged as the load is moved through the backlash region. However, even in the absence of significant torsional resonance, a high-gain control system can “buzz” in the backlash region. Friction is often sufficient to eliminate this small-amplitude, high-frequency component.

The difficulty with direct-drive control systems lies in matching motor to load. Most electric motors deliver rated power at higher speeds than are required by the driven load, so that load power must be delivered by the direct-drive motor operating at a slow and relatively inefficient speed. Shaft power at low speed involves a correspondingly high torque, which requires a large motor and a high-power controller. Motor copper loss (heating) is high in delivering the high motor torque. However, direct-drive motors provide maximum load velocity and acceleration, and can position massive loads within seconds of arc (rotational) or tenths of thousandths of an inch (linear) under dynamic conditions.

Where performance requirements are moderate, the required load torque can be traded off against speed by using a speed-changing transmission, typically, a gear train. The transmission effectively matches the best operating region of the motor to the required operating region of the load, and both motor and controller can be much smaller than would be needed for a comparable direct drive.

Torsional Vibration.—Control system instabilities can result from insufficient stiffness between the motor and the inertia of the driven load. The behavior of such a system is similar to that of a torsional pendulum, easily excited by commanded motions of the control system. If frictional losses are moderate to low, sustained oscillations will occur. In spite of the complex dynamics of the closed-loop system, the resonant frequency, as for a torsional pendulum, is given to a high degree of accuracy by the formula:

$$f_n = \frac{1}{2\pi} \times \sqrt{\frac{K}{J_L}}$$

where f_n is in hertz, K is torsional stiffness in in.-lb/rad, and J_L is load inertia in in.-lb-sec^2/rad. If this resonant frequency falls within the bandwidth of the control system, self-sustained oscillations are likely to occur. These oscillations are often overlooked by control systems analysts because they do not appear in simple control systems, and they are very difficult to correct.

Friction inherently reduces the oscillation by dissipating the energy in the system inertia. If there is backlash between motor and load, coulomb friction (opposing motion but independent of speed) is especially effective in damping out the oscillation. However, the required friction for satisfactory damping can be excessive, introducing positioning error and adding to motor (and controller) power requirements. Friction also varies with operating conditions and time.

The most common method of eliminating torsional oscillation is to introduce a filter in the error channel of the control system to shape the gain characteristic as a function of frequency. If the torsional resonance is within the required system bandwidth, little can be done except stiffening the mechanical system and increasing the resonant frequency. If the filter reduces the gain within the required bandwidth, it will reduce performance. This method will work only if the natural resonance is above the minimum required performance bandwidth.

The simplest shaping network is the notch network (Table 4, network damping), which, in effect, is a band-rejection filter that sharply reduces gain at the notch frequency. By locating the notch frequency so as to balance out the torsional resonance peak, the oscillation can be eliminated. Where there are several modes of oscillation, several filter networks can be connected in series.

Electric Motors.—Electric motors for control systems must suit the application. Motors used in open-loop systems (excluding stepper motors) need not respond quickly to input command changes. Where the command is set by a human, response times of hundreds of milliseconds to several seconds may be acceptable. Slow response does not lead to the instabilities that time delays can introduce into closed-loop systems.

Closed-loop systems need motors with fast response, of which the best are permanent-magnet dc units, used where wide bandwidth, efficient operation, and high power output are required. Table 2 lists some types of control motors and their characteristics. An important feature of high-performance, permanent-magnet motors using high-energy, rare-earth magnets is that their maximum torque output capacity can be 10 to 20 or more times higher than their rated torque. In intermittent or low-duty-cycle applications, very high torque loads can be driven by a given motor. However, when rare-earth magnets (samarium cobalt or neodymium) are not used, peak torque capability may be limited by the possibility of demagnetization. Rare-earth magnets are relatively expensive, so it is important to verify peak torque capabilities for lower-cost motors that may use weaker Alnico or ferrite magnets.

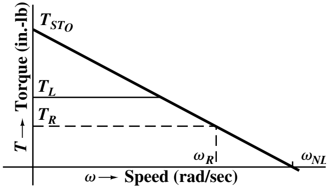


Fig. 2. Idealized Control Motor Characteristics for a Consistent Set of Units

Duty-cycle calculations are an aspect of thermal analysis that are well understood and are not covered here. Motor manufacturers usually supply information on thermal characteristics including thermal time constants and temperature rise per watt of internal power dissipation.

Characteristics of permanent-magnet motors are defined with fair accuracy by relatively few parameters. The most important characteristics are: D_M motor damping in lb-in.-sec/rad; J_M motor inertia in lb-in.-sec$_2$/rad; and R winding resistance in ohms. Fig. 1 shows other control motor characteristics, T_{STo} stall torque with no current limiting; T_L maximum torque with current limiting; ω_{NL} no-load speed; ω_R rated speed. Other derived motor parameters include V rated voltage in volts; $I_{STo} = V/R$ current in amperes at stall with no current limiting; I_L ampere limit, adjusted in amplifier; I_R rated current; $K_T = T_{STo}/I_{STo}$ torque constant in in.-lb/ampere; $K_E = V/\omega_{NL}$ voltage constant in volt/rad/sec; $K_M = K_T/\sqrt{R}$, torque per square root of winding resistance; $D_M = T_{STo}/\omega_{NL}$ motor damping in in.-lb/rad/sec; and $T_M = J_M/D_M$ motor mechanical time constant in seconds.

Stepper Motors.—In a stepper motor, power is applied to a wound stator, causing the brushless rotor to change position to correspond with the internal magnetic field. The rotor maintains its position relative to the internal magnetic field at all times. In its most common mode of operation, the stepper motor is energized by an electronic controller whose current output to the motor windings defines the position of the internally generated magnetic field. Applying a command pulse to the controller will change the motor currents to reposition the rotor. A series of pulses, accompanied by a direction command, will cause rotation in uniformly spaced steps in the specified direction.

If the pulses are applied at a sufficiently high frequency, the rotor will be carried along with the system's inertia and will rotate relatively uniformly but with a modulated velocity. At the other extreme, the response to a single pulse will be a step followed by an overshoot and a decaying oscillation. Where the application cannot permit the oscillation, damping can be included in the controller.

Stepper motors are often preferred because positions of the rotor are known from the number of pulses and the step size. An initial index point is required as an output position reference, and care is required in the electronic circuits to avoid introducing random pulses that will cause false positions. As a minimum, the output index point on an appropriate shaft can verify the step count during operation.

Gearing.—In a closed-loop system, gearing may be used to couple a high-speed, low-torque motor to a lower-speed, higher-torque load. The gearing must meet requirements for accuracy, strength, and reliability to suit the application. In addition, the closed loop requires minimum backlash at the point where the feedback sensor is coupled. In a velocity-controlled system, the feedback sensor is a tachometer that is usually coupled directly to the rotor shaft. Backlash between motor and tachometer, as well as torsional compliance, must be minimized for stable operation of a high-performance system. Units combining motor and tachometer on a single shaft can usually be purchased as an assembly.

By contrast, a positioning system may use a position feedback sensor that is closely coupled to the shaft being positioned. As with the velocity system, backlash between the motor and feedback sensor must be minimized for closed-loop stability. Antibacklash gearing is frequently used between the gearing and the position feedback sensor. When the position feedback sensor is a limited rotation device, it may be coupled to a gear that turns faster than the output gear to allow use of its full range. Although this step-up gearing enhances it, accuracy is ultimately limited by the errors in the intermediate gearing between the position sensor and the output.

When an appreciable load inertia is being driven, it is important that the mechanical stiffness between the position sensor coupling point and the load be high enough to avoid natural torsional resonances in the passband.

Feedback Transducers.—Controlled variables are measured by feedback transducers and are the key to accuracy in operation of closed-loop systems. When the accuracy of a carefully designed control system approaches the accuracy of the feedback transducer, the need for precision in the other system components is reduced.

Transducers may measure the quantity being controlled in digital or analog form, and are available for many different parameters such as pressure and temperature, as well as distance traveled or degrees of rotation. Machine designers generally need to measure and control linear or rotary motion, velocity, position, and sometimes acceleration. Although some transducers are nonlinear, a linear relationship between the measured variable and the (usually electrical) output is most common.

Output characteristics of an analog linear-position transducer are shown in Fig. 2. By dividing errors into components, accuracy can be increased by external adjustments, and slope error and zero offsets are easily trimmed in. Nonlinearity is controlled by the manufacturer. In Fig. 2 are seen the discrete error components that can be distinguished because of the ease with which they can be canceled out individually by external adjustments. The most common compensation is for zero-position alignment, so that when the machine has been set to the start position for a sequence, the transducer can be positioned to read zero output. Alternatively, with all components in fixed positions, a small voltage can be inserted in series with the transducer output for a very accurate alignment of mechanical and electrical zeros. This method helps in canceling long-term drift, particularly in the mechanical elements.

The second most common adjustment of a position transducer is of its output gradient, that is, transducer output volts per degree. Depending on the type of analog transducer, it is usually possible to add a small adjustment to the electrical input, to introduce a proportional change in output gradient. As with the zero-position adjustment, the gradient may be set very accurately initially and during periodic maintenance. The remaining errors shown in Fig. 2, such as intrinsic nonlinearity or nonconformity, result from limitations in design and manufacture of the transducer.

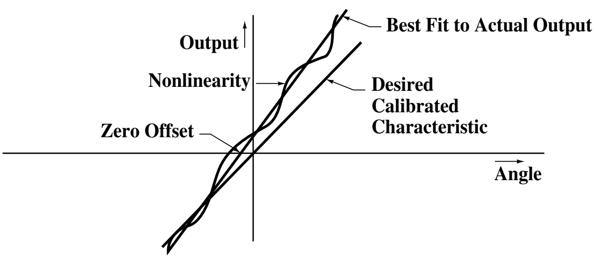


Fig. 3. Output Characteristics of a General Linear Position Transducer

Greater accuracy can be achieved in computer-controlled systems by using the computer to cancel out transducer errors. The system's mechanical values and corresponding transducer values are stored in a lookup table in the computer and referred to as necessary. Accuracies approaching the inherent repeatability and stability of the system can thus be secured. If necessary, recalibration can be performed at frequent intervals.

Analog Transducers.—The simplest analog position transducer is the resistance potentiometer, the resistance element in which is usually a deposited-film rather than a wire-wound type. Very stable resistance elements based on conductive plastics, with resolution to a few microinches and operating lives in the 100 million rotations, are available, capable of working in severe environments with high vibrations and shock and at temperatures of 150 to 200°C. Accuracies of a few hundredths, and stability of thousandths, of a per cent, can be obtained from these units by trimming the plastics resistance element as a function of angle.

Performance of resistance potentiometers deteriorates when they operate at high speeds, and prolonged operation at speeds above 10 rpm causes excessive wear and increasing output noise. An alternative to the resistance potentiometer is the variable differential transformer, which uses electrical coupling between ac magnetic elements to measure angular or linear motion without sliding contacts. These units have unlimited resolution with accuracy comparable to the best resistance potentiometers but are more expensive and require compatible electronic circuits.

A variable differential transformer needs ac energization, so an ac source is required. A precision demodulator is frequently used to change the ac output to dc. Sometimes the ac output is balanced against an ac command signal whose input is derived from the same ac source. In dealing with ac signals, phase-angle matching and an accurate amplitude-scale factor are required for proper operation. Temperature compensation also may be required, primarily due to changes in resistance of the copper windings. Transducer manufacturers will supply full sets of compatible electronic controls.

Synchros and Resolvers.—Synchros and resolvers are transducers that are widely used for sensing of angles at accuracies down to 10 to 20 arc-seconds. More typically, and at much lower cost, their accuracies are 1 to 2 arc-minutes. Cost is further reduced when accuracies of 0.1 degree or higher are acceptable.

Synchros used as angle-position transducers are made as brush types with slip rings and in brushless types. These units can rotate continuously at high speeds, the operating life of brushless designs being limited only by the bearing life. Synchros have symmetrical three-wire stator windings that facilitate transmission of angle data over long distances (thousands of feet). Such a system is also highly immune to noise and coupled signals. Practically the only trimming required for very long line systems is matching the line-to-line capacitances.

Because synchros can rotate continuously, they can be used in multispeed arrangements, where, for example, full-scale system travel may be represented by 36 or 64 full rotations. When reduced by gearing to a single, full-scale turn, a synchro's electrical inaccuracy is the typical 0.1° error divided by 36 or 64 or whatever gear ratio is used. This error is insignificant compared with the error of the gearing coupling the high-speed synchro and the single speed (1 rotation for full scale) output shaft. The accuracy is dependable and stable, using standard synchros and gearing.

Hydraulic and Pneumatic Systems

In Fig. 1 is shown a schematic of a hydraulic cylinder and the relationships between force and area that govern all hydraulic systems. Hydraulic actuators that drive the load may be cylinders or motors, depending on whether linear or rotary motion is required. The load must be defined by its torque–speed characteristics and inertia, and a suitable hydraulic actuator selected before the remaining system components can be chosen. Fluid under pressure and suitable valves are needed to control motion. Both single- and double-acting hydraulic cylinders are available, and the latter type is seen in Fig. 1.

Pressure can be traded off against velocity, if desired, by placing a different effective area at each side of the piston. The same pressure on a smaller area will move the piston at a higher speed but lower force for a given rate of fluid delivery. The cylinder shown in Fig. 1 can drive loads in either direction. The simple formulas of plane geometry relate cylinder areas, force, fluid flow, and rate of movement. Other configurations can develop equal forces and speeds in both directions.

The rotary equivalent of the cylinder is the hydraulic motor, which is defined by the fluid displacement required to turn the output shaft through one revolution, by the output torque, and by the load requirements of torque and speed. Output torque is proportional to fluid pressure, which can be as high as safety permits. Output speed is defined by the number of gallons per minute supplied to the motor. As an example, if 231 cu. in. = 1 gallon, an input of 6 gallons/min (gpm) with a 5-cu. in. displacement gives a mean speed of $6 \times 231/5 = 277$ rpm. The motor torque must be defined by lb-in. per 100 $lb_f/in.^2$ (typically) from which the required pressure can be determined. Various motor types are available.

Hydraulic Pumps.—The most-used hydraulic pump is the positive-displacement type, which delivers a fixed amount of fluid for every cycle. These pumps are also called hydrostatic because they deliver energy by static pressure rather than by the kinetic energy of a moving fluid. Positive-displacement pumps are rated by the gpm delivered at a stated speed and by the maximum pressure, which are the key parameters defining the power capacity of the hydraulic actuator. Delivered gpm are reduced under load due to leakage, and the reduction is described by the volumetric efficiency, which is the ratio of actual to theoretical output.

Hydraulic Fluids.—The hydraulic fluid is the basic means of transmitting power, and it also provides lubrication and cooling when passed through a heat exchanger. The fluid must be minimally compressible to avoid springiness and delay in response. The total system inertia reacts with fluid compliance to generate a resonant frequency, much as inertia and mechanical compliance react in an electromechanical system. Compliance must be low enough that resonances do not occur in the active bandwidth of the servomechanism, and that unacceptable transients do not occur under shock loads. Seal friction and fluid viscosity tend to damp out resonant vibrations. Shock-absorbing limit stops or cushions are usually located at the travel limits to minimize transient impact forces.

F = force (lb) = pressure × area

P_1 and P_2 = line pressure on either side of piston in $lb_f/in.^2$

d_1 and d_2 = diameters of piston rod and piston in in.

$$F_1 = \frac{\pi}{4}(d_2^2 - d_1^2) \times P_1 = 0.7854 P_1 (d_2^2 - d_1^2)$$

$$F_2 = \frac{\pi}{4} d_2^2 P_2 = 0.7854 P_2 d_2^2$$

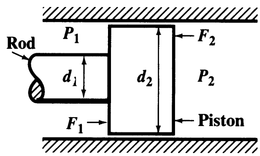


Fig. 1. Elementary Hydraulic Force/Area Formulas

Hydraulic fluids with special additives for lubrication minimize wear between moving parts. An auxiliary function is prevention of corrosion and pitting. Hydraulic fluids must also be compatible with gaskets, seals, and other nonmetallic materials.

Viscosity is another critical parameter of hydraulic fluids as high viscosity means high resistance to fluid flow with a corresponding power loss and heating of the fluid, pressure drop in the hydraulic lines, difficulty in removing bubbles, and sometimes overdamped operation. Unfortunately, viscosity falls very rapidly with increasing temperature, which can lead to reduction of the lubrication properties and excessive wear as well as increasing leakage. For hydraulic actuators operating at very low temperatures, the fluid pour point is important. Below this temperature, the hydraulic fluid will not flow. Design guidelines similar to those used with linear or rotating bearings are applicable in these conditions. Fire-resistant fluids are available for use in certain conditions such as in die casting, where furnaces containing molten metal are often located near hydraulic systems.

A problem with hydraulic systems that is absent in electromechanical systems is that of dirt, air bubbles, and contaminants in the fluid. Enclosed systems are designed to keep out contaminants, but the main problem is with the reservoir or fluid storage unit. A suitable sealer must be used in the reservoir to prevent corrosion and a filter should be used during filling. Atmospheric pressure is required on the fluid surface in the reservoir except where a pressurized reservoir is used. Additional components include coarse and fine filters to remove contaminants and these filters may be rated to remove micron sized particles (1 micron = 0.00004 in.).

Very fine filters are sometimes used in high-pressure lines, where dirt might interfere with the operation of sensitive valves. Where a high-performance pump is used, a fine filter is a requirement. Usually, only coarse filters are used on fluid inlet lines because fine filters might introduce excessive pressure drop.

Aside from the reservoir used for hydraulic fluid storage, line connections, fittings, and couplings are needed. Expansion of these components under pressure increases the mechanical compliance of the system, reducing the frequencies of any resonances and possibly interfering with the response of wide-band systems.

Formulas relating fluid flow and mechanical power follow. These formulas supplement the general force, torque, speed, and power formulas of mechanical systems.

$$F = P \times A$$

$$A = 0.7854 \times d^2$$

$$hp = 0.000583q \times \text{pressure in lb}_f/\text{in.}^2$$

1 gallon of fluid flow/min at 1 $\text{lb}_f/\text{in.}^2$ pressure = 0.000582 hp.

For rotary outputs,

$$hp = \text{torque} \times \text{rpm}/63{,}025$$

where torque is in lb-in. (Theoretical hp output must be multiplied by the efficiency of the hydraulic circuits to determine actual output.)

In the preceding equations,

P = pressure in $\text{lb}_f/\text{in.}^2$

A = piston area in in.^2

F = force in lb

q = fluid flow in gallons/min

d = piston diameter in inches

Hydraulic and Pneumatic Control Systems.—Control systems for hydraulic and pneumatic circuits are more mature than those for electromechanical systems because they have been developed over many more years. Hydraulic components are available at moderate prices from many sources. Although their design is complex, application and servicing of these systems are usually more straightforward than with electromechanical systems.

Electromechanical and hydraulic/pneumatic systems may be analyzed by similar means. The mathematical requirements for accuracy and stability are analogous, as are most performance features, although nonlinearities are caused by different physical attributes. Nonlinear friction, backlash, and voltage and current limiting are common to both types of system, but hydraulic/pneumatic systems also have the behavior characteristics of fluid-driven systems such as thermal effects and fluid flow dynamics including turbulence, leakage caused by imperfect seals, and contamination.

Both control types require overhead equipment that does not affect performance but adds to overall cost and complexity. For instance, electromechanical systems require electrical power sources and power control components, voltage regulators, fuses and circuit breakers, relays and switches, connectors, wiring and related devices. Hydraulic/pneumatic systems require fluid stored under pressure, motor-driven pumps or compressors, valves, pressure regulators/limiters, piping and fasteners, as well as hydraulic/pneumatic motors and cylinders. Frequently, the optimum system is selected on the basis of overhead equipment already available.

Electromechanical systems are generally slower and heavier than hydraulic systems and less suited to controlling heavy loads. The bandwidths of hydraulic control systems can respond to input signals of well over 100 Hz as easily as an electromechanical system can respond to, say, 10 to 20 Hz. Hydraulic systems can drive very high torque loads without intermediate transmissions such as the gear trains often used with electromechanical systems. Also, hydraulic/pneumatic systems using servo valves and piston/cylinder arrangements are inherently suited to linear motion operation, whereas electromechanical controls based on conventional electrical machines are more naturally suited to driving rotational loads.

Until recently, electromechanical systems were limited to system bandwidths of about 10 Hz, with power outputs of a few hundred watts. However, their capabilities have now been sharply extended through the use of rare-earth motor magnets having much higher energies than earlier designs. Similarly, semiconductor power components deliver much higher output power at lower prices than earlier equipment. Electromechanical control systems are now suited to applications of more than 100 hp with bandwidths up to 40 Hz and sometimes up to 100 Hz.

Although much depends on the specific design, the edge in reliability, even for high-power, fast-response needs, is shifting toward electromechanical systems. Basically, there are more things that can go wrong in hydraulic/pneumatic systems, as indicated by the shift to more electrical systems in aircraft.

Hydraulic Control Systems.—Using essentially incompressible fluid, hydraulic systems are suited to a wide range of applications, whereas pneumatic power is generally limited to simpler uses. In Fig. 2 are shown the essential features of a simple linear hydraulic control system and a comparable system for driving a rotating load.

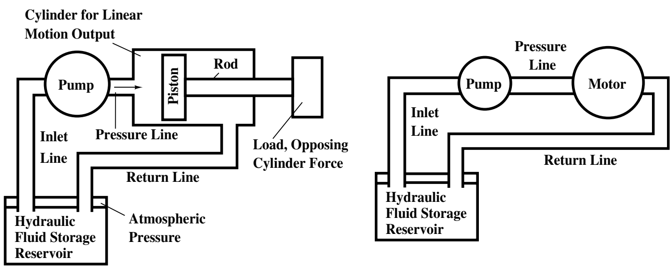


Fig. 2. (left) A Simple Linear Hydraulic Control System in Which the Load Force Returns the Piston and (right) a Comparable System for Driving a Rotating Load

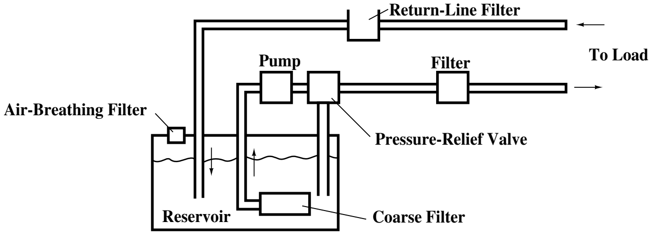


Fig. 3. Some of the Auxiliary Components Used in a Practical Hydraulic System

Hydraulic controls of the type shown have fast response and very high load capacities. In a linear actuator, for example, each $lb_f/in.^2$ of system pressure acts against the area of the piston to generate the force applied. Hydraulic pressures of up to 3000 $lb_f/in.^2$ are readily obtained from hydraulic pumps, so that cylinders can exert forces of hundreds of tons without the need for speed-reducing transmission systems to increase the force. The hydraulic fluid distributes heat, so it helps cool the system.

Systems similar to those in Fig. 2 can be operated in open- or closed-loop modes. Open-loop operation can be controlled by programming units that initiate each step by operating relays, limit switches, solenoid valves, and other components to generate the forces over the required travel ranges. Auxiliary components are used to ensure safe operation and make such systems flexible and reliable, as shown in Fig. 3.

In the simplest mode, whether open- or closed-loop, hydraulic system operation may be discontinuous or proportional. Discontinuous operation, sometimes called bang-bang, or on–off, works well, is widely used in low- to medium-accuracy systems, and is easy to maintain. In this closed-loop mode, accuracy is limited; if the response to error is set too high, the system will oscillate between on–off modes, with average output at about the desired value. This oscillation, however, can be noisy, introduces system transients, and may cause rapid wear of system components.

Another factor to be considered in on–off systems is the shock caused by sudden opening and closing of high-pressure valves, which introduce transient pulses in the fluid flow and can cause high stresses in components. These problems can be addressed by the use of pressure-limiting relief valves and other units.

Proportional Control Systems.—Where the highest accuracy is required, perhaps in two directions, and with aiding or opposing forces or torques, a more sophisticated proportional control, closed-loop system is preferred. As shown in Fig. 4, the amplifier and electric servomotor used in electromechanical closed-loop systems is replaced in the closed-loop hydraulic system by an electronically controlled servo-valve. In its simplest form, the valve uses a linear motor to position the spool that determines the flow path for the hydraulic fluid. In some designs, the linear motor may be driven by a solenoid against a bias spring on the value spool. In other arrangements, the motor may be a bidirectional unit that permits a fluid flow depending on the polarity and amplitude of the voltage supplied to the motor.

Such designs can be used in proportional control systems to achieve smooth operation and minimum nonlinearities, and will give the maximum accuracy required by the best machine tool applications. Where very high power must be controlled, use is often made of a two-stage valve in which the output from the first stage is used to drive the second-stage valve, as shown in Fig. 5.

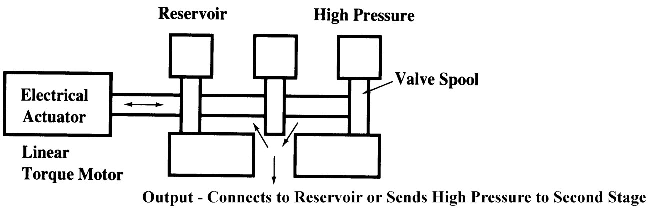


Fig. 4.

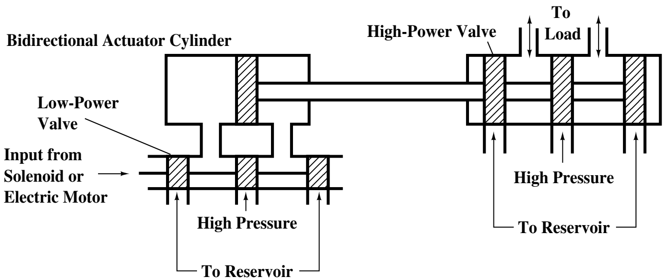


Fig. 5. Two-Stage Valve for Large-Power Control from a Low-Power Input

Electronic Controls.—An error-sensing electronic amplifier drives the solenoid motor of Fig. 5, which provides automatic output correction in a closed-loop system. The input is an ideal place to introduce electrical control features, adding greatly to the versatility of the control system. The electronic amplifier can provide the necessary driving power using pulse-width modulation as required, for minimum heating. The output can respond to signals in the low-microvolt range.

A major decision is whether to use analog or digital control. Although analog units are simple, they are much less versatile than their digital counterparts. Digital systems can be readjusted for total travel, speed, and acceleration by simple reprogramming. Use of appropriate feedback sensors can match accuracy to any production requirement, and a single digital system can be easily adapted to a great variety of similar applications. This adaptability is an important cost-saving feature for moderate-sized production runs. Modern microprocessors can integrate the operation of sets of systems.

Because nonlinearities and small incremental motions are easy to implement, digital systems are capable of very smooth acceleration, which avoids damaging shocks and induced leaks, and enhances reliability so that seals and hose connections last longer. The accuracy of digital control systems depends on transducer availability, and a full range of such devices has been developed and is now available.

Other features of digital controls are their capacity for self-calibration, easy digital readout, and periodic self-compensation. For example, it is easy to incorporate backlash compensation. Inaccuracies can be corrected by using lookup tables that may themselves be updated as necessary. Digital outputs can be used as part of an inspection plan, to indicate need for tool changing, adjustment or sharpening, or for automatic record keeping. Despite continuing improvements in analog systems, digital control of hydraulic systems is favored in large plants.

Pneumatic Systems.—Hydraulic systems transmit power by means of the flow of an essentially incompressible fluid. Pneumatic systems use a highly compressible gas. For this reason, a pneumatic system is slower in responding to loads, especially sudden output loads, than a hydraulic system. Similarly, torque or force requires time and output motion to build up. Response to sudden output loads shows initial overshoot. Much more complex networks or other damping means are required to develop stable response in closed-loop systems. On the other hand, there are no harmful shock waves analogous to the transients that can occur in hydraulic systems, and pneumatic system components last comparatively longer.

Notwithstanding their performance deficiencies, pneumatic systems have numerous desirable features. Pneumatic systems avoid some fire hazards compared with the most preferred hydraulic fluids. Air can be vented to the atmosphere so a flow line only is needed, reducing the complexity, cost, and weight of the overall system. Pneumatic lines, couplings, and fittings are lighter than their hydraulic counterparts, often a significant advantage. The gaseous medium also is lighter than hydraulic fluid, and pneumatic systems are usually easier to clean, assemble, and generally maintain. Fluid viscosity and its temperature variations are virtually negligible with pneumatic systems.

Among drawbacks with pneumatics are that lubrication must be carefully designed in, and more power is needed to achieve a desired pressure when the fluid medium is a compressible gas. Gas under high pressure can cause an explosion if its storage tank is damaged, so storage must have substantial safety margins. Gas compressibility makes pneumatic systems 1 or 2 orders of magnitude slower than hydraulic systems.

The low stiffness of pneumatic systems is another indicator of the long response time. Resonances occur between the compressible gas and equivalent system inertias at lower frequencies. Even the relatively low speed of sound in connecting lines contributes to response delay, adding to the difficulty of closed-loop stabilization. Fortunately, it is possible to construct pneumatic analogs to electrical networks to simplify stabilization at the exact point of the delays. Such pneumatic stabilizing means are commercially available and are important elements of closed-loop pneumatic control systems.

In contrast with hydraulic systems, where speed may be controlled by varying pump output, pneumatic system control is almost exclusively by valves, which control the flow from a pneumatic accumulator or pressure source. The pressure is maintained between limits by an intermittently operated pump. Low-pressure outlet ports must be large enough to accommodate the high volume of the expanded gas. In Fig. 6 is shown a simplified system for closed-loop position control applied to an air cylinder, in which static accuracy is controlled by the position sensor. Proper design requires a good theoretical analysis and attention to practical design if good, stable, closed-loop response is to be achieved.

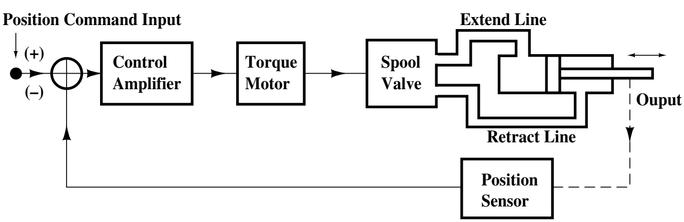


Fig. 6. A Pneumatic Closed-Loop Linear Control System

O-RINGS

An O-ring is a one-piece molded elastomeric seal with a circular cross-section that seals by distortion of its resilient elastic compound. Dimensions of O-rings are given in ANSI/SAE AS568A, Aerospace Size Standard for O-rings. The standard ring sizes have been assigned identifying dash numbers that, in conjunction with the compound (ring material), completely specifies the ring. Although the ring sizes are standardized, ANSI/SAE AS568A does not cover the compounds used in making the rings; thus, different manufacturers will use different designations to identify various ring compounds. For example, 230-8307 represents a standard O-ring of size 230 (2.484 in. ID by 0.139 in. width) made with compound 8307, a general-purpose nitrile compound. O-ring material properties are discussed at the end of this section.

When properly installed in a groove, an O-ring is normally slightly deformed so that the naturally round cross-section is squeezed diametrically out of round prior to the application of pressure. This compression ensures that under static conditions, the ring is in contact with the inner and outer walls enclosing it, with the resiliency of the rubber providing a zero-pressure seal. When pressure is applied, it tends to force the O-ring across the groove, causing the ring to further deform and flow up to the fluid passage and seal it against leakage, as in Fig. 1(a). As additional pressure is applied, the O-ring deforms into a D shape, as in Fig. 1(b). If the clearance gap between the sealing surface and the groove corners is too large or if the pressure exceeds the deformation limits of the O-ring material (compound), the O-ring will extrude into the clearance gap, reducing the effective life of the seal. For very low-pressure static applications, the effectiveness of the seal can be improved by using a softer durometer compound or by increasing the initial squeeze on the ring, but at higher pressures, the additional squeeze may reduce the ring's dynamic sealing ability, increase friction, and shorten ring life.

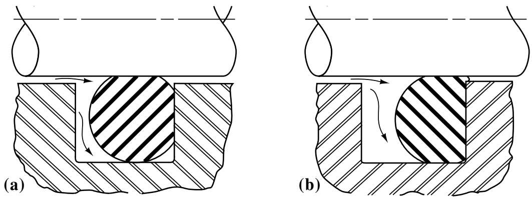


Fig. 1.

The initial diametral squeeze of the ring is very important in the success of an O-ring application. The squeeze is the difference between the ring width W and the gland depth F (Fig. 2) and has a great effect on the sealing ability and life of an O-ring application.

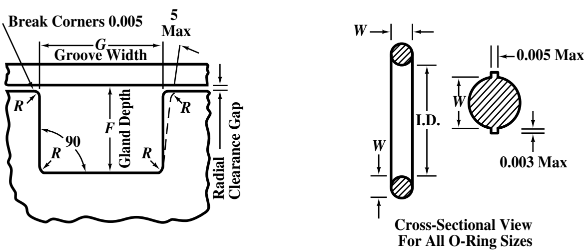


Fig. 2. Groove and Ring Details

The ideal squeeze varies according to the ring cross-section, with the average being about 20 per cent, i.e., the ring's cross-section *W* is about 20 per cent greater than the gland depth *F* (groove depth plus clearance gap). The groove width is normally about 1.5 times larger than the ring width *W*. When installed, an O-ring compresses slightly and distorts into the free space within the groove. Additional expansion or swelling may also occur due to contact of the ring with fluid or heat. The groove must be large enough to accommodate the maximum expansion of the ring or the ring may extrude into the clearance gap or rupture the assembly. In a dynamic application, the extruded ring material will quickly wear and fray, severely limiting seal life.

To prevent O-ring extrusion or to correct an O-ring application, reduce the clearance gap by modifying the dimensions of the system, reduce the system operating pressure, install antiextrusion backup rings in the groove with the O-ring, as in Fig. 3, or use a harder O-ring compound. A harder compound may result in higher friction and a greater tendency of the seal to leak at low pressures. Backup rings, frequently made of leather, Teflon, metal, phenolic, hard rubber, and other hard materials, prevent extrusion and nibbling where large clearance gaps and high pressure are necessary.

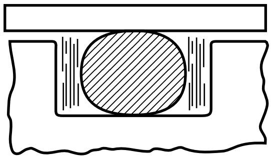


Fig. 3. Preferred Use of Backup Washers

The most effective and reliable sealing is generally provided by using the diametrical clearances given in manufacturers' literature. However, the information in Table 1 may be used to estimate the gland depth (groove depth plus radial clearance) required in O-ring applications. The radial clearance used (radial clearance equals one-half the diametral clearance) also depends on the system pressure, the ring compound and hardness, and specific details of the application.

Table 1. Gland Depth for O-Ring Applications

Standard O-Ring Cross-Sectional Diameter (in.)	Gland Depth (in.)	
	Reciprocating Seals	Static Seals
0.070	0.055 to 0.057	0.050 to 0.052
0.103	0.088 to 0.090	0.081 to 0.083
0.139	0.121 to 0.123	0.111 to 0.113
0.210	0.185 to 0.188	0.170 to 0.173
0.275	0.237 to 0.240	0.226 to 0.229

Source: Auburn Manufacturing Co. When possible, use manufacturer recommendations for clearance gaps and groove depth.

Fig. 4 indicates conditions where O-ring seals may be used, depending on the fluid pressure and the O-ring hardness. If the conditions of use fall to the right of the curve, extrusion of the O-ring into the surrounding clearance gap will occur, greatly reducing the life of the ring. If conditions fall to the left of the curve, no extrusion of the ring will occur, and the ring may be used under these conditions. For example, in an O-ring application with a 0.004-in. diametral clearance and 2500-psi pressure, extrusion will occur with a 70 durometer O-ring but not with an 80 durometer O-ring. As the graph indicates, high-pressure applications require lower clearances and harder O-rings for effective sealing.

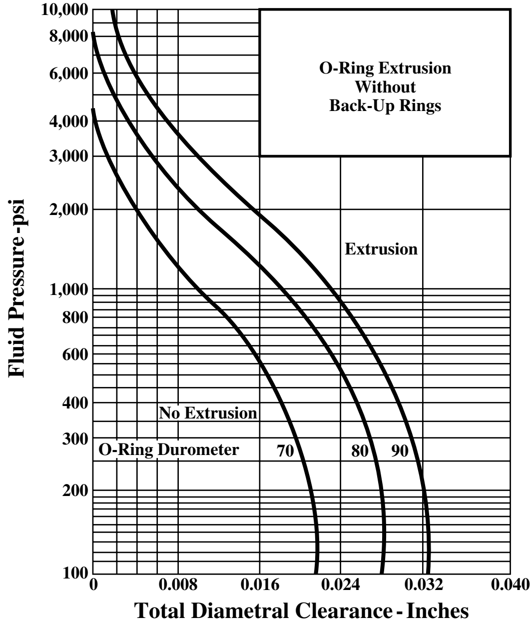


Fig. 4. Extrusion Potential of O-Rings as a Function of Hardness and Clearance

Recommended groove width, clearance dimensions, and bottom-of-groove radius for O-ring numbers up to 475 (25.940-in. ID by 0.275-in. width) can be found using Table 2 in conjunction with Fig. 5. In general, except for ring cross-sections smaller than $\frac{1}{16}$ in., the groove width is approximately 1.5W, where W is the ring cross-sectional diameter. Straight-sided grooves are best for preventing extrusion of the ring or nibbling; however, for low-pressure applications (less than 1500 psi) sloped sides with an angle up to 5° can be used to simplify machining of the groove. The groove surfaces should be free of burrs, nicks, or scratches. For static seals (i.e., no contact between the O-ring and any moving parts), the groove surfaces should have a maximum roughness of 32 to 63 µin. rms for liquid-sealing applications and 16 to 32 µin. rms for gaseous-sealing applications. In dynamic seals, relative motion exists between the O-ring and one or more parts and the maximum groove surface roughness should be 8 to 16 µin. rms for sliding contact applications (reciprocating seals, for example) and 16 to 32 µin. rms for rotary contact applications (rotating and oscillating seals).

In dynamic seal applications, the roughness of surfaces in contact with O-rings (bores, pistons, and shafts, for example) should be 8 to 16 µin. rms, without longitudinal or circumferential scratches. Surface finishes of less than 5 µin. rms are too smooth to give a good seal life because they wipe too clean, causing the ring to wear against the housing in the absence of a lubricating film. The best-quality surfaces are honed, burnished, or hard chromium plated. Soft and stringy metals such as aluminum, brass, bronze, Monel, or free machining stainless steel should not be used in contact with moving seals. In static applications, O-ring contacting surfaces should have a maximum surface roughness of 64 to 125 μin. rms.

Table 2. Diametral Clearance and Groove Sizes for O-Ring Applications

ANSI/SAE AS568 Number	Tolerances: *A*	Tolerances: *B*	Diametral Clearance, *D*: Reciprocating & Static Seals	Diametral Clearance, *D*: Rotary Seals	Groove Width, *G*: Backup Rigs: None	Groove Width, *G*: Backup Rigs: One	Groove Width, *G*: Backup Rigs: Two	Bottom of Groove Radius, *R*
001	+0.001 −0.000	+0.000 −0.001	0.002 to 0.004	0.012 to 0.016	0.063			0.005 to 0.015
002					0.073			
003					0.083			
004 to 012					0.094	0.149	0.207	
013 to 050	+0.002 −0.000	+0.000 −0.002						
102 to 129			0.002 to 0.005		0.141	0.183	0.245	
130 to 178								
201 to 284			0.002 to 0.006	0.016 to 0.020	0.188	0.235	0.304	0.010 to 0.025
309 to 395			0.003 to 0.007		0.281	0.334	0.424	
425 to 475	+0.003 −0.000	+0.000 −0.003	0.004 to 0.010		0.375	0.475	0.579	0.020 to 0.035

Source: Auburn Manufacturing Co. All dimensions are in inches. Clearances listed are minimum and maximum values; standard groove widths may be reduced by about 10 per cent for use with ring compounds that free swell less than 15 per cent. Dimension *A* is the ID of any surface contacted by the outside circumference of the ring; *B* is the OD of any surface contacted by the inside circumference of the ring.

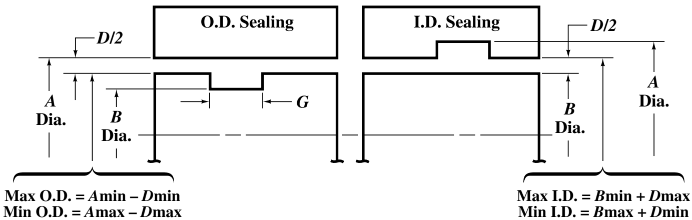


Fig. 5. Installation data for use with Table 2. Max and Min are maximum and minimum piston and bore diameters for O.D. and I.D., respectively.

The preferred bore materials are steel and cast iron, and pistons should be softer than the bore to avoid scratching them. The bore sections should be thick enough to resist expansion and contraction under pressure so that the radial clearance gap remains constant, reducing the chance of damage to the O-ring by extrusion and nibbling. Some compatibility problems may occur when O-rings are used with plastics parts because certain compounding ingredients may attack the plastics, causing crazing of the plastics surface.

O-rings are frequently used as driving belts in round bottom or V-grooves with light tension for low-power drive elements. Special compounds are available with high resistance to stress relaxation and fatigue for these applications. Best service is obtained in drive belt applications when the initial belt tension is between 80 and 200 psi and the initial installed stretch is between 8 and 25 per cent of the circumferential length. Most of the compounds used for drive belts operate best between 10 and 15 per cent stretch, although polyurethane has good service life when stretched as much as 20 to 25 per cent.

Table 3. Typical O-Ring Compounds

Nitrile	General-purpose compound for use with most petroleum oils, greases, gasoline, alcohols and glycols, LP gases, propane and butane fuels. Also for food service to resist vegetable and animal fats. Effective temperature range is about −40° to 250°F. Excellent compression set, tear and abrasion resistance, but poor resistance to ozone, sunlight and weather. Higher-temperature nitrile compounds with similar properties are also available.
Hydrogenated Nitrile	Similar to general-purpose nitrile compounds with improved high-temperature performance, resistance to aging, and petroleum product compatibility.
Polychloroprene (Neoprene)	General-purpose compound with low compression set and good resistance to elevated temperatures. Good resistance to sunlight, ozone, and weathering, and fair oil resistance. Frequently used for refrigerator gases such as Freon. Effective temperature range is about −40° to 250°F.
Ethylene Propylene	General-purpose compound with excellent resistance to polar fluids such as water, steam, ketones, and phosphate esters, and brake fluids, but not resistant to petroleum oils and solvents. Excellent resistance to ozone and flexing. Recommended for belt-drive applications. Continuous duty service in temperatures up to 250°F.
Silicon	Widest temperature range (−150° to 500°F) and best low-temperature flexibility of all elastomeric compounds. Not recommended for dynamic applications, due to low strength, or for use with most petroleum oils. Shrinkage characteristics similar to organic rubber, allowing existing molds to be used.
Polyurethane	Toughest of the elastomers used for O-rings, characterized by high tensile strength, excellent abrasion resistance, and tear strength. Compression set and heat resistance are inferior to nitrile. Suitable for hydraulic applications that anticipate abrasive contaminants and shock loads. Temperature service range of −65° to 212°F.
Fluorosilicone	Wide temperature range (−80° to 450°F) for continuous duty and excellent resistance to petroleum oils and fuels. Recommended for static applications only, due to limited strength and low abrasion resistance.
Polyacrylate	Heat resistance better than nitrile compounds, but inferior low temperature, compression set, and water resistance. Often used in power steering and transmission applications due to excellent resistance to oil, automatic transmission fluids, oxidation, and flex cracking. Temperature service range of −20° to 300°F.
Fluorocarbon (Viton)	General-purpose compound suitable for applications requiring resistance to aromatic or halogenated solvents or to high temperatures (−20° to 500°F with limited service to 600°F). Outstanding resistance to blended aromatic fuels, straight aromatics, and halogenated hydrocarbons and other petroleum products. Good resistance to strong acids (temperature range in acids (−20° to 250°F), but not effective for use with very hot water, steam, and brake fluids.

Ring Materials.—Thousands of O-ring compounds have been formulated for specific applications. Some of the most common types of compounds and their typical applications are given in Table 3. The Shore A durometer is the standard instrument used for measuring the hardness of elastomeric compounds. The softest O-rings are 50 and 60 Shore A and stretch more easily, exhibit lower breakout friction, seal better on rough surfaces, and need less clamping pressure than harder rings. For a given squeeze, the higher the durometer hardness of a ring, the greater the associated friction because a greater compressive force is exerted by hard rings than soft rings.

The most widely used rings are medium-hard O-rings with 70 Shore A hardness, which have the best wear resistance and frictional properties for running seals. Applications that involve oscillating or rotary motion frequently use 80 Shore A materials. Rings with a hardness above 85 Shore A often leak more because of less effective wiping action. These harder rings have a greater resistance to extrusion, but for small sizes may break easily during installation. O-ring hardness varies inversely with temperature, but when used for continuous service at high temperatures, the hardness may eventually increase after an initial softening of the compound.

O-ring compounds have thermal coefficients of expansion in the range of 7 to 20 times that of metal components, so shrinkage or expansion with temperature change can pose problems of leakage past the seal at low temperatures and excessive pressures at high temperatures when a ring is installed in a tight-fitting groove. Likewise, when an O-ring is immersed in a fluid, the compound usually absorbs some of the fluid and consequently increases in volume. Manufacturer's data give volumetric increase data for compounds completely immersed in various fluids. For confined rings (those with only a portion of the ring exposed to fluid), the size increase may be considerably lower than for rings completely immersed in fluid. Certain fluids can also cause ring shrinkage during “idle” periods, i.e., when the seal has a chance to dry out. If this shrinkage is more than 3 to 4 per cent, the seal may leak.

Excessive swelling due to fluid contact and high temperatures softens all compounds approximately 20 to 30 Shore A points from room temperature values and designs should anticipate the expected operating conditions. At low temperatures, swelling may be beneficial because fluid absorption may make the seal more flexible. However, the combination of low temperature and low pressure makes a seal particularly difficult to maintain. A soft compound should be used to provide a resilient seal at low temperatures. Below −65°F, only compounds formulated with silicone are useful; other compounds are simply too stiff, especially for use with air and other gases.

Compression set is another material property and a very important sealing factor. It is a measure of the shape memory of the material, that is, the ability to regain shape after being deformed. Compression set is a ratio, expressed as a percentage, of the unrecovered to original thickness of an O-ring compressed for a specified period of time between two heated plates and then released. O-rings with excessive compressive set will fail to maintain a good seal because, over time, the ring will be unable to exert the necessary compressive force (squeeze) on the enclosing walls. Swelling of the ring due to fluid contact tends to increase the squeeze and may partially compensate for the loss due to compression set. Generally, compression set varies by compound and ring cross-sectional diameter, and increases with the operating temperature.

ROLLED STEEL SECTIONS, WIRE AND SHEET-METAL GAGES

Rolled Steel Sections

Lengths of Angles Bent to Circular Shape.—To calculate the length of an angle-iron used either inside or outside of a tank or smokestack, the following table of constants may be used: Assume, for example, that a stand-pipe, 20 feet inside diameter, is provided with a 3 by 3 by $\frac{3}{8}$ inch angle-iron on the inside at the top. The circumference of a circle 20 feet in diameter is 754 inches. From the table of constants, find the constant for a 3 by 3 by $\frac{3}{8}$ inch angle-iron, which is 4.319. The length of the angle then is 754 − 4.319 = 749.681 inches. Should the angle be on the outside, add the constant instead of subtracting it; thus, 754 + 4.319 = 758.319 inches.

Size of Angle	Const.	Size of Angle	Const.	Size of Angle	Const.
$\frac{1}{4} \times 2 \times 2$	2.879	$\frac{5}{16} \times 3 \times 3$	4.123	$\frac{1}{2} \times 5 \times 5$	6.804
$\frac{5}{16} \times 2 \times 2$	3.076	$\frac{3}{8} \times 3 \times 3$	4.319	$\frac{3}{8} \times 6 \times 6$	7.461
$\frac{3}{8} \times 2 \times 2$	3.272	$\frac{1}{2} \times 3 \times 3$	4.711	$\frac{1}{2} \times 6 \times 6$	7.854
$\frac{1}{4} \times 2\frac{1}{2} \times 2\frac{1}{2}$	3.403	$\frac{3}{8} \times 3\frac{1}{2} \times 3\frac{1}{2}$	4.843	$\frac{3}{4} \times 6 \times 6$	8.639
$\frac{5}{16} \times 2\frac{1}{2} \times 2\frac{1}{2}$	3.600	$\frac{1}{2} \times 3\frac{1}{2} \times 3\frac{1}{2}$	5.235	$\frac{1}{2} \times 8 \times 8$	9.949
$\frac{3}{8} \times 2\frac{1}{2} \times 2\frac{1}{2}$	3.796	$\frac{3}{8} \times 4 \times 4$	5.366	$\frac{3}{4} \times 8 \times 8$	10.734
$\frac{1}{2} \times 2\frac{1}{2} \times 2\frac{1}{2}$	4.188	$\frac{1}{2} \times 4 \times 4$	5.758	$1 \times 8 \times 8$	11.520
$\frac{1}{4} \times 3 \times 3$	3.926	$\frac{3}{8} \times 5 \times 5$	6.414	…	…

Standard Designations of Rolled Steel Shapes.—Through a joint effort, the American Iron and Steel Institute (AISI) and the American Institute of Steel Construction (AISC) have changed most of the designations for their hot-rolled structural steel shapes. The present designations, standard for steel producing and fabricating industries, should be used when designing, detailing, and ordering steel. The accompanying table compares the present designations with the previous descriptions.

Hot-Rolled Structural Steel Shape Designations (AISI and AISC)

Present Designation	Type of Shape	Previous Designation
W 24 × 76	W shape	24 WF 76
W 14 × 26	W shape	14 B 26
S 24 × 100	S shape	24 I 100
M 8 × 18.5	M shape	8 M 18.5
M 10 × 9	M shape	10 JR 9.0
M 8 × 34.3	M shape	8 × 8 M 34.3
C 12 × 20.7	American Standard Channel	12 [20.7
MC 12 × 45	Miscellaneous Channel	12 × 4 [45.0
MC 12 × 10.6	Miscellaneous Channel	12 JR [10.6
HP 14 × 73	HP shape	14 BP 73
L 6 × 6 × $\frac{3}{4}$	Equal Leg Angle	∠ 6 × 6 × $\frac{3}{4}$
L 6 × 4 × $\frac{5}{8}$	Unequal Leg Angle	∠ 6 × 4 × $\frac{5}{8}$
WT 12 × 38	Structural Tee cut from W shape	ST 12 WF 38
WT 7 × 13	Structural Tee cut from W shape	ST 7 B 13
St 12 × 50	Structural Tee cut from S shape	ST 12 I 50
MT 4 × 9.25	Structural Tee cut from M shape	ST 4 M 9.25
MT 5 × 4.5	Structural Tee cut from M shape	ST 5 JR 4.5
MT 4 × 17.15	Structural Tee cut from M shape	ST 4 M 17.15
PL $\frac{1}{2}$ × 18	Plate	PL 18 × $\frac{1}{2}$
Bar 1	Square Bar	Bar 1
Bar 1 $\frac{1}{4}$∅	Round Bar	Bar 1$\frac{1}{4}$∅
Bar 2$\frac{1}{2}$ × $\frac{1}{2}$	Flat Bar	Bar 2$\frac{1}{2}$ × $\frac{1}{2}$
Pipe 4 Std.	Pipe	Pipe 4 Std.
Pipe 4 X-Strong	Pipe	Pipe 4 X-Strong
Pipe 4 XX-Strong	Pipe	Pipe 4 XX-Strong
TS 4 × 4 × .375	Structural Tubing: Square	Tube 4 × 4 × .375
TS 5 × 3 × .375	Structural Tubing: Rectangular	Tube 5 × 3 × .375
TS 3 OD × .250	Structural Tubing: Circular	Tube 3 OD × .250

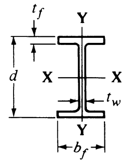
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Wide-Flange Sections—1

Wide-flange sections are designated, in order, by a section letter, nominal depth of the member in inches, and the nominal weight in pounds per foot; thus:

W 18 × 64

indicates a wide-flange section having a nominal depth of 18 inches, and a nominal weight per foot of 64 pounds. Actual geometry for each section can be obtained from the values below.



Designation	Area, A	Depth, d	Flange: Width, b_f	Flange: Thickness, t_f	Web Thickness, t_w	Axis X-X: I	Axis X-X: S	Axis X-X: r	Axis Y-Y: I	Axis Y-Y: S	Axis Y-Y: r
	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
[a]W 27 × 178	52.3	27.81	14.085	1.190	0.725	6990	502	11.6	555	78.8	3.26
× 161	47.4	27.59	14.020	1.080	0.660	6280	455	11.5	497	70.9	3.24
× 146	42.9	27.38	13.965	0.975	0.605	5630	411	11.4	443	63.5	3.21
× 114	33.5	27.29	10.070	0.930	0.570	4090	299	11.0	159	31.5	2.18
× 102	30.0	27.09	10.015	0.830	0.515	3620	267	11.0	139	27.8	2.15
× 94	27.7	26.92	9.990	0.745	0.490	3270	243	10.9	124	24.8	2.12
× 84	24.8	26.71	9.960	0.640	0.460	2850	213	10.7	106	21.2	2.07
W 24 × 162	47.7	25.00	12.955	1.220	0.705	5170	414	10.4	443	68.4	3.05
× 146	43.0	24.74	12.900	1.090	0.650	4580	371	10.3	391	60.5	3.01
× 131	38.5	24.48	12.855	0.960	0.605	4020	329	10.2	340	53.0	2.97
× 117	34.4	24.26	12.800	0.850	0.550	3540	291	10.1	297	46.5	2.94
× 104	30.6	24.06	12.750	0.750	0.500	3100	258	10.1	259	40.7	2.91
× 94	27.7	24.31	9.065	0.875	0.515	2700	222	9.87	109	24.0	1.98
× 84	24.7	24.10	9.020	0.770	0.470	2370	196	9.79	94.4	20.9	1.95
× 76	22.4	23.92	8.990	0.680	0.440	2100	176	9.69	82.5	18.4	1.92
× 68	20.1	23.73	8.965	0.585	0.415	1830	154	9.55	70.4	15.7	1.87
× 62	18.2	23.74	7.040	0.590	0.430	1550	131	9.23	34.5	9.80	1.38
× 55	16.2	23.57	7.005	0.505	0.395	1350	114	9.11	29.1	8.30	1.34
W 21 × 147	43.2	22.06	12.510	1.150	0.720	3630	329	9.17	376	60.1	2.95
× 132	38.8	21.83	12.440	1.035	0.650	3220	295	9.12	333	53.5	2.93
× 122	35.9	21.68	12.390	0.960	0.600	2960	273	9.09	305	49.2	2.92
× 111	32.7	21.51	12.340	0.875	0.550	2670	249	9.05	274	44.5	2.90
× 101	29.8	21.36	12.290	0.800	0.500	2420	227	9.02	248	40.3	2.89
× 93	27.3	21.62	8.420	0.930	0.580	2070	192	8.70	92.9	22.1	1.84
× 83	24.3	21.43	8.355	0.835	0.515	1830	171	8.67	81.4	19.5	1.83
× 73	21.5	21.24	8.295	0.740	0.455	1600	151	8.64	70.6	17.0	1.81
× 68	20.0	21.13	8.270	0.685	0.430	1480	140	8.60	64.7	15.7	1.80
× 62	18.3	20.99	8.240	0.615	0.400	1330	127	8.54	57.5	13.9	1.77
× 57	16.7	21.06	6.555	0.650	0.405	1170	111	8.36	30.6	9.35	1.35
× 50	14.7	20.83	6.530	0.535	0.380	984	94.5	8.18	24.9	7.64	1.30
× 44	13.0	20.66	6.500	0.450	0.350	843	81.6	8.06	20.7	6.36	1.26
W 18 × 119	35.1	18.97	11.265	1.060	0.655	2190	231	7.90	253	44.9	2.69
× 106	31.1	18.73	11.200	0.940	0.590	1910	204	7.84	220	39.4	2.66
× 97	28.5	18.59	11.145	0.870	0.535	1750	188	7.82	201	36.1	2.65
× 86	25.3	18.39	11.090	0.770	0.480	1530	166	7.77	175	31.6	2.63
× 76	22.3	18.21	11.035	0.680	0.425	1330	146	7.73	152	27.6	2.61
× 71	20.8	18.47	7.635	0.810	0.495	1170	127	7.50	60.3	15.8	1.70
× 65	19.1	18.35	7.590	0.750	0.450	1070	117	7.49	54.8	14.4	1.69
× 60	17.6	18.24	7.555	0.695	0.415	984	108	7.47	50.1	13.3	1.69
× 55	16.2	18.11	7.530	0.630	0.390	890	98.3	7.41	44.9	11.9	1.67
× 50	14.7	17.99	7.495	0.570	0.355	800	88.9	7.38	40.1	10.7	1.65
× 46	13.5	18.06	6.060	0.605	0.360	712	78.8	7.25	22.5	7.43	1.29
× 40	11.8	17.90	6.015	0.525	0.315	612	68.4	7.21	19.1	6.35	1.27
× 35	10.3	17.70	6.000	0.425	0.300	510	57.6	7.04	15.3	5.12	1.22

[a] Consult the AISC Manual, noted above, for W steel shapes having nominal depths greater than 27 in.

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration.

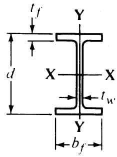
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Wide-Flange Sections–2

Wide-flange sections are designated, in order, by a section letter, nominal depth of the member in inches, and the nominal weight in pounds per foot; thus:

W 16 × 78

indicates a wide-flange section having a nominal depth of 16 inches, and a nominal weight per foot of 78 pounds. Actual geometry for each section can be obtained from the values below.



Designation	Area, A	Depth, d	Flange Width, b_f	Flange Thick-ness, t_f	Web Thick-ness, t_w	Axis X-X I	Axis X-X S	Axis X-X r	Axis Y-Y I	Axis Y-Y S	Axis Y-Y r
	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
W 16 × 100	29.4	16.97	10.425	0.985	0.585	1490	175	7.10	186	35.7	2.51
× 89	26.2	16.75	10.365	0.875	0.525	1300	155	7.05	163	31.4	2.49
× 77	22.6	16.52	10.295	0.760	0.455	1110	134	7.00	138	26.9	2.47
× 67	19.7	16.33	10.235	0.665	0.395	954	117	6.96	119	23.2	2.46
× 57	16.8	16.43	7.120	0.715	0.430	758	92.2	6.72	43.1	12.1	1.60
× 50	14.7	16.26	7.070	0.630	0.380	659	81.0	6.68	37.2	10.5	1.59
× 45	13.3	16.13	7.035	0.565	0.345	586	72.7	6.65	32.8	9.34	1.57
× 40	11.8	16.01	6.995	0.505	0.305	518	64.7	6.63	28.9	8.25	1.57
× 36	10.6	15.86	6.985	0.430	0.295	448	56.5	6.51	24.5	7.00	1.52
× 31	9.12	15.88	5.525	0.440	0.275	375	47.2	6.41	12.4	4.49	1.17
× 26	7.68	15.69	5.500	0.345	0.250	301	38.4	6.26	9.59	3.49	1.12
W 14 × 730	215.0	22.42	17.890	4.910	3.070	14300	1280	8.17	4720	527	4.69
× 665	196.0	21.64	17.650	4.520	2.830	12400	1150	7.98	4170	472	4.62
× 605	178.0	20.92	17.415	4.160	2.595	10800	1040	7.80	3680	423	4.55
× 550	162.0	20.24	17.200	3.820	2.380	9430	931	7.63	3250	378	4.49
× 500	147.0	19.60	17.010	3.500	2.190	8210	838	7.48	2880	339	4.43
× 455	134.0	19.02	16.835	3.210	2.015	7190	756	7.33	2560	304	4.38
× 426	125.0	18.67	16.695	3.035	1.875	6600	707	7.26	2360	283	4.34
× 398	117.0	18.29	16.590	2.845	1.770	6000	656	7.16	2170	262	4.31
× 370	109.0	17.92	16.475	2.660	1.655	5440	607	7.07	1990	241	4.27
× 342	101.0	17.54	16.360	2.470	1.540	4900	559	6.98	1810	221	4.24
× 311	91.4	17.12	16.230	2.260	1.410	4330	506	6.88	1610	199	4.20
× 283	83.3	16.74	16.110	2.070	1.290	3840	459	6.79	1440	179	4.17
× 257	75.6	16.38	15.995	1.890	1.175	3400	415	6.71	1290	161	4.13
× 233	68.5	16.04	15.890	1.720	1.070	3010	375	6.63	1150	145	4.10
× 211	62.0	15.72	15.800	1.560	0.980	2660	338	6.55	1030	130	4.07
× 193	56.8	15.48	15.710	1.440	0.890	2400	310	6.50	931	119	4.05
× 176	51.8	15.22	15.650	1.310	0.830	2140	281	6.43	838	107	4.02
× 159	46.7	14.98	15.565	1.190	0.745	1900	254	6.38	748	96.2	4.00
× 145	42.7	14.78	15.500	1.090	0.680	1710	232	6.33	677	87.3	3.98

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration.

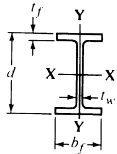
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Wide-Flange Sections—3

Wide-flange sections are designated, in order, by a section letter, nominal depth of the member in inches, and the nominal weight in pounds per foot; thus:

W 14 × 38

indicates a wide-flange section having a nominal depth of 14 inches, and a nominal weight per foot of 38 pounds. Actual geometry for each section can be obtained from the values below.



			Flange		Web	Axis X-X			Axis Y-Y		
	Area, A	Depth, d	Width, b_f	Thick-ness, t_f	Thick-ness, t_w	I	S	r	I	S	r
Designation	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
W 14 × 132	38.8	14.66	14.725	1.030	0.645	1530	209	6.28	548	74.5	3.76
× 120	35.3	14.48	14.670	0.940	0.590	1380	190	6.24	495	67.5	3.74
× 109	32.0	14.32	14.605	0.860	0.525	1240	173	6.22	447	61.2	3.73
× 99	29.1	14.16	14.565	0.780	0.485	1110	157	6.17	402	55.2	3.71
× 90	26.5	14.02	14.520	0.710	0.440	999	143	6.14	362	49.9	3.70
× 82	24.1	14.31	10.130	0.855	0.510	882	123	6.05	148	29.3	2.48
× 74	21.8	14.17	10.070	0.785	0.450	796	112	6.04	134	26.6	2.48
×68	20.0	14.04	10.035	0.720	0.415	723	103	6.01	121	24.2	2.46
× 61	17.9	13.89	9.995	0.645	0.375	640	92.2	5.98	107	21.5	2.45
× 53	15.6	13.92	8.060	0.660	0.370	541	77.8	5.89	57.7	14.3	1.92
× 48	14.1	13.79	8.030	0.595	0.340	485	70.3	5.85	51.4	12.8	1.91
× 43	12.6	13.66	7.995	0.530	0.305	428	62.7	5.82	45.2	11.3	1.89
× 38	11.2	14.10	6.770	0.515	0.310	385	54.6	5.87	26.7	7.88	1.55
× 34	10.0	13.98	6.745	0.455	0.285	340	48.6	5.83	23.3	6.91	1.53
× 30	8.85	13.84	6.730	0.385	0.270	291	42.0	5.73	19.6	5.82	1.49
× 26	7.69	13.91	5.025	0.420	0.255	245	35.3	5.65	8.91	3.54	1.08
× 22	6.49	13.74	5.000	0.335	0.230	199	29.0	5.54	7.00	2.80	1.04
W 12 × 336	98.8	16.82	13.385	2.955	1.775	4060	483	6.41	1190	177	3.47
× 305	89.6	16.32	13.235	2.705	1.625	3550	435	6.29	1050	159	3.42
× 279	81.9	15.85	13.140	2.470	1.530	3110	393	6.16	937	143	3.38
× 252	74.1	15.41	13.005	2.250	1.395	2720	353	6.06	828	127	3.34
× 230	67.7	15.05	12.895	2.070	1.285	2420	321	5.97	742	115	3.31
× 210	61.8	14.71	12.790	1.900	1.180	2140	292	5.89	664	104	3.28
× 190	55.8	14.38	12.670	1.735	1.060	1890	263	5.82	589	93.0	3.25
× 170	50.0	14.03	12.570	1.560	0.960	1650	235	5.74	517	82.3	3.22
× 152	44.7	13.71	12.480	1.400	0.870	1430	209	5.66	454	72.8	3.19
× 136	39.9	13.41	12.400	1.250	0.790	1240	186	5.58	398	64.2	3.16
× 120	35.3	13.12	12.320	1.105	0.710	1070	163	5.51	345	56.0	3.13
× 106	31.2	12.89	12.220	0.990	0.610	933	145	5.47	301	49.3	3.11
× 96	28.2	12.71	12.160	0.900	0.550	833	131	5.44	270	44.4	3.09
× 87	25.6	12.53	12.125	0.810	0.515	740	118	5.38	241	39.7	3.07
× 79	23.2	12.38	12.080	0.735	0.470	662	107	5.34	216	35.8	3.05
× 72	21.1	12.25	12.040	0.670	0.430	597	97.4	5.31	195	32.4	3.04
× 65	19.1	12.12	12.000	0.605	0.390	533	87.9	5.28	174	29.1	3.02
× 58	17.0	12.19	10.010	0.640	0.360	475	78.0	5.28	107	21.4	2.51
× 53	15.6	12.06	9.995	0.575	0.345	425	70.6	5.23	95.8	19.2	2.48
× 50	14.7	12.19	8.080	0.640	0.370	394	64.7	5.18	56.3	13.9	1.96
× 45	13.2	12.06	8.045	0.575	0.335	350	58.1	5.15	50.0	12.4	1.94
× 40	11.8	11.94	8.005	0.515	0.295	310	51.9	5.13	44.1	11.0	1.93
× 35	10.3	12.50	6.560	0.520	0.300	285	45.6	5.25	24.5	7.47	1.54
× 30	8.79	12.34	6.520	0.440	0.260	238	38.6	5.21	20.3	6.24	1.52
× 26	7.65	12.22	6.490	0.380	0.230	204	33.4	5.17	17.3	5.34	1.51
× 22	6.48	12.31	4.030	0.425	0.260	156	25.4	4.91	4.66	2.31	0.847
× 19	5.57	12.16	4.005	0.350	0.235	130	21.3	4.82	3.76	1.88	0.822
× 16	4.71	11.99	3.990	0.265	0.220	103	17.1	4.67	2.82	1.41	0.773
× 14	4.16	11.91	3.970	0.225	0.200	88.6	14.9	4.62	2.36	1.19	0.753

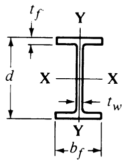
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Wide-Flange Sections—4

Wide-flange sections are designated, in order, by a section letter, nominal depth of the member in inches, and the nominal weight in pounds per foot; thus:

W 8 × 67

indicates a wide-flange section having a nominal depth of 8 inches, and a nominal weight per foot of 67 pounds. Actual geometry for each section can be obtained from the values below.



			Flange		Web	Axis X-X			Axis Y-Y		
Designation	Area, A	Depth, d	Width, b_f	Thickness, t_f	Thickness, t_w	I	S	r	I	S	r
	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
W 10 × 112	32.9	11.36	10.415	1.250	0.755	716	126	4.66	236	45.3	2.68
× 100	29.4	11.10	10.340	1.120	0.680	623	112	4.60	207	40.0	2.65
× 88	25.9	10.84	10.265	0.990	0.605	534	98.5	4.54	179	34.8	2.63
× 77	22.6	10.60	10.190	0.870	0.530	455	85.9	4.49	154	30.1	2.60
× 68	20.0	10.40	10.130	0.770	0.470	394	75.7	4.44	134	26.4	2.59
× 60	17.6	10.22	10.080	0.680	0.420	341	66.7	4.39	116	23.0	2.57
× 54	15.8	10.09	10.030	0.615	0.370	303	60.0	4.37	103	20.6	2.56
× 49	14.4	9.98	10.000	0.560	0.340	272	54.6	4.35	93.4	18.7	2.54
× 45	13.3	10.10	8.020	0.620	0.350	248	49.1	4.32	53.4	13.3	2.01
× 39	11.5	9.92	7.985	0.530	0.315	209	42.1	4.27	45.0	11.3	1.98
× 33	9.71	9.73	7.960	0.435	0.290	170	35.0	4.19	36.6	9.20	1.94
× 30	8.84	10.47	5.810	0.510	0.300	170	32.4	4.38	16.7	5.75	1.37
× 26	7.61	10.33	5.770	0.440	0.260	144	27.9	4.35	14.1	4.89	1.36
× 22	6.49	10.17	5.750	0.360	0.240	118	23.2	4.27	11.4	3.97	1.33
× 19	5.62	10.24	4.020	0.395	0.250	96.3	18.8	4.14	4.29	2.14	0.874
× 17	4.99	10.11	4.010	0.330	0.240	81.9	16.2	4.05	3.56	1.78	0.844
× 15	4.41	9.99	4.000	0.270	0.230	68.9	13.8	3.95	2.89	1.45	0.810
× 12	3.54	9.87	3.960	0.210	0.190	53.8	10.9	3.90	2.18	1.10	0.785
W 8 × 67	19.7	9.00	8.280	0.935	0.570	272	60.4	3.72	88.6	21.4	2.12
× 58	17.1	8.75	8.220	0.810	0.510	228	52.0	3.65	75.1	18.3	2.10
× 48	14.1	8.50	8.110	0.685	0.400	184	43.3	3.61	60.9	15.0	2.08
× 40	11.7	8.25	8.070	0.560	0.360	146	35.5	3.53	49.1	12.2	2.04
× 35	10.3	8.12	8.020	0.495	0.310	127	31.2	3.51	42.6	10.6	2.03
× 31	9.13	8.00	7.995	0.435	0.285	110	27.5	3.47	37.1	9.27	2.02
× 28	8.25	8.06	6.535	0.465	0.285	98.0	24.3	3.45	21.7	6.63	1.62
× 24	7.08	7.93	6.495	0.400	0.245	82.8	20.9	3.42	18.3	5.63	1.61
× 21	6.16	8.28	5.270	0.400	0.250	75.3	18.2	3.49	9.77	3.71	1.26
× 18	5.26	8.14	5.250	0.330	0.230	61.9	15.2	3.43	7.97	3.04	1.23
× 15	4.44	8.11	4.015	0.315	0.245	48.0	11.8	3.29	3.41	1.70	0.876
× 13	3.84	7.99	4.000	0.255	0.230	39.6	9.91	3.21	2.73	1.37	0.843
× 10	2.96	7.89	3.940	0.205	0.170	30.8	7.81	3.22	2.09	1.06	0.841
W 6 × 25	7.34	6.38	6.080	0.455	0.320	53.4	16.7	2.70	17.1	5.61	1.52
× 20	5.87	6.20	6.020	0.365	0.260	41.4	13.4	2.66	13.3	4.41	1.50
× 16	4.74	6.28	4.030	0.405	0.260	32.1	10.2	2.60	4.43	2.20	0.966
× 15	4.43	5.99	5.990	0.260	0.230	29.1	9.72	2.56	9.32	3.11	1.46
× 12	3.55	6.03	4.000	0.280	0.230	22.1	7.31	2.49	2.99	1.50	0.918
× 9	2.68	5.90	3.940	0.215	0.170	16.4	5.56	2.47	2.19	1.11	0.905
W 5 × 19	5.54	5.15	5.030	0.430	0.270	26.2	10.2	2.17	9.13	3.63	1.28
× 16	4.68	5.01	5.000	0.360	0.240	21.3	8.51	2.13	7.51	3.00	1.27
W 4 × 13	3.83	4.16	4.060	0.345	0.280	11.3	5.46	1.72	3.86	1.90	1.00

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration.

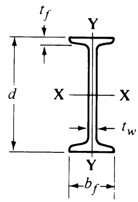
Data taken from the “Manual of Steel Construction,” 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel S Sections

"S" is the section symbol for "I" Beams. S shapes are designated, in order, by their section letter, actual depth in inches, and nominal weight in pounds per foot. Thus:

S 5 × 14.75

indicates an S shape (or I beam) having a depth of 5 inches and a nominal weight of 14.75 pounds per foot.



Designation	Area A	Depth, d	Flange Width, b_f	Flange Thickness, t_f	Web Thickness, t_w	Axis-X-X I	Axis-X-X S	Axis-X-X r	Axis Y-Y I	Axis Y-Y S	Axis Y-Y r
	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
S 24 × 121	35.6	24.50	8.050	1.090	0.800	3160	258	9.43	83.3	20.7	1.53
× 106	31.2	24.50	7.870	1.090	0.620	2940	240	9.71	77.1	19.6	1.57
×100	29.3	24.00	7.245	0.870	0.745	2390	199	9.02	47.7	13.2	1.27
× 90	26.5	24.00	7.125	0.870	0.625	2250	187	9.21	44.9	12.6	1.30
× 80	23.5	24.00	7.000	0.870	0.500	2100	175	9.47	42.2	12.1	1.34
S 20 × 96	28.2	20.30	7.200	0.920	0.800	1670	165	7.71	50.2	13.9	1.33
× 86	25.3	20.30	7.060	0.920	0.660	1580	155	7.89	46.8	13.3	1.36
× 75	22.0	20.00	6.385	0.795	0.635	1280	128	7.62	29.8	9.32	1.16
× 66	19.4	20.00	6.255	0.795	0.505	1190	119	7.83	27.7	8.85	1.19
S 18 × 70	20.6	18.00	6.251	0.691	0.711	926	103	6.71	24.1	7.72	1.08
× 54.7	16.1	18.00	6.001	0.691	0.461	804	89.4	7.07	20.8	6.94	1.14
S 15 × 50	14.7	15.00	5.640	0.622	0.550	486	64.8	5.75	15.7	5.57	1.03
× 42.9	12.6	15.00	5.501	0.622	0.411	447	59.6	5.95	14.4	5.23	1.07
S 12 × 50	14.7	12.00	5.477	0.659	0.687	305	50.8	4.55	15.7	5.74	1.03
× 40.8	12.0	12.00	5.252	0.659	0.462	272	45.4	4.77	13.6	5.16	1.06
× 35	10.3	12.00	5.078	0.544	0.428	229	38.2	4.72	9.87	3.89	0.980
× 31.8	9.35	12.00	5.000	0.544	0.350	218	36.4	4.83	9.36	3.74	1.00
S 10 × 35	10.3	10.00	4.944	0.491	0.594	147	29.4	3.78	8.36	3.38	0.901
× 25.4	7.46	10.00	4.661	0.491	0.311	124	24.7	4.07	6.79	2.91	0.954
S 8 × 23	6.77	8.00	4.171	0.426	0.441	64.9	16.2	3.10	4.31	2.07	0.798
× 18.4	5.41	8.00	4.001	0.426	0.271	57.6	14.4	3.26	3.73	1.86	0.831
S 7 × 20	5.88	7.00	3.860	0.392	0.450	42.4	12.1	2.69	3.17	1.64	0.734
× 15.3	4.50	7.00	3.662	0.392	0.252	36.7	10.5	2.86	2.64	1.44	0.766
S 6 × 17.25	5.07	6.00	3.565	0.359	0.465	26.3	8.77	2.28	2.31	1.30	0.675
× 12.5	3.67	6.00	3.332	0.359	0.232	22.1	7.37	2.45	1.82	1.09	0.705
S 5 × 14.75	4.34	5.00	3.284	0.326	0.494	15.2	6.09	1.87	1.67	1.01	0.620
× 10	2.94	5.00	3.004	0.326	0.214	12.3	4.92	2.05	1.22	0.809	0.643
S 4 × 9.5	2.79	4.00	2.796	0.293	0.326	6.79	3.39	1.56	0.903	0.646	0.569
× 7.7	2.26	4.00	2.663	0.293	0.193	6.08	3.04	1.64	0.764	0.574	0.581
S 3 × 7.5	2.21	3.00	2.509	0.260	0.349	2.93	1.95	1.15	0.586	0.468	0.516
× 5.7	1.67	3.00	2.330	0.260	0.170	2.52	1.68	1.23	0.455	0.390	0.522

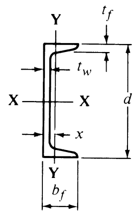
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

American Standard Steel Channels

American Standard Channels are designated, in order, by a section letter, actual depth in inches, and nominal weight per foot in pounds. Thus:

C 7 × 14.75

indicates an American Standard Channel with a depth of 7 inches and a nominal weight of 14.75 pounds per foot.



			Flange		Web	Axis X-X			Axis Y-Y			
	Area, A	Depth, d	Width, b_f	Thick-ness, t_f	Thick-ness, t_w	I	S	r	I	S	r	x
Designation	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.	in.
C 15 × 50	14.7	15.00	3.716	0.650	0.716	404	53.8	5.24	11.0	3.78	0.867	0.798
× 40	11.8	15.00	3.520	0.650	0.520	349	46.5	5.44	9.23	3.37	0.886	0.777
× 33.9	9.96	15.00	3.400	0.650	0.400	315	42.0	5.62	8.13	3.11	0904	0.787
C 12 × 30	8.82	12.00	3.170	0.501	0.510	162	27.0	4.29	5.14	2.06	0.763	0.674
× 25	7.35	12.00	3.047	0.501	0.387	144	24.1	4.43	4.47	1.88	0.780	0.674
× 20.7	6.09	12.00	2.942	0.501	0.282	129	21.5	4.61	3.88	1.73	0.799	0.698
C 10 × 30	8.82	10.00	3.033	0.436	0.673	103	20.7	3.42	3.94	1.65	0.669	0.649
× 25	7.35	10.00	2.886	0.436	0.526	91.2	18.2	3.52	3.36	1.48	0.676	0.617
× 20	5.88	10.00	2.739	0.436	0.379	78.9	15.8	3.66	2.81	1.32	0.692	0.606
× 15.3	4.49	10.00	2.600	0.436	0.240	67.4	13.5	3.87	2.28	1.16	0.713	0.634
C 9 × 20	5.88	9.00	2.648	0.413	0.448	60.9	13.5	3.22	2.42	1.17	0.642	0.583
× 15	4.41	9.00	2.485	0.413	0.285	51.0	11.3	3.40	1.93	1.01	0.661	0.586
× 13.4	3.94	9.00	2.433	0.413	0.233	47.9	10.6	3.48	1.76	0.962	0.669	0.601
C 8 × 18.75	5.51	8.00	2.527	0.390	0.487	44.0	11.0	2.82	1.98	1.01	0.599	0.565
× 13.75	4.04	8.00	2.343	0.390	0.303	36.1	9.03	2.99	1.53	0.854	0.615	0.553
× 11.5	3.38	8.00	2.260	0.390	0.220	32.6	8.14	3.11	1.32	0.781	0.625	0.571
C 7 × 14.75	4.33	7.00	2.299	0.366	0.419	27.2	7.78	2.51	1.38	0.779	0.564	0.532
× 12.25	3.60	7.00	2.194	0.366	0.314	24.2	6.93	2.60	1.17	0.703	0.571	0.525
× 9.8	2.87	7.00	2.090	0.366	0.210	21.3	6.08	2.72	0.968	0.625	0.581	0.540
C 6 × 13	3.83	6.00	2.157	0.343	0.437	17.4	5.80	2.13	1.05	0.642	0.525	0.514
× 10.5	3.09	6.00	2.034	0.343	0.314	15.2	5.06	2.22	0.866	0.564	0.529	0.499
× 8.2	2.40	6.00	1.920	0.343	0.200	13.1	4.38	2.34	0.693	0.492	0.537	0.511
C 5 × 9	2.64	5.00	1.885	0.320	0.325	8.90	3.56	1.83	0.632	0.450	0.489	0.478
× 6.7	1.97	5.00	1.750	0.320	0.190	7.49	3.00	1.95	0.479	0.378	0.493	0.484
C 4 × 7.25	2.13	4.00	1.721	0.296	0.321	4.59	2.29	1.47	0.433	0.343	0.450	0.459
× 5.4	1.59	4.00	1.584	0.296	0.184	3.85	1.93	1.56	0.319	0.283	0.449	0.457
C 3 × 6	1.76	3.00	1.596	0.273	0.356	2.07	1.38	1.08	0.305	0.268	0.416	0.455
× 5	1.47	3.00	1.498	0.273	0.258	1.85	1.24	1.12	0.247	0.233	0.410	0.438
× 4.1	1.21	3.00	1.410	0.273	0.170	1.66	1.10	1.17	0.197	0.202	0.404	0.436

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration; x = distance from center of gravity of section to outer face of structural shape.

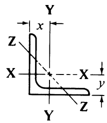
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Angles with Equal Legs

These angles are commonly designated by section symbol, width of each leg, and thickness, thus:

L 3 × 3 × 1/4

indicates a 3 × 3-inch angle of 1/4-inch thickness.



Size	Thickness	Weight per Foot	Area	Axis X-X & Y-Y			Z-Z
				I	*r*	*x* or *y*	r
in.	in.	lb.	in.^{2}	in.^{4}	in.	in.	in.
8 × 8	1 1/8	56.9	16.7	98.0	2.42	2.41	1.56
	1	51.0	15.0	89.0	2.44	2.37	1.56
	7/8	45.0	13.2	79.6	2.45	2.32	1.57
	3/4	38.9	11.4	69.7	2.47	2.28	1.58
	5/8	32.7	9.61	59.4	2.49	2.23	1.58
	9/16	29.6	8.68	54.1	2.50	2.21	1.59
	1/2	26.4	7.75	48.6	2.50	2.19	1.59
6 × 6	1	37.4	11.00	35.5	1.80	1.86	1.17
	7/8	33.1	9.73	31.9	1.81	1.82	1.17
	3/4	28.7	8.44	28.2	1.83	1.78	1.17
	5/8	24.2	7.11	24.2	1.84	1.73	1.18
	9/16	21.9	6.43	22.1	1.85	1.71	1.18
	1/2	19.6	5.75	19.9	1.86	1.68	1.18
	7/16	17.2	5.06	17.7	1.87	1.66	1.19
	3/8	14.9	4.36	15.4	1.88	1.64	1.19
	5/16	12.4	3.65	13.0	1.89	1.62	1.20
5 × 5	7/8	27.2	7.98	17.8	1.49	1.57	.973
	3/4	23.6	6.94	15.7	1.51	1.52	.975
	5/8	20.0	5.86	13.6	1.52	1.48	.978
	1/2	16.2	4.75	11.3	1.54	1.43	.983
	7/16	14.3	4.18	10.0	1.55	1.41	.986
	3/8	12.3	3.61	8.74	1.56	1.39	.990
	5/16	10.3	3.03	7.42	1.57	1.37	.994
4 × 4	3/4	18.5	5.44	7.67	1.19	1.27	.778
	5/8	15.7	4.61	6.66	1.20	1.23	.779
	1/2	12.8	3.75	5.56	1.22	1.18	.782
	7/16	11.3	3.31	4.97	1.23	1.16	.785
	3/8	9.8	2.86	4.36	1.23	1.14	.788
	5/16	8.2	2.40	3.71	1.24	1.12	.791
	1/4	6.6	1.94	3.04	1.25	1.09	.795
3 1/2 × 3 1/2	1/2	11.1	3.25	3.64	1.06	1.06	.683
	7/16	9.8	2.87	3.26	1.07	1.04	.684
	3/8	8.5	2.48	2.87	1.07	1.01	.687
	5/16	7.2	2.09	2.45	1.08	.990	.690
	1/4	5.8	1.69	2.01	1.09	.968	.694
3 × 3	1/2	9.4	2.75	2.22	.898	.932	.584
	7/16	8.3	2.43	1.99	.905	.910	.585
	3/8	7.2	2.11	1.76	.913	.888	.587
	5/16	6.1	1.78	1.51	.922	.865	.589
	1/4	4.9	1.44	1.24	.930	.842	.592
	3/16	3.71	1.09	.962	.939	.820	.596
2 1/2 × 2 1/2	1/2	7.7	2.25	1.23	.739	.806	.487
	3/8	5.9	1.73	.984	.753	.762	.487
	5/16	5.0	1.46	.849	.761	.740	.489
	1/4	4.1	1.19	.703	.769	.717	.491
	3/16	3.07	.902	.547	.778	.694	.495
2 × 2	3/8	4.7	1.36	.479	.594	.636	.389
	5/16	3.92	1.15	.416	.601	.614	.390
	1/4	3.19	.938	.348	.609	.592	.391
	3/16	2.44	.715	.272	.617	.569	.394
	1/8	1.65	.484	.190	.626	.546	.398

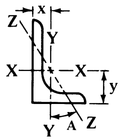
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Angles with Unequal Legs

These angles are commonly designated by section symbol, width of each leg, and thickness, thus:

L 7 × 4 × ½

indicates a 7 × 4-inch angle of ½-inch thickness.



Size	Thick-ness	Weight per Ft.	Area	Axis-X-X				Axis Y-Y			Axis Z-Z		
				I	*S*	*r*	*y*	*I*	*S*	*r*	*x*	*r*	Tan *A*
in.	in.	lb.	in.²	in.⁴	in.³	in.	in.	in.⁴	in.³	in.	in.	in.	
9 × 4	5/8	26.3	7.73	64.9	11.5	2.90	3.36	8.32	2.65	1.04	.858	.847	.216
	9/16	23.8	7.00	59.1	10.4	2.91	3.33	7.63	2.41	1.04	.834	.850	.218
	1/2	21.3	6.25	53.2	9.34	2.92	3.31	6.92	2.17	1.05	.810	.854	.220
8 × 6	1	44.2	13.0	80.8	15.1	2.49	2.65	38.8	8.92	1.73	1.65	1.28	.543
	7/8	39.1	11.5	72.3	13.4	2.51	2.61	34.9	7.94	1.74	1.61	1.28	.547
	3/4	33.8	9.94	63.4	11.7	2.53	2.56	30.7	6.92	1.76	1.56	1.29	.551
	5/8	28.5	8.36	54.1	9.87	2.54	2.52	26.3	5.88	1.77	1.52	1.29	.554
	9/16	7	7.56	49.3	8.95	2.55	2.50	24.0	5.34	1.78	1.50	1.30	.556
	1/2	23.0	6.75	44.3	8.02	2.56	2.47	21.7	4.79	1.79	1.47	1.30	.558
	7/16	20.2	5.93	39.2	7.07	2.57	2.45	19.3	4.23	1.80	1.45	1.31	.560
8 × 4	1	37.4	11.0	69.6	14.1	2.52	3.05	11.6	3.94	1.03	1.05	.846	.247
	3/4	28.7	8.44	54.9	10.9	2.55	2.95	9.36	3.07	1.05	.953	.852	.258
	9/16	21.9	6.43	42.8	8.35	2.58	2.88	7.43	2.38	1.07	.882	.861	.265
	1/2	19.6	5.75	38.5	7.49	2.59	2.86	6.74	2.15	1.08	.859	.865	.267
7 × 4	3/4	26.2	7.69	37.8	8.42	2.22	2.51	9.05	3.03	1.09	1.01	.860	.324
	5/8	22.1	6.48	32.4	7.14	2.24	2.46	7.84	2.58	1.10	.963	.865	.329
	1/2	17.9	5.25	26.7	5.81	2.25	2.42	6.53	2.12	1.11	.917	.872	.335
	3/8	13.6	3.98	20.6	4.44	2.27	2.37	5.10	1.63	1.13	.870	.880	.340
6 × 4	7/8	27.2	7.98	27.7	7.15	1.86	2.12	9.75	3.39	1.11	1.12	.857	.421
	3/4	23.6	6.94	24.5	6.25	1.88	2.08	8.68	2.97	1.12	1.08	.860	.428
	5/8	20.0	5.86	21.1	5.31	1.90	2.03	7.52	2.54	1.13	1.03	.864	.435
	9/16	18.1	5.31	19.3	4.83	1.90	2.01	6.91	2.31	1.14	1.01	.866	.438
	1/2	16.2	4.75	17.4	4.33	1.91	1.99	6.27	2.08	1.15	.987	.870	.440
	7/16	14.3	4.18	15.5	3.83	1.92	1.96	5.60	1.85	1.16	.964	.873	.443
	3/8	12.3	3.61	13.5	3.32	1.93	1.94	4.90	1.60	1.17	.941	.877	.446
	5/16	10.3	3.03	11.4	2.79	1.94	1.92	4.18	1.35	1.17	.918	.882	.448
6 × 3½	1/2	15.3	4.50	16.6	4.24	1.92	2.08	4.25	1.59	.972	.833	.759	.344
	3/8	11.7	3.42	12.9	3.24	1.94	2.04	3.34	1.23	.988	.787	.676	.350
	5/16	9.8	2.87	10.9	2.73	1.95	2.01	2.85	1.04	.996	.763	.772	.352
5 × 3½	3/4	19.8	5.81	13.9	4.28	1.55	1.75	5.55	2.22	.977	.996	.748	.464
	5/8	16.8	4.92	12.0	3.65	1.56	1.70	4.83	1.90	.991	.951	.751	.472
	1/2	13.6	4.00	9.99	2.99	1.58	1.66	4.05	1.56	1.01	.906	.755	.479
	7/16	12.0	3.53	8.90	2.64	1.59	1.63	3.63	1.39	1.01	.883	.758	.482
	3/8	10.4	3.05	7.78	2.29	1.60	1.61	3.18	1.21	1.02	.861	.762	.486
	5/16	8.7	2.56	6.60	1.94	1.61	1.59	2.72	1.02	1.03	.838	.766	.489
	1/4	7.0	2.06	5.39	1.57	1.62	1.56	2.23	.830	1.04	.814	.770	.492
5 × 3	5/8	15.7	4.61	11.4	3.55	1.57	1.80	3.06	1.39	.815	.796	.644	.349

Steel Angles with Unequal Legs

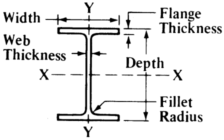
5 × 3	½	12.8	3.75	9.45	2.91	1.59	1.75	2.58	1.15	.829	.750	.648	.357
	7⁄16	11.3	3.31	8.43	2.58	1.60	1.73	2.32	1.02	.837	.727	.651	.361
	⅜	9.8	2.86	7.37	2.24	1.61	1.70	2.04	.888	.845	.704	.654	.364
	5⁄16	8.2	2.40	6.26	1.89	1.61	1.68	1.75	.753	.853	.681	.658	.368
	¼	6.6	1.94	5.11	1.53	1.62	1.66	1.44	.614	.861	.657	.663	.371
4 × 3½	⅝	14.7	4.30	6.37	2.35	1.22	1.29	4.52	1.84	1.03	1.04	.719	.745
	½	11.9	3.50	5.32	1.94	1.23	1.25	3.79	1.52	1.04	1.00	.722	.750
	7⁄16	10.6	3.09	4.76	1.72	1.24	1.23	3.40	1.35	1.05	.978	.724	.753
	⅜	9.1	2.67	4.18	1.49	1.25	1.21	2.95	1.17	1.06	.955	.727	.755
	5⁄16	7.7	2.25	3.56	1.26	1.26	1.18	2.55	.994	1.07	.932	.730	.757
	¼	6.2	1.81	2.91	1.03	1.27	1.16	2.09	.808	1.07	.909	.734	.759
4 × 3	⅝	13.6	3.98	6.03	2.30	1.23	1.37	2.87	1.35	.849	.871	.637	.534
	½	11.1	3.25	5.05	1.89	1.25	1.33	2.42	1.12	.864	.827	.639	.543
	7⁄16	9.8	2.87	4.52	1.68	1.25	1.30	2.18	.992	.871	.804	.641	.547
	⅜	8.5	2.48	3.96	1.46	1.26	1.28	1.92	.866	.879	.782	.644	.551
	5⁄16	7.2	2.09	3.38	1.23	1.27	1.26	1.65	.734	.887	.759	.647	.554
	¼	5.8	1.69	2.77	1.00	1.28	1.24	1.36	.599	.896	.736	.651	.558
3½ × 3	½	10.2	3.00	3.45	1.45	1.07	1.13	2.33	1.10	.881	.875	.621	.714
	7⁄16	9.1	2.65	3.10	1.29	1.08	1.10	2.09	.975	.889	.853	.622	.718
	⅜	7.9	2.30	2.72	1.13	1.09	1.08	1.85	.851	.897	.830	.625	.721
	5⁄16	6.6	1.93	2.33	.954	1.10	1.06	1.58	.722	.905	.808	.627	.724
	¼	5.4	1.56	1.91	.776	1.11	1.04	1.30	.589	.914	.785	.631	.727
3½ × 2½	½	9.4	2.75	3.24	1.41	1.09	1.20	1.36	.760	.704	.705	.534	.486
	7⁄16	8.3	2.43	2.91	1.26	1.09	1.18	1.23	.677	.711	.682	.535	.491
	⅜	7.2	2.11	2.56	1.09	1.10	1.16	1.09	.592	.719	.660	.537	.496
	5⁄16	6.1	1.78	2.19	.927	1.11	1.14	.939	.504	.727	.637	.540	.501
	¼	4.9	1.44	1.80	.755	1.12	1.11	.777	.412	.735	.614	.544	.506
3 × 2½	½	8.5	2.50	2.08	1.04	.913	1.00	1.30	.744	.722	.750	.520	.667
	7⁄16	7.6	2.21	1.88	.928	.920	.978	1.18	.664	.729	.728	.521	.672
	⅜	6.6	1.92	1.66	.810	.928	.956	1.04	.581	.736	.706	.522	.676
	5⁄16	5.6	1.62	1.42	.688	.937	.933	.898	.494	.744	.683	.525	.680
	¼	4.5	1.31	1.17	.561	.945	.911	.743	.404	.753	.661	.528	.684
	3⁄16	3.39	.996	.907	.430	.954	.888	.577	.310	.761	.638	.533	.688
3 × 2	½	7.7	2.25	1.92	1.00	.924	1.08	.672	.474	.546	.583	.428	.414
	7⁄16	6.8	2.00	1.73	.894	.932	1.06	.609	.424	.553	.561	.429	.421
	⅜	5.9	1.73	1.53	.781	.940	1.04	.543	.371	.559	.539	.430	.428
	5⁄16	5.0	1.46	1.32	.664	.948	1.02	.740	.317	.567	.516	.432	.435
	¼	4.1	1.19	1.09	.542	.957	.993	.392	.260	.574	.493	.435	.440
	3⁄16	3.07	.902	.842	.415	.966	.970	.307	.200	.583	.470	.439	.446
2½ × 2	⅜	5.3	1.55	.912	.547	.768	.831	.514	.363	.577	.581	.420	.614
	5⁄16	4.5	1.31	.788	.466	.776	.809	.446	.310	.584	.559	.422	.620
	¼	3.62	1.06	.654	.381	.784	.787	.372	.254	.592	.537	.424	.626
	3⁄16	2.75	.809	.509	.293	.793	.764	.291	.196	.600	.514	.427	.631

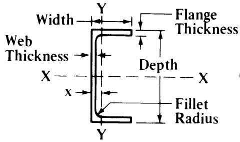
Symbols: I = moment of inertia; S = section modulus; r = radius of gyration; x = distance from center of gravity of section to outer face of structural shape.

Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Aluminum Association Standard Structural Shapes

I-BEAMS





CHANNELS

Depth	Width	Weight per Foot	Area	Flange Thickness	Web Thickness	Fillet Radius	Axis *X-X*			Axis *Y-Y*			
							I	*S*	*r*	*I*	*S*	*r*	*x*
in.	in.	lb.	$in.^2$	in.	in.	in.	$in.^4$	$in.^3$	in.	$in.^4$	$in.^3$	in.	in.
I-BEAMS													
3.00	2.50	1.637	1.392	0.20	0.13	0.25	2.24	1.49	1.27	0.52	0.42	0.61	…
3.00	2.50	2.030	1.726	0.26	0.15	0.25	2.71	1.81	1.25	0.68	0.54	0.63	…
4.00	3.00	2.311	1.965	0.23	0.15	0.25	5.62	2.81	1.69	1.04	0.69	0.73	…
4.00	3.00	2.793	2.375	0.29	0.17	0.25	6.71	3.36	1.68	1.31	0.87	0.74	…
5.00	3.50	3.700	3.146	0.32	0.19	0.30	13.94	5.58	2.11	2.29	1.31	0.85	…
6.00	4.00	4.030	3.427	0.29	0.19	0.30	21.99	7.33	2.53	3.10	1.55	0.95	…
6.00	4.00	4.692	3.990	0.35	0.21	0.30	25.50	8.50	2.53	3.74	1.87	0.97	…
7.00	4.50	5.800	4.932	0.38	0.23	0.30	42.89	12.25	2.95	5.78	2.57	1.08	…
8.00	5.00	6.181	5.256	0.35	0.23	0.30	59.69	14.92	3.37	7.30	2.92	1.18	…
8.00	5.00	7.023	5.972	0.41	0.25	0.30	67.78	16.94	3.37	8.55	3.42	1.20	…
9.00	5.50	8.361	7.110	0.44	0.27	0.30	102.02	22.67	3.79	12.22	4.44	1.31	…
10.00	6.00	8.646	7.352	0.41	0.25	0.40	132.09	26.42	4.24	14.78	4.93	1.42	…
10.00	6.00	10.286	8.747	0.50	0.29	0.40	155.79	31.16	4.22	18.03	6.01	1.44	…
12.00	7.00	11.672	9.925	0.47	0.29	0.40	255.57	42.60	5.07	26.90	7.69	1.65	…
12.00	7.00	14.292	12.153	0.62	0.31	0.40	317.33	52.89	5.11	35.48	10.14	1.71	…
CHANNELS													
2.00	1.00	0.577	0.491	0.13	0.13	0.10	0.288	0.288	0.766	0.045	0.064	0.303	0.298
2.00	1.25	1.071	0.911	0.26	0.17	0.15	0.546	0.546	0.774	0.139	0.178	0.391	0.471
3.00	1.50	1.135	0.965	0.20	0.13	0.25	1.41	0.94	1.21	0.22	0.22	0.47	0.49
3.00	1.75	1.597	1.358	0.26	0.17	0.25	1.97	1.31	1.20	0.42	0.37	0.55	0.62
4.00	2.00	1.738	1.478	0.23	0.15	0.25	3.91	1.95	1.63	0.60	0.45	0.64	0.65
4.00	2.25	2.331	1.982	0.29	0.19	0.25	5.21	2.60	1.62	1.02	0.69	0.72	0.78
5.00	2.25	2.212	1.881	0.26	0.15	0.30	7.88	3.15	2.05	0.98	0.64	0.72	0.73
5.00	2.75	3.089	2.627	0.32	0.19	0.30	11.14	4.45	2.06	2.05	1.14	0.88	0.95
6.00	2.50	2.834	2.410	0.29	0.17	0.30	14.35	4.78	2.44	1.53	0.90	0.80	0.79
6.00	3.25	4.030	3.427	0.35	0.21	0.30	21.04	7.01	2.48	3.76	1.76	1.05	1.12
7.00	2.75	3.205	2.725	0.29	0.17	0.30	22.09	6.31	2.85	2.10	1.10	0.88	0.84
7.00	3.50	4.715	4.009	0.38	0.21	0.30	33.79	9.65	2.90	5.13	2.23	1.13	1.20
8.00	3.00	4.147	3.526	0.35	0.19	0.30	37.40	9.35	3.26	3.25	1.57	0.96	0.93
8.00	3.75	5.789	4.923	0.471	0.25	0.35	52.69	13.17	3.27	7.13	2.82	1.20	1.22
9.00	3.25	4.983	4.237	0.35	0.23	0.35	54.41	12.09	3.58	4.40	1.89	1.02	0.93
9.00	4.00	6.970	5.927	0.44	0.29	0.35	78.31	17.40	3.63	9.61	3.49	1.27	1.25
10.00	3.50	6.136	5.218	0.41	0.25	0.35	83.22	16.64	3.99	6.33	2.56	1.10	1.02
10.00	4.25	8.360	7.109	0.50	0.31	0.40	116.15	23.23	4.04	13.02	4.47	1.35	1.34
12.00	4.00	8.274	7.036	0.47	0.29	0.40	159.76	26.63	4.77	11.03	3.86	1.25	1.14
12.00	5.00	11.822	10.053	0.62	0.35	0.45	239.69	39.95	4.88	25.74	7.60	1.60	1.61

Structural sections are available in 6061-T6 aluminum alloy. Data supplied by The Aluminum Association.

Wire and Sheet-metal Gages

The thicknesses of sheet metals and the diameters of wires conform to various gaging systems. These gage sizes are indicated by numbers, and the following tables give the decimal equivalents of the different gage numbers. Much confusion has resulted from the use of gage numbers, and in ordering materials it is preferable to give the exact dimensions in decimal fractions of an inch. While the dimensions thus specified should conform to the gage ordinarily used for a given class of material, any error in the specification due, for example, to the use of a table having “rounded off” or approximate equivalents, will be apparent to the manufacturer at the time the order is placed. Furthermore, the decimal method of indicating wire diameters and sheet metal thicknesses has the advantage of being self-explanatory, whereas arbitrary gage numbers are not. The decimal system of indicating gage sizes is now being used quite generally, and gage numbers are gradually being discarded. Unfortunately, there is considerable variation in the use of different gages. For example, a gage ordinarily used for copper, brass and other non-ferrous materials, may at times be used for steel, and vice versa. The gages specified in the following are the ones ordinarily employed for the materials mentioned, but there are some minor exceptions and variations in the different industries.

Wire Gages.—The wire gage system used by practically all of the steel producers in the United States is known by the name Steel Wire Gage or to distinguish it from the Standard Wire Gage (S.W.G.) used in Great Britain it is called the United States Steel Wire Gage. It is the same as the Washburn and Moen, American Steel and Wire Company, and Roebling Wire Gages. The name has the official sanction of the Bureau of Standards at Washington but is not legally effective. The only wire gage which has been recognized in Acts of Congress is the Birmingham Gage (also known as Stub's Iron Wire). The Birmingham Gage is, however, nearly obsolete in both the United States and Great Britain, where it originated. Copper and aluminum wires are specified in decimal fractions. They were formerly universally specified in the United States by the American or Brown & Sharpe Wire Gage. Music spring steel wire, one of the highest quality wires of several types used for mechanical springs, is specified by the piano or music wire gage.

In Great Britain one wire gage has been legalized. This is called the Standard Wire Gage (S.W.G.), formerly called Imperial Wire Gage.

Gages for Rods.—Steel wire rod sizes are designated by fractional or decimal parts of an inch and by the gage numbers of the United States Steel Wire Gage. Copper and aluminum rods are specified by decimal fractions and fractions. Drill rod may be specified in decimal fractions but in the carbon and alloy tool steel grades may also be specified in the Stub's Steel Wire Gage and in the high-speed steel drill rod grade may be specified by the Morse Twist Drill Gage (Manufacturers' Standard Gage for Twist Drills). For gage numbers with corresponding decimal equivalents see the tables of American Standard Straight Shank Twist Drills.

Gages for Wall Thicknesses of Tubing.—At one time the Birmingham or Stub's Iron Wire Gage was used to specify the wall thickness of the following classes of tubing: seamless brass, seamless copper, seamless steel, and aluminum. The Brown & Sharpe Wire Gage was used for brazed brass and brazed copper tubing. Wall thicknesses are now specified by decimal parts of an inch but the wall thickness of steel pressure tubes and steel mechanical tubing may be specified by the Birmingham or Stub's Iron Wire Gage. In Great Britain the Standard Wire Gage (S.W.G.) is used to specify the wall thickness of some kinds of steel tubes.

Table 1. Wire Gages in Approximate Decimals of an Inch

No. of Wire Gage	American Wire or Brown & Sharpe Gage	Steel Wire Gage (U.S.)[a]	British Standard Wire Gage (Imperial Wire Gage)	Music or Piano Wire Gage	Birmingham or Stub's Iron Wire Gage	Stub's Steel Wire Gage	No. of Wire Gage	Stub's Steel Wire Gage
7/0	…	0.4900	0.5000	…	…	…	51	0.066
6/0	0.5800	0.4615	0.4640	0.004	…	…	52	0.063
5/0	0.5165	0.4305	0.4320	0.005	0.5000	…	53	0.058
4/0	0.4600	0.3938	0.4000	0.006	0.4540	…	54	0.055
3/0	0.4096	0.3625	0.3720	0.007	0.4250	…	55	0.050
2/0	0.3648	0.3310	0.3480	0.008	0.3800	…	56	0.045
1/0	0.3249	0.3065	0.3240	0.009	0.3400	…	57	0.042
1	0.2893	0.2830	0.3000	0.010	0.3000	0.227	58	0.041
2	0.2576	0.2625	0.2760	0.011	0.2840	0.219	59	0.040
3	0.2294	0.2437	0.2520	0.012	0.2590	0.212	60	0.039
4	0.2043	0.2253	0.2320	0.013	0.2380	0.207	61	0.038
5	0.1819	0.2070	0.2120	0.014	0.2200	0.204	62	0.037
6	0.1620	0.1920	0.1920	0.016	0.2030	0.201	63	0.036
7	0.1443	0.1770	0.1760	0.018	0.1800	0.199	64	0.035
8	0.1285	0.1620	0.1600	0.020	0.1650	0.197	65	0.033
9	0.1144	0.1483	0.1440	0.022	0.1480	0.194	66	0.032
10	0.1019	0.1350	0.1280	0.024	0.1340	0.191	67	0.031
11	0.0907	0.1205	0.1160	0.026	0.1200	0.188	68	0.030
12	0.0808	0.1055	0.1040	0.029	0.1090	0.185	69	0.029
13	0.0720	0.0915	0.0920	0.031	0.0950	0.182	70	0.027
14	0.0641	0.0800	0.0800	0.033	0.0830	0.180	71	0.026
15	0.0571	0.0720	0.0720	0.035	0.0720	0.178	72	0.024
16	0.0508	0.0625	0.0640	0.037	0.0650	0.175	73	0.023
17	0.0453	0.0540	0.0560	0.039	0.0580	0.172	74	0.022
18	0.0403	0.0475	0.0480	0.041	0.0490	0.168	75	0.020
19	0.0359	0.0410	0.0400	0.043	0.0420	0.164	76	0.018
20	0.0320	0.0348	0.0360	0.045	0.0350	0.161	77	0.016
21	0.0285	0.0318	0.0320	0.047	0.0320	0.157	78	0.015
22	0.0253	0.0286	0.0280	0.049	0.0280	0.155	79	0.014
23	0.0226	0.0258	0.0240	0.051	0.0250	0.153	80	0.013
24	0.0201	0.0230	0.0220	0.055	0.0220	0.151	…	…
25	0.0179	0.0204	0.0200	0.059	0.0200	0.148	…	…
26	0.0159	0.0181	0.0180	0.063	0.0180	0.146	…	…
27	0.0142	0.0173	0.0164	0.067	0.0160	0.143	…	…
28	0.0126	0.0162	0.0149	0.071	0.0140	0.139	…	…
29	0.0113	0.0150	0.0136	0.075	0.0130	0.134	…	…
30	0.0100	0.0140	0.0124	0.080	0.0120	0.127	…	…
31	0.00893	0.0132	0.0116	0.085	0.0100	0.120	…	…
32	0.00795	0.0128	0.0108	0.090	0.0090	0.115	…	…
33	0.00708	0.0118	0.0100	0.095	0.0080	0.112	…	…
34	0.00630	0.0104	0.0092	0.100	0.0070	0.110	…	…
35	0.00561	0.0095	0.0084	0.106	0.0050	0.108	…	…
36	0.00500	0.0090	0.0076	0.112	0.0040	0.106	…	…
37	0.00445	0.0085	0.0068	0.118	…	0.103	…	…
38	0.00396	0.0080	0.0060	0.124	…	0.101	…	…
39	0.00353	0.0075	0.0052	0.130	…	0.099	…	…
40	0.00314	0.0070	0.0048	0.138	…	0.097	…	…
41	0.00280	0.0066	0.0044	0.146	…	0.095	…	…
42	0.00249	0.0062	0.0040	0.154	…	0.092	…	…
43	0.00222	0.0060	0.0036	0.162	…	0.088	…	…
44	0.00198	0.0058	0.0032	0.170	…	0.085	…	…
45	0.00176	0.0055	0.0028	0.180	…	0.081	…	…
46	0.00157	0.0052	0.0024	…	…	0.079	…	…
47	0.00140	0.0050	0.0020	…	…	0.077	…	…
48	0.00124	0.0048	0.0016	…	…	0.075	…	…
49	0.00111	0.0046	0.0012	…	…	0.072	…	…
50	0.00099	0.0044	0.0010	…	…	0.069	…	…

[a] Also known as Washburn and Moen, American Steel and Wire Co. and Roebling Wire Gages. A greater selection of sizes is available and is specified by what are known as split gage numbers. They can be recognized by ½ fractions which follow the gage number; i.e., 4½. The decimal equivalents of split gage numbers are in the Steel Products Manual entitled: *Wire and Rods, Carbon Steel* published by the American Iron and Steel Institute, Washington, DC.

Sheet-Metal Gages.—Thicknesses of steel sheets are based upon a weight of 41.82 pounds per square foot per inch of thickness, which is known as the Manufacturers' Standard Gage for Sheet Steel. This gage differs from the older United States Standard Gage for iron and steel sheets and plates, established by Congress in 1893, based upon a weight of 40 pounds per square foot per inch of thickness which is the weight of wrought-iron plate.

Thicknesses of aluminum, copper, and copper-base alloys were formerly designated by the American or Brown & Sharpe Wire Gage but now are specified in decimals or fractions of an inch. American National Standard B32.1-1952 (R1988) entitled Preferred Thicknesses for Uncoated Thin Flat Metals (see accompanying Table 2) gives thicknesses that are based on the 20- and 40-series of preferred numbers in American National Standard Preferred Numbers — ANSI Z17.1 (see Handbook page 19) and are applicable to uncoated, thin, flat metals and alloys. Each number of the 20-series is approximately 12 percent greater than the next smaller one and each number of the 40-series is approximately 6 percent greater than the next smaller one.

Table 2. Preferred Thicknesses for Uncoated Metals and Alloys—Under 0.250 Inch in Thickness *ANSI B32.1-1952 (R1994)*

Preferred Thickness, Inches							
Based on 20-Series	Based on 40-Series	Based on 20-Series	Based on 40-Series	Based on 20-Series	Based on 40-Series	Based on 20-Series	Based on 40-Series
…	0.236	0.100	0.100	…	0.042	0.018	0.018
0.224	0.224	…	0.095	0.040	0.040	…	0.017
…	0.212	0.090	0.090	…	0.038	0.016	0.016
0.200	0.200	…	0.085	0.036	0.036	…	0.015
…	0.190	0.080	0.080	…	0.034	0.014	0.014
0.180	0.180	…	0.075	0.032	0.032	…	0.013
…	0.170	0.071	0.071	…	0.030	0.012	0.012
0.160	0.160	…	0.067	0.028	0.028	0.011	0.011
…	0.150	0.063	0.063	…	0.026	0.010	0.010
0.140	0.140	…	0.060	0.025	0.025	0.009	0.009
…	0.132	0.056	0.056	…	0.024	0.008	0.008
0.125	0.125	…	0.053	0.022	0.022	0.007	0.007
…	0.118	0.050	0.050	…	0.021	0.006	0.006
0.112	0.112	…	0.048	0.020	0.020	0.005	0.005
…	0.106	0.045	0.045	…	0.019	0.004	0.004

The American National Standard ANSI B32.1-1952 (R1994) lists preferred thicknesses that are based on the 20- and 40-series of preferred numbers and states that those based on the 40-series should provide adequate coverage. However, where intermediate thicknesses are required, the Standard recommends that thicknesses be based on the 80-series of preferred numbers (see Handbook page 19).

Thicknesses for copper and copper-base alloy flat products below ¼ inch thick are specified by the 20-series of American National Standard Preferred Numbers given in ANSI B32.1. Although the table in ANSI B32.1 gives only the 20- and 40-series of numbers, it states that when intermediate thicknesses are required they should be selected from thicknesses based on the 80-series of numbers (see Handbook page 19).

Zinc sheets are usually ordered by specifying decimal thickness although a zinc gage exists and is shown in Table 3.

Table 3. Sheet-Metal Gages in Approximate Decimals of an Inch

Gage No.	Steel Gage	B.G.[a]	Galvanized Sheet	Zinc Gage	Gage No.	Steel Gage	B.G.[a]	Galvanized Sheet	Zinc Gage
15/0	…	1.000	…	…	20	0.0359	0.0392	0.0396	0.070
14/0	…	0.9583	…	…	21	0.0329	0.0349	0.0366	0.080
13/0	…	0.9167	…	…	22	0.0299	0.03125	0.0336	0.090
12/0	…	0.8750	…	…	23	0.0269	0.02782	0.0306	0.100
11/0	…	0.8333	…	…	24	0.0239	0.02476	0.0276	0.125
10/0	…	0.7917	…	…	25	0.0209	0.02204	0.0247	…
9/0	…	0.7500	…	…	26	0.0179	0.01961	0.0217	…
8/0	…	0.7083	…	…	27	0.0164	0.01745	0.0202	…
7/0	…	0.6666	…	…	28	0.0149	0.01562	0.0187	…
6/0	…	0.6250	…	…	29	0.0135	0.01390	0.0172	…
5/0	…	0.5883	…	…	30	0.0120	0.01230	0.0157	…
4/0	…	0.5416	…	…	31	0.0105	0.01100	0.0142	…
3/0	…	0.5000	…	…	32	0.0097	0.00980	0.0134	…
2/0	…	0.4452	…	…	33	0.0090	0.00870	…	…
1/0	…	0.3964	…	…	34	0.0082	0.00770	…	…
1	…	0.3532	…	…	35	0.0075	0.00690	…	…
2	…	0.3147	…	…	36	0.0067	0.00610	…	…
3	0.2391	0.2804	…	0.006	37	0.0064	0.00540	…	…
4	0.2242	0.2500	…	0.008	38	0.0060	0.00480	…	…
5	0.2092	0.2225	…	0.010	39	…	0.00430	…	…
6	0.1943	0.1981	…	0.012	40	…	0.00386	…	…
7	0.1793	0.1764	…	0.014	41	…	0.00343	…	…
8	0.1644	0.1570	0.1681	0.016	42	…	0.00306	…	…
9	0.1495	0.1398	0.1532	0.018	43	…	0.00272	…	…
10	0.1345	0.1250	0.1382	0.020	44	…	0.00242	…	…
11	0.1196	0.1113	0.1233	0.024	45	…	0.00215	…	…
12	0.1046	0.0991	0.1084	0.028	46	…	0.00192	…	…
13	.0897	0.0882	0.0934	0.032	47	…	0.00170	…	…
14	0.0747	0.0785	0.0785	0.036	48	…	0.00152	…	…
15	0.0673	0.0699	0.0710	0.040	49	…	0.00135	…	…
16	0.0598	0.0625	0.0635	0.045	50	…	0.00120	…	…
17	0.0538	0.0556	0.0575	0.050	51	…	0.00107	…	…
18	.0478	0.0495	0.0516	0.055	52	…	0.00095	…	…
19	0.0418	0.0440	0.0456	0.060	…	…	…	…	…

[a] B.G. is the Birmingham Gage for sheets and hoops.

The United States Standard Gage (not shown above) for iron and steel sheets and plates was established by Congress in 1893 and was primarily a *weight* gage rather than a thickness gage. The equivalent thicknesses were derived from the weight of wrought iron. The weight per cubic foot was taken at 480 pounds, thus making the weight of a plate 12 inches square and 1 inch thick, 40 pounds. In converting weight to equivalent thickness, gage tables formerly published contained thicknesses equivalent to the basic weights just mentioned. For example, a No. 3 U.S. gage represents a wrought-iron plate having a weight of 10 pounds per square foot; hence, if the weight per square foot per inch thick is 40 pounds, the plate thickness for a No. 3 gage = 10 ÷ 40 = 0.25 inch, which was the original thickness equivalent for this gage number. Because this and the other thickness equivalents were derived from the weight of wrought iron, they are not correct for steel.

Most sheet-metal products in Great Britain are specified by the British Standard Wire Gage (Imperial Wire Gage). Black iron and steel sheet and hooping, and galvanized flat and corrugated steel sheet, however, are specified by the Birmingham Gage (B.G.), which was legalized in 1914. This Birmingham Gage should not be confused with the Birmingham or Stub's Iron Wire Gage mentioned previously.

Metric Sizes for Flat Metal Products.— American National Standard B32.3 M-1984 establishes a preferred series of metric thicknesses, widths, and lengths for flat metal products of rectangular cross section; the thickness and width values are also applicable to base metals that may be coated in later operations. Table 4 lists the preferred thicknesses. Whenever possible, the Preferred Thickness values should be used, with the Second or Third Preference chosen only if no suitable Preferred size is available. Since not all metals and grades are produced in each of the sizes given in Table 4, producers or distributors should be consulted to determine a particular product and size combination's availability.

Table 4. Preferred Metric Thicknesses for All Flat Metal Products
ANSI/ASME B32.3M-1984

Preferred Thickness	Second Preference	Third Preference	Preferred Thickness	Second Preference	Third Preference
0.050	…	…	…	…	3.6
0.060	…	…	…	3.8	…
0.080	…	…	4.0	…	…
0.10	…	…	…	4.2	…
0.12	…	…	…	4.5	…
…	0.14	…	…	4.8	…
0.16	…	…	5.0	…	…
…	0.18	…	…	5.5	…
0.20	…	…	6.0	…	…
…	0.22	…	…	…	6.5
0.25	…	…	…	7.0	…
…	0.28	…	…	…	7.5
0.30	…	…	8.0	…	…
…	0.35	…	…	9.0	…
0.40	…	…	10	…	…
…	0.45	…	…	11	…
0.50	…	…	12	…	…
…	0.55	…	…	14	…
0.60	…	…	16	…	…
…	0.65	…	…	18	…
…	0.70	…	20	…	…
…	…	0.75	…	22	…
0.80	…	…	25	…	…
…	…	0.85	…	28	…
…	0.90	…	30	…	…
…	…	0.95	…	32	…
1.0	…	…	35	…	…
…	…	1.05	…	38	…
…	1.1	…	40	…	…
1.2	…	…	…	45	…
…	…	1.3	50	…	…
…	1.4	…	…	55	…
…	…	1.5	60	…	…
1.6	…	…	…	70	…
…	…	1.7	80	…	…
…	1.8	…	…	90	…
…	…	1.9	100	…	…
2.0	…	…	…	110	…
…	…	2.1	120	…	…
…	2.2	…	…	130	…
…	…	2.4	140	…	…
2.5	…	…	…	150	…
…	…	2.6	160	…	…
…	2.8	…	180	…	…
3.0	…	…	200	…	…
…	3.2	…	250	…	…
…	…	3.4	300	…	…
3.5	…	…			

All dimensions are in millimeters.

PIPE AND PIPE FITTINGS

Wrought Steel Pipe.—ANSI/ASME B36.10M-1995 covers dimensions of welded and seamless wrought steel pipe, for high or low temperatures or pressures.

The word *pipe* as distinguished from *tube* is used to apply to tubular products of dimensions commonly used for pipelines and piping systems. Pipe dimensions of sizes 12 inches and smaller have outside diameters numerically larger than the corresponding nominal sizes whereas outside diameters of tubes are identical to nominal sizes.

Size: The size of all pipe is identified by the nominal pipe size. The manufacture of pipe in the nominal sizes of ⅛ inch to 12 inches, inclusive, is based on a standardized outside diameter (OD). This OD was originally selected so that pipe with a standard OD and having a wall thickness which was typical of the period would have an inside diameter (ID) approximately equal to the nominal size. Although there is now no such relation between the existing standard thicknesses, ODs and nominal sizes, these nominal sizes and standard ODs continue in use as "standard."

The manufacture of pipe in nominal sizes of 14-inch OD and larger proceeds on the basis of an OD corresponding to the nominal size.

Weight: The nominal weights of steel pipe are calculated values and are tabulated in Table 1. They are based on the following formula:

$$W_{pe} = 10.68(D - t)t$$

where W_{pe} = nominal plain end weight to the nearest 0.01 lb/ft.
D = outside diameter to the nearest 0.001 inch
t = specified wall thickness rounded to the nearest 0.001 inch

Wall thickness: The nominal wall thicknesses are given in Table 1 which also indicates the wall thicknesses in API Standard 5L.

The wall thickness designations "Standard," "Extra-Strong," and "Double Extra-Strong" have been commercially used designations for many years. The Schedule Numbers were subsequently added as a convenient designation for use in ordering pipe. "Standard" and Schedule 40 are identical for nominal pipe sizes up to 10 inches, inclusive. All larger sizes of "Standard" have ⅜-inch wall thickness. "Extra-Strong" and Schedule 80 are identical for nominal pipe sizes up to 8 inch, inclusive. All larger sizes of "Extra-Strong" have ½-inch-wall thickness.

Wall Thickness Selection: When the selection of wall thickness depends primarily on capacity to resist internal pressure under given conditions, the designer shall compute the exact value of wall thickness suitable for conditions for which the pipe is required as prescribed in the "ASME Boiler and Pressure Vessel Code," "ANSI B31 Code for Pressure Piping," or other similar codes, whichever governs the construction. A thickness can then be selected from Table 1 to suit the value computed to fulfill the conditions for which the pipe is desired.

Metric Weights and Mass: Standard SI metric dimensions in millimeters for outside diameters and wall thicknesses may be found by multiplying the inch dimensions by 25.4. Outside diameters converted from those shown in Table 1 should be rounded to the nearest 0.1 mm and wall thicknesses to the nearest 0.01 mm.

The following formula may be used to calculate the SI metric plain end mass in kg/m using the converted metric diameters and thicknesses:

$$W_{pe} = 0.02466(D - t)t$$

where W_{pe} = nominal plain end mass rounded to the nearest 0.01 kg/m.
D = outside diameter to the nearest 0.1 mm for sizes shown in Table 1.
t = specified wall thickness rounded to the nearest 0.01 mm.

Table 1. American National Standard Weights and Dimensions of Welded and Seamless Wrought Steel Pipe *ANSI/ASME B36.10M-1995*

Nom. Size and (O.D.), in.	Wall Thick., in.	Plain End Wgt., lb/ft	Identification: Sch. No.	Identification: Other	
1/8 (0.405)	0.057	0.21	30	…	…
	0.068	0.24	40	5L	STD
	0.095	0.31	80	5L	XS
1/4 (0.540)	0.073	0.36	30	…	…
	0.088	0.42	40	5L	STD
	0.119	0.54	80	5L	XS
3/8 (0.675)	0.073	0.47	30	…	…
	0.091	0.57	40	5L	STD
	0.126	0.74	80	5L	XS
1/2 (0.840)	0.095	0.76	30	…	…
	0.109	0.85	40	5L	STD
	0.147	1.09	80	5L	XS
	0.188	1.31	160	…	…
	0.294	1.71	…	5L	XXS
3/4 (1.050)	0.095	0.97	30	…	…
	0.113	1.13	40	5L	STD
	0.154	1.47	80	5L	XS
	0.219	1.94	160	…	…
	0.308	2.44	…	5L	XXS
1 (1.315)	0.114	1.46	30	…	…
	0.133	1.68	40	5L	STD
	0.179	2.17	80	5L	XS
	0.250	2.84	160	…	…
	0.358	3.66	…	5L	XXS
1¼ (1.660)	0.117	1.93	30	…	…
	0.140	2.27	40	5L	STD
	0.191	3.00	80	5L	XS
	0.250	3.76	160	…	…
	0.382	5.21	…	5L	XXS
1½ (1.900)	0.125	2.37	30	…	…
	0.145	2.72	40	5L	STD
	0.200	3.63	80	5L	XS
	0.281	4.86	160	…	…
	0.400	6.41	…	5L	XXS
2 (2.375)	0.083	2.03	…	5L	…
	0.109	2.64	…	5L	…
	0.125	3.00	…	5L	…
	0.141	3.36	…	5L	…
	0.154	3.65	40	5L	STD
	0.172	4.05	…	5L	…
	0.188	4.39	…	5L	…
	0.218	5.02	80	5L	XS
	0.250	5.67	…	5L	…
	0.281	6.28	…	…	…
	0.344	7.46	160	…	…
	0.436	9.03	…	…	XXS
2½ (2.875)	0.083	2.47	…	5L	…
	0.109	3.22	…	5L	…
	0.125	3.67	…	5L	…
	0.141	4.12	…	5L	…
	0.156	4.53	…	5L	…
	0.172	4.97	…	5L	…
	0.188	5.40	…	5L	…
	0.203	5.79	40	5L	STD
	0.216	6.13	…	5L	…
	0.250	7.01	…	5L	…
	0.276	7.66	80	5L	XS
	0.375	10.01	160	…	…
	0.552	13.69	…	5L	XXS
3 (3.500)	0.083	3.03	…	5L	…
	0.109	3.95	…	5L	…
	0.125	4.51	…	5L	…
3 (3.500)	0.141	5.06	…	5L	…
	0.156	5.57	…	5L	…
	0.172	6.11	…	5L	…
	0.188	6.65	…	5L	…
	0.216	7.58	40	5L	STD
	0.250	8.68	…	5L	…
	0.281	9.66	…	5L	…
	0.300	10.25	80	5L	XS
	0.438	14.32	160	…	…
	0.600	18.58	…	5L	XXS
3½ (4.000)	0.083	3.47	…	5L	…
	0.109	4.53	…	5L	…
	0.125	5.17	…	5L	…
	0.141	5.81	…	5L	…
	0.156	6.40	…	5L	…
	0.172	7.03	…	5L	…
	0.188	7.65	…	5L	…
	0.226	9.11	40	5L	STD
	0.250	10.01	…	5L	…
	0.281	11.16	…	5L	…
	0.318	12.50	80	5L	XS
4 (4.500)	0.083	3.92	…	5L	…
	0.109	5.11	…	5L	…
	0.125	5.84	…	5L	…
	0.141	6.56	…	5L	…
	0.156	7.24	…	5L	…
	0.172	7.95	…	5L	…
	0.188	8.66	…	5L	…
	0.203	9.32	…	5L	…
	0.219	10.01	…	5L	…
	0.237	10.79	40	5L	STD
	0.250	11.35	…	5L	…
	0.281	12.66	…	5L	…
	0.312	13.96	…	5L	…
	0.337	14.98	80	5L	XS
	0.438	19.00	120	5L	…
	0.531	22.51	160	5L	…
	0.674	27.54	…	5L	XXS
5 (5.563)	0.083	4.86	…	5L	…
	0.125	7.26	…	5L	…
	0.156	9.01	…	5L	…
	0.188	10.79	…	5L	…
	0.219	12.50	…	5L	…
	0.258	14.62	40	5L	STD
	0.281	15.85	…	5L	…
	0.312	17.50	…	5L	…
	0.344	19.17	…	5L	…
	0.375	20.78	80	5L	XS
	0.500	27.04	120	5L	…
	0.625	32.96	160	5L	…
	0.750	38.55	…	5L	XXS
6 (6.625)	0.083	5.80	…	5L	…
	0.109	7.59	…	5L	…
	0.125	8.68	…	5L	…
	0.141	9.76	…	5L	…
	0.156	10.78	…	5L	…
	0.172	11.85	…	5L	…

Table 1. *(Continued)* **American National Standard Weights and Dimensions of Welded and Seamless Wrought Steel Pipe** *ANSI/ASME B36.10M-1995*

Nom. Size and (O.D.), in.	Wall Thick., in.	Plain End Wgt., lb/ft	Identification: Sch. No.	Identification: Other	
6 (6.625)	0.188	12.92	…	5L	…
	0.203	13.92	…	5L	…
	0.219	14.98	…	5L	…
	0.250	17.02	…	5L	…
	0.280	18.97	40	5L	STD
	0.312	21.04	…	5L	…
	0.344	23.08	…	5L	…
	0.375	25.03	…	5L	…
	0.432	28.57	80	5L	XS
	0.500	32.71	…	5L	…
	0.562	36.39	120	5L	…
	0.625	40.05	…	5L	…
	0.719	45.35	160	5L	…
	0.750	47.06	…	5L	…
	0.864	53.16	…	5L	XXS
	0.875	53.73	…	5L	…
8 (8.625)	0.125	11.35	…	5L	…
	0.156	14.11	…	5L	…
	0.188	16.94	…	5L	…
	0.203	18.26	…	5L	…
	0.219	19.66	…	5L	…
	0.250	22.36	20	5L	…
	0.277	24.70	30	5L	…
	0.312	27.70	…	5L	…
	0.322	28.55	40	5L	STD
	0.344	30.42	…	5L	…
	0.375	33.04	…	5L	…
	0.406	35.64	60	…	…
	0.438	38.30	…	5L	…
	0.500	43.39	80	5L	XS
	0.562	48.40	…	5L	…
	0.594	50.95	100	…	…
	0.625	53.40	…	5L	…
	0.719	60.71	120	5L	…
	0.750	63.08	…	5L	…
	0.812	67.76	140	5L	…
	0.875	72.42	…	5L	XXS
	0.906	74.69	160	…	…
	1.000	81.44	…	5L	…
10 (10.750)	0.156	17.65	…	5L	…
	0.188	21.21	…	5L	…
	0.203	22.87	…	5L	…
	0.219	24.63	…	5L	…
	0.250	28.04	20	5L	…
	0.279	31.20	…	5L	…
	0.307	34.24	30	5L	…
	0.344	38.23	…	5L	…
	0.365	40.48	40	5L	STD
	0.438	48.24	…	5L	…
	0.500	54.74	60	5L	XS
	0.562	61.15	…	5L	…
	0.594	64.43	80	…	…
	0.625	67.58	…	5L	…
	0.719	77.03	100	5L	…
	0.812	86.18	…	5L	…
	0.844	89.29	120	…	…
	0.875	92.28	…	5L	…
	0.938	98.30	…	5L	…
	1.000	104.13	140	5L	XXS
10 (10.750)	1.125	115.64	160	…	…
	1.250	126.83	…	5L	…
12 (12.750)	0.172	23.11	…	5L	…
	0.188	25.22	…	5L	…
	0.203	27.20	…	…	…
	0.219	29.31	…	5L	…
	0.250	33.38	20	5L	…
	0.281	37.42	…	5L	…
	0.312	41.45	…	5L	…
	0.330	43.77	30	5L	…
	0.344	45.58	…	5L	…
	0.375	49.56	…	5L	STD
	0.406	53.52	40	5L	…
	0.438	57.59	…	5L	…
	0.500	65.42	…	5L	XS
	0.562	73.15	60	5L	…
	0.625	80.93	…	5L	…
	0.688	88.63	80	5L	…
	0.750	96.12	…	5L	…
	0.812	103.53	…	5L	…
	0.844	107.32	100	…	…
	0.875	110.97	…	5L	…
	0.938	118.33	…	5L	…
	1.000	125.49	120	5L	XXS
	1.062	132.57	…	5L	…
	1.125	139.67	140	5L	…
	1.250	153.53	…	5L	…
	1.312	160.27	160	5L	…
14 (14.000)	0.188	27.73	…	5L	…
	0.203	29.91	…	5L	…
	0.210	30.93	…	5L	…
	0.219	32.23	…	5L	…
	0.250	36.71	10	5L	…
	0.281	41.17	…	5L	…
	0.312	45.61	20	5L	…
	0.344	50.17	…	5L	…
	0.375	54.57	30	5L	STD
	0.406	58.94	…	5L	…
	0.438	63.44	40	5L	…
	0.469	67.78	…	5L	…
	0.500	72.09	…	5L	XS
	0.562	80.66	…	5L	…
	0.594	85.05	60	…	…
	0.625	89.28	…	5L	…
	0.688	97.81	…	5L	…
	0.750	106.13	80	5L	…
	0.812	114.37	…	5L	…
	0.875	122.65	…	5L	…
	0.938	130.85	100	5L	…
	1.000	138.84	…	5L	…
	1.062	146.74	…	5L	…
	1.094	150.79	120	…	…
	1.125	154.69	…	5L	…
	1.250	170.21	140	5L	…
	1.406	189.11	160	…	…
	2.000	256.32	…	…	…
	2.125	269.50	…	…	…
	2.200	277.25	…	…	…
	2.500	307.05	…	…	…

Table 2. Properties of American National Standard Schedule 40 Welded and Seamless Wrought Steel Pipe

Diameter, Inches			Wall Thickness, Inches	Cross-Sectional Area of Metal	Weight per Foot, Pounds		Capacity per Foot of Length		Length of Pipe in Feet to Contain		Properties of Sections		
Nominal	Actual Inside	Actual Outside			Of Pipe	Of Water in Pipe	In Cubic Inches	In Gallons	One Cubic Foot	One Gallon	Moment of Inertia	Radius of Gyration	Section Modulus
1/8	0.269	0.405	0.068	0.072	0.24	0.025	0.682	0.003	2532.	338.7	0.00106	0.122	0.00525
1/4	0.364	0.540	0.088	0.125	0.42	0.045	1.249	0.005	1384.	185.0	0.00331	0.163	0.01227
3/8	0.493	0.675	0.091	0.167	0.57	0.083	2.291	0.010	754.4	100.8	0.00729	0.209	0.02160
1/2	0.622	0.840	0.109	0.250	0.85	0.132	3.646	0.016	473.9	63.35	0.01709	0.261	0.4070
3/4	0.824	1.050	0.113	0.333	1.13	0.231	6.399	0.028	270.0	36.10	0.03704	0.334	0.07055
1	1.049	1.315	0.133	0.494	1.68	0.374	10.37	0.045	166.6	22.27	0.08734	0.421	0.1328
1 1/4	1.380	1.660	0.140	0.669	2.27	0.648	17.95	0.078	96.28	12.87	0.1947	0.539	0.2346
1 1/2	1.610	1.900	0.145	0.799	2.72	0.882	24.43	0.106	70.73	9.456	0.3099	0.623	0.3262
2	2.067	2.375	0.154	1.075	3.65	1.454	40.27	0.174	42.91	5.737	0.6658	0.787	0.5607
2 1/2	2.469	2.875	0.203	1.704	5.79	2.074	57.45	0.249	30.08	4.021	1.530	0.947	1.064
3	3.068	3.500	0.216	2.228	7.58	3.202	88.71	0.384	19.48	2.604	3.017	1.163	1.724
3 1/2	3.548	4.000	0.226	2.680	9.11	4.283	118.6	0.514	14.56	1.947	4.788	1.337	2.394
4	4.026	4.500	0.237	3.174	10.79	5.515	152.8	0.661	11.31	1.512	7.233	1.510	3.215
5	5.047	5.563	0.258	4.300	14.62	8.666	240.1	1.04	7.198	0.9622	15.16	1.878	5.451
6	6.065	6.625	0.280	5.581	18.97	12.52	346.7	1.50	4.984	0.6663	28.14	2.245	8.496
8	7.981	8.625	0.322	8.399	28.55	21.67	600.3	2.60	2.878	0.3848	72.49	2.938	16.81
10	10.020	10.750	0.365	11.91	40.48	34.16	946.3	4.10	1.826	0.2441	160.7	3.674	29.91
12	11.938	12.750	0.406	15.74	53.52	48.49	1343.	5.81	1.286	0.1720	300.2	4.364	47.09
16	15.000	16.000	0.500	24.35	82.77	76.55	2121.	9.18	0.8149	0.1089	732.0	5.484	91.50
18	16.876	18.000	0.562	30.79	104.7	96.90	2684.	11.62	0.6438	0.0861	1172.	6.168	130.2
20	18.812	20.000	0.594	36.21	123.1	120.4	3335.	14.44	0.5181	0.0693	1706.	6.864	170.6
24	22.624	24.000	0.688	50.39	171.3	174.1	4824.	20.88	0.3582	0.0479	3426.	8.246	285.5
32	30.624	32.000	0.688	67.68	230.1	319.1	8839.	38.26	0.1955	0.0261	8299.	11.07	518.7

Note: Torsional section modulus equals twice section modulus.

Table 3. Properties of American National Standard Schedule 80 Welded and Seamless Wrought Steel Pipe

Diameter, Inches			Wall Thickness, Inches	Cross-Sectional Area of Metal	Weight per Foot, Pounds		Capacity per Foot of Length		Length of Pipe in Feet to Contain		Properties of Sections		
Nominal	Actual Inside	Actual Outside			Of Pipe	Of Water in Pipe	In Cubic Inches	In Gallons	One Cubic Foot	One Gallon	Moment of Inertia	Radius of Gyration	Section Modulus
1/8	0.215	0.405	0.095	0.093	0.315	0.016	0.436	0.0019	3966.	530.2	0.00122	0.115	0.00600
1/4	0.302	0.540	0.119	0.157	0.537	0.031	0.860	0.0037	2010.	268.7	0.00377	0.155	0.01395
3/8	0.423	0.675	0.126	0.217	0.739	0.061	1.686	0.0073	1025.	137.0	0.00862	0.199	0.02554
1/2	0.546	0.840	0.147	0.320	1.088	0.101	2.810	0.0122	615.0	82.22	0.02008	0.250	0.04780
3/4	0.742	1.050	0.154	0.433	1.474	0.187	5.189	0.0225	333.0	44.52	0.04479	0.321	0.08531
1	0.957	1.315	0.179	0.639	2.172	0.312	8.632	0.0374	200.2	26.76	0.1056	0.407	0.1606
1 1/4	1.278	1.660	0.191	0.881	2.997	0.556	15.39	0.0667	112.3	15.01	0.2418	0.524	0.2913
1 1/2	1.500	1.900	0.200	1.068	3.631	0.766	21.21	0.0918	81.49	10.89	0.3912	0.605	0.4118
2	1.939	2.375	0.218	1.477	5.022	1.279	35.43	0.1534	48.77	6.519	0.8680	0.766	0.7309
2 1/2	2.323	2.875	0.276	2.254	7.661	1.836	50.86	0.2202	33.98	4.542	1.924	0.924	1.339
3	2.900	3.500	0.300	3.016	10.25	2.861	79.26	0.3431	21.80	2.914	3.895	1.136	2.225
3 1/2	3.364	4.000	0.318	3.678	12.50	3.850	106.7	0.4617	16.20	2.166	6.280	1.307	3.140
4	3.826	4.500	0.337	4.407	14.98	4.980	138.0	0.5972	12.53	1.674	9.611	1.477	4.272
5	4.813	5.563	0.375	6.112	20.78	7.882	218.3	0.9451	7.915	1.058	20.67	1.839	7.432
6	5.761	6.625	0.432	8.405	28.57	11.29	312.8	1.354	5.524	0.738	40.49	2.195	12.22
8	7.625	8.625	0.500	12.76	43.39	19.78	548.0	2.372	3.153	0.422	105.7	2.878	24.52
10	9.562	10.750	0.594	18.95	64.42	31.11	861.7	3.730	2.005	0.268	245.2	3.597	45.62
12	11.374	12.750	0.688	26.07	88.63	44.02	1219.	5.278	1.417	0.189	475.7	4.271	74.62
14	12.500	14.000	0.750	31.22	106.1	53.16	1473.	6.375	1.173	0.157	687.4	4.692	98.19
16	14.312	16.000	0.844	40.19	136.6	69.69	1931.	8.357	0.895	0.120	1158.	5.366	144.7
18	16.124	18.000	0.938	50.28	170.9	88.46	2450.	10.61	0.705	0.094	1835.	6.041	203.9
20	17.938	20.000	1.031	61.44	208.9	109.5	3033.	13.13	0.570	0.076	2772.	6.716	277.2
22	19.750	22.000	1.125	73.78	250.8	132.7	3676.	15.91	0.470	0.063	4031.	7.391	366.4

Note: Torsional section modulus equals twice section modulus.

Volume of Flow at 1 Foot Per-Minute Velocity in Pipe and Tube

Nominal Dia., Inches	Schedule 40 Pipe			Schedule 80 Pipe			Type K Copper Tube			Type L Copper Tube		
	Cu. Ft. per Minute	Gallons per Minute	Pounds 60 F Water per Min.	Cu. Ft. per Minute	Gallons per Minute	Pounds 60 F Water per Min.	Cu. Ft. per Minute	Gallons per Minute	Pounds 60 F Water per Min.	Cu. Ft. per Minute	Gallons per Minute	Pounds 60 F Water per Min.
1⁄8	0.0004	0.003	0.025	0.0003	0.002	0.016	0.0002	0.0014	0.012	0.0002	0.002	0.014
1⁄4	0.0007	0.005	0.044	0.0005	0.004	0.031	0.0005	0.0039	0.033	0.0005	0.004	0.034
3⁄8	0.0013	0.010	0.081	0.0010	0.007	0.061	0.0009	0.0066	0.055	0.0010	0.008	0.063
1⁄2	0.0021	0.016	0.132	0.0016	0.012	0.102	0.0015	0.0113	0.094	0.0016	0.012	0.101
3⁄4	0.0037	0.028	0.232	0.0030	0.025	0.213	0.0030	0.0267	0.189	0.0034	0.025	0.210
1	0.0062	0.046	0.387	0.0050	0.037	0.312	0.0054	0.0404	0.338	0.0057	0.043	0.358
1 1⁄4	0.0104	0.078	0.649	0.0088	0.067	0.555	0.0085	0.0632	0.53	0.0087	0.065	0.545
1 1⁄2	0.0141	0.106	0.882	0.0123	0.092	0.765	0.0196	0.1465	1.22	0.0124	0.093	0.770
2	0.0233	0.174	1.454	0.0206	0.154	1.280	0.0209	0.1565	1.31	0.0215	0.161	1.34
2 1⁄2	0.0332	0.248	2.073	0.0294	0.220	1.830	0.0323	0.2418	2.02	0.0331	0.248	2.07
3	0.0514	0.383	3.201	0.0460	0.344	2.870	0.0461	0.3446	2.88	0.0473	0.354	2.96
3 1⁄2	0.0682	0.513	4.287	0.0617	0.458	3.720	0.0625	0.4675	3.91	0.0640	0.479	4.00
4	0.0884	0.660	5.516	0.0800	0.597	4.970	0.0811	0.6068	5.07	0.0841	0.622	5.20
5	0.1390	1.040	8.674	0.1260	0.947	7.940	0.1259	0.9415	7.87	0.1296	0.969	8.10
6	0.2010	1.500	12.52	0.1820	1.355	11.300	0.1797	1.3440	11.2	0.1862	1.393	11.6
8	0.3480	2.600	21.68	0.3180	2.380	19.800	0.3135	2.3446	19.6	0.3253	2.434	20.3
10	0.5476	4.10	34.18	0.5560	4.165	31.130	0.4867	3.4405	30.4	0.5050	3.777	21.6
12	0.7773	5.81	48.52	0.7060	5.280	44.040	0.6978	5.2194	43.6	0.7291	5.454	45.6
14	0.9396	7.03	58.65	0.8520	6.380	53.180	—	—	—	—	—	—
16	1.227	9.18	76.60	1.1170	8.360	69.730	—	—	—	—	—	—
18	1.553	11.62	96.95	1.4180	10.610	88.500	—	—	—	—	—	—
20	1.931	14.44	120.5	1.7550	13.130	109.510	—	—	—	—	—	—

To obtain volume of flow at any other velocity, multiply values in table by velocity in feet per minute.

Plastics Pipe.—Shortly after World War II, plastics pipe became an acceptable substitute, under certain service conditions, for other piping materials. Now, however, plastics pipe is specified on the basis of its own special capabilities and limitations. The largest volume of application has been for water piping systems.

Besides being light in weight, plastics pipe performs well in resisting deterioration from corrosive or caustic fluids. Even if the fluid borne is harmless, the chemical resistance of plastics pipe offers protection against a harmful exterior environment, such as when buried in a corrosive soil.

Generally, plastics pipe is limited by its temperature and pressure capacities. The higher the operating pressure of the pipe system, the less will be its temperature capability. The reverse is true, also. Since it is formed from organic resins, plastics pipe will burn. For various piping compositions, ignition temperatures vary from 700° to 800°F (370° to 430°C).

The following are accepted methods for joining plastics pipe:

Solvent Welding is usually accomplished by brushing a solvent cement on the end of the length of pipe and into the socket end of a fitting or the flange of the next pipe section. A chemical weld then joins and seals the pipe after connection.

Threading is a procedure not recommended for thin-walled plastics pipe or for specific grades of plastics. During connection of thicker-walled pipe, strap wrenches are used to avoid damaging and weakening the plastics.

Heat Fusion involves the use of heated air and plastics filler rods to weld plastics pipe assemblies. A properly welded joint can have a tensile strength equal to 90 percent that of the pipe material.

Elastomeric Sealing is used with bell-end piping. It is a recommended procedure for large diameter piping and for underground installations. The joints are set quickly and have good pressure capabilities.

Table 1. Dimensions and Weights of Thermoplastics Pipe

Nominal Pipe Size		Outside Diameter		Schedule 40				Schedule 80			
				Nom. Wall Thickness		Nominal Weight		Nom. Wall Thickness		Nominal Weight	
in.	cm	in.	cm	in.	cm	lb/100′	kg/m	in.	cm	lb/100′	kg/m
1/8	0.3	0.405	1.03	0.072	0.18	3.27	0.05	0.101	0.256	4.18	0.06
1/4	0.6	0.540	1.37	0.093	0.24	5.66	0.08	0.126	0.320	7.10	0.11
3/8	1.0	0.675	1.71	0.096	0.24	7.57	0.11	0.134	0.340	9.87	0.15
1/2	1.3	0.840	2.13	0.116	0.295	11.4	0.17	0.156	0.396	14.5	0.22
3/4	2.0	1.050	2.67	0.120	0.305	15.2	0.23	0.163	0.414	19.7	0.29
1	2.5	1.315	3.34	0.141	0.358	22.5	0.33	0.190	0.483	29.1	0.43
1 1/4	3.2	1.660	4.22	0.148	0.376	30.5	0.45	0.202	0.513	40.1	0.60
1 1/2	3.8	1.900	4.83	0.154	0.391	36.6	0.54	0.212	0.538	48.7	0.72
2	5.1	2.375	6.03	0.163	0.414	49.1	0.73	0.231	0.587	67.4	1.00
2 1/2	6.4	2.875	7.30	0.215	0.546	77.9	1.16	0.293	0.744	103	1.5
3	7.6	3.500	8.89	0.229	0.582	102	1.5	0.318	0.808	138	2.1
3 1/2	8.9	4.000	10.16	0.240	0.610	123	1.8	0.337	0.856	168	2.5
4	10.2	4.500	11.43	0.251	0.638	145	2.2	0.357	0.907	201	3.0
5	12.7	5.563	14.13	0.273	0.693	197	2.9	0.398	1.011	280	4.2
6	15.2	6.625	16.83	0.297	0.754	256	3.8	0.458	1.163	385	5.7
8	20.3	8.625	21.91	0.341	0.866	385	5.7	0.530	1.346	584	8.7
10	25.4	10.75	27.31	0.387	0.983	546	8.1	0.629	1.598	867	12.9
12	30.5	12.75	32.39	0.430	1.09	722	10.7	0.728	1.849	1192	17.7

The nominal weights of plastics pipe given in this table are based on an empirically chosen material density of 1.00 g/cm^3. The nominal unit weight for a specific plastics pipe formulation can be obtained by multiplying the weight values from the table by the density in g/cm^3 or by the specific gravity of the particular platics composition.

The following are ranges of density factors for various plastics pipe materials: PE, 0.93 to 0.96; PVC, 1.35 to 1.40; CPVC, 1.55; ABS, 1.04 to 1.08; SR, 1.05; PB, 0.91 to 0.92; and PP, 0.91. For meanings of abbreviations see Table 2.

Information supplied by the Plastics Pipe Institute.

Insert Fitting is particularly useful for PE and PB pipe. For joining pipe sections, insert fittings are pushed into the pipe and secured by stainless steel clamps.

Transition Fitting involves specially designed connectors to join plastic pipe with other materials, such as cast iron, steel, copper, clay, and concrete.

Plastic pipe can be specified by means of Schedules 40, 80, and 120, which conform dimensionally to metal pipe, or through a Standard Dimension Ratio (SDR). The SDR is a rounded value obtained by dividing the average outside diameter of the pipe by the wall thickness. Within an individual SDR series of pipe, pressure ratings are uniform, regardless of pipe diameter.

Table 1 provides the weights and dimensions for Schedule 40 and 80 thermoplastic pipe, Table 2 gives properties of plastics pipe, Table 3 gives maximum non-shock operating pressures for several varieties of Schedule 40 and 80 plastics pipe at 73°F, and Table 4 gives correction factors to pressure ratings for elevated temperatures.

Table 2. General Properties and Uses of Plastic Pipe

Plastic Pipe Material	Properties	Common Uses	Operating Temperature[a]		Joining Methods
			With Pressure	Without Pressure	
ABS (Acrylonitrilebutadiene styrene)	Rigid; excellent impact strength at low temperatures; maintains rigidity at higher temperatures.	Water, Drain, Waste, Vent, Sewage.	100°F (38°C)	180°F (82°C)	Solvent cement, Threading, Transition fitting.
PE (Polyethylene)	Flexible; excellent impact strength; good performance at low temperatures.	Water, Gas, Chemical, Irrigation.	100°F (38°C)	180°F (82°C)	Heat fusion, Insert and Transition fitting.
PVC (Polyvinylchloride)	Rigid; fire self-extinguishing; high impact and tensile strength.	Water, Gas, Sewage, Industrial process, Irrigation.	100°F (38°C)	180°F (82°C)	Solvent cement, Elastomeric seal, Mechanical coupling, Transition fitting.
CPVC (Chlorinated polyvinyl chloride)	Rigid; fire self-extinguishing; high impact and tensile strength.	Hot and cold water, Chemical.	180°F (82°C) at 100 psig (690kPa) for SDR-11		Solvent cement, Threading, Mechanical coupling, Transition fitting.
PB (Polybutylene)	Flexible; good performance at elevated temperatures.	Water, Gas, Irrigation.	180°F (82°C)	200°F (93°C)	Insert fitting, Heat fusion, Transition fitting.
PP (Polypropylene)	Rigid; very light; high chemical resistance, particularly to sulfur-bearing compounds.	Chemical waste and processing.	100°F (38°C)	180°F (82°C)	Mechanical coupling, Heat fusion, Threading.
SR (Styrene rubber plastic)	Rigid; moderate chemical resistance; fair impact strength.	Drainage, Septic fields.	150°F (66°C)	…	Solvent cement, Transition fitting, Elastomeric seal.

[a] The operating temperatures shows are general guide points. For specific operating temperature and pressure data for various grades of the types of plastic pipe given, please consult the pipe manufacturer or the Plastics Pipe Institute.

From information supplied by the Plastics Pipe Institute.

Table 3. Maximum Nonshock Operating Pressure (psi) for Thermoplastic Piping at 73°F

Nominal Pipe Size (inch)	Schedule 40 PVC & CPVC (Socket End)	Schedule 40 ABS	Schedule 80 PVC & CPVC Socket End	Schedule 80 PVC & CPVC Threaded End	Schedule 80 Polypropylene Thermo-seal Joint	Schedule 80 Polypropylene Threaded End[a]	Schedule 80 PVDF Thermoseal Joint	Schedule 80 PVDF Threaded End	Schedule 80 ABS
½	600	476	850	420	410	20	580	290	678
¾	480	385	690	340	330	20	470	230	550
1	450	360	630	320	310	20	430	210	504
1¼	370	294	520	260	260	20	…	…	416
1½	330	264	470	240	230	20	326	160	376
2	280	222	400	200	200	…	270	140	323
2½	300	243	420	210	…	20	…	…	340
3	260	211	370	190	160	20	250	NR	297
4	220	177	320	160	140	NR	220	NR	259
6	180	141	280	NR	…	…	190	NR	222
8	160	…	250[b]	NR	…	…	…	…	…
10	140	…	230	NR	…	…	…	…	…
12	130	…	230	NR	…	…	…	…	…

[a] Recommended for intermittent drainage pressure not exceeding 20 psi.
[b] 8-inch CPVC Tee, 90° Ell, and 45° Ell are rated at half the pressure shown.

ABS pressures refer to unthreaded pipe only.

For service at higher temperature, multiply the pressure obtained from this table by the correction factor from Table 2.

NR is not recommended.

Table 4. Temperature-Correction Factors for Thermoplastic Piping Operating Pressures

Operating Temperature,°F	Pipe Material PVC	Pipe Material CPVC	Pipe Material Polypropylene	Pipe Material PVDF
70	1	1	1	1
80	0.90	0.96	0.97	0.95
90	0.75	0.92	0.91	0.87
100	0.62	0.85	0.85	0.80
110	0.50	0.77	0.80	0.75
115	0.45	0.74	0.77	0.71
120	0.40	0.70	0.75	0.68
125	0.35	0.66	0.71	0.66
130	0.30	0.62	0.68	0.62
140	0.22	0.55	0.65	0.58
150	NR	0.47	0.57	0.52
160	NR	0.40	0.50	0.49
170	NR	0.32	0.26	0.45
180	NR	0.25	[a]	0.42
200	NR	0.18	NR	0.36
210	NR	0.15	NR	0.33
240	NR	NR	NR	0.25
280	NR	NR	NR	0.18

[a] Recommended for intermittent drainage pressure not exceeding 20 psi.

NR = not recommended.

For more detailed information concerning the properties of a particular plastic pipe formulation, consult the pipe manufacturer or The Plastics Pipe Institute, 355 Lexington Ave., New York, N.Y. 10017.