INTRODUCTION

Ball and roller bearings are used widely in instruments and machines in order to minimize friction and power loss. While the concept of the ball bearing dates back at least to Leonardo da Vinci, their design and manufacture has become remarkably sophisticated.

This technology was brought to its present state of perfection only after a long period of research and development. The benefits of such specialized research can be obtained when it is possible to use a standardized bearing of the proper size and type. However, such bearings cannot be used indiscriminately without a careful study of the loads and operating conditions. In addition, the bearing must be provided with adequate mounting, lubrication and sealing.

Design engineers have usually two possible sources for obtaining information which they can use to select a bearing for their particular application:

a) Textbooks
b) Manufacturers' catalogs

Textbooks are excellent sources; however, they tend to be overly detailed and aimed at the student of the subject matter rather than the practicing designer. They, in most cases, contain information on how to design rather than how to select a bearing for a particular application.

Manufacturers' catalogs, in turn, are also excellent and contain a wealth of information which relates to the products of the particular manufacturer. These catalogs, however, fail to provide alternatives – which may divert the designer's interest to products not manufactured by them.

Our Company, however, provides the broadest selection of many types of bearings made by different manufacturers. For this reason, we are interested in providing a condensed overview of the subject matter in an objective manner, using data obtained from different texts, handbooks and manufacturers' literature. This information will enable the reader to select the proper bearing in an expeditious manner.

If the designer's interest exceeds the scope of the presented material, a list of references is provided at the end of the Technical Section.

At the same time, we are expressing our thanks and are providing credit to the sources which supplied the material presented here.

The information deals with:

a) Rolling Contact Bearings
b) Sintered-Metal Sliding Contact Bearings
and c) Plastic and Nonmetallic Sliding Contact Bearings

1.0 ROLLING CONTACT BEARINGS

1.1 General

Rolling contact bearings can be divided into three basic groups:

a) Ball Bearings
b) Thrust Bearings
and c) Roller Bearings

Each of these groups can further be divided into subgroups. Rather than enumerating the subgroups, they will be shown in Fig. 1-1 and Fig. 1-2.

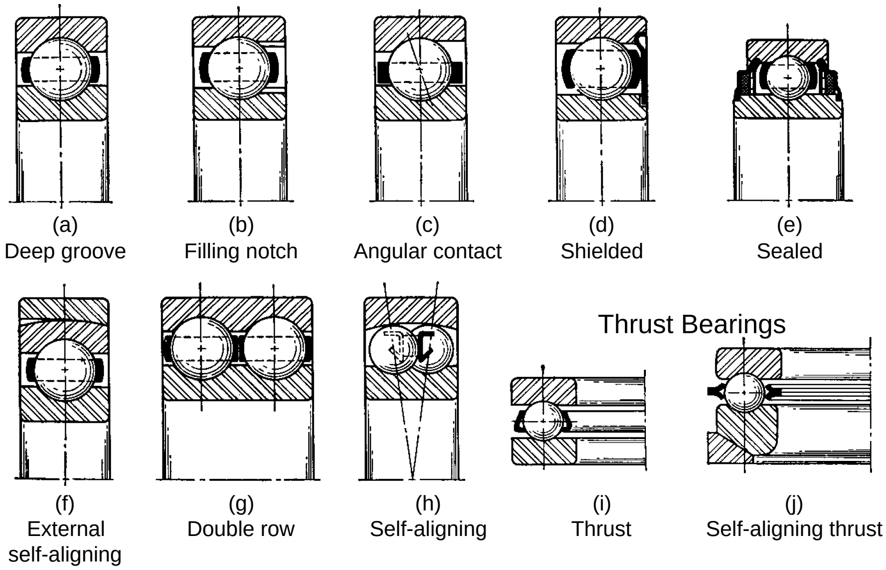


Fig. 1-1 Radial and Thrust Ball Bearings

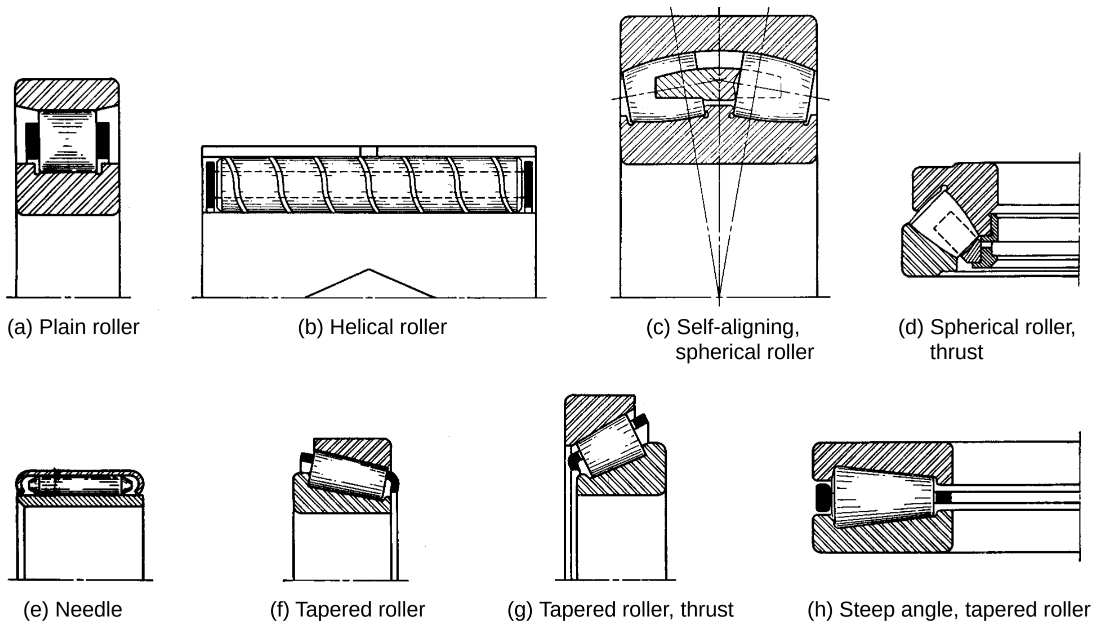


Fig. 1-2 Radial and Thrust Roller Bearings

1.2 Construction and Types of Ball Bearings

A ball bearing usually consists of four parts: an inner ring, an outer ring, the balls and the cage or separator. To increase the contact area and permit larger loads to be carried, the balls run in curvilinear grooves in the rings. The radius of the groove is slightly larger than the radius of the ball, and a very slight amount of radial play must be provided. The bearing is thus permitted to adjust itself to small amounts of angular misalignment between the assembled shaft and mounting. The separator keeps the balls evenly spaced and prevents them from touching each other on the sides where their relative velocities are the greatest.

Ball bearings are made in a wide variety of types and sizes. Single-row radial bearings are made in four series, extra light, light, medium, and heavy, for each bore, as illustrated in Fig. 1-3(a), (b), and (c). The heavy series of bearings is designated by 400. Most, but not all, manufacturers use a numbering system so devised that if the last two digits are multiplied by 5, the result will be the bore in millimeters. The digit in the third place from the right indicates the series number. Thus, bearing 307 signifies a medium-series bearing of 35-mm bore. For additional digits, which may be present in the catalog number of a bearing, refer to manufacturer's details. Some makers list deep groove bearings and bearings with two rows of balls. For bearing designations of **Quality Bearings & Components** (**QBC**), see special pages devoted to this purpose.

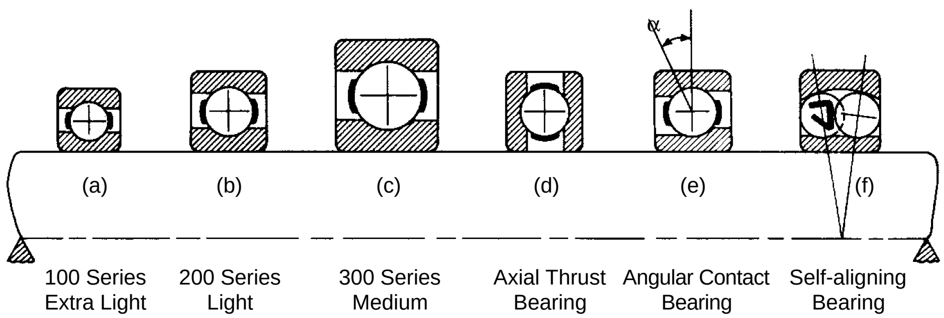
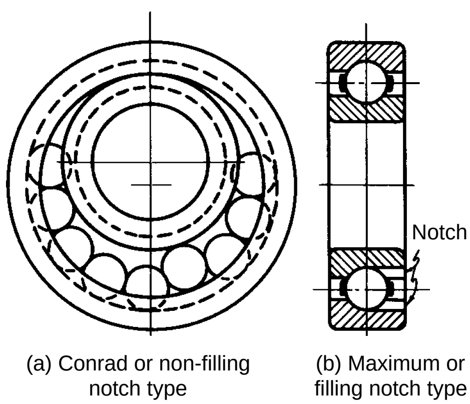


Fig. 1-3 Types of Ball Bearings

The radial bearing is able to carry a considerable amount of axial thrust. However, when the load is directed entirely along the axis, the thrust type of bearing should be used. The angular contact bearing will take care of both radial and axial loads. The self-aligning ball bearing will take care of large amounts of angular misalignment. An increase in radial capacity may be secured by using rings with deep grooves, or by employing a double-row radial bearing.

Radial bearings are divided into two general classes, depending on the method of assembly. These are the Conrad, or nonfilling-notch type, and the maximum, or filling-notch type. In the Conrad bearing, the balls are placed between the rings as shown in Fig. 1-4(a). Then they are evenly spaced and the separator is riveted in place. In the maximum-type bearing, the balls are



(a) Conrad or non-filling notch type

(b) Maximum or filling notch type

Fig. 1-4 Methods of Assembly for Ball Bearings

inserted through a filling notch ground into each ring, as shown in Fig. 1-4(b). Because more balls can be placed in such bearings, their load capacity is greater than that of the Conrad type. However, the presence of the notches limits the load-carrying capacity of these bearings in the axial direction.

High-carbon chromium steel 52100 and 440C stainless steel are used for balls and rings, and are treated to high strength and hardness. The surfaces are smoothly ground and polished. The commonly accepted minimum hardness for bearing components is 58 Rockwell C. This material is not suitable for temperatures over 350° F. For higher temperatures, steels especially developed for high-temperature service should be used. The dimensional tolerances are very small; the balls must be very uniform in size. The stresses are extremely high because of the small contact areas, and the yield point of the material may be exceeded at certain points. Because of the high values of the fluctuating stresses, antifriction bearings are not designed for unlimited life, but for some finite period of service determined by the fatigue strength of the materials. A specified speed and number of hours of expected service must therefore accompany the given load values for these bearings.

1.3 Bearing Selection Factors

Bearings are basically antifriction devices. For this reason, the friction characteristics of different bearing types have to be examined.

In addition to the rolling resistance, other factors which contribute to the friction are as follows:

1. Sliding between the rolling elements and the race. When the rolling elements are curved, all points in contact do not have the same linear velocity, because of their differing radii of rotation. In Fig. 1-5, for example, a point A on the ball will have a definite linear velocity if no sliding occurs. However, a second point B on the ball will have less linear velocity than A because of its smaller radius of rotation. But point B on the race actually has a slightly greater linear velocity than A. This introduces sliding in both backward and forward directions. Other factors which introduce sliding are the inevitable inaccuracies in geometry and other deviations from true rolling.
2. The sliding action between the rolling element and the separator. Although contact takes place at the poles, where the velocity is lowest, some sliding action is present.
3. In roller bearings, the sliding action between the rolling elements and the guide flanges.
4. The losses between the bearing parts and the lubricant and between the different particles of the lubricant.

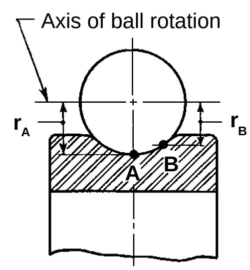


Fig. 1-5 Sliding caused by Geometry of Bearing

Palmgren[1] gives the following frictional coefficients for antifriction bearings:

Self-aligning ball bearings .... f = 0.0010
Cylindrical roller bearing .... f = 0.0011
Thrust ball bearings .... f = 0.0013
Single-row deep-groove ball bearings .... f = 0.0015
Tapered and spherical roller bearings .... f = 0.0018
Needle bearings .... f = 0.0045

All these coefficients are referred to the bearing bore. They are for run-in bearings, under

[1] See reference at the end of the Technical Section.

Quality Bearings & Components

normal conditions, with good lubrication. When determining the total losses in a given application, the seal friction must not be ignored since it may be considerable.

In addition to considerations related to friction, attention must be given to speed requirements. Permissible speeds are influenced by bearing size, properties, lubrication detail and operating temperatures. The permissible speed varies inversely with mean bearing diameter.

Some guidelines for selecting bearings can be summarized as follows:

- Ball bearings are the less expensive choice in the smaller sizes and under lighter loads, while roller bearings are less expensive for larger sizes and heavier loads.
- Roller bearings are more satisfactory under shock or impact loading than ball bearings.
- Ball-thrust bearings are for pure thrust loading only. At high speeds, a deep-groove or angular-contact ball bearing usually will be a better choice, even for pure thrust loads.
- Self-aligning ball bearings and cylindrical roller bearings have very low friction coefficients.
- Deep-groove ball bearings are available with seals built into the bearing so that the bearing can be prelubricated to operate for long periods without attention.

The following Table 1-1 attempts to summarize and tabulate various considerations which influence the selection of the appropriate bearings:

Table 1-1 Bearing Selection Factors*

Relative rating

- ■ good
- ◩ intermediate
- • poor
- ▬ none
- ☒ does not apply
- (blank) not available or known

NOTE: This table is only a general guide – ratings may change when considering special types or treatments.

Bearing Types

Selection Factors	Ball 2a deep groove	Ball 2b deep groove – fill slot	Ball 2c angular contact	Ball 2d 2-row angular contact	Ball 2e self-aligning	Ball 2f thrust	Roller 3a cylindrical	Roller 3b cylindrical – locating	Roller 3c barrel	Roller 3d spherical thrust	Roller 3e tapered	Roller 3f needle	Journal 8a full	Journal 8b partial	Journal 8c axial slotted	Journal 8d elliptical	Journal 8e three-lobed	Journal 8f pivoted shoe	Thrust 12a parallel	Thrust 12b step	Thrust 12c tapered land	Thrust 12d tilt shoe	Ext. press. ⑤ capillary ⑥	Ext. press. ⑤ variable flow dam ⑥	Gas ⑤ pneumostatic	Gas ⑤ pneumodynamic
Low starting friction	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	•	•	•	•	•	•	•	•	•	•	■	■	■	•
Low running friction	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	•	◩	◩	◩	■	◩	■	■
Low noise	•	•	•	•	•	•	•	•	•	•	•	•	■	■	■	■	■	■	■	■	■	■	■	■	■	■
Small diameter ①	•	•	•	•	•	•	◩	◩	◩	◩	◩	■	■	■	■	■	■	◩	•	◩	◩	◩	◩	◩	•	•
Short length ①	■	■	■	◩	■	■	■	■	■	■	■	◩	◩	◩	◩	◩	◩	◩	■	■	■	◩	◩	◩	•	•
High accuracy	◩	•	■	■	◩	◩	◩	◩	◩	◩	◩	◩	•	•	•	•	•	◩	•	•	•	•	■	■	■	•
Most available	■	■	■	■	■	■	■	■	■	◩	■	■	■	■	•	◩	•	•	■	•	◩	•	•	•	•	•
High radial load ②	◩	◩	◩	◩	◩	▬	■	■	■	▬	■	▬	■	■	◩	◩	◩	◩	▬	▬	▬	▬	◩	◩	•	•
High thrust load ②	◩	•	■	■	•	■	▬	•	◩	■	■	▬	▬	▬	▬	▬	▬	▬	•	■	■	■	◩	◩	•	•
High dynamic load ②	◩	◩	◩	◩	◩	◩	■	■	■	■	■	■	■	◩	◩	◩	•	◩	■	◩	◩	◩	◩	◩	•	•
Tolerate misalignment	◩	◩	◩	•	■	◩	•	•	■	■	•	•	◩	◩	◩	◩	◩	■	◩	◩	◩	■	◩	◩	•	•
Tolerate dirt	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	■	■	■	■	■	■	■	■	■	■	•	◩	•	•
Low initial cost	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	■	■	■	◩	◩	•	■	■	■	•	•	•	•	◩
High speed	■	•	■	◩	◩	•	■	■	•	•	•	•	◩	◩	◩	■	■	■	•	■	■	■	■	◩	■	■
High temperature ③	◩						◩						•												■	◩
Simple lube system	■	■	■	■	■	■	■	■	■	■	■	■	◩	◩	◩	◩	◩	◩	◩	◩	◩	◩	•	•	•	■
High stability ④	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	•	•	◩	■	■	■	■	■	■	■	■	◩	•	•
Easy for designer	■	■	■	■	■	■	■	■	■	■	■	■	■	■	◩	◩	◩	•	■	◩	◩	◩	•	•	•	•

① with a given load (radial or thrust) ② with a given size ③ above 450°F ④ applies to high speed fluid-film bearings
⑤ journal or thrust type ⑥ restrictor controlled (liquid)

* See reference at the end of the Technical Section.

1.4 Bearing Loads

The first step in sizing a suitable ball bearing for a given application is the determination of the loads which it has to support. In this section, we list some of the most frequently occurring mechanical configurations and the bearing loads imposed by them.

(a) Radial Shaft Load Between Bearings

P = radial load
R_1, R_2 = bearing loads
l_1, l_2 = distances from radial load to bearings

$$R_1 = \frac{l_2 P}{l_1 + l_2} \qquad (1)$$

$$R_2 = \frac{l_1 P}{l_1 + l_2} \qquad (2)$$

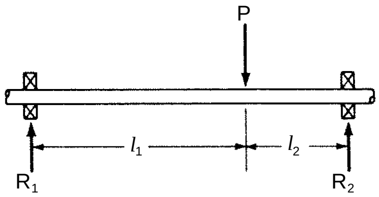


Fig. 1-6 Radial Load Between Bearings

(b) Overhung Radial Load

Notation same as in paragraph **(a)**.

$$R_1 = \frac{l_2 P}{l_1 - l_2} \qquad (3)$$

$$R_2 = \frac{l_1 P}{l_1 - l_2} \qquad (4)$$

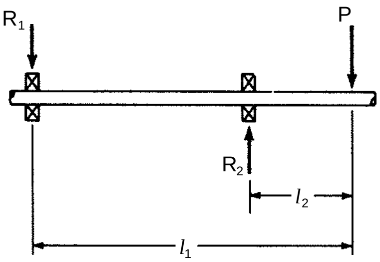


Fig. 1-7 Overhung Radial Load

For cases other than those shown above, the rules of static distribution of loads on a beam should be considered. The shaft which is supported by bearings is nothing else but a beam subjected to forces which result in radial loading of bearings.

1.5 Determination of Bearing Size

(a) Basic Definitions

In the course of many years of experience with ball bearings and extensive testing, it has been found that the prediction of the load capacity of a ball bearing is a statistical event related to the fatigue life of the bearing. This makes the sizing of ball bearings more difficult than that of many other machine elements.

A basic phenomenon in ball bearings is that ball bearing life has been found to be inversely proportional to the cube of the bearing load. This means that when the load is doubled, the life expectancy of the bearing is reduced by a factor of eight. This phenomenon has been studied extensively and has led to the adoption of an industry-wide national standard for rating ball bearings pioneered by the American Bearing Manufacturers Association (formerly Anti-Friction Bearing Manufacturers Association, Inc.), 1200 19th Street, N.W., Suite 300, Washington, D.C. 20036-2433.

The following represents a summary of the load rating of ball bearings of less than one inch in diameter, according to ANSI-AFBMA Standard 9-1978: “Load Rating and Fatigue Life for Ball Bearings” – reprinted with the permission of the American National Standards Institute, Inc., 11 West 42nd Street, 13th Floor, New York, N.Y. 10036.

Ball bearings were formerly rated on the basis of the compressive stress in the most heavily loaded ball. Except for static loads, experience has shown that the actual cause of failure is fatigue. Fatigue characteristics are thus used for load rating and are dependent to a large extent on experimental results.

The life of a ball bearing is the life in hours at some known speed, or the number of revolutions, that the bearing will attain before the first evidence of fatigue appears on any of the moving elements. Experience has shown that the life of an individual ball bearing cannot be precisely predicted. Fatigue characteristics are thus used for load ratings.

Even if ball bearings are properly mounted, adequately lubricated, protected from foreign matter, and are not subject to extreme operating conditions, they can ultimately fatigue. Under ideal conditions, the repeated stresses developed in the contact areas between the balls and the raceways eventually can result in fatigue of the material which manifests itself as spalling of the load carrying surfaces. In most applications, the fatigue life is the maximum useful life of a bearing. This fatigue is the criterion of life used as the basis for the first part of this standard.

The material in the standard which follows assumes bearings having nontruncated contact area, hardened good quality steel as the bearing material, adequate lubrication, proper ring support and alignment, nominal internal clearances, and adequate groove radii. In addition, certain high-speed effects such as ball centrifugal forces and gyroscopic moments are not considered.

The following nomenclature and definitions are used in life testing of bearings. A multitude of identical bearings are tested under same conditions:

RATING LIFE is the life at which 10 percent of bearings have failed and 90 percent of them are still good. This value is designated as L_{10} and is expressed in millions of revolutions.

LIFE of an individual ball bearing is the number of revolutions (or hours at some given constant speed) designated as L which the bearing runs before the first evidence of fatigue develops in the material of either ring (or washer) or of any of the rolling elements.

MEDIAN LIFE is the life at which 50 percent of bearings failed and 50 percent are still good. It is designated as L_{50}, which is generally not more than five times the RATING LIFE, L_{10}.

BASIC LOAD RATING "C" for a radial or angular contact ball bearing is the calculated, constant, radial load which a group of apparently identical bearings with stationary outer ring can theoretically endure for a RATING LIFE of one million revolutions of the inner ring. For a thrust ball bearing, it is the calculated, constant, centric, thrust load which a group of apparently identical bearings can theoretically endure for a RATING LIFE of one million revolutions of one of the bearing washers. The basic load rating is a reference value only of the base value of one million revolutions RATING LIFE having been chosen for ease of calculation. Since applied loading as great as the basic load rating tends to cause local plastic deformation of the rolling surfaces, it is not anticipated that such heavy loading would normally be applied.

(b) Determination of Basic Load Rating

The basic load rating C for a rating life of one million revolutions for radial and angular contact ball bearings, except filling slot bearings, with balls not larger than 1 in. diameter, is given by the equation:

$$C = f_c(i \cos \alpha)^{0.7} Z^{2/3} D^{1.8} \quad \text{(lbs.)} \qquad (5)$$

where:

- i = number of rows of balls in the bearing
- α = nominal angle of contact (angle between line of action of ball load and plane perpendicular to bearing axis)
- Z = number of balls per row

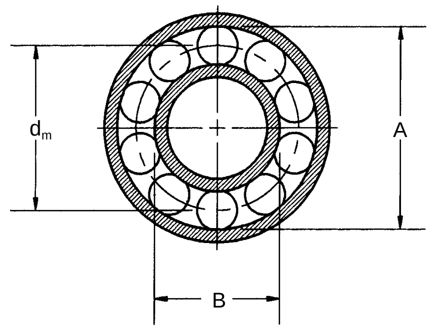
D = ball diameter
f_c = a constant from Table 1-2, as determined by the value of $(D \cos \alpha)/d_m$
d_m = pitch diameter of ball races

NOTE: For balls larger than 1 inch diameter, the exponent for D is 1.4.

To get a better feel for the meaning of one million revolutions, it is attained in 8 hrs at a speed of 2,084 rpm. Most ball bearings, however, may have intended life many times exceeding one million revolutions.

In the above formula, d_m represents the pitch diameter of the ball races. It can be expressed as follows:

$$d_m = \frac{A - B}{2} + B = \frac{A + B}{2} \qquad (6)$$



A and B are dimensions as shown. However, assuming that inner ring and outer ring wall thicknesses are the same, A becomes outside diameter, and B the bore of the bearing.

Values of f_c are shown in Table 1-2 for different values of $(D \cos \alpha)/d_m$.

RATING LIFE L_{10} in millions of revolutions for a ball bearing application can be calculated from:

$$L_{10} = \left(\frac{C}{P}\right)^3 \qquad (7)$$

where:

C = the basic load rating as previously defined
and P = the load.

(c) Illustrative Examples

Example 1

Consider an ABEC 3 single row, radial ball bearing having 10 balls of 1/16" diameter, 0.300" inner race diameter and 0.452" outer race diameter in a single shield configuration.

$\alpha = 0°$ (radial bearing)
Z = 10 (number of balls)
D = 1/16" (ball diameter)

and $d_m = \frac{1}{2}(0.330 + 0.452) = 0.391$" (pitch diameter of ball races).

Therefore, $$\left(\frac{D \cos \alpha}{d_m}\right) = \frac{0.062 \times 1}{0.391} = 0.16$$

From Table 1-2 this value yields (from third column) a value of f_c=4530. Substituting these values in Equation (5) for C, we obtain:

$$C = 4530 \times 1 \times 10^{2/3} \times 0.062^{1.8} = 143 \text{ lbs}$$

This means that for a load of P = 143 lbs, the rating life of this ball bearing will be one million revolutions and 90% of a group of such ball bearings will be expected to complete or exceed this value.

Table 1-2* Values of f_c

$\frac{D \cos \alpha}{d_m}$	Single Row Radial Contact; Single & Double Row Angular Contact, Groove Type(1)		Double Row Radial Contact Groove Type		Self-Aligning	
	Metric(2)	Inch(3)	Metric(2)	Inch(3)	Metric(2)	Inch(3)
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

NOTES:

(1) 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 (1) c. does not apply.

(2) Use to obtain C in newtons when D is given in mm.

(3) Use to obtain C in pounds when D is given in inches.

Suppose now it is desired to determine the "L" life of this bearing when operating at 200 rpm and a load of 50 lbs, the life being evaluated in hours of operation.

Let the life in hours be denoted by L, and let N denote the rpm of the bearing. We then have:

$$L = \frac{10^6 L_{10}}{60\ N} = \left(\frac{C}{P}\right)^3 \frac{10^6}{60\ N} \qquad (8)$$

Substituting N = 200, P = 50 and C = 143 into Equation (8), we obtain L = 1949 hours.

NOTE: L_{10} is bearing life in millions of revolutions; L is bearing life in hours.

A table showing required life at constant operating speed has been given by N. Chironis ("Today's Ball Bearings", *Product Engineering*, December 12, 1960, pp. 63-77, table on p. 68). This table is reproduced below with the permission of McGraw-Hill Book Company, New York, N.Y.

Table 1-3 Required Life at Constant Operating Speed
(data from SKF Industries)

Type of Machine	Life in Hours of Operation
Instruments and apparatus which are only infrequently used. *Ex.:* demonstration apparatus, devices for operation of sliding doors.	500
Aircraft Engines.	500–2000
Machines for service of short duration or intermittent operation, where service interruptions are of minor importance. *Ex.:* hand tools, lifting tackle in machinery shops, hand-driven machines in general, farm machinery, assembly cranes, charging machines, foundry cranes, household machines.	4000–8000
Machines for intermittent service where dependable operation is of great importance. *Ex.*: auxiliary machines in power stations, conveying-equipment in production lines, elevators, general-cargo cranes, machine tools less frequently used.	8000–12000
Machines for 8-hour service which are not always fully utilized. *Ex.:* machines in general in the mechanical industries, cranes for continuous service, blowers, jackshafts.	20000–30000
Machines for continuous operation (24-hour service). *Ex.:* separators, compressors, pumps, mainline shafting, roller beds and conveyor rollers, mine hoists, stationary electric motors.	40000–60000
Machines for 24-hour service where dependability is of great importance. *Ex.:* pulp and paper machines, public power stations, mine pumps, public pumping stations, machines for continuous service aboard ships.	100000–200000

In order to provide data for larger size bearings as well as additional examples, Table 1-4 is given.

Table 1-4 Dimensions and Basic Load Ratings for Conrad-Type Single-Row Radial Ball Bearings

Bearing No.	Bore		Outside Diameter		Width		Balls		Capacity, lbs	
	mm	inch	mm	inch	mm	inch	No. Z	Dia. D	Dynamic C	Static P_{st}
102			32	1.2598	9	0.3543	9	3/16	965	550
202	15	0.5906	35	1.3780	11	0.4331	7	1/4	1340	760
302			42	1.6535	13	0.5118	8	17/64	1660	930
103			35	1.3780	10	0.3937	10	3/16	1040	640
203	17	0.6693	40	1.5748	12	0.4724	7	5/16	1960	1040
303			47	1.8504	14	0.5512	6	3/8	2400	1240
104			42	1.6535	12	0.4724	9	1/4	1620	980
204	20	0.7874	47	1.8504	14	0.5512	8	5/16	2210	1280
304			52	2.0472	15	0.5906	7	3/8	2760	1530
105			47	1.8504	12	0.4724	10	1/4	1740	1140
205	25	0.9843	52	2.0472	15	0.5906	9	5/16	2420	1520
305			62	2.4409	17	0.6693	8	13/32	3550	2160
106			55	2.1654	13	0.5118	11	9/32	2290	1590
206	30	1.1811	62	2.4409	16	0.6299	9	3/8	3360	2190
306			72	2.8346	19	0.7480	8	1/2	5120	3200
107			62	2.4409	14	0.5512	11	5/16	2760	2010
207	35	1.3780	72	2.8346	17	0.6693	9	7/16	4440	2980
307			80	3.1496	21	0.8268	8	17/32	5750	3710
108			68	2.6772	15	0.5906	13	5/16	3060	2450
208	40	1.5748	80	3.1496	18	0.7087	9	1/2	5640	3870
308			90	3.5433	23	0.9055	8	5/8	7670	5050
109			75	2.9528	16	0.6299	13	11/32	3630	2970
209	45	1.7717	85	3.3465	19	0.7480	9	1/2	5660	3980
309			100	3.9370	25	0.9843	8	11/16	9120	6150
110			80	3.1496	16	0.6299	14	11/32	3770	3260
210	50	1.9685	90	3.5433	20	0.7874	10	1/2	6070	4540
310			110	4.3307	27	1.0630	8	3/4	10680	7350
111			90	3.5433	18	0.7087	13	13/32	4890	3950
211	55	2.1654	100	3.9370	21	0.8268	10	9/16	7500	5710
311			120	4.7244	29	1.1417	8	13/16	12350	8660
112			95	3.7402	18	0.7087	14	13/32	5090	4560
212	60	2.3622	110	4.3307	22	0.8661	10	5/8	9070	6890
312			130	5.1181	31	1.2205	8	7/8	14130	10100
113			100	3.9370	18	0.7087	15	13/32	5280	4950
213	65	2.5591	120	4.7244	23	0.9055	10	11/16	10770	8460
313			140	5.5118	33	1.2992	8	15/16	16010	11600
114			110	4.3307	20	0.7874	14	15/32	6580	6080
214	70	2.7559	125	4.9213	24	0.9449	10	11/16	10760	8740
314			150	5.9055	35	1.3780	8	1	18000	13260

Example 2:

Find the value of C for a 207 radial bearing.

Solution:

By Table 1-4: $d_m = \frac{1}{2}(2.8346 + 1.3780) = 2.1063$ in

$$\frac{D\cos\alpha}{d_m} = \frac{0.4375}{2.1063} = 0.208$$

By Table 1-2: $f_c = 4550$

By Table 1-4: $D = \frac{7}{16} = 0.4375$ in

$\log D = 9.64098 - 10$
$1.8 \log D = 9.35376 - 10$
$D^{1.8} = 0.2258$
$Z = 9, \quad Z^{2/3} = \sqrt[3]{9^2} = 4.327$

From Equation (5) for C: $C = 4550 \times 4.327 \times 0.2258 = 4440$ lbs, load for 1 million revolutions with 90 percent probability that it will be attained or exceeded.

(d) Relationship between Load and Number of Revolutions

In some cases, it is needed to determine the new value of the permitted loading when the number of revolutions N is changed.

Experimentally, it was proven that:

$$\frac{N_1}{N_2} = \frac{P_2^{\,3}}{P_1^{\,3}} \qquad (9)$$

where N is number of revolutions and P is radial load.

Furthermore, it was established that

$$10^6\, C^3 = N_1\, P_1^{\,3} = N_2\, P_2^{\,3} = N_3\, P_3^{\,3} \text{ ... is a constant,}$$

or subsequently:

$$N_1 = \frac{10^6\, C^3}{P_1^{\,3}} \qquad (10)$$

It has to be made clear that C is the basic load rating in lbs. for a rating life of 1 million revolutions, and this fact establishes the above relationship.

If a bearing has a rating life expressed in number of revolutions designated by N, the life of the bearing expressed in hours, designated by L, can be found from:

$$N = 60\, n\, L$$

where n is the actual speed in rpm of the bearing.

Example 3

For Example 2 where we found C = 4440 lbs., find the radial load P_1 for a rating life of 500 hours, at 1500 rpm.

$$P_1^3 = \frac{10^6\,C^3}{N_1} = \frac{10^6\,C^3}{60\,n\,L}$$

Apply: C = 4.440 lbs., n = 1,500 rpm, and L = 500 hrs

$$P_1^3 = \frac{10^6 \times 4.440^3}{60 \times 1{,}500 \times 500} = 1.945 \bullet 10^6$$

$$P_1 = 10^2 \times \sqrt[3]{1.945} = 1250 \text{ lbs.}$$

(e) Combined Axial and Radial Loads

This condition is dealt with by ANSI-AFBMA Standard 9-1978 which defines the combined load to be expressed as:

$$P = C_1\,(X \bullet i \bullet F_r + Y \bullet F_a) \qquad (11)$$

Table 1-5 Shock and Impact Factors

Type of Load	C_1
Constant or steady	1.0
Light shocks	1.5
Moderate shocks	2.0
Heavy shocks	3.0 and up

where value C_1 is a service factor which is shown in Table 1-5.

In the above equation:

i = race rotation factor equal 1 for inner ring rotation, 1.2 for outer ring rotation.

F_r and F_a are radial and axial components, respectively, of the load.

X and Y are factors to be used as shown in Table 1-6.

NOTE: Y is the axial or thrust factor determined from the value of

$$\frac{F_a}{i\,Z\,D^2}$$

Quality Bearings & Components

Table 1-6 Values of X and Y

Bearing Type				Single Row Bearings (Fa/Fr) > e X	Single Row Bearings (Fa/Fr) > e Y	Double Row Bearings (Fa/Fr) ≤ e X	Double Row Bearings (Fa/Fr) ≤ e Y	Double Row Bearings (Fa/Fr) > e X	Double Row Bearings (Fa/Fr) > e Y	e
	$\frac{F_a}{C_o}$	$\frac{F_a}{i\,Z\,D^2}$ Newtons	$\frac{F_a}{i\,Z\,D^2}$ lbf							
	0.014	0.172	25		2.30				2.30	0.19
	0.028	0.345	50		1.99				1.99	0.22
	0.056	0.689	100		1.71				1.71	0.26
	0.084	1.03	150		1.56				1.55	0.28
Radial Contact Groove Ball Bearings	0.11	1.38	200	0.56	1.45	1	0	0.56	1.45	0.30
	0.17	2.07	300		1.31				1.31	0.34
	0.28	3.45	500		1.15				1.15	0.38
	0.42	5.17	750		1.04				1.04	0.42
	0.56	6.89	1000		1.00				1.00	0.44
	$\frac{F_a}{C_o}$	$\frac{F_a}{i\,Z\,D^2}$ Newtons	$\frac{F_a}{i\,Z\,D^2}$ lbf	For this type use the X, Y and e values applicable to single row radial contact bearings.						
	0.014	0.172	25				2.78		3.74	0.23
	0.028	0.345	50				2.40		3.23	0.26
	0.056	0.689	100				2.07		2.78	0.30
	0.085	1.03	150				1.87		2.52	0.34
Angular Contact Groove Ball Bearings with Contact Angle 5°	0.11	1.38	200			1	1.75	0.78	2.36	0.36
	0.17	2.07	300				1.58		2.13	0.40
	0.28	3.45	500				1.39		1.87	0.45
	0.42	5.17	750				1.26		1.69	0.50
	0.56	6.89	1000				1.21		1.63	0.52
	0.014	0.172	25		1.88		2.18		3.06	0.29
	0.029	0.345	50		1.71		1.98		2.78	0.32
	0.057	0.689	100		1.52		1.76		2.47	0.36
	0.086	1.03	150		1.41		1.63		2.20	0.38
10°	0.11	1.38	200	0.46	1.34	1	1.55	0.75	2.18	0.40
	0.17	2.07	300		1.23		1.42		2.00	0.44
	0.29	3.45	500		1.10		1.27		1.79	0.49
	0.43	5.17	750		1.01		1.17		1.64	0.54
	0.57	6.89	1000		1.00		1.16		1.63	0.54
	0.015	0.172	25		1.47		1.65		2.39	0.38
	0.029	0.345	50		1.40		1.57		2.28	0.40
	0.058	0.689	100		1.30		1.46		2.11	0.43
	0.087	1.03	150		1.23		1.38		2.00	0.46
15°	0.12	1.38	200	0.44	1.19	1	1.34	0.72	1.93	0.47
	0.17	2.07	300		1.12		1.26		1.82	0.50
	0.29	3.45	500		1.02		1.14		1.66	0.55
	0.44	5.17	750		1.00		1.12		1.63	0.56
	0.58	6.89	1000		1.00		1.12		1.63	0.56
20°				0.43	1.00	1	1.09	0.70	1.63	0.57
25°				0.41	0.87	1	0.92	0.67	1.41	0.68
30°				0.39	0.76	1	0.78	0.63	1.24	0.80
35°				0.37	0.66	1	0.66	0.60	1.07	0.95
40°				0.35	0.57	1	0.55	0.57	0.98	1.14
Self-aligning Ball Bearings				0.40	0.40 cot ∞	1	0.42 cot ∞	0.65	0.65 cot ∞	1.5 tan α

(1) 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.
(2) Values of X, Y and e for a load or contact angle other than shown in Table 5-5 are obtained by linear interpolation.
(3) Values of X, Y and e shown in Table 5-5 do not apply to filling slot bearings for applications in which ball-raceway contact areas project substantially into the filling slot under load.
(4) For single row bearings, when Fa/Fr ≤ e, use X = 1 and Y = 0.

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Example 4

For a bearing dealt with in Example 2, assume that it carries a combined load of 400 lbs radially and 300 lbs axially at 1200 rpm. The outer ring rotates, and the bearing is subjected to moderate shock. Find the rating life of this bearing in hours.

Solution:

$$\frac{F_a}{i\,Z\,D^2} = \frac{300}{9 \times 0.4375^2} = 174$$

$Y = 1.50$

$C_1 = 2$

$P = 2(0.56 \times 1.2 \times 400 + 1.5 \times 300) = 1440$ lbs equivalent radial load

$$N = \frac{10^6\,C^3}{P^3} = 60\,n\,L$$

$$L = \frac{10^6\,C^3}{60\,n\,P^3} = \frac{10^6 \times 4440^3}{60 \times 1200 \times 1440^3} = 410 \text{ hr,}$$ life with 90 percent probability that it will be attained or exceeded.

NOTE: The impact load on a bearing should not exceed the static capacity as given by Table 1-4 or the race may be damaged by Brinelling from the balls. This load may be exceeded somewhat if the bearing is rotating and the duration of the load is sufficient for the bearing to make one or more complete revolutions while the load is acting.

Example 5

What change in the loading of a ball bearing will cause the expected life to be doubled?

Solution:

Let N_1 and P_1 be the original life and load for the bearing. Let N_2 and P_2 be the new life and load.

Then: $N_2 = 2N_1$

By Equation (9):

$$P_2^3 = \frac{N_1\,P_1^3}{N_2} = \frac{N_1\,P_1^3}{2N_1} = 0.5\,P_1^3$$

$$P_2 = \sqrt[3]{0.5} \times P_1 \quad \text{or}$$

$$P_2 = 0.794\,P_1$$

Hence a reduction of the load to 79 percent of its original value will cause a doubling of the expected life of a ball bearing.

(f) Variable Loading of Bearings

Ball bearings frequently operate under conditions of variable load and speed. Design calculations should take into account all portions of the work cycle and should not be based solely on the most severe operating conditions. The work cycle should be divided into a number of portions in each of which the speed and load can be considered as constant.

Suppose P_1, P_2,... are the loads on the bearing for successive intervals of the work cycle. Let

N_1 be the life of the bearing, in revolutions, if operated exclusively at the constant load P_1. Let there be N_1' applications of load P_1. Then N_1'/N_1 represents the proportion of the life consumed in this portion of the cycle.

Let N_2 be the life of the bearing, in revolutions, if operated exclusively at load P_2. Let there be N_2' applications of load P_2. Then N_2'/N_2 represents the proportion of the life consumed by load P_2.

A corresponding statement can be made for each portion of the work cycle. The sum of these proportions represents the total life of the bearing or unity. Then:

$$\frac{N_1'}{N_1} + \frac{N_2'}{N_2} + \frac{N_3'}{N_3} + \ldots = 1 \qquad (12)$$

Let N_c be the life of the bearing under the combined loading. Let $N_1' = \alpha_1 N_c$ where α_1 represents the proportion of the total life, consumed under load P_1. In a similar way, $N_2' = \alpha_2 N_c$, $N_3' = \alpha_3 N_c$, and so on. Substitution in Equation (12) yields:

$$\frac{\alpha_1}{N_1} + \frac{\alpha_2}{N_2} + \frac{\alpha_3}{N_3} + \ldots = \frac{1}{N_c}$$

Using Equation (10):

$$N_1 = \frac{10^6\, C^3}{P_1^{\,3}}, \quad N2 = \frac{10^6\, C^3}{P_2^{\,3}}, \ldots \text{ and so on.}$$

Combining these last two equations we can obtain:

$$\frac{1}{N_c} = \frac{\alpha_1\, P_1^{\,3}}{10^6\, C^3} + \frac{\alpha_2\, P_2^{\,3}}{10^6\, C^3} + \ldots, \text{ or multiplying both sides of the equation by } 10^6\, C^3$$

$$\frac{10^6\, C^3}{N_c} = \alpha_1\, P_1^{\,3} + \alpha_2\, P_2^{\,3} + \ldots \qquad (13)$$

From previous definition of α it is obvious that $\alpha_1 + \alpha_2 + \ldots$ must equal unity. The application of this equation will be demonstrated by the following examples.

Example 6

A ball bearing is to operate on the following work cycle:

Radial load of 1400 lbs at 200 rpm for 25% of the time
Radial load of 2000 lbs at 500 rpm for 20% of the time
Radial load of 800 lbs at 400 rpm for 55% of the time

Total rpm is to be 1100.

Additional conditions:

The inner ring rotates; loads are steady. Find the minimum value of the basic rating load C for a suitable bearing for this application if the required life is 7 years at 4 hours per day.

Since both the load as well as the speed for the particular load varies, we have to establish the actual work cycle per minute.

	Assumed interval, min	*rpm*	*In assumed interval, rev.*
P_1 = 1400 lbs	0.25	200	50
P_2 = 2000 lbs	0.20	500	100
P_3 = 800 lbs	0.55	400	220
	1.00		370 rpm

The following table should be constructed:

Then $\alpha_1 = \frac{50}{370}, \ \alpha_2 = \frac{100}{370}, \ \alpha_3 = \frac{220}{370}$

A working year is assumed to consist of 250 days.

Total life duration of the bearing expressed in hours will become 7 x 250 x 4 = 7000 hours, whereas this expressed in number of revolutions becomes:

$$N_c = 7000 \times 60 \times 370 = 1554 \times 10^5 \text{ revolutions.}$$

Inputing this data in the formula (13), previously derived in **1.5 (f)**:

$$\frac{10^6\ C^3}{N_c} = \alpha_1 P_1^3 + \alpha_2 P_2^3 + \alpha_3 P_3^3 ...,$$

we obtain: $\frac{50}{370} \times 1400^3 + \frac{100}{370} \times 2000^3 + \frac{220}{370} \times 800^3 = (3708 + 21622 + 3044) \times 10^5$

$$\frac{10^6\ C^3}{N_c} = 28374 \times 10^5$$

$$\frac{C^3}{N_c} = 2837.4$$

$$C^3 = 2837.4 \times N_c = 2837.4 \times 1554 \times 10^5 = 44093 \times 10^7$$

$$C = 7610 \text{ lbs}$$

In order to choose the appropriate bearing, we refer to Table 1-4 from which we find that a bearing such as No. 308 should be satisfactory, keeping in mind there is but a 90 percent probability that the required life will be attained or exceeded.

Example 7

A 306 radial ball bearing with inner ring rotation has a 10-sec work cycle as follows:

For 2 seconds	**For 8 seconds**
F_r = 800 lbs	F_r = 600 lbs
F_a = 400 lbs	F_a = 0 lbs
n = 900 rpm	n = 1200 rpm
Light shock	*Steady load*

Find the rating life of this bearing in hours and in years of 250 working days of 2 hours each.

Solution:

Since the bearing chosen is No. 306, from Table 1-4:

Z = 8, D = 0.5 and i = 1.

$$\frac{F_a}{i\ Z\ D^2} = \frac{400}{1 \times 8 \times 0.5^2} = 200$$

From Table 1-6 for this value of 200, a value for Y will be 1.45 and X will be 0.56.

From Equation (11) and Table 1-5, for the combined axial and radial loads with light shock and 2-second duration:

$$P_1 = C_1 (X\ i\ F_r + Y\ F_a) = 1.5\ (0.56 \times 1 \times 800 + 1.45 \times 400)$$

$$P_1 = 1542 \text{ lbs (equivalent radial load)}$$

Since P_2 is a pure radial load:

$$P_2 = F_r = 600 \text{ lbs}$$

The number of revolutions for the 2-second time duration will be:

$$\frac{900}{60} \times 2 = 30$$

whereas for the 8-second time duration will be:

$$\frac{1200}{60} \times 8 = 160$$

The combined total number of revolutions in 10 seconds is:

$$30 + 160 = 190$$

then,

$$\alpha_1 = \frac{30}{190} = \frac{3}{19}, \qquad \alpha_2 = \frac{160}{190} = \frac{16}{19}$$

From formula (13)

$$\frac{10^6\ C^3}{N_c} = \alpha_1\ P_1^3 + \alpha_2\ P_2^3$$

Using C = 5120 in Table 1-4 for bearing No. 306:

$$\frac{10^6 \times 5120^3}{N_c} = \frac{3}{19} \times 1542^3 + \frac{16}{19} \times 600^3 = 578.9 \times 10^6 + 181.9 \times 10^6 = 760.8 \times 10^6$$

$$N_c = \frac{5120^3}{760.8} = \frac{134218 \times 10^6}{760.8} = 176 \times 10^6 = L_{10} \times 10^6$$

This is the number of revolutions the bearing will endure. The total number of revolutions during the 10-second operation was established as being 190. Therefore, the number of revolutions per minute will be:

190

$$n = \frac{}{10} \times 60 = 1140 \text{ rpm}$$

From Equation (8):

$$L = \frac{10^6 L_{10}}{60 \times n} = \frac{175 \times 10^6}{60 \times 1140} = 2558 \text{ hours}$$

This expressed in years of operation will become

$$\frac{2558}{2 \times 250} = 5.12 \text{ years}$$ of life with 90 percent probability of service, assuming 2 hours of operation per day

(g) Static Loading of Bearings

Up to this point we have been dealing with dynamic loading of bearings. This is the condition when there is relative motion between the rings of the bearings and the balls that are rotating. If this is not the case, as a result of static concentrated loads of the balls against the races, the depressions of the balls into the races will gradually enlarge, and permanent indentations will remain. The static capacity is ordinarily defined as the maximum allowable static load that does not impair the running characteristics of the bearing to make it unusable.

This permanent deformation under the balls is known as Brinnelling and takes place at moderate to high loads. The magnitude of the permissible load is found by methods given in the standards. Calculations for the bearings of Table 1-4 have been made and are shown in the column headed P_{st}.

When very smooth and quiet operation is required, the loading should be no more than about one-half the static capacity.

Back and forth rotation of the shaft through small angles can cause early failure of bearings unless the load is very light. Lubrication is difficult because the oil or grease may not be replenished back of a ball or roller before the motion is reversed.

(h) Effect of Increased Confidence Levels

When a bearing is installed there is no way of knowing whether it is one of the 90 percent that are good or one of the 10 percent that will not attain the rating life. In other words, one can have but 90 percent confidence that the bearing will achieve or exceed its rating life, usually designated L_{10}.

In some cases a greater degree of reliability is required. The expected life will of course be reduced as the reliability is made higher. Let an adjusting factor a_1 be taken such that life L_n is equal to $a_1 L_{10}$. Factors a_1 for different values of the reliability are given in Table 1-7. Life L_{10} is the rating life.

Table 1-7 Constants for Designing at Different Confidence Levels

Reliability (%)	L_n	Life Adjustment Factor, a_1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

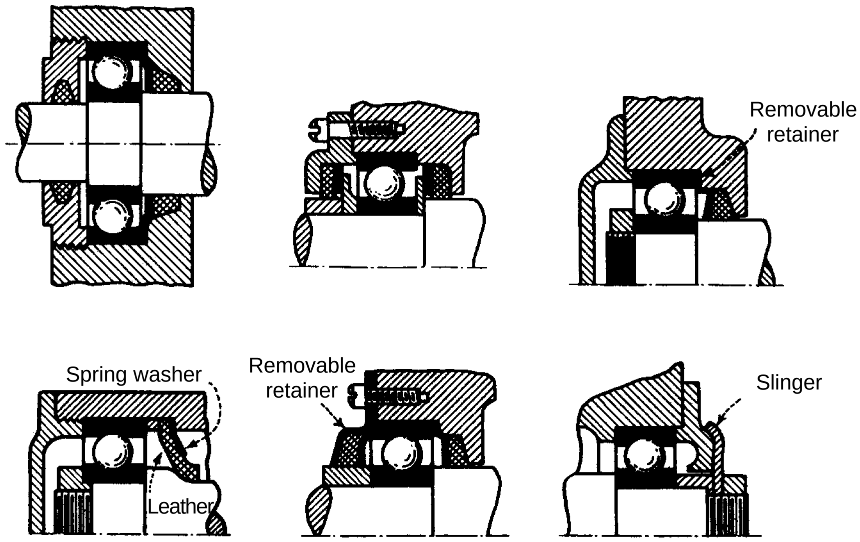

Fig. 1-8

1.6 Mounting of Ball Bearings

For a rotating shaft, relative rotation between shaft and bearing is usually prevented by mounting the inner ring with a press fit and securing it with a nut threaded on the shaft. Excessive interference of metal must be avoided in press fits, or the stretching of the inner ring may decrease the small but requisite internal looseness of the bearing.

The tolerances for shafts and housings as a function of their respective sizes are given in Tables 1-8 and 1-9. Please note that the nominal sizes are given in millimeters, however, the tolerances themselves are given in inches.

Although the outer ring, when the shaft rotates, is mounted more loosely than the inner ring, rotational creep between the ring and housing should be prevented. When two bearings are mounted on the same shaft, the outer ring of one of them should be permitted to shift axially to care for any differential expansion between shaft and housing. Several examples of typical mounting details with oil retainers are shown in Fig. 1-8. The catalogs of the various manufacturers contain useful illustrations of this kind, as well as other practical information.

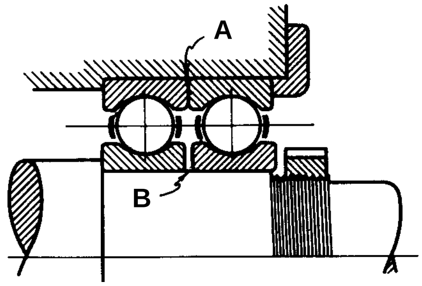

Fig. 1-9 Method for obtaining preloading in ball bearings

Shafts or spindles in machine tools and precision equipment that must rotate without play or clearance in either the radial or axial directions can be mounted on preloaded ball bearings. The preloading, which removes all play from the bearing, can be secured in a number of different ways. For example, suppose the outer rings of the bearings at A in Fig. 1-9 project a small but controlled amount beyond the inner rings. When the inner rings are brought into contact at B by means of the locknut, the

Table 1-8 Deviation of Shaft Diameters from Nominal Dimensions (inches)

Fit inner ring to shaft		Push fit	Push fit to wringing fit	Wringing fit	Drive fit		Light force fit		Force fit		Heavy force fit	
Nominal diam (mm)		g6	h6	h5	j5	j6	k5	k6	m5	m6	n6	p6
Over	Incl.											
3	6	–0.0002 –0.0005	0 –0.0003	0 –0.0002	+0.0002 –0.0000							
6	10	–0.0002 –0.0006	0 –0.0004	0 –0.0002	–0.0002 –0.0006	–0.0003 –0.0001						
10	18	–0.0002 –0.0007	0 –0.0004	0 –0.0003	–0.0002 –0.0007	–0.0003 –0.0001	–0.0004 –0.0000	–0.0005 –0.0000				
18	30	–0.0003 –0.0008	0 –0.0005	0 –0.0004	–0.0003 –0.0008	–0.0004 –0.0002	–0.0004 –0.0001	–0.0006 –0.0001	–0.0007 –0.0003	–0.0008 –0.0003	–0.0011 –0.0006	
30	50	–0.0004 –0.0010	0 –0.0006	0 –0.0004	–0.0004 –0.0010	–0.0004 –0.0002	–0.0005 –0.0001	–0.0007 –0.0001	–0.0008 –0.0004	–0.0010 –0.0004	–0.0013 –0.0007	–0.0017 –0.0010
50	80	–0.0004 –0.0011	0 –0.0007	0 –0.0005	–0.0004 –0.0011	–0.0005 –0.0003	–0.0006 –0.0001	–0.0008 –0.0001	–0.0009 –0.0004	–0.0012 –0.0004	–0.0015 –0.0008	–0.0020 –0.0013
80	120	–0.0005 –0.0013	0 –0.0009	0 –0.0006	–0.0005 –0.0013	–0.0005 –0.0004	–0.0007 –0.0001	–0.0010 –0.0001	–0.0011 –0.0005	–0.0014 –0.0005	–0.0018 –0.0009	–0.0023 –0.0015
120	180	–0.0006 –0.0015	0 –0.0010	0 –0.0007	–0.0006 –0.0015	–0.0006 –0.0004	–0.0008 –0.0001	–0.0011 –0.0001	–0.0013 –0.0006	–0.0016 –0.0006	–0.0020 –0.0011	–0.0027 –0.0017

Table 1-9 Deviation of Housing Bores from Nominal Dimensions (inches)

Fit inner ring to shaft		Close running fit	Slide fit		Push fit		Wringing fit		Drive fit		Heavy drive fit		Light force fit	
Nominal diam (mm)		G7	H8	H7	J7	J6	K6	K7	M6	M7	N6	N7	P6	P7
Over	Incl.													
10	18	–0.0002 –0.0009	0 –0.0011	0 –0.0007	–0.0003 –0.0004	–0.0002 –0.0002	–0.0004 –0.0001	–0.0005 –0.0002	–0.0006 –0.0002	–0.0007 0	–0.0008 –0.0004	–0.0009 –0.0002	–0.0010 –0.0006	–0.0011 –0.0004
18	30	–0.0003 –0.0011	0 –0.0013	0 –0.0008	–0.0004 –0.0005	–0.0002 –0.0003	–0.0004 –0.0001	–0.0006 –0.0002	–0.0007 –0.0002	–0.0008 0	–0.0009 –0.0004	–0.0011 –0.0003	–0.0012 –0.0007	–0.0014 –0.0006
30	50	–0.0004 –0.0013	0 –0.0015	0 –0.0010	–0.0004 –0.0006	–0.0002 –0.0004	–0.0005 –0.0001	–0.0007 –0.0003	–0.0008 –0.0002	–0.0010 0	–0.0011 –0.0005	–0.0013 –0.0003	–0.0015 –0.0008	–0.0017 –0.0007
50	80	–0.0004 –0.0016	0 –0.0018	0 –0.0012	–0.0005 –0.0007	–0.0002 –0.0005	–0.0006 –0.0002	–0.0008 –0.0004	–0.0009 –0.0002	–0.0012 0	–0.0013 –0.0006	–0.0015 –0.0004	–0.0018 –0.0010	–0.0020 –0.0008
80	120	–0.0005 –0.0019	0 –0.0021	0 –0.0014	–0.0005 –0.0009	–0.0002 –0.0006	–0.0007 –0.0002	–0.0010 –0.0004	–0.0011 –0.0002	–0.0014 0	–0.0015 –0.0004	–0.0018 –0.0004	–0.0020 –0.0012	–0.0023 –0.0009
120	180	–0.0006 –0.0021	0 –0.0025	0 –0.0016	–0.0006 –0.0010	–0.0003 –0.0007	–0.0008 –0.0002	–0.0011 –0.0005	–0.0013 –0.0003	–0.0016 0	–0.0018 –0.0008	–0.0020 –0.0005	–0.0024 –0.0014	–0.0027 –0.0011
180	250	–0.0006 –0.0024	0 –0.0028	0 –0.0018	–0.0006 –0.0012	–0.0003 –0.0009	–0.0009 –0.0002	–0.0013 –0.0005	–0.0015 –0.0003	–0.0018 0	–0.0020 –0.0009	–0.0024 –0.0006	–0.0028 –0.0016	–0.0031 –0.0013
250	315	–0.0007 –0.0027	0 –0.0032	0 –0.0020	–0.0006 –0.0014	–0.0003 –0.0010	–0.0011 –0.0002	–0.0014 –0.0006	–0.0016 –0.0004	–0.0020 0	–0.0022 –0.0010	–0.0026 –0.0006	–0.0031 –0.0019	–0.0035 –0.0014
315	400	–0.0007 –0.0030	0 –0.0035	0 –0.0022	–0.0007 –0.0015	–0.0003 –0.0011	–0.0011 –0.0003	–0.0016 –0.0007	–0.0018 –0.0004	–0.0022 0	–0.0024 –0.0010	–0.0029 –0.0006	–0.0034 –0.0020	–0.0039 –0.0016
400	500	–0.0008 –0.0033	0 –0.0038	0 –0.0025	–0.0008 –0.0017	–0.0003 –0.0013	–0.0013 –0.0003	–0.0018 –0.0007	–0.0020 –0.0004	–0.0025 0	–0.0026 –0.0011	–0.0031 –0.0007	–0.0037 –0.0022	–0.0043 –0.0018
500	630	–0.0009 –0.0035	0 –0.0041	0 –0.0027	–0.0009 –0.0018	–0.0003 –0.0014	–0.0014 –0.0003	–0.0019 –0.0008	–0.0022 –0.0005	–0.0027 0	–0.0029 –0.0012	–0.0034 –0.0007	–0.0041 –0.0024	–0.0046 –0.0020

balls will be displaced in the rings an amount sufficient to remove all looseness from the bearing. Close attention must be paid to dimensions and tolerances to secure just enough projection of the ring to remove the play, but not so much as to induce excessive pressure or binding of the balls. The bearing at the other end of the shaft must be arranged for free axial movement of the outer ring. The bearings in Fig. 1-9 can be separated if desired with one bearing at each end of the shaft. Although this arrangement will remove the looseness from both ends of the shaft, serious stresses may be induced by a temperature difference between shaft and housing. Preloaded, double-row radial bearings are made by some manufacturers.

1.7 Unground Ball Bearings

The foregoing discussion has referred to ball bearings of the highest quality of materials and workmanship. Other bearings of lower quality can be purchased for installations requiring less accuracy or where cost is the controlling factor. The rings are made on automatic screw machines and are hardened but not ground.

Different types of construction are in use. The bearing of Fig. 1-10(a) has the outer ring split by a plane perpendicular to the axis. The bearing is assembled by spinning the edges of the bushing, which is slipped over the outer rings. The bearing of Fig. 1-10(b) has a split inner ring, and is made by staking the bore of the inner ring as shown. Various additional features, such as pulleys, gears, castor wheels, and so on, can be incorporated as an integral part of the outer ring. Fig. 1-10 (c) shows a sheave-idler in which the outer ring is formed by the stampings comprising the sheave. Unground ball bearings are frequently cheaper than an equivalent plain bushing.

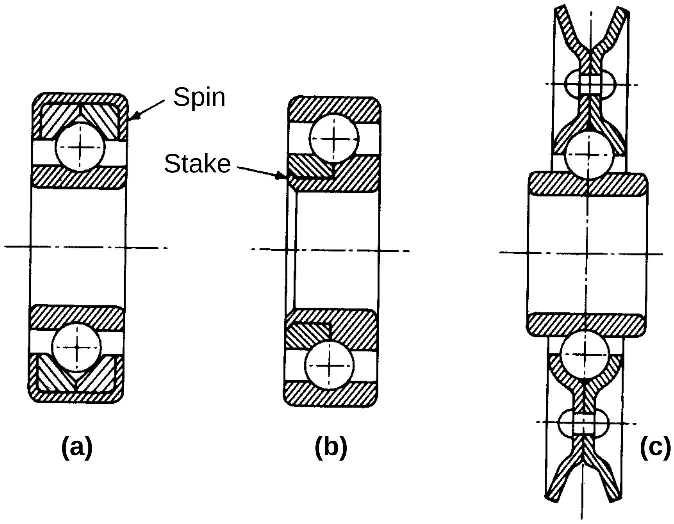

Fig. 1-10 Types of Unground Ball Bearings

1.8 Roller Bearings

Several roller bearings are shown in Fig. 1-2 as well as in Fig. 1-11. These types of bearings are usually used when shock and impact loads are present, or when large bearings are needed.

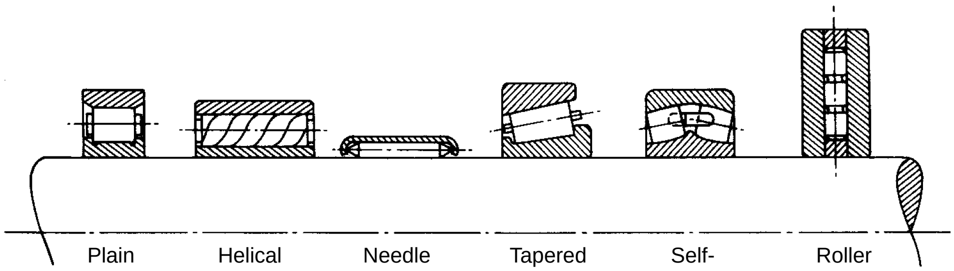

Fig. 1-11 Types of Roller Bearings

A roller bearing in general consists of the same four elements as a ball bearing: the two rings, the cage, and the rollers. Some typical examples of roller bearings are shown in Fig. 1-11. Means of mounting roller bearings are shown in Fig. 1-12.

In a plain roller bearing, the flanges on the rings serve to guide the rollers in the proper direction. When the flanges are omitted from one of the rings, as shown in Fig. 1-11, the rings can then be displaced axially with respect to each other, and no thrust component can be carried.

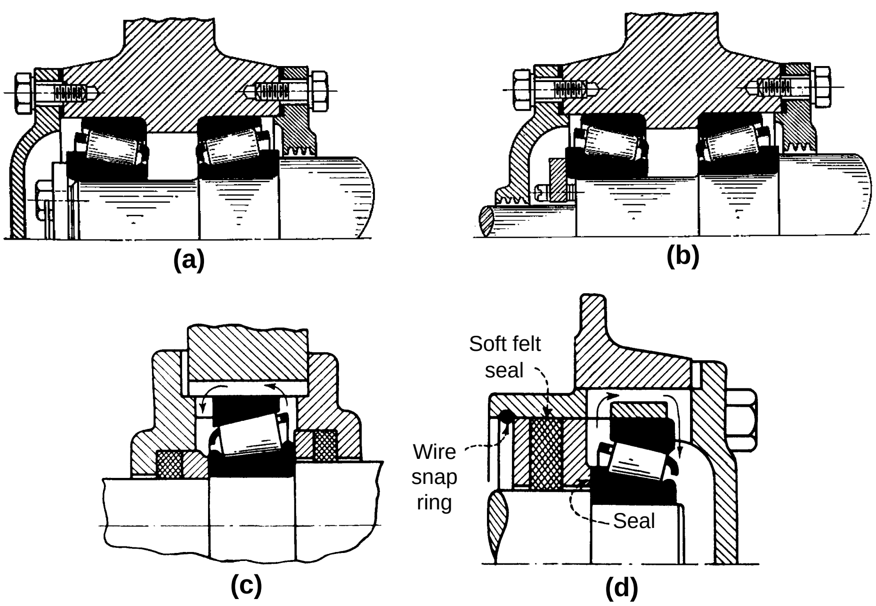

Fig. 1-12 Details of Mounting of Roller Bearings

In addition to the radial load, the tapered roller bearing can carry a large axial component whose magnitude depends on the angularity of the rollers. The radial load will also produce a thrust component. The outer ring is separable from the remainder of the bearing. In this type of bearing, it is possible to make adjustment for the radial clearance. Two bearings are usually mounted opposed to each other, and the clearance is controlled by adjusting one bearing against the other. Double-row tapered roller bearings are also available.

Roller bearings in general can be applied only where the angular misalignment caused by shaft deflection is very slight. This deficiency is not present in the spherical roller bearing. It has excellent load capacity and can carry a thrust component in either direction.

In the helical roller bearing, the rollers are wound from strips of spring steel, and afterwards are hardened and ground to size. If desired, the rollers can bear directly on the shaft without an inner ring, particularly if the shaft surface has been locally hardened. This bearing has been successfully applied under conditions of dirty environment.

Fig. 1-13 Self-Aligning Spherical Roller Bearing for Radial and Thrust Loads

The needle bearing has rollers that are very long as compared to their diameters. Cages are frequently not used, and the inner ring may or may not be present. The outer ring may consist of hardened thin-walled metal as shown in Fig. 1-13; the housing in which the bearing is mounted must have sufficient thickness to give adequate support. The friction of needle bearings is several times as great as for ordinary cylindrical roller bearings. Because of the tendency of the unguided rollers to skew, needle bearings are particularly adapted to oscillating loads, as in wrist pins, rocker arms, and universal joints. For continuous rotation, needle bearings are usually suitable where the loading is intermittent and variable so that the needles will be frequently unloaded and thus tend to return to their proper locations. When the application involves angular misalignment of the shaft, two short bearings end to end usually are better than one bearing with long rollers. The needle bearing is low-priced and requires very little radial space.

Spherical roller bearings, Figures 1-12 and 1-13, can be used when the shaft has angular misalignment.

Thrust bearings can be constructed by the use of straight or tapered rollers.

Roller bearings are selected by a process similar to that used for ball bearings. They must be chosen, however, in accordance with the recommendations given in the catalog of the manufacturer of the particular type of bearing under consideration.

Roller bearings are usually made of case-hardened steels. The carburized case or exterior should have a hardness of 58-63 R_c. The core is softer with a hardness of 25-40 R_c. Certain plain-carbon and alloy steels have been found suitable for roller bearing service. The maximum temperature is limited to about 350° F.

The separator, cage, or retainer for conventional bearings is usually a stamping of low-carbon steel. For higher speeds or precision service, the separator is machined from a suitable copper alloy, such as bronze. Cages are also made of a solid lubricant material for use where conventional types of lubrication cannot be used.

1.9 Lubrication and Surface Finishes

Rolling contact bearings have to be lubricated in addition to having exceedingly good surface finishes.

The life of a rolling element bearing depends to a large extent on the smoothness of the contacting surfaces – the balls, rollers, and races. Typical surface roughness dimensions for production bearings are as follows:

Balls	2– 3 µin. rms
Ball races	6–10 µin. rms
Rollers	8–12 µin. rms
Roller races	10–20 µin. rms

These are in terms of microinches or millionths of an inch, usually written µin.

The unit of measurement of the surface roughness is rms which stands for “root-mean-square height”. This value is obtained by drawing a diamond point instrument over the surface with a magnified readout. These measurements are taken at equidistant points on the profile, squaring these values, adding them, dividing the sum by the number of readings taken and taking the square root of this average.

There are calibrated specimens available and surface roughness can be established by comparison to the specimen.

Surface finishes of bearings vary considerably from manufacturer to manufacturer. They are usually not given specifically for each product.

As far as lubrication is concerned, in general, the application environment will usually dictate the proper lube required. Today’s lubrication selection has varied greatly over the past few years. Modern methods of mixing, compounding and blending various additives and bases has become a very exact science, a far cry from late 1940 when almost all lubricants were a refined petroleum product.

Operational conditions such as temperature, loads, speed, environment and torque available, will determine what type should be used – oil, grease or dry films. Oil fluid is the base lubricant for nearly all bearings, whereas grease is an oil that has been thickened. The use of lubrication will reduce friction and wear, prevent corrosion or oxidation and help to prevent heat buildup within the bearing. Other benefits that result from proper lubrication are quietness, lower torque and extended life. Lubrication selection is very important to good bearing performance. The following tables of lubrications shown are the most widely used by bearing users today. Due to the constant change of product demand and scientific technology, we recommend that a QBC engineer be consulted if you cannot locate a suitable lubrication in the following charts.

Unless otherwise specified by the customer, QBC will supply bearings with an oil lubrication meeting military specifications (MIL-L-6085A) or grease meeting (MIL-G-23827A).

The numbering system developed by QBC incorporates a lubrication code. This gives the user an opportunity to specify the lubricant required as per Table 1-10. In case the code numbers assigned do not cover the lubricant required, Tables 1-11 thru 1-13 list the lubricants available on special orders.

Quality Bearings & Components

Table 1-10 Available Lubricants

Lubricant Code	Brand Name	Basic Type Oil	Operating Temp. °F	Uses
01	*Windsor L245X (MIL-L-6085A)	Synthetic oil	–65 to +300	Light general purpose instrument oil
15	DuPont Krytox 143 AC	Fluorinated oil	–30 to +550	High temperature stability with good lubricity properties
49	AeroShell #7 (MIL-G-23827A)	Diester	–100 to +300	Wide temperature range; good general purpose grease
54	Texaco Low Temp EP (MIL-G-23827A)	Synthetic Ester	–65 to +250	Low torque at cold temperature
20	*Exxon Beacon 325	Synthetic grease	–65 to +250	General purpose grease
39	*Exxon Andok C	Channeling petroleum grease	–20 to +250	Smooth running, long life with minimum migration
13	Toray SH44M	Silicone grease	–25 to +350	Higher temperature stability
48	*Mobil 28 (MIL-G-81322)	Synthetic hydrocarbon	–65 to +350	Wide temperature range, good low temperature torque, general purpose
72	Multemp PS No. 2	Petroleum grease	–60 to +250	Low torque, general purpose grease
75	Chevron SRI-2	Mineral grease	–20 to +350	High speed, high load grease
83	*Shell Alvania X2	Mineral grease	–30 to +250	Long life
10	DuPont Krytox 240AC (MIL-G-27617)	Fluorinated grease	–30 to +550	High temperature stability with good lubricity properties
12	KYODO SRL	Synthetic grease	–40 to +300	Low noise and low torque applications
25	NIG-ACE W	Synthetic grease	0 to +300	Low noise and low torque applications
40	Isoflex JL 032R	Synthetic grease	–60 to +250	High speed, low torque grease
04	U-1494	Synthetic grease	–40 to +350	High speed, high load applications

*Most popular and readily available lubrication.
If no lubrication is called out, QBC will ship bearings with one of these general purpose lubricants.

Quality Bearings & Components

Table 1-11 Oil Lubricants Available on Special Order

Manufacturer & Trade Name	Mil. Spec	Oper. Range, °F	Type	Pouring Point, °F	Flash Point, °F	Viscosity CS 75°F/210°F
Anderson Oil Co.						
LS252	MIL-17353A	–65/250	Diester	–75	340	7.6/1.9
Bendix Corp.						
P10	MIL-L-6085A	–70/350	Diester	–80	420	23.4/3.8
Bray Oil Co.						
NPT3A		–65/175	Diester	–90	400	19/3.5
885	MIL-L-6085	–50/400	Diester	–85	410	1875/9
NPT9		–30/350	Ester	–50	495	710/55
Dow Corning						
DC200	VVL1078	–40/550	Silicone	–50	600	Various
DC510	MIL-L-27694	–70/500	Silicone	–80	600	Various
DC550		–40/450	Silicone	–50	600	125/20
FS1265		–50/300	Silicone	–30	500	Various
DuPont, E.I.						
Krytox 143 AB		–45/450	Perfluor	–45	500	85/10.3
Exxon Corp.						
P15A	MIL-L-7808	–65/300	Diester	–75	450	22/3.5
Aviation Inst. Oil	MIL-L-7870	–65/290	Petroleum	–70	300	17/2.6
Univis P12	MIL-L-6085A	–75/300	Diester	–90	410	30/3.6
Univis P38	MIL-L-6085	–65/300	Diester	–70	415	72/37
General Electric						
Versilube F44		–100/500	Silicone	–100	550	70/15
Versilube F50	MIL-S-81087	–100/400	Silicone	–100	550	75/22
Versilube SF81		–40/400	Silicone	–55	600	Various
Versilube SF96		–40/400	Silicone	–50	600	40/16.5
Gulf Oil Co.						
Synthetic Fluid#6		–50/275	Mineral	–90	295	3200/12
Houghton Oil						
Cosmolube 270A	MIL-L-6085A	–65/250	Diester	–70	365	15/3.5
Mobil Oil						
SHC824		–50/350	Synthetic	–65	455	100/6.5
XRL743A		–50/350	Synthetic	–65	520	100/6.5
MPS Corp.						
MO119		–30/250	Synthetic	–80	455	119 @ 100°F
Shell Oil Co.						
Aeroshell #3	MIL-L-7870	–70/240	Petroleum	–75	275	16.5/2.3
Aeroshell #12	MIL-L-6085A	–70/300	Diester	–70	365	21.5/3.5
Aeroshell #4	MIL-H-5606	–70/500	Petroleum	–85	215	859/10.4
Tenneco Chemical						
Anderol L401D	MIL-L-6085A	–75/260	Diester	–80	430	19.7/3.4
Anderol L423		–80/350	Synthetic	–100	370	200/5.1

Table 1-12 Greases Available on Special Order

Manufacturer & Trade Name	Mil. Spec	Oper. Range, °F	Base Oil	Thickener	Color
American Oil Co.					
Rykon Premium #2		–10/200	Mineral	Arylurea	Reddish
Rykon Premium #3		–20/250	Mineral	Arylurea	Pink
Supermil ASU31052	MIL-G-25013	–100/450	Silicone	Arylurea	Lavender
Supermil ASU72832	MIL-G-23827A	–100/250	Diester	Litium	Amber
Bray Oil Co.					
Braycote 627S	MIL-G-23827	–100/300	Ester	Organic	Lt. Brown
Braycote 637S	MIL-G-25537	–65/260	Mineral	Calcium Soap	Lt. Brown
601		–100/390	Polyether	Tetrafluor	Off White
Chevron Oil Co.					
BRB-2	MIL-G-3545C	–20/350	Mineral	Polyurea	Blue Green
OHT		+20/300	Mineral	Sodium	Greenish
NRR335		–65/300	Synthetic Aeromatic	Sodium	Maroon
Dow Corning					
Molykote BR2 Plus		–20/300	Mineral	Lithium	Black
Molykote 33		–100/350	Silicone	Lithium	Gray
Molykote 41		–0/550	Silicone	Lithium	Black
Molykote 44	MIL-G-46886A	–100/400	Silicone	Lithium	Dark Amber
Molykote 55M	MIL-G-4343	–65/350	Silicone	Lithium	Tan
DuPont, E.I.					
Krytox 240AA	MIL-G-27617	–30/450	Fluor Carbon	Vidax	White
Krytox 240AB	MIL-G-27617	–30/450	Fluor Carbon	Vidax	White
Krytox 240AZ	MIL-G-27617	–65/300	Fluor Carbon	Vidax	White
Krytox 240AC	MIL-G-27617	–30/550	Perfluor	Tetrafluor	White
Exxon Corp.					
Andok B	MIL-G-18709A	–20/250	Mineral	Sodium	Brown
Andok 260	MIL-G-3545C	–20/250	Mineral	Sodium	Amber
General Electric					
Versilube G351	MIL-L-15719A	–40/400	Silicone	Lithium	Cream
Houghton E.F.					
Cosmolube 615	MIL-L-4343	–65/375	Silicone	Lithium	Lt. Brown
Kyodo Yushi					
PS #2		–60/230	Diester	Lithium	White
Mobil Oil					
BRB #23	MIL-L-7711	–0/250	Petroleum	Sodium	Tan
Mobil 24	MIL-G-25013	–100/550	Silicone	Organic	Reddish
Mobil 27	MIL-G-23827	–65/325	Carbon	Non Soap	Tan

Table 1-13 Greases Available on Special Order

Manufacturer & Trade Name	Mil. Spec	Oper. Range, °F	Base Oil	Thickener	Color
NYE Rheolube					
703A		–30/250	Mineral	Sodium	Tan
716B		–60/300	Polyol Ester	Lithium	Tan
781D		–95/390	Silicone	Lithium	Off White
899RP		–130/480	Fluorether	PTFE	White
2000		–60/260	Hydrocarbon	Organic	Red
Rheo Temp 500	MIL-G-3278A	–65/350	Diester	Sodium	Blue
Shell Oil					
Aeroshell #5	MIL-G-3545C	–20/300	Petroleum	Microgel	Dark Brown
Aeroshell #6	MIL-G-24139	–40/250	Mineral	Microgel	Amber
Aeroshell #7	MIL-G-23827A	–100/300	Diester	Microgel	Amber
Aeroshell #14	MIL-G-23827	–65/250	Mineral	Calcium Soap	Tan
Aeroshell #17	MIL-G-21164	–100/300	Diester	Microgel	Dark Grey
Aeroshell #22	MIL-G-81322A	–80/350	Diester	Microgel	Dark Grey
Alvania #3	MIL-G-81322C	–30/275	Mineral	Lithium	Amber
Cyprina #3	MIL-G-18709	–0/250	Mineral	Lithium	Lt. Tan
Dolium R #2		–30/300	Mineral	Ashless	Amber
Darina	MIL-G-18709	–0/300	Mineral	Microgel	Amber
Royal Lubricant					
Royco 13D	MIL-G-25013	–100/450	Silicone	Teflon	Lavender
Royco 21	MIL-G-7421	–100/250	Diester	Lithium	Brownish
Royco 22MS	MIL-G-81827	–80/360	Diester	Clay	Black
Royco 27A	MIL-G-23827	–100/300	Diester	Lithium	Brownish
Royco 37	MIL-G-25537	–65/250	Mineral	Calcium Soap	Tan
Royco 64C	MIL-G-21164	–65/250	Diester	Lithium	Black
Tenneco Chem. (Huls)					
Anderol 753A		–40/300	Diester	Lithium	Lt. Brown
Anderol 757		–40/300	Diester	Lithium	Lt. Brown
Anderol 761		–40/400	Diester	Silica	Lt. Brown
Anderol 793A	MIL-G-3278A	–65/300	Diester	Lithium	Lt. Amber
Anderol 794		–65/250	Diester	Lithium	Lt. Amber
Anderol 795		–65/300	Diester	Lithium	Off White
Texaco Oil Co.					
Premium RB		–30/325	Mineral	Lithium	Orange
Low Temp EP	MIL-G-23827	–65/250	Synthetic Mineral	Lithium	Purplish Brown
Regal AFB #2	MIL-G-18709	–40/250	Parafin	Lithium	Green
Unitemp 500	MIL-G-3278A	–65/350	Diester	Sodium	Blue

2.0 SINTERED-METAL BEARINGS

2.1 General Properties

Sintered-metal self-lubricating bearings are based on powder-metallurgy technology. They are economical, suitable for high production rates and can be manufactured to precision tolerances.

General properties of porous-metal bearing materials have been described in Machine Design magazine (Vol. 54, #14, June 17, 1982, pp. 131-132), with whose permission the following material is reprinted:

Sintered-metal self-lubricating bearings "are widely used in home appliances, small motors, machine tools, aircraft and automotive accessories, business machines, instruments and farm and construction equipment.

Most porous-metal bearings consist of either bronze or iron which has interconnecting pores. These voids take up to 10% to 35% of the total volume. In operation, lubricating oil is stored in these voids and feeds through the interconnected pores to the bearing surface. Any oil which is forced from the loaded zone of the bearing is reabsorbed by capillary action. Since these bearings can operate for long periods of time without additional supply of lubricant, they can be used in inaccessible or inconvenient places where relubrication would be difficult.

Many variations are possible to meet specific requirements. From 1% to 3.5% graphite is frequently added to enhance self-lubricating properties. High porosity with a maximum amount of lubricating oil is used for high-speed light-load applications, such as fractional-horsepower motor bearings. A low-oil-content low-porosity material with a high graphite content is more satisfactory for oscillating and reciprocating motions where it is hard to build up an oil film.

Powder producers can control powder characteristics such as purity, hydrogen loss, particle size and distribution, and particle shape. Each of these properties in some way affects performance. In the bronze system, for example, shrinkage increases as particle size of tin or copper powder in the mix decreases. Graphite additions result in growth but always lower the strength of the bearings. Lubricants used in the mix have only a slight influence on dimensional change, but a more pronounced effect on the apparent density and flow rate.

After sintering, the bearing must be sized to the specific dimensions. Sizing reduces interconnected porosity and produces greater strength, lower ductility and a smooth finish.

Bronze: The most common porous bearing material. It contains 90% copper and 10% tin. These bearings are wear-resistant, ductile, conformable, and corrosion-resistant. Their lubricity, imbeddability and low cost give them a wide range of applications from home appliances to farm machinery.

Leaded Bronzes: Have a 20% reduction of the tin content of the usual 90-10 bronze and 4% reduction in copper. Lead content is 14% to 16% of total composition and results in a lower coefficient of friction and good resistance to galling in case the lubricant supply is interrupted. These alloys also have higher conformability than the 90-10 bronzes.

Copper-Iron: The inclusion of iron in the composition boosts compressive strength although the speed limit drops accordingly. These materials are useful in applications involving shock and heavy loads, and should be used with hardened shafts.

Hardenable Copper-Iron: The addition of 1.5% free carbon to copper-iron materials allows them to be heat treated to a particle hardness of Rockwell C65. They provide high impact resistance and should be used with hardened and ground shafts.

Iron: Combine low cost with good bearing qualities, widely used in automotive applications, toys, farm equipment, and machine tools. Powdered-iron is frequently blended with up to 10% copper for improved strength. These materials have a relatively low limiting value of PV (on the V side), but have high oil-volume capacity because of the high porosity. They have good resistance to wear, but should be used with hardened and ground steel shafts.

Leaded Iron: Provide improved speed capability, but are still low-cost bearing materials.

Aluminum: In some applications they provide cooler operation, greater tolerance for misalignment, lower weight and longer oil life than porous bronze or iron. The limiting PV value is 50,000, the same as for porous bronze and porous iron.

2.2 Sizing Sintered Bearings

The load-carrying capacity of porous-metal bearings can be measured by a friction/wear criterion, which is a measure of the heat generated by the bearing. It is called the PV factor. The PV factor, as its name implies, is the product of the bearing load, P, expressed in pounds per square inch of projected bearing area, and the surface velocity of the shaft expressed in feet per minute.

If d = inside bearing diameter (in.)
l = length of bearing (in.)
F = bearing load (lbs.)
and N = shaft speed (rpm), then:

$$P = \frac{F}{l d} \quad (\text{lbs/in}^2) \tag{14}$$

$$V = \frac{\pi d N}{12} \quad (\text{ft/min}) \tag{15}$$

and hence,

$$PV = \left(\frac{F}{l\,d}\right)\left(\frac{\pi d N}{12}\right) = \frac{\pi F N}{12\,l} = \frac{0.262\ FN}{l} \tag{16}$$

Most engineering data relating to the PV factor lists an upper limit to the factor; i.e., a value which should not be exceeded for satisfactory bearing operation. The working value of the PV factor, however, is often less than this upper limit, such as in the case where the sliding velocity is not sufficiently high to maintain an adequate lubricating film. In addition, the PV limit is affected by the static load-carrying capacity of the material, which should not be exceeded. The latter is a function of environmental factors, bearing clearances, geometry and the nature of the load (continuous, intermittent or shock loading). Detailed information on these considerations is usually furnished by the metal manufacturer. General guidelines are summarized in Table 2-1.

2.3 Clearances

As in all bearings, satisfactory operation of porous-metal bearings require suitable clearances between shaft and housing. While guidelines depend on the materials used and the nature of the application, a representative chart showing recommended bearing clearances for porous-bronze and porous-iron bearings is given in Figure 2-1.

We carry a full line of both thick and thin wall bushings. Please consult the tables in this section of the handbook for information on *recommended shaft size* and *bore diameter* to be used with various bushing sizes.

Table 2-1 General Guidelines for the PV Factor in Porous-Metal Bearings

Limiting conditions for operation of porous bearings can be expressed as a PV factor. Since P = load, psi; V = surface velocity, fpm; the PV value gives an index of frictional heat generated on a unit area of the bearing surface. A maximum value of 50000 is common for porous bearings. For long-time running with no additional lubrication, 20000 should be a limit in selecting loads for various speeds. For thrust bearings, a maximum PV of 10000 should be used.

Provision to replenish the oil supply is desirable when the PV factor approaches the maximum under continuous operation for extended periods of time, or for high temperatures. For such cases, oil can be applied to the OD or ends of the bearing. From there it is drawn, by capillary action, into the bearing and metered to the shaft. A reservoir of grease next to the bearing also can be helpful.

Material	PV	Static P (psi)	Dynamic P (psi)	V (fpm)
Bronze	50000	8000	2000	1200
Lead-Bronze	60000	3500	800	1500
Copper-Iron	35000	20000	4000	225
Hardenable Copper-Iron	75000	50000	8000	35
Iron	30000	10000	3000	400
Bronze-Iron	35000	10500	2500	800
Lead-Iron	50000	4000	1000	800
Aluminum	50000	4000	2000	1200

Under certain conditions these recommended values can be exceeded but with a sacrifice in service life.

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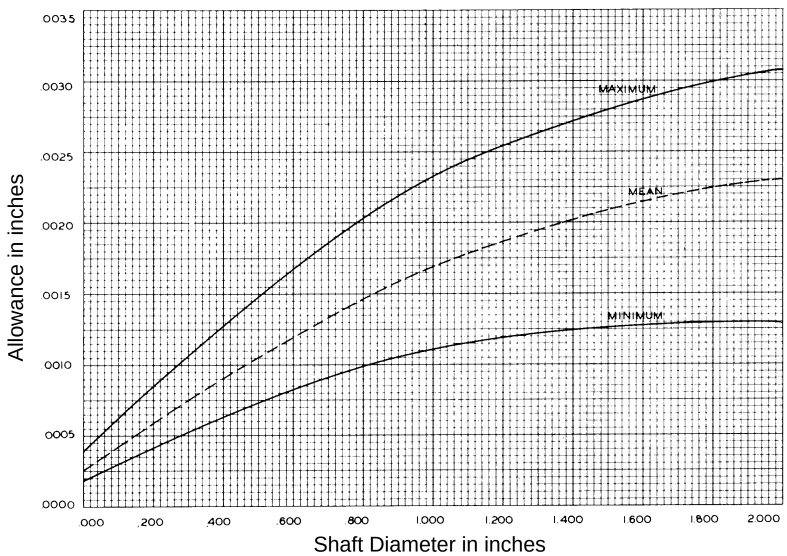


The upper curve (maximum) and all allowances above the mean are suggested for iron-based bearings only. The chart is representative of average conditions, and each application needs to be evaluated individually.

Fig. 2-1 Recommended Bearing Clearances*

* Reprinted with the permission of Keystone Carbon Company, St. Mary's, PA, from *Keystone Porous Bronze and Porous Iron Bearings*, Fig. B-34, p. 9.

SINTERED BEARINGS INSTALLATION DATA

Table 2-2 Thin Wall Bearings

Nominal Hole Size		Hole to Accommodate Bearing		Bearing Outside Diameter		Interference	
Fractional	Decimal	Min.	Max.	Min.	Max.	Min.	Max.
3/16	.1875	.1875	.1885	.1895	.1905	.0010	.0030
1/4	.2500	.2500	.2510	.2520	.2530		
5/16	.3125	.3125	.3135	.3145	.3155		
3/8	.3750	.3750	.3760	.3770	.3780		
7/16	.4375	.4375	.4385	.4395	.4405		
1/2	.5000	.5000	.5010	.5020	.5030		
9/16	.5625	.5625	.5635	.5645	.5655		
5/8	.6250	.6250	.6260	.6270	.6280		
11/16	.6875	.6875	.6685	.6890	.6905		
3/4	.7500	.7500	.7510	.7525	.7535	.0015	.0035
13/16	.8125	.8125	.8135	.8150	.8160		
7/8	.8750	.8750	.8760	.8775	.8785		

Nominal Hole Size		Bearing Hole Size After Close - In		Shaft Size		Clearance	
Fractional	Decimal	Min.	Max.	Min.	Max.	Min.	Max.
1/8	.1250	.1250	.1260	.1235	.1245	.0005	.0025
3/16	.1875	.1875	.1885	.1860	.1870		
1/4	.2500	.2500	.2510	.2485	.2495		
5/16	.3125	.3125	.3135	.3105	.3115	.0010	.0030
3/8	.3750	.3750	.3760	.3730	.3740		
7/16	.4375	.4375	.4385	.4355	.4365		
1/2	.5000	.5000	.5010	.4980	.4990		
9/16	.5625	.5625	.5635	.5605	.5615		
5/8	.6250	.6250	.6260	.6230	.6240		

Table 2-3 Thick Wall Bearings

Nominal Hole Size		Hole to Accommodate Bearing		Bearing Outside Diameter		Interference	
Fractional	Decimal	Min.	Max.	Min.	Max.	Min.	Max.
1/4	.2500	.249	.250	.251	.252	.001	.003
5/16	.3125	.311	.312	.313	.314		
3/8	.3750	.374	.375	.376	.377		

Nominal Hole Size		Bearing Hole Size After Close - In		Shaft Size		Clearance	
Fractional	Decimal	Min.	Max.	Min.	Max.	Min.	Max.
1/8	.1250	.1245	.1255	.1230	.1240	.0005	.0025
3/16	.1375	.1875	.1885	.1860	.1870		
1/4	.2500	.2500	.2510	.2485	.2495		

2.4 Press Fits

A press fit is used when available space and torque to be transmitted is limited. Tolerances of mating parts have to be closely controlled to assure a minimum and avoid all excessive interference.

Formulas for press fit are:

$$p = \frac{eE}{2d}\left[1 - \left(\frac{d}{D}\right)^2\right] \text{ and } T = \frac{\pi}{2}\, p\mu d^2 L,$$

from here

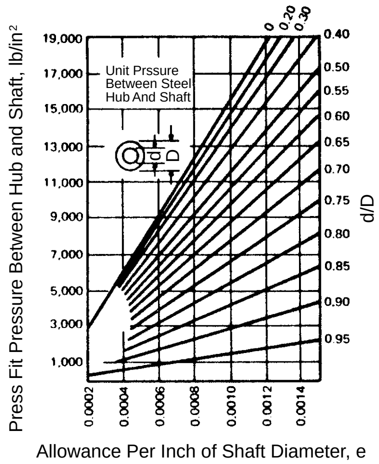
$$T = 0.785\mu dLeE\left[1 - \left(\frac{d}{D}\right)^2\right] \quad \text{(lb in)} \qquad (17)$$

$$S = \frac{2P}{1 - \left(\frac{d}{D}\right)^2} \quad \text{or } S = \frac{eE}{d} \qquad (18)$$

where:

- p = unit pressure on the interfering surfaces (lb/in^2)
- e = amount of interference (in)
- E = modulus of elasticity (psi)
- d = shaft diameter (in)
- μ = coefficient of friction (assume 0.1-0.2)
- L = length of interference surfaces (in)
- S = combined stress resulting from circumferential tension and radial compression (psi)
- T = slip torque (to be divided by safety factor of 2) (lb in)

Graph gives value of p for different d/D ratios and different values of e.



By permission, *Product Engineering*, May 1960

Table 2-4

Running Clearance

Proper running clearance for bearings depends to a great extent on the particular item. Only minimum recommended clearances for oil impregnated bearings used with ground steel shafting are listed.

Running Clearance

Shaft Size in.	Clearance, min. in.	
	Bronze Base	Iron Base
Up to 0.760	0.0005	0.001
0.761 to 1.150	0.001	0.0015
1.511 to 2.510	0.0015	0.002
Over 2.510	0.002	0.0025

Press Fits

Plain cylindrical journal bearings are commonly installed by press fitting the bearing into a housing with an insertion arbor. For housings rigid enough to withstand the press fit without appreciable distortion and for bearings with thickness approximately one-eighth of the bearing outside diameter, the press fits shown are recommended.

Recommended Press Fits

Outside Dia. in.	Press Fit, in.	
	Minimum	Maximum
Up to 0.760	0.001	0.003
0.761 to 1.150	0.0015	0.004
1.511 to 2.510	0.002	0.005
2.511 to 3.010	0.002	0.006
Over 3.010	0.002	0.007

Oil impregnated sintered bearings are manufactured under strictly controlled conditions, and are subjected to in-process inspection. They are tested for radial crushing strength of magnitude:

$$P = \frac{K\,L\,T^2}{D - T} \quad \text{(lbs)}$$

where:

D = O.D. of bearing (in)
T = wall thickness of bearing (in)
L = bearing length (in)
K = strength constant = 22500

Should additional shaft to bearing clearance be required, a ball burnishing operation should be used for the following reasons:

a) to maintain concentricity
b) to maintain surface finish of I.D.
c) to reduce contamination of surface

The required size of the bearing can be determined from equations:

1) $P = \dfrac{W}{L\,d}$ (lbs/in^2) (load on projected bearing areas not to exceed 1000 psi)

2) $V = \dfrac{d\,\pi\,N}{12}$ (ft/min) (surface speed at bearing I.D. not to exceed 1000 ft/min)

3) $PV = \dfrac{W\,N\,\pi}{12L}$ (PV factor – not to exceed 50000)

where:

W = bearing load (lbs)
L = bearing length (in)
N = shaft speed (rpm)
d = bearing I.D. (in)

Above values are reasonable for the following conditions: continuous rotation, oil impregnation without additional lubrication.

2.5 Standardization

American Society for Testing of Materials (ASTM, 100 Ban-Harbor Drive, W. Conchohocken, PA 19428, Tel. 610-832-9500) publishes detailed specifications dealing with Sintered Bronze Bearings. It is designated as B438-83 (published in 1983). The most significant data pertaining to products listed in this catalog can be summarized as follows:

Table 2-5 Material Composition

Material	%
Copper	87.5 – 90.5
Iron	1.0 Max.
Lead	(a)
Carbon (Graphite Max.)	1.75 Max.
Tin	9.5 – 10.5
Zinc	——
Acid Insolubles	——
Total Other Elements	0.5

(a) included in other elements

Table 2-6 Physical & Mechanical Properties

Characteristic	Value
Density (g/cm³)	6.4 – 6.8
Porosity (% by volume)	19 min.
"K" Strength Constant	26500
Tensile Strength (psi)	14000
Elongation (% per in)	1
Yield Strength in Compression (psi)	11000

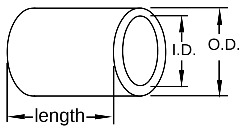
Table 2-7 Miscellaneous Designations

Organization	Designation
ASTM	B-438-83 Grade 1, Type 2
Military	MIL-B-5687D Type 1,Grade 1
MPIF Standard 35	CT-1000-K26
SAE	
New	841
Old	Type 1 Class A
AMS	4805

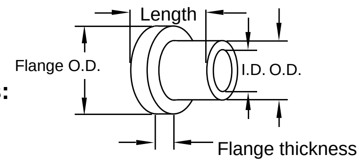
Quality Bearings & Components

Table 2-8 Tolerances of Plain and Flanged Bearings

Plain Bearings:



Flanged Bearings:



	Over (in)	Up to & Including	Tolerance
Inside & Outside Diameters (in)	— 1/2 1 1-1/2 2-1/2 3-1/2	1/2 1 1-1/2 2-1/2 3-1/2 4-1/2	+.000 – .001 +.000 – .001 +.000 – .001 +.000 – .0015 +.000 – .002 +.000 – .0025
Length (in)	— 1-1/2 3	1-1/2 3 4-1/2	± .005 ± .0075 ± .010
Flange Diameter, Based on Flange OD	— 1-1/4 2-1/2 4	1-1/4 2-1/2 4 4-1/2	± .005 ± .010 ± .015 ± .025
Flange Thickness, Based on Flange OD	— 1-1/4	1-1/4 2-1/2	± .0025 ± .005
Flange Fillets, Radii, Based on Body OD	— 1 2 2-1/2	1 2 2-1/2 4	1/32 ± .010 3/64 ± .010 1/16 ± .010 3/32 ± 1/64
Concentricity, ID with Respect to OD (Maximum Total Dial Indicator Reading) Based on ID	— 1 1-1/2 3	1 1-1/2 3 4-1/2	.003 .003 .004 .005

2.6 Conclusion

Sintered bearings are used widely in instruments and general machinery, in which their self-lubricating characteristics and load-carrying ability is very desirable. When properly designed, they can be both economical and highly functional.

Their manufacturing method consists of briquetting the metal powder mixtures to the proper density. Subsequently, they are sintered for different duration subject to the temperatures. Sintered bearings are then sized to obtain the required dimensional characteristics. This is followed by inspection and impregnation with a lubricating oil.

3.0 PLASTIC AND NONMETALLIC BEARINGS

3.1 General Characteristics

Among the significant characteristics of plastic bearings, the following are noteworthy:

- Self-lubricating
- Low wear rates
- Relatively high performance rating (PV) among sleeve bearing materials
- Bearing O.D.'s compatible with standard sintered bronze sizes for upgrading existing equipment
- Kinetic and static coefficient of friction virtually the same under heavy loads
- Extremely low coefficient of friction, as shown in Figure 3-1
- Lightweight
- Ability to conform under load
- Resistance to chemicals

The design characteristics of plastic and nonmetallic bearings bear both similarities and differences relative to those of porous-metal bearings. This will now be described in greater detail.

3.2 Properties of Plastic and Nonmetallic Bearing Materials

Plastics (such as acetal, nylon, PTFE), carbon graphite and other nonmetallic materials have been increasingly used as self-lubricating bearings. Their composition has been refined over many years so as to obtain favorable bearing characteristics. These include low friction, corrosion resistance, ability to conform under load (plastic bearings), ability to function over wide temperature ranges and substantial load-carrying capability. Although temperature ranges, dimensional stability and load limitations of plastic gears are in general less than for metallic bearings, plastic bearings are remarkably versatile and economical.

A summary of characteristics of representative plastic and nonmetallic materials has been given by *Machine Design Magazine* (Vol. 54, #14, June 17, 1982, p. 132) with whose permission the following material is reprinted.

Phenolics: Composite materials consisting of cotton fabric, asbestos, or other fillers bonded with phenolic resin. The good compatibility of the phenolics makes them easily lubricated by various fluids.

They have replaced wood bearings and metals in such applications as propeller and rubber-shaft bearings in ships, and electrical switch-gear, rolling-mill and water-turbine bearings. In small instruments and clock motors, laminated phenolics serve as structural members as well as a bearing material. They have excellent strength and shock resistance, coupled with resistance to water, acid, and alkali solutions.

Some precautions must be observed with phenolic bearings. Thermal conductivity is low, so heat generated by bearing friction cannot readily be transmitted through the bearing liner. Consequently, larger, heavily loaded bearings must have a generous feed of water or lubricating oil to carry away heat. Some swelling and warping of these bearings occurs in the larger sizes, so larger-than-normal shaft clearances are required.

Nylon: Although the phenolics have predominated in heavy-duty applications, they are frequently replaced by nylon, which has the widest use in bearings. Nylon bushings exhibit low friction and require no lubrication. Nylon is quiet in operation, resists abrasion, wears at a low rate, and is easily molded, cast, or machined to close tolerances. Possible problems with cold flow at high loads can be minimized by using a thin liner of the material in a well-supported metal sleeve.

Improvement in mechanical properties, rigidity, and wear-resistance is obtained by adding fillers such as graphite and molybdenum disulfide to nylon. While the maximum recommended continuous service temperature for ordinary nylon is 170°F, and 250°F for heat-stabilized compositions, filled-nylon parts resist distortion at temperatures up to 300°F.

PTFE: Has an exceptionally low coefficient of friction and high self-lubricating characteristics, resistance to attack by almost any chemical, and an ability to operate under a wide temperature range. High cost combined with low load capacity has frequently caused PTFE resin to be selected only in some modified form. PTFE is used as a bearing material in automotive knuckle and ball joints, chemical and food processing equipment, aircraft accessories, textile machinery, and business machines.

Although unmodified PTFE can be used to a PV value of only 1000, PTFE filled with glass fiber, graphite, or other inert materials, can be used at PV values up to 10000 or more. In general, higher PV values can be used with PTFE bearings at low speeds where its coefficient of friction may be as low as 0.05 to 0.1.

One bearing material combines the low friction and good wear resistance of lead-filled PTFE with the strength and thermal conductivity of a bronze and steel supporting structure. A plated steel backing is covered with a thin layer of sintered, spherical, bronze particles. The porous bronze is then impregnated with a mixture of PTFE and lead to provide a thin surface layer. Service temperatures of –330°F to +536°F are possible.

Woven PTFE fabrics are often readily handled and applied. With their resistance to cold flow, they are used as bearings in a wide variety of high-load applications as automotive thrust washers, ball-and-socket joints, aircraft controls and accessories, bridge bearings, and electrical switch gear. To provide a strong bond to either steel or other rigid backing material, a secondary fiber such as polyester, cotton, or glass is commonly interwoven with the PTFE. The woven fabric then is bonded to a steel backing.

Improved versions of this type of bearing have woven or braided “socks” (of PTFE and a bondable material). The bearing sleeve is then filament wound with a fiberglass-epoxy shell. These bearings have been reported to carry dynamic loads as high as 50000 psi.

Acetal: Components made from acetal rod are dimensionally stable even under extremely wet or humid conditions and will not swell like nylon in these conditions. Additionally, it resists most organic solvents. Natural white acetal is an USDA/FDA approved material for food processing applications. Acetal is relatively easy to machine and does not burr easily. Acetal is a generic descriptive name for two polymers: *Celcon®* – a copolymer made by Celanese – and *Delrin®* – a homopolymer made by E. I. DuPont Nemours. Both types are tough enough and strong enough to replace metal for many applications.

Acetron® NS: is a patented acetal-based compound containing special solid lubricants which help provide superior performance in bearing and wear applications. These lubricants are uniformly dispersed in the base acetal, providing a premium, internally lubricated compound with high Pressure Velocity (PV) capabilities, a low coefficient of friction, and an extremely good "k" factor.

Table 3-1 Wear Rate, Coefficient of Friction and Limiting PV Data

Acetal	Wear Factor "k" (1)	Comparative Wear Rate to Acetron® NS	Coefficient of Friction		Limiting PV (4)
			Static (2)	Dynamic (3)	
Acetron® NS	48	1.0	.18 – .19	.20 – .21	8750
Delrin AF Blend	57	1.2	.18 – .19	.19 – .20	8300
Delrin AF	65	1.4	.18 – .19	.19 – .20	11000
Delrin 500 CL (a)	176	3.7	.22 – .24	.23 – .25	3500
Acetron® GP	200	4.2	.22 – .25	.22 – .28	2700
Turcite A	213	4.4	.29 – .34	.20 – .23	6560

(1) Measured on 1/2" I.D. journal at 5000 PV (118 fpm & 42.2 psi)
K= h/PVT x 10^{-10} (in^3 min/ft lb hr) where:
h = radial wear (in)
P = normal pressure (psi)
V = sliding speed (fm)
T = test duration (hrs)
(2) Measured on thrust washer bearing under a normal load of 50 lbs. Gradually increasing torque was applied until the bearing completed a 90° rotation in about one second.
(3) Measured on thrust washer testing machine, unlubricated @ 20 fpm & 250 psi.
(4) Limiting PV (Test valued — unlubricated @ 100 fpm (lb ft/in^2 min)
(a) Equivalent to DSM's MC® 901.

The additive system which delivers the lubrication is a patented composite. With it, the solid lubricants firmly locked in the acetal matrix are always exposed to the bearing surface. It's this constant source of lubrication which enables Acetron® NS acetal to outperform other bearing materials. It also provides lubrication during break-in of bearings and for enhanced wear-resistance.

Because the acetal and solid lubrication do not absorb significant quantities of moisture, Acetron® NS acetal is stable in both wet and dry environments. It is highly recommended for precision, close tolerance parts.

The presence of the lubricant system in the acetal matrix also allows very free machining. The result is a very competitively priced product which will outperform other filled acetals in most bearing and wear applications, and give it a noticeable advantage over more expensive, premium-priced, internally lubricated acetal compositions.

Polyamide, Polysulfone, Polyphenylene Sulfide: High-temperature materials with excellent resistance to both chemical attack and burning. With suitable fillers, these moldable plastics are useful for PV factors to 20000 and 30000. Polyamide molding compounds employing graphite as a self-lubricating filler show promise in bearing, seal, and piston ring applications at temperatures to 500°F. Polyphenylene sulfide can be applied as a coating through use of a slurry spray, dry powder, or fluidized bed. These coating techniques require a final bake at about 700°F.

Ultrahigh-Molecular-Weight Polyethylene: Resists abrasion and has a smooth, low-friction surface. Often an ideal material for parts commonly made from acetal, nylon, or PTFE materials.

Carbon-Graphite: The self-lubricating properties of carbon bearings, their stability at temperatures up to 750°F, and their resistance to attack by chemicals and solvents, give them important advantages in fields where other bearing materials are unsatisfactory. Carbon-graphite bearings are used where contamination by oil or grease is undesirable, as in textile machinery, food handling machinery, and pharmaceutical processing equipment. They are used as bearings in and around ovens, furnaces, boilers and jet engines where temperatures are too high for conventional lubricants. They are also used with low-viscosity and corrosive liquids in such applications as metering devices or pumps for gasoline, kerosene, hot and cold water, sea water, chemical process streams, acids, alkalis, and solvents.

The composition and processing used with carbon bearings can be varied to provide characteristics required for particular applications. Carbon-graphite has from 5% to 20% porosity. These pores can be filled with a phenolic or epoxy resin for improved strength and hardness, or with oil or metals (such as silver, copper, bronze, cadmium, or babbitt) to improve compatibility properties.

3.3 Load Carrying Ability of Plastic Bearings

In **Section 2.2** of sintered metal bearings, the meaning and formulas for calculation of PV factor was dealt with.

For different plastic materials, the following values of PV and load capacities apply:

Table 3-2

Bearing Material	Load Capacity (psi)	Max. Temp. (°F)	Max. Speed (fpm)	PV Limit (Unlubricated)
Phenolics	6000	200	2500	15000
Nylon	2000	200	600	3000
PTFE	500	500	50	1000
Filled PTFE	2500	500	1000	10000
PTFE fabric	60000	500	150	25000
Polycarbonate	1000	220	1000	3000
Acetal	2000	200	600	3000
Carbon-graphite	600	750	2500	15000
Rubber	50	150	4000	—
Wood	2000	160	2000	12000

A PV limit of 15000 ordinarily can be used for dry operation of carbon bearings. This should be reduced for continuous running with a steady load over a long period of time to avoid excessive wear. When operating with liquids which permit the development of a supporting fluid film, much higher PV values can be used.

A hard, rust-resistant shaft with at least a 10 µin. finish should be used. Hardened tool steel or chrome plate is recommended for heavy loads and high-speed applications. Steel having a hardness over Rockwell C50, bronzes, 18-8 stainless steels, and various carbides and ceramics also can be used.

Certain precautions should be observed in applying carbon-graphite. Since this material is brittle, it is chipped or cracked easily if struck on an edge or a corner, or if subjected to high thermal, tensile, or bending stresses. Edges should be relieved with a chamfer. Sharp corners, thin sections, keyways and blind holes should be avoided wherever possible. Because of brittleness and low coefficient of expansion (about 1/4 that of steel), carbon-graphite bearings are often shrunk into a steel sleeve. This minimizes changes in shaft clearance with temperature variations and provides mechanical support for the carbon-graphite elements.

The PV factor, used as a load-speed limit also provides a basis for estimating relative wear rates. The total volume of material worn away is approximately proportional to the total normal load multiplied by the distance traveled in a length of time.

Thus,

$$R = K(PV)\,T$$

where:

R = radial wear in a sleeve bearing (in)
K = wear factor (in^3•min/ft•lb•hr)
P = load (psi)
V = surface velocity (fpm)
T = time (hrs)

This equation does not always provide accurate absolute values for wear rate, but it is useful for estimating relative wear rates for alternative materials. In general, K wear values with fillers are lower than unfilled materials. If wear values are important for specific components, life tests should be made. These might employ moderately accelerated load and speed conditions to obtain a K value representative of the plastic, the shaft and its finish, and the application conditions.

K values should be increased by 50% for cast iron and bronze shafts, and more than 5 times with soft stainless steel or aluminum alloys. Increased surface hardness can markedly reduce wear, while surface roughness of the shaft often has an optimum value in the 4 to 14 µin. rms range. Lubrication also has a pronounced influence on wear. With oil impregnation, wear rates commonly drop to negligible values with plastics, wood, and porous metals.

The wear factor K values are shown as follows:

Table 3-3

Material	Wear Factor K (in^3 min/ft lb hr)	
	Filled*	No Filler
Nylon	16×10^{-10}	200×10^{-10}
Polyester	20×10^{-10}	——
Polycarbonate	30×10^{-10}	2500×10^{-10}
Polyurethane	35×10^{-10}	——
Polypropylene	36×10^{-10}	——
Styrene Acrylonitrile	65×10^{-10}	——
Polysulfone	70×10^{-10}	——
Acetal	200×10^{-10}	65×10^{-10}

For 40 psi load at 2000 PV operating against carbon steel of hardness 20 Rc with a 6–12 µm finish.
* Filled with 30% (by weight) glass fiber, 15% (by weight) PTFE.

Comparative values for plastics often used as bearing materials are given in the following table:

Table 3-4

Property	Graphitar (Carbon-Graphite)	Oilon PV® – 80 (TFE)	Rulon® (TFE)
Coefficient of friction	0.04 to 0.25	0.05 to 0.10	0.15 to 0.20
Temperature range	Cryogenic to 1000°F in some grades	–40°F to +250°F	–400°F to +550°F
Approx. max PV (unlubricated)	15000	18000	10000 (sleeve bearing)
Max. P	*	3000 psi	1000 psi
Max. V	*	1700 ft/min	400 ft/min
Recommended shaft surface finish	≤ 30 rms	*	8 to 32 rms
Recommended shaft clearance	0.003 in/in for most unlubricated applications	$(tw)10^{-4}$ + 0.004" t = temp. °F w= bearing wall thickness (in)	*
Typical elastic modulus	$(0.5 \text{ to } 3.5)\,10^6$ psi	$(3.5 – 3.8)10^6$ psi	*
Tensile strength	1000 – 9500 psi, depending on grade	7200 psi	*

* Consult manufacturer

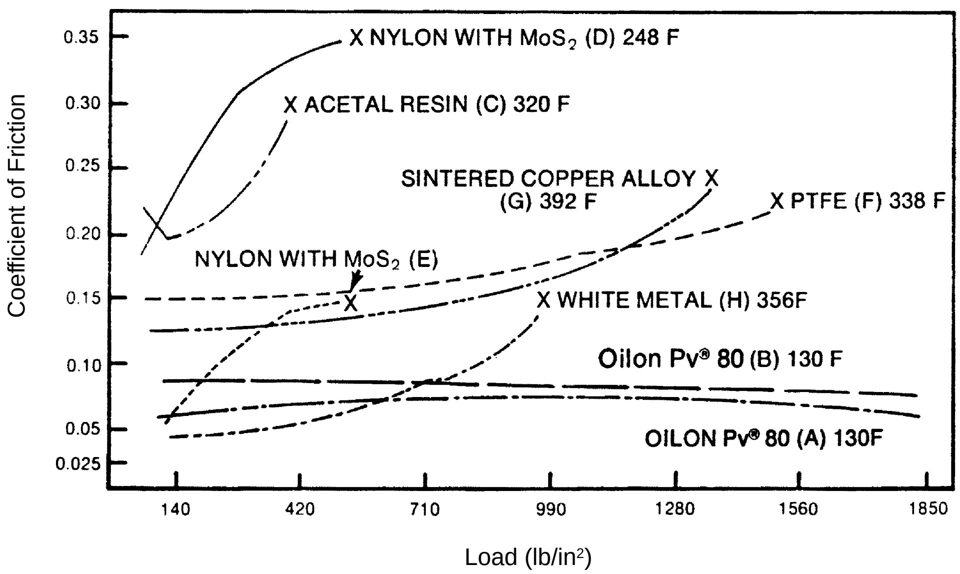
Data reprinted with the permission of the following manufacturers:

(i) "Graphitar" Wickes, 1621 Holland Ave., Saginaw, MI 48601;
(ii) "Oilon PV®" – 80 Design Guide", TFE Industries, 148 Parkway Kalamazoo, MI 49006
(iii) "Rulon® Standard Stock Bearings, Engineering Manual, Cat. 75", Dixon Corp., Div. of Dixon Industries, Bristol, RI 02809.

TECHNICAL SECTION

3.4 Coefficient of Friction vs. Load for Various Materials

The coefficient of friction varies with the bearing unit load. The following graph depicts this relationship for various plastic materials.



(A) Oilon Pv®–80—Class I
(B) Oilon Pv®–80—Class II
(C) Acetal—Class I
(D) Nylon MoS_2—Class II
(E) Nylon MoS_2—Class I
(F) PTFE glass filled—Class II
(G) Oil Impregnated Sintered Copper Alloy— Class II
(H) White Metal—Class I

Class I: Grease applied externally, prior to start-up.
Class II: No grease applied prior to start-up.

Test conditions:
Velocity: 46 ft/min (350 rpm)
Load: 140 lbs/in^2, addition applied at 10 min. intervals
Dimensions of Test Specimen: 5/8” OD x 3/8” ID x 3/8” long
Mating Material: Steel 113° F HR-B 90

Fig. 3-1 Coefficient of Friction vs. Load

A comparison of frictional characteristics of various metallic and plastic materials is given in Figure 3-1. In some plastic materials the coefficient of friction decreases with load, thereby greatly reducing or eliminating the stick-slip in the start-up of machinery.

In recent years the properties of plastic bearing materials have been materially enhanced by the addition of fillers (such as fiber, powder, graphite and molybdenum disulfide) and composites (metal or other backings). If the cost is warranted the mechanical properties of such bearings can be dramatically improved.

3.5 Example

A shaft of 1/2" in diameter is supported by two plastic bearings. The force equals 10 lbs. The bearing length is 3/4". The shaft rotates at 750 rpm.

$$PV = \frac{0.262 \cdot F \cdot rpm}{l} = \frac{0.262 \times 10 \times 750}{0.75} = 2619 \text{ fpm} \cdot \text{psi}$$

From the tables showing the maximum PV values the proper material can be chosen. If the

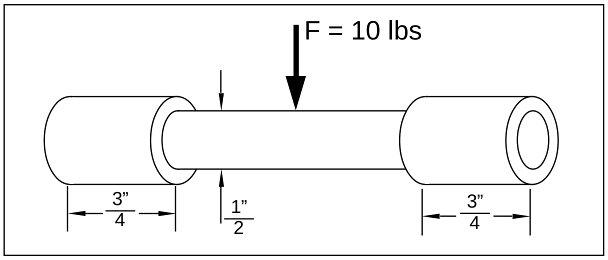


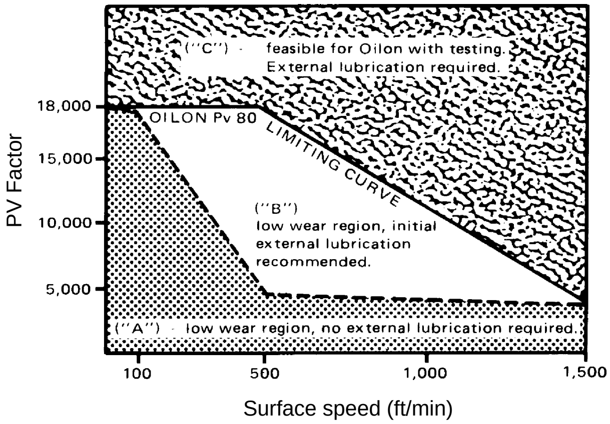
Fig. 3-2

computed value exceeds the value in the table for the chosen material, the dimensions of the shaft and of the bearing should be changed.

3.6 Lubrication

Lubricants reduce the static and dynamic coefficients of friction and permit materials to operate at higher PV's than without lubrication. While most plastics do not require lubrication, some type of lubricant will generally enhance bearing performance. In many cases, water will provide sufficient lubrication and cooling during bearing operation. At the time a plastic bearing is installed, it is a good idea to apply a light film of grease on the ID of the bearing prior to mounting on the shaft.

The effect of lubrication on the factor of a particular material (in this case, Oilon PV–80) is shown on the following graph:



(A) — low wear region, no external lubrication required
(B) — low wear region, initial external lubrication recommended
(C) — feasible for Oilon with testing. External lubrication required.

Fig. 3-3

3.7 Conclusion

Plastic and nonmetallic bearings are widely used in appliances, toys, general machinery and applications ranging from cameras and toys to office machinery and automobiles. When properly designed their light weight and economy can be highly attractive.

Calculations of bearing loads as shown in the example are also applicable on plastic and nonmetallic bearings.

EQUIVALENT PRECISION CLASSES OF DIFFERENT STANDARDS

ABEC	RBEC	ISO	DIN
1	1	0	Normal
3	3	6	P6
5	5	5	P5
7	None	4	P4

ABEC – Anti-Friction Bearing Manufacturers Association (Ball Bearings)
RBEC – Anti-Friction Bearing Manufacturers Association (Roller Bearings)
ISO – International Organization for Standardization
DIN – Deutsche Industrie Normen

ACKNOWLEDGMENTS

We are expressing our thanks and we are crediting the following publications for providing material for this text, by the permission of the publisher:

1. *Design of Machine Elements*, M. F. Spotts, Late Professor Emeritus of Mechanical Engineering Department, Northwestern University and T. E. Shoup, Dean of Engineering, Santa Clara University, Prentice Hall, Upper Saddle River, New Jersey 07458, 1998.
2. *Machine Design*, Joseph Edward Shigley, Associate Professor, Dept. of Mechanical Engineering – University of Michigan, McGraw Hill Book Company, 1956.
3. *Standard Handbook of Machine Design*, Chapter 27 “Rolling Contact Bearings”, Charles Mischke, Ph.D., P.E., Professor Emeritus of Mechanical Engineering, Iowa State University, McGraw Hill Book Company, 1996.
4. *Handbook of Shafts, Bearings and Couplings*, Technical Staff of Stock Drive Products and Sterling Instrument (SDP/SI), 1993.

For further study of this subject matter the following references are provided:

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3. *Bearings Reference Issue*, *Machine Design*, No. 42, June 18, 1970.
4. *Boundary Lubrication – An Appraisal of World Literature*, New York: American Society of Mechanical Engineers, 1969.
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11. Palmgren, A., *Ball and Roller Bearings*, 3rd ed., Philadelphia: SKF Industries, Inc., 1959.
12. Rothbart, H. A., Ed., *Mechanical Design and Systems Handbook*, New York: McGraw Hill Book Company, 1964, Section 13.
13. Shaw, M. C., and E. F. Macks, *Analysis and Lubrication of Bearings*, New York, McGraw Hill Book Company, 1949, Chapter 10.
14. *SKF Engineering Data*, SKF Industries, Inc., Philadelphia, 1979.