Bearing

Bearing Types

Bearings are manufactured to take pure radial loads, pure thrust loads, or a combination of the two kinds of loads. The nomenclature of a ball bearing is illustrated in Fig. 7.1, which also shows the four essential parts of a bearing. These are the outer ring, the inner ring, the balls or rolling elements, and the separator. In low-priced bearings, the separator is sometimes omitted, but it has the important function of separating the elements so that rubbing contact will not occur.

Figure 7.1

Nomenclature of a ball bearing. *(General Motors Corp. Used with permission, GM Media Archives.)*

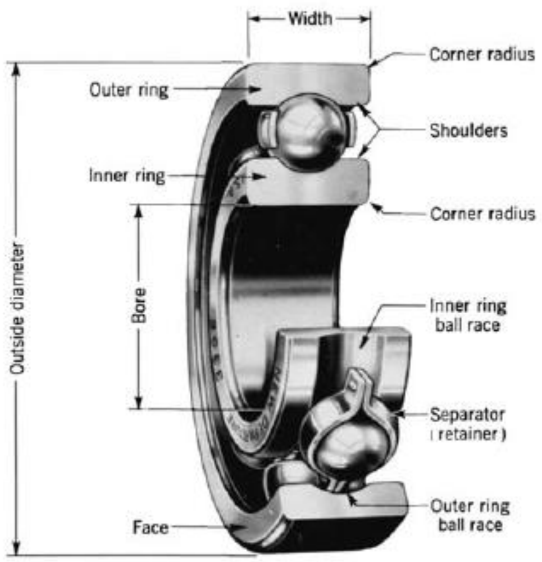


Figure 7.2

Various types of ball bearings.

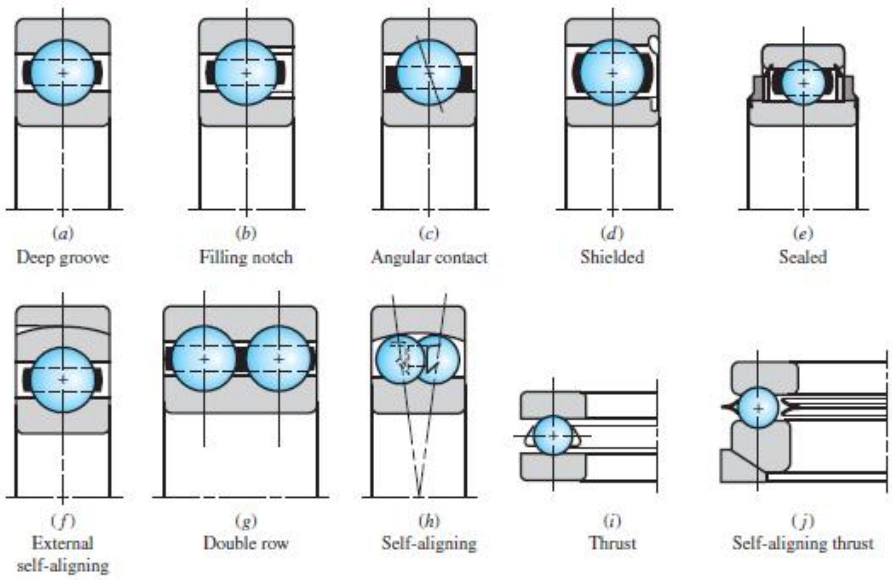


Fig. Basic types bearing and bearing block

In this section we include a selection from the many types of standardized bearings that are manufactured. Most bearing manufacturers provide engineering manuals and brochures containing lavish descriptions of the various types available. In the small space available here, only a meager outline of some of the most common types can be given. So you should include a survey of bearing manufacturers' literature in your studies of this section.

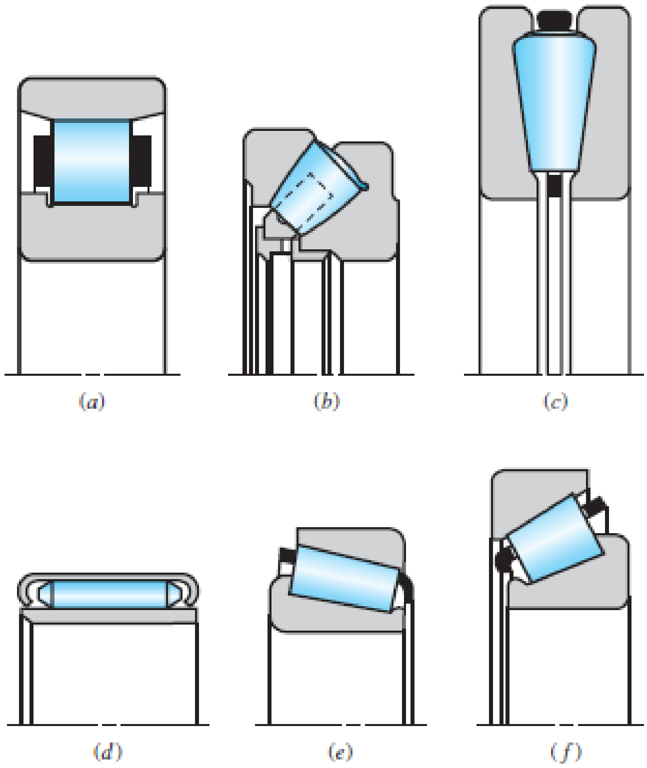
Some of the various types of standardized bearings that are manufactured are shown in Fig. 7.2. The single-row deep-groove bearing will take radial load as well as some thrust load. The balls are inserted into the grooves by moving the inner ring thrust capacity is decreased, however, because of the bumping of the balls against the edge of the notch when thrust loads are present. The angular-contact bearing (Fig. 7.2c) provides a greater thrust capacity. All these bearings may be obtained with shields on one or both sides. The shields are not a complete closure but do offer a measure of protection against dirt. A variety of bearings are manufactured with seals on one or both sides. When the seals are on both sides, the bearings are lubricated at the factory. Although a sealed bearing is supposed to be lubricated for life, a method of relubrication is sometimes provided.

Single-row bearings will withstand a small amount of shaft misalignment of deflection, but where this is severe, self-aligning bearings may be used. Double-row bearings are made in a variety of types and sizes to carry heavier radial and thrust loads. Sometimes two single-row bearings are used together for the same reason, although a double-row bearing will generally require fewer parts and occupy less space.

The one-way ball thrust bearings (Fig. 7.2i) are made in many types and sizes. Some of the large variety of standard roller bearings available are illustrated in Fig. 7.3. Straight roller bearings (Fig. 7.3a) will carry a greater radial load than ball bearings of the same size because of the greater contact area. However, they have the disadvantage of requiring almost perfect geometry of the raceways and rollers. A slight misalignment will cause the rollers to skew and get out of line. For this reason, the retainer must be heavy. Straight roller bearings will not, of course, take thrust loads. Helical rollers are made by winding rectangular material into rollers, after which they are hardened and ground. Because of the inherent flexibility, they will take considerable misalignment. If necessary, the shaft and housing can be used for raceways instead of separate inner and outer races. This is especially important if radial space is limited.

Figure 7.3

Types of roller bearings: (*a*) straight roller; (*b*) spherical roller, thrust; (*c*) tapered roller, thrust; (*d*) needle; (*e*) tapered roller; (*f*) steep-angle tapered roller. *(Courtesy of The Timken Company.)*



The spherical-roller thrust bearing (Fig. 7.3b) is useful where heavy loads and misalignment occur. The spherical elements have the advantage of increasing their contact area as the load is increased.

Needle bearings (Fig. 7.3d) are very useful where radial space is limited. They have a high load capacity when separators are used, but may be obtained without separators. They are furnished both with and without races. Tapered roller bearings (Fig. 7.3e, f) combine the advantages of ball and straight roller bearings, since they can take either radial or thrust loads or any combination of the two, and in addition, they have the high load-carrying capacity of straight roller bearings. The tapered roller bearing is designed so that all elements in the roller surface and the raceways intersect at a common point on the bearing axis.

The bearings described here represent only a small portion of the many available for selection. Many special-purpose bearings are manufactured, and bearings are also made for particular classes of machinery. Typical of these are:

• Instrument bearings, which are high-precision and are available in stainless steel and high-temperature materials
• Nonprecision bearings, usually made with no separator and sometimes having split or stamped sheet-metal races
• Ball bushings, which permit either rotation or sliding motion or both
• Bearings with flexible rollers

Bearing Life

When the ball or roller of rolling-contact bearings rolls, contact stresses occur on the inner ring, the rolling element, and on the outer ring. Because the curvature of the contacting elements in the axial direction is different from that in the radial direction, the equations for these stresses are more involved than in the Hertz equations. If a bearing is clean and properly lubricated, is mounted and sealed against the entrance of dust and dirt, is maintained in this condition, and is operated at reasonable temperatures, then metal fatigue will be the only cause of failure.

Inasmuch as metal fatigue implies many millions of stress applications successfully endured, we need a quantitative life measure. Common life measures are
• Number of revolutions of the inner ring (outer ring stationary) until the first tangible evidence of fatigue
• Number of hours of use at a standard angular speed until the first tangible evidence of fatigue

The commonly used term is bearing life, which is applied to either of the measures just mentioned. It is important to realize, as in all fatigue, life as defined above is a stochastic variable and, as such, has both a distribution and associated statistical parameters. The life measure of an individual bearing is defined as the total number of revolutions (or hours at a constant speed) of bearing operation until the failure criterion is developed. Under ideal conditions, the fatigue failure consists of spalling of the load-carrying surfaces.

The American Bearing Manufacturers Association (ABMA) standard states that the failure criterion is the first evidence of fatigue. The fatigue criterion used by the Timken Company laboratories is the spalling or pitting of an area of 0.01 in2. Timken also observes that the useful life of the bearing may extend considerably beyond this point. This is an operational definition of fatigue failure in rolling bearings. The rating life is a term sanctioned by the ABMA and used by most manufacturers.

The rating life of a group of nominally identical ball or roller bearings is defined as the number of revolutions (or hours at a constant speed) that 90 percent of a group of bearings will achieve or exceed before the failure criterion develops. The terms minimum life, L_{10} life, and B_{10} life are also used as synonyms for rating life. The rating life is the 10th percentile location of the bearing group's revolutions-to-failure distribution. Median life is the 50th percentile life of a group of bearings. The term average life has been used as a synonym for median life, contributing to confusion. When many groups of bearings are tested, the median life is between 4 and 5 times the L_{10} life.

Each bearing manufacturer will choose a specific rating life for which load ratings of its bearings are reported. The most commonly used rating life is 10^6 revolutions. The Timken Company is a well-known exception, rating its bearings at 3 000 hours at 500 rev/min, which is $90.(10^6)$ revolutions. These levels of rating life are actually quite low for today's bearings, but since rating life is an arbitrary reference point, the traditional values have generally been maintained.

Bearing Load Life at Rated Reliability

When nominally identical groups are tested to the life-failure criterion at different loads. To establish a single point, load F_1 and the rating life of group one $(L_{10})_1$ are the coordinates that are logarithmically transformed. The reliability associated with this point, and all other points, is 0,90. Thus we gain a glimpse of the load-life function at 0,90 reliability. Using a regression equation of the form

L_N – number of cycles

the result of many tests for various kinds of bearings result in
• m = 3 for ball bearings
• m = 10/3 for roller bearings (cylindrical and tapered roller)

A catalog load rating is defined as the radial load that causes 10 percent of a group of bearings to fail at the bearing manufacturer's rating life. We shall denote the catalog load rating as C_{10}. The catalog load rating is often referred to as a Basic Dynamic Load Rating, or sometimes just Basic Load Rating, if the manufacturer's rating life is 10^6 revolutions. The radial load that would be necessary to cause failure at such a low life would be unrealistically high. Consequently, the Basic Load Rating should be viewed as a reference value, and not as an actual load to be achieved by a bearing.

Combined Radial and Thrust Loading

A ball bearing is capable of resisting radial loading and a thrust loading. Furthermore, these can be combined. Consider F_a and F_r to be the axial thrust and radial loads, respectively, and F_e to be the equivalent radial load that does the same damage as the combined radial and thrust loads together. A rotation factor V is defined such that V = 1 when the inner ring rotates and V = 1,2 when the outer ring rotates. Two dimensionless groups can now be formed: $F_e/(VF_r)$ and $F_a/(VF_r)$. When these two dimensionless groups are plotted, the data fall in a gentle curve that is well approximated by two straight-line segments. The abscissa *e* is defined by the intersection of the two lines. The equations for the two lines are:

The X and Y factors depend upon the geometry and construction of the specific bearing. Table 7.1 lists representative values of X_1, Y_1, X_2, and Y_2 as a function of *e*, which in turn is a function of F_a/C_0, where C_0 is the basic static load rating. The basic static load rating is the load that will produce a total permanent deformation in the raceway and rolling element at any contact point of 0,0001 times the diameter of the rolling element. The basic static load rating is typically tabulated, along with the basic dynamic load rating C_{10}, in bearing manufacturers' publications. See Handbook, or catalogue of bearings for example.

In these equations, the rotation factor V is intended to correct for the rotating-ring conditions. The factor of 1,2 for outer-ring rotation is simply an acknowledgment that the fatigue life is reduced under these conditions. Self-aligning bearings are an exception: they have V = 1 for rotation of either ring. Since straight or cylindrical roller bearings will take no axial load, or very little, the Y factor is always zero.

Table 7.1

Equivalent Radial Load Factors for Ball Bearings

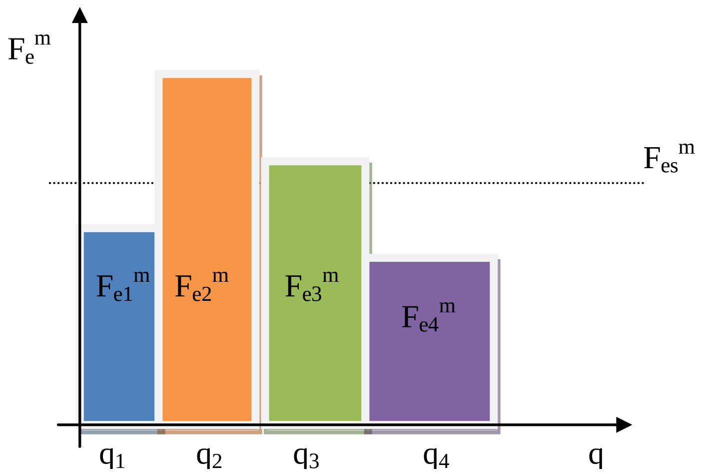
		$F_a/(VF_r) \le e$		$F_a/(VF_r) > e$	
F_a/C_0	e	X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Variable load of bearing

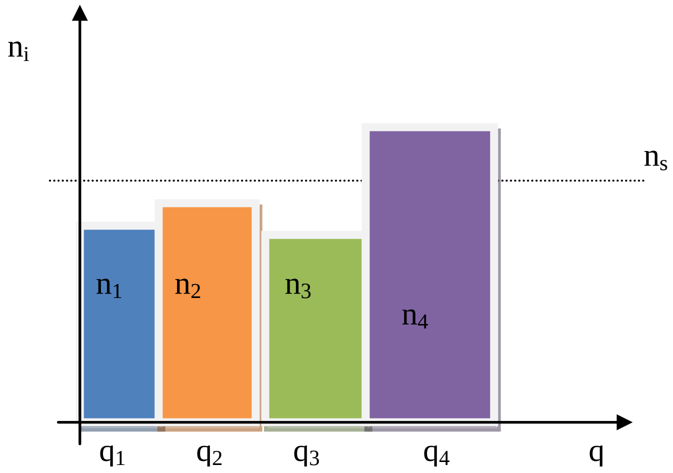
Many of the operating rotary systems are those whose operating state is not constant. Three basic operating states occur:

1. Constant speed and variable load operations.
2. Constant load and variable speed operation.
3. Operation at variable load and variable speed. In order to determine the service life of a bearing, it is necessary to approximate the variable nature of the load to a substitute value (medium) which will cause the same amount of load as would cause a variable load.



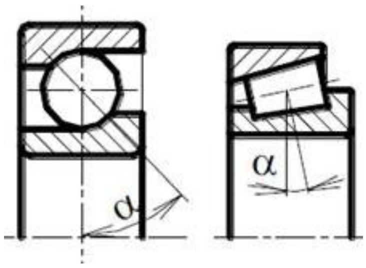
$q_1 ... q_k$ – proportion of partial load action $q_1 + q_2 + ... + q_k = 100\%$
F_{es}^m – medium non-variable equivalent load by which we replace the action of variable equivalent load
m – bearing loading type
k – number of changes

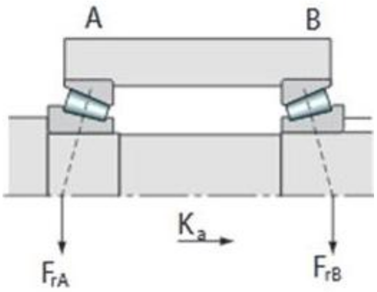
If only the speed changes and the load is constant

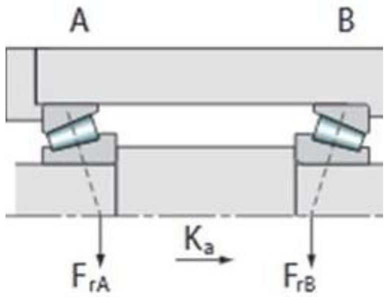


Reaction forces in bearing

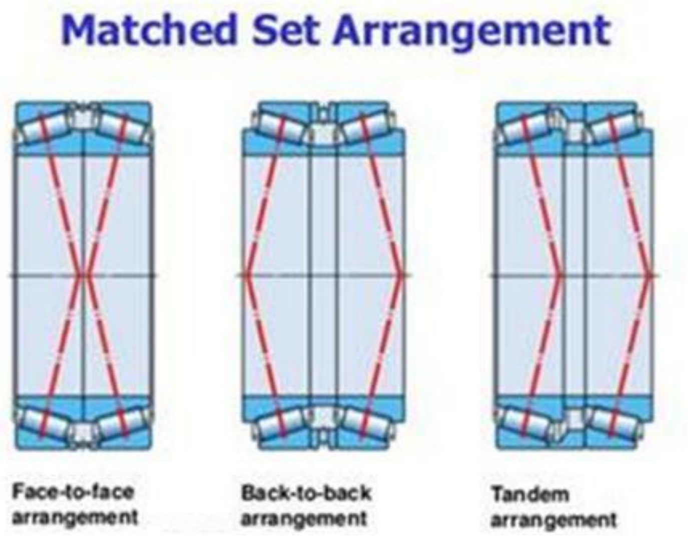
The reaction forces in the bearing is assumed in the axis of the pressure angle and the bearing. This means that in the case of ball, roller, needle and barrel bearings, it is usually in the middle of the bearing. In the case of angular contact ball bearings and tapered roller bearings, this is according to FIG. It is important to choose the arrangement of the bearings so that the internal axial forces of the bearing are eliminated by the mutual arrangement of the bearings.







Back to Back – „O“ Face to Face – „X“



Friction torque of Bearing

The friction torque of the rolling bearing depends on the method of bearing lubrication and the load. The friction moment of an unloaded rotating bearing can be expressed:

υ– kinematic viscosity of the lubricant at operating temperature
n – revolution per min^{-1}
d_s – middle diameter of bearing - mm
f_0 – factor depending on bearing design and oil flow

The friction moment of a loaded rotating bearing can be expressed:

F_{e0} – equivalents static load - N
C_0 – basic static load rating - N
p – coefficient of bearing type
f_1,f_2 – coefficients depending on the design and oil flow

Total bearing friction torque:

Under certain conditions (mean working speed) we can modify the equation as follows:

Fe – equivalents dynamic load -N
d – diameter of inert ring of bearing - mm
f –friction coefficient
f=0,0010 – self - aligning ball bearing
f=0,0011 – cylindrical roller bearing
f=0,0015 – ball bearing
f=0,0018 – tapered roller bearing
f=0,0025 – needler roller bearing.

Assuming that **P = Mk.ω,** we get the power dissipation of the bearing: