

P · A · R · T · 5

BEARINGS AND LUBRICATION

CHAPTER 18

ROLLING-CONTACT BEARINGS

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18.1 INTRODUCTION / 18.4
18.2 LOAD-LIFE RELATION FOR CONSTANT RELIABILITY / 18.9
18.3 SURVIVAL RELATION AT STEADY LOAD / 18.10
18.4 RELATING LOAD, LIFE, AND RELIABILITY GOAL / 18.11
18.5 COMBINED RADIAL AND THRUST LOADINGS / 18.14
18.6 APPLICATION FACTORS / 18.15
18.7 VARIABLE LOADING / 18.15
18.8 MISALIGNMENT / 18.18
REFERENCES / 18.19

GLOSSARY OF SYMBOLS

a	Exponents; $a = 3$ for ball bearings; $a = 10/3$ for roller bearings
AF	Application factor
b	Weibull shape parameter
C_s	Static load rating
C_{10}	Basic load rating or basic dynamic load rating
f	Fraction
F	Load
F_a	Axial load
F_{eq}	Equivalent radial load
F_i	i th equivalent radial load
F_r	Radial load
I	Integral
L	Life measure, r or h
L_D	Desired or design life measure
L_R	Rating life measure
L_{10}	Life measure exceeded by 90 percent of bearings tested

n	Design factor
n_D	Desired or design rotative speed, r/min
n_i	Application or design factor at i th level
n_R	Rating rotative speed, r/min
R	Reliability
V	Rotation factor; inner ring rotations, $V = 1$; outer ring, $V = 1.20$
x	Life measure in Weibull survival equation
x_0	Weibull guaranteed life parameter
X	Radial factor for equivalent load prediction
Y	Thrust factor for equivalent load prediction
θ	Weibull characteristic life parameter, rotation angle
ϕ	Period of cyclic variation, rad

18.1 INTRODUCTION

Figures 18.1 to 18.12 illustrate something of the terminology and the wide variety of rolling-contact bearings available to the designer. Catalogs and engineering manuals can be obtained from bearing manufacturers, and these are very comprehensive and of excellent quality. In addition, most manufacturers are anxious to advise designers on specific applications. For this reason the material in this chapter is concerned mostly with providing the designer an independent viewpoint.

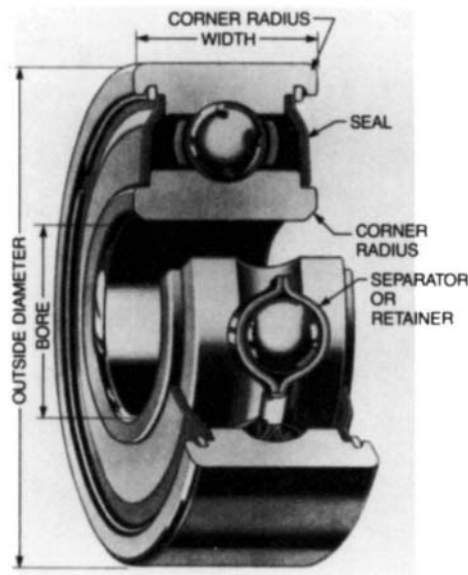


FIGURE 18.1 Photograph of a deep-groove precision ball bearing with metal two-piece cage and dual seals to illustrate rolling-bearing terminology. (*The Barden Corporation.*)

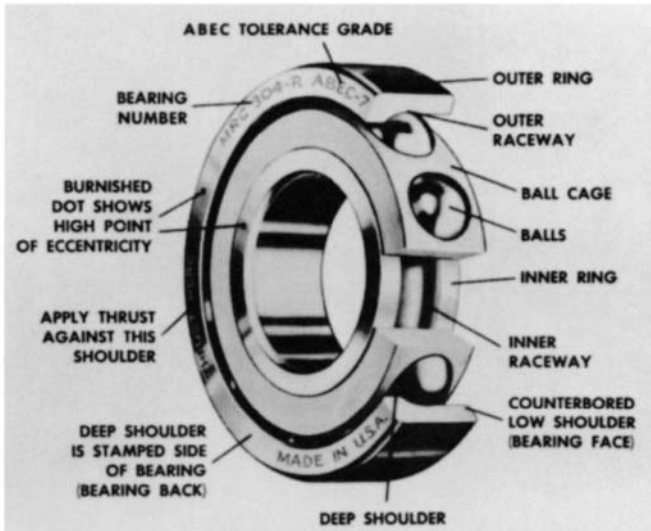


FIGURE 18.2 Photograph of a precision ball bearing of the type generally used in machine-tool applications to illustrate terminology. (*Bearings Division, TRW Industrial Products Group.*)



FIGURE 18.3 Rolling bearing with spherical rolling elements to permit misalignment up to $\pm 3^\circ$ with an unsealed design. The sealed bearing, shown above, permits misalignment to $\pm 2^\circ$. (*McGill Manufacturing Company, Inc.*)



FIGURE 18.4 A heavy-duty cage-guided needle roller bearing with machined race. Note the absence of an inner ring, but standard inner rings can be obtained. (*McGill Manufacturing Company, Inc.*)

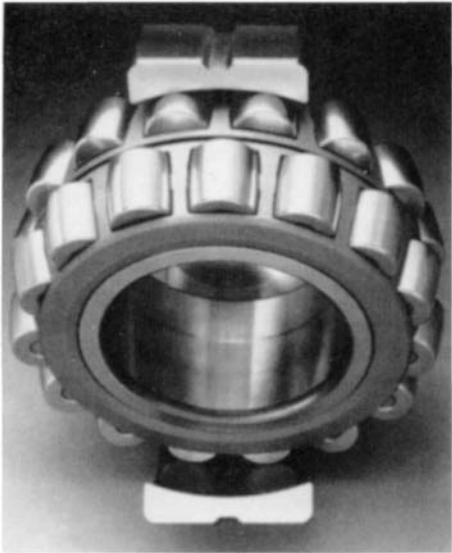


FIGURE 18.5 A spherical roller bearing with two rows of rollers running on a common sphered raceway. These bearings are self-aligning to permit misalignment resulting from either mounting or shaft deflection under load. (*SKF Industries, Inc.*)

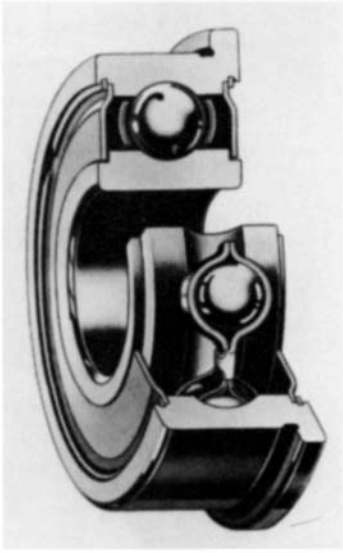


FIGURE 18.6 Shielded, flanged, deep-groove ball bearing. Shields serve as dirt barriers; flange facilitates mounting the bearing in a through-bored hole. (*The Barden Corporation.*)



FIGURE 18.7 Ball thrust bearing. (*The Torrington Company.*)



FIGURE 18.8 Spherical roller thrust bearing. (*The Torrington Company.*)



FIGURE 18.9 Tapered-roller thrust bearing.
(The Torrington Company.)



FIGURE 18.10 Tapered-roller bearing; for axial loads, thrust loads, or combined axial and thrust loads. (The Timken Company.)

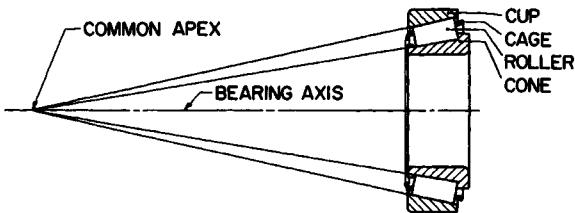


FIGURE 18.11 Basic principle of a tapered-roller bearing with nomenclature. (The Timken Company.)

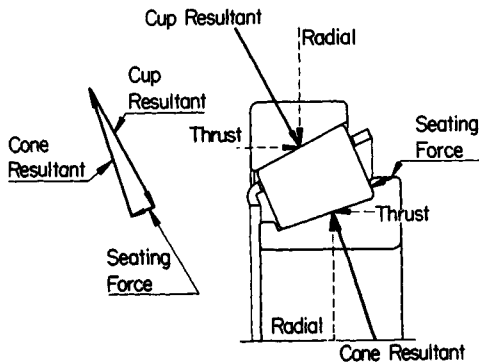


FIGURE 18.12 Force analysis of a Timken bearing.
(The Timken Company.)

TABLE 18.1 Coefficients of Friction

Bearing type	Coefficient of friction μ
Self-aligning ball	0.0010
Cylindrical roller with flange-guided short rollers	0.0011
Ball thrust	0.0013
Single-row ball	0.0015
Spherical roller	0.0018
Tapered roller	0.0018

SOURCE: Ref. [18.1].

Rolling-contact bearings use balls and rollers to exploit the small coefficients of friction when hard bodies roll on each other. The balls and rollers are kept separated and equally spaced by a separator (cage, or retainer). This device, which is essential to proper bearing functioning, is responsible for additional friction. Table 18.1 gives friction coefficients for several types of bearings [18.1]. Consult a manufacturer's catalog for equations for estimating friction torque as a function of bearing mean diameter, load, basic load rating, and lubrication detail. See also Chap. 20.

Permissible speeds are influenced by bearing size, properties, lubrication detail, and operating temperatures. The speed varies inversely with mean bearing diameter. For additional details, consult any manufacturer's catalog.

Some of the guidelines for selecting bearings, which are valid more often than not, are as follows:

- Ball bearings are the less expensive choice in the smaller sizes and under lighter loads, whereas 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.
- Although rolling-contact bearings are "standardized" and easily selected from vendor catalogs, there are instances of cooperative development by customer and vendor involving special materials, hollow elements, distorted raceways, and novel applications. Consult your bearing specialist.

It is possible to obtain an estimate of the basic static load rating C_s . For ball bearings,

$$C_s = Mn_b d_b^2 \quad (18.1)$$

For roller bearings,

$$C_s = Mn_r l_r d \quad (18.2)$$

where C_s = basic static loading rating, pounds (lb) [kilonewtons (kN)]
 n_b = number of balls
 n_r = number of rollers
 d_b = ball diameter, inches (in) [millimeters (mm)]
 d = roller diameter, in (mm)
 l_e = length of single-roller contact line, in (mm)

Values of the constant M are listed in Table 18.2.

TABLE 18.2 Value of Constant M for Use in Eqs. (18.1) and (18.2)

Type of bearing	Constant M	
	C_s , lb	C_s , kN
Radial ball	1.78×10^3	5.11×10^3
Ball thrust	7.10×10^3	20.4×10^3
Radial roller	3.13×10^3	8.99×10^3
Roller thrust	14.2×10^3	40.7×10^3

18.2 LOAD-LIFE RELATION FOR CONSTANT RELIABILITY

When proper attention is paid to a rolling-contact bearing so that fatigue of the material is the only cause of failure, then nominally identical bearings exhibit a reliability-life-measure curve, as depicted in Fig. 18.13. The *rating life* is defined as the life measure (revolutions, hours, etc.) which 90 percent of the bearings will equal or exceed. This is also called the L_{10} life or the B_{10} life. When the radial load is adjusted so that the L_{10} life is 1 000 000 revolutions (r), that load is called the *basic load rating* C (SKF Industries, Inc.). The Timken Company rates its bearings at 90 000 000. Whatever the rating basis, the life L can be normalized by dividing by the rating life L_{10} . The *median life* is the life measure equaled or exceeded by half of the bearings. Median life is roughly 5 times rating life.

For steady radial loading, the life at which the first tangible evidence of surface fatigue occurs can be predicted from

$$F^a L = \text{constant} \quad (18.3)$$

where $a = 3$ for ball bearings and $a = \frac{10}{3}$ for cylindrical and tapered-roller bearings. At constant reliability, the load and life at condition 1 can be related to the load and life at condition 2 by Eq. (18.3). Thus

$$F_1^a L_1 = F_2^a L_2 \quad (18.4)$$

If F_1 is the basic load rating C_{10} , then L_1 is the rating life L_{10} , and so

$$C_{10} = \left(\frac{L}{L_{10}} \right)^{1/a} (F) \quad (18.5)$$

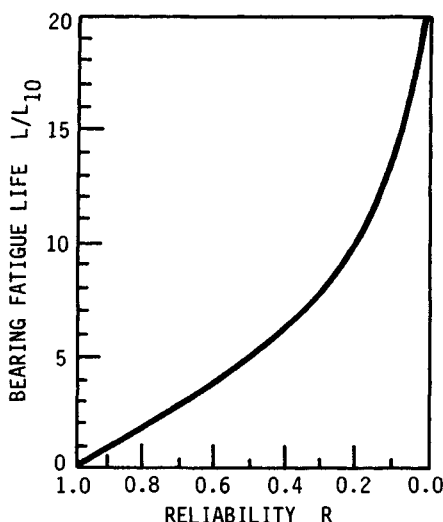


FIGURE 18.13 Survival function representing endurance tests on rolling-contact bearings from data accumulated by SKF Industries, Inc. (From Ref. [18.2].)

If L_R is in hours and n_R is in revolutions per minute, then $L_{10} = 60L_R n_R$. It follows that

$$C_{10} = F_D \left(\frac{L_D n_D}{L_R n_R} \right)^{1/a} \quad (18.6)$$

where the subscript D refers to desired (or design) and the subscript R refers to rating conditions.

18.3 SURVIVAL RELATION AT STEADY LOAD

Figure 18.14 shows how reliability varies as the loading is modified [18.2]. Equation (18.5) allows the ordinate to be expressed as either F/C_{10} or L/L_{10} . Figure 18.14 is based on more than 2500 SKF bearings. If Figs. 18.13 and 18.14 are scaled for recovery of coordinates, then the reliability can be tabulated together with L/L_{10} . Machinery applications use reliabilities exceeding 0.94. An excellent curve fit can be realized by using the three-parameter Weibull distribution. For this distribution the reliability can be expressed as

$$R = \exp \left[- \left(\frac{x - x_0}{\theta - x_0} \right)^b \right] \quad (18.7)$$

where x = life measure, x_0 = Weibull guaranteed life measure, θ = Weibull characteristic life measure, and b = Weibull shape factor. Using the 18 points in Table 18.3 with $x_0 = 0.02$, $\theta = 4.459$, and $b = 1.483$, we see that Eq. (18.7) can be particularized as

$$R = \exp \left[- \left(\frac{L/L_{10} - 0.02}{4.439} \right)^{1.483} \right] \quad (18.8)$$

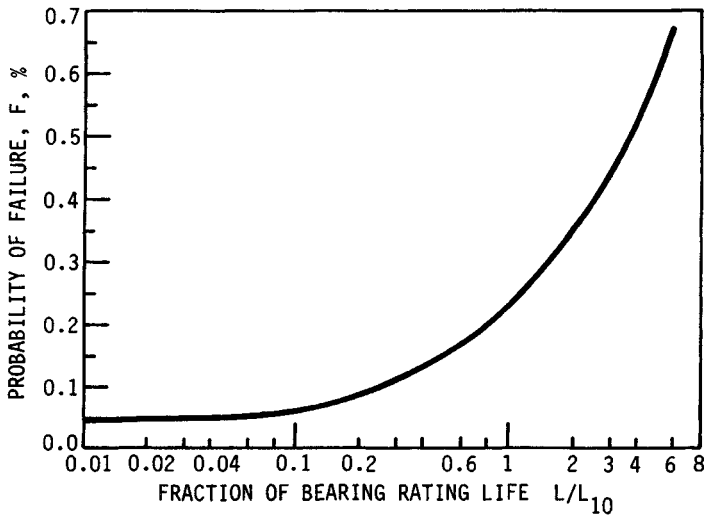


FIGURE 18.14 Survival function at higher reliabilities based on more than 2500 endurance tests by SKF Industries, Inc. (From Ref. [18.2].) The three-parameter Weibull constants are $\theta = 4.459$, $b = 1.483$, and $x_0 = 0.02$ when $x = L/L_{10} = Ln/(L_R n_R)$.

For example, for $L/L_{10} = 0.1$, Eq. (18.8) predicts $R = 0.9974$.

18.4 RELATING LOAD, LIFE, AND RELIABILITY GOAL

If Eq. (18.3) is plotted on log-log coordinates, Fig. 18.15 results. The FL loci are rectified, while the parallel loci exhibit different reliabilities. The coordinates of point A are the rating life and the basic load rating. Point D represents the desired (or design) life and the corresponding load. A common problem is to select a bearing which will provide a life L_D while carrying load F_D and exhibit a reliability R_D . Along line BD , constant reliability prevails, and Eq. (18.4) applies:

TABLE 18.3 Survival Equation Points at Higher Reliabilities[†]

Reliability R	Life measure L/L_{10}	Reliability R	Life measure L/L_{10}
0.94	0.67	0.994	0.17
0.95	0.60	0.995	0.15
0.96	0.52	0.996	0.13
0.97	0.435	0.997	0.11
0.975	0.395	0.9975	0.095
0.98	0.35	0.998	0.08
0.985	0.29	0.9985	0.07
0.99	0.23	0.999	0.06
0.992	0.20	0.9995	0.05

[†]Scaled from Ref. [18.2], Fig. 2.

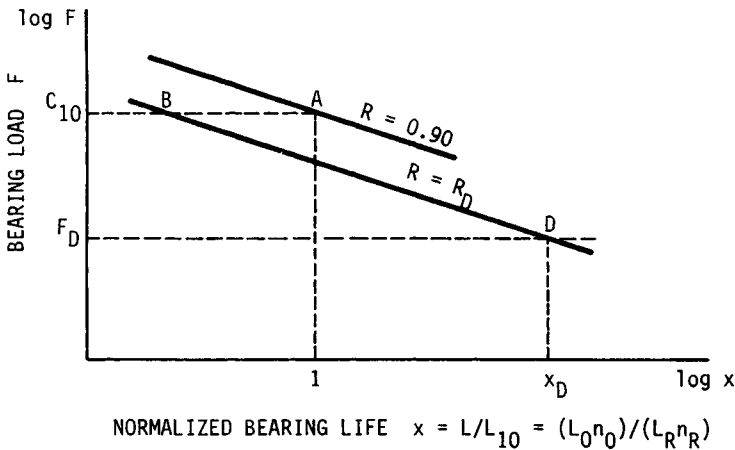


FIGURE 18.15 Reliability contours on a load-life plot useful for relating catalog entry, point A, to design goal, point D.

$$F_B = F_D \left(\frac{x_D}{x_B} \right)^{1/a} \quad (18.9)$$

Along line AB the reliability changes, but the load is constant and Eq. (18.7) applies. Thus

$$R = \exp \left[- \left(\frac{x - x_0}{\theta - x_0} \right)^b \right] \quad (18.10)$$

Now solve this equation for x and particularize it for point B , noting that $R_D = R_B$.

$$x_B = x_0 + (\theta - x_0) \left(\ln \frac{1}{R_D} \right)^{1/b} \quad (18.11)$$

Substituting Eq. (18.11) into Eq. (18.9) yields

$$F_B = C_{10} = F_D \left\{ \frac{x_D}{x_0 + (\theta - x_0) [\ln (1/R_D)]^{1/b}} \right\}^{1/a} \quad (18.12)$$

For reliabilities greater than 0.90, which is the usual case, $\ln (1/R) \cong 1 - R$ and Eq. (18.12) simplifies as follows:

$$C_{10} = F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R)^{1/b}} \right]^{1/a} \quad (18.13)$$

The desired life measure x_D can be expressed most conveniently in millions of revolutions (for SKF).

Example 1. If a ball bearing must carry a load of 800 lb for 50×10^6 and exhibit a reliability of 0.99, then the basic load rating should equal or exceed

$$C_{10} = 800 \left[\frac{50}{0.02 + (4.439)(1 - 0.99)^{1/1.483}} \right]^{1/3}$$

$$= 4890 \text{ lb}$$

This is the same as 21.80 kN, which corresponds to the capability of a 02 series 35-mm-bore ball bearing. Since selected bearings have different basic load ratings from those required, a solution to Eq. (18.13) for reliability extant after specification is useful:

$$R = 1 - \left[\frac{x_D - x_0(C_{10}/F_D)^a}{(\theta - x_0)(C_{10}/F_D)^a} \right]^b \quad (18.14)$$

Example 2. If the bearing selected for Example 1, a 02 series 50-mm bore, has a basic load rating of 26.9 kN, what is the expected reliability? And $C_{10} = (26.9 \times 10^3)/445 = 6045 \text{ lb}$. So

$$R = 1 - \left[\frac{50 - 0.02(6045/800)^3}{(4.439)(6045/800)^3} \right]^{1.483} = 0.9966$$

The previous equations can be adjusted to a two-parameter Weibull survival equation by setting x_0 to zero and using appropriate values of θ and b . For bearings rated at a particular speed and time, substitute $L_D n_D / (L_R n_R)$ for x_D .

The survival relationship for Timken tapered-roller bearings is shown graphically in Fig. 18.16, and points scaled from this curve form the basis for Table 18.4. The survival equation turns out to be the two-parameter Weibull relation:

$$R = \exp \left[- \left(\frac{x}{\theta} \right)^b \right] = \exp \left[- \left(\frac{L/L_{10}}{4.890} \right)^{1.4335} \right] \quad (18.15)$$

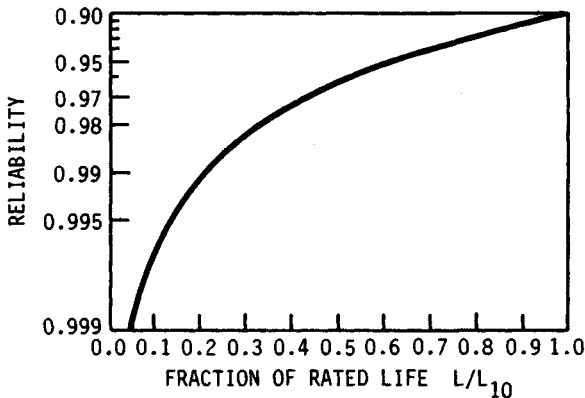


FIGURE 18.16 Survival function at higher reliabilities based on the Timken Company tapered-roller bearings. The curve fit is a two-parameter Weibull function with constants $\theta = 4.48$ and $b = \frac{1}{2}$ ($x_0 = 0$) when $x = Ln/(L_R n_R)$. (From Ref. [18.3].)

TABLE 18.4 Survival Equation Points for Tapered-Roller Bearings[†]

Reliability R	Life measure L/L_{10}	Reliability R	Life measure L/L_{10}
0.90	1.00	0.96	0.53
0.91	0.92	0.97	0.43
0.92	0.86	0.98	0.325
0.93	0.78	0.99	0.20
0.94	0.70	0.995	0.13
0.95	0.62	0.999	0.04

[†]Scaled from Fig. 4 of *Engineering Journal*, Sec. 1, The Timken Company, Canton, Ohio, rev. 1978.

The equation corresponding to Eq. (18.13) is

$$\begin{aligned}
 C_{10} &= F_D \left[\frac{x_D}{\theta(1-R)^{1/b}} \right]^{1/a} \\
 &= F_D \left(\frac{x_D}{\theta} \right)^{1/a} (1-R)^{-1/ab}
 \end{aligned} \quad (18.16)$$

And the equation corresponding to Eq. (18.14) is

$$R = 1 - \left(\frac{x_D}{\theta} \right)^b \left(\frac{C_{10}}{F_D} \right)^{-ab} \quad (18.17)$$

Example 3. A Timken tapered-roller bearing is to be selected to carry a radial load of 4 kN and have a reliability of 0.99 at 1200 hours (h) and a speed of 600 revolutions per minute (r/min). Thus

$$x_D = \frac{L_D n_D}{L_R n_R} = \frac{1200(600)}{3000(500)} = 0.480$$

and

$$C_{10} = 4 \left[\frac{0.48}{4.48(1-0.99)^{1/1.5}} \right]^{3/10} = 5141 \text{ N}$$

Timken bearings are rated in U.S. Customary System (USCS) units or in newtons; therefore, a basic load rating of 5141 N or higher is to be sought.

For any bearings to be specified, check with the manufacturer's engineering manual for survival equation information. This is usually in the form of graphs, nomograms, or equations of available candidates. Check with the manufacturer on cost because production runs materially affect bearing cost.

18.5 COMBINED RADIAL AND THRUST LOADINGS

Ball bearings can resist some thrust loading simultaneously with a radial load. The equivalent radial load is the constant pure radial load which inflicts the same dam-

age on the bearing per revolution as the combination. A common form for weighting the radial load F_r and the axial load F_a is

$$F_e = VXF_r + YF_a \quad (18.18)$$

where F_e = equivalent radial load. The weighting factors X and Y are given for each bearing type in the manufacturer's engineering manual. The parameter V distinguishes between inner-ring rotation, $V = 1$, and outer-ring rotation, $V = 1.20$. A common form of Eq. (18.18) is

$$F_e = \max(VF_r, X_1VF_r + Y_1F_a, X_2VF_r + Y_2F_a, \dots) \quad (18.19)$$

18.6 APPLICATION FACTORS

In machinery applications the peak radial loads on a bearing are different from the nominal or average load owing to a variation in torque or other influences. For a number of situations in which there is a body of measurement and experience, bearing manufacturers tabulate application factors that are used to multiply the average load to properly account for the additional fatigue damage resulting from the fluctuations. Such factors perform the same function as a design factor. In previous equations, F_D is replaced by nF_D or $AF(F_D)$, where AF is the application factor.

18.7 VARIABLE LOADING

At constant reliability the current $F^a L$ product measures progress toward failure. The area under the F^a versus L curve at failure is an index to total damage resulting in failure. The area under the $F^a L$ locus at any time prior to failure is an index to damage so far. If the radial load or equivalent radial load varies during a revolution or several revolutions in a periodic fashion, then the equivalent radial load is related to the instantaneous radial load by

$$F_{eq} = \left(\frac{1}{\phi} \int_0^\phi F^a d\theta \right)^{1/a} \quad (18.20)$$

where ϕ = period of the variation— 2π for repetition every revolution, 4π for repetition every second revolution, etc. (see Fig. 18.17).

Example 4. A bearing load is given by $F(\theta) = 1000 \sin \theta$ in pounds force. Estimate the equivalent load by using Simpson's rule,

$$F_{eq} = \left[\frac{1}{\pi} \int_0^\pi (1000 \sin \theta)^{10/3} d\theta \right]^{3/10} = 762 \text{ lb}$$

When equivalent loads are applied in a stepwise fashion, the equivalent radial load is expressible by

$$F_{eq} = \left[\sum_{i=1}^k f_i (n_i F_i)^a \right]^{1/a} \quad (18.21)$$

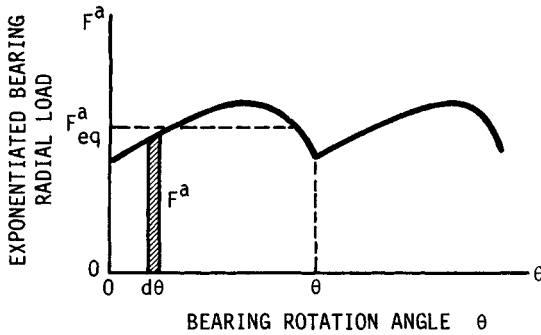


FIGURE 18.17 Equivalent radial load when load varies periodically with angular position.

where f_i = fraction of revolution at load F_i
 n_i = application or design factor
 F_i = i th equivalent radial load
 a = applicable exponent—3 for ball bearings and 10 for roller bearings

Example 5. A four-step loading cycle is applied to a ball bearing. For one-tenth of the time, the speed is 1000 rpm, $F_r = 800$ lb, and $F_a = 400$ lb; for two-tenths of the time, the speed is 1200 rpm, $F_r = 1000$ lb, and $F_a = 500$ lb; for three-tenths of the time, the speed is 1500 rpm, $F_r = 1500$ lb, and $F_a = 700$ lb; for four-tenths of the time, the speed is 800 rpm, $F_r = 1100$ lb, and $F_a = 500$ lb. For this shallow-angle, angular-contact ball bearing, $X_1 = 1$, $Y_1 = 1.25$, $X_2 = 0.45$, $Y_2 = 1.2$, and $V = 1$. This loading cycle is also depicted in Fig. 18.18.

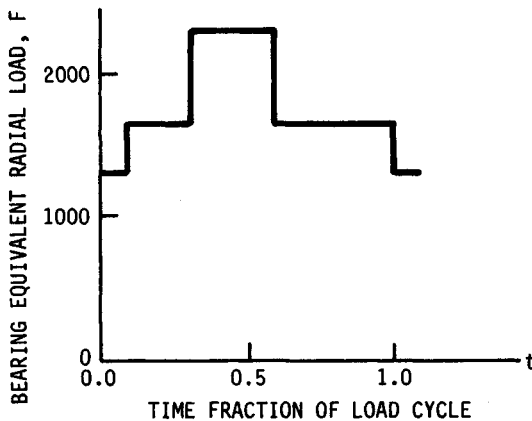


FIGURE 18.18 Loading cycle: one-tenth of time at 1000 rpm, $F_r = 800$, $F_a = 400$; two-tenths of time at 1200 rpm, $F_r = 1000$, $F_a = 500$; three-tenths of time at 1500 rpm, $F_r = 1500$, $F_a = 700$; four-tenths of time at 800 rpm, $F_r = 1100$, $F_a = 500$; $X_1 = 1$, $Y_1 = 1.25$, $X_2 = 0.45$, $Y_2 = 1.2$, $V = 1$.

TABLE 18.5 Tabulation for Example 5

Time fraction	Speed, rpm	Product	Revolution fraction f_i	Radial load F_r	Axial load F_a	Equivalent load F_e	Application factor AF	Product (AF)(F_e)
0.1	1000	100	0.090	800	400	1300	1.1	1430
0.2	1200	240	0.216	1000	500	1625	1.25	2031
0.3	1500	450	0.405	1500	700	2375	1.25	2969
0.4	800	<u>320</u>	0.288	1100	500	1725	1.50	2588
		1110						

The first step in the solution is to create Table 18.5. The equivalent radial load is

$$F_{eq} = [0.090(1430)^3 + 0.216(2031)^3 + 0.405(2969)^3 + 0.288(2588)^3]^{1/3} = 2604 \text{ lb}$$

Without the use of design factors, the equivalent radial load is

$$F_{eq} = [0.090(1300)^3 + 0.216(1625)^3 + 0.405(2375)^3 + 0.288(1725)^3]^{1/3} = 2002 \text{ lb}$$

The overall design factor is 2604/2002, or 1.30. If this sequence were common in a machinery application, a bearing manufacturer might recommend an application factor of 1.30 for this particular application.

18.8 MISALIGNMENT

The inner ring of a rolling-contact bearing is tightly fitted to the shaft, and the axis of rotation is oriented, as is the shaft centerline. The outer ring is held by some form of housing, and its axis is oriented as demanded by the housing. As the shaft deflects under load, these two axes lie at an angle to each other. This misalignment for very small angles is accommodated in "slack," and no adverse life consequences are exhibited. As soon as the slack is exhausted, the intended deflection is resisted and the bearing experiences unintended loading. Life is reduced below prediction levels. A shaft design which is too limber does not fail, but bearings are replaced with much greater frequency. It is too easy to be critical of bearings when the problem lies in the shaft design.

Figure 18.19 shows the dramatic fractional life reduction owing to misalignment in line-contact bearings [18.4].

If there is misalignment, it should not exceed 0.001 radian (rad) in cylindrical and tapered-roller bearings, 0.0087 rad for spherical ball bearings, or about 0.004 rad for

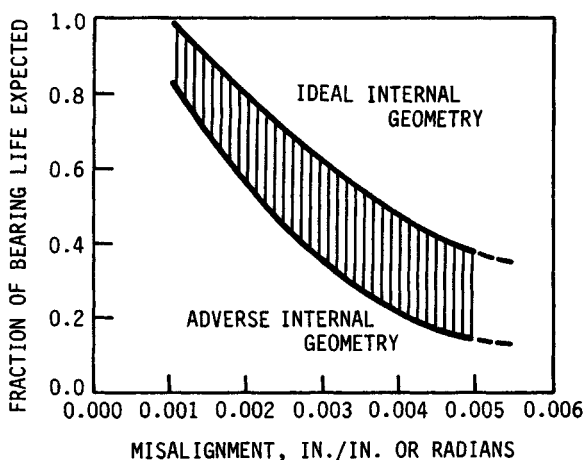


FIGURE 18.19 Fractional bearing life to be expected as a function of misalignment in line-contact bearings. (From Ref. [18.4], Fig. 11.)

deep-groove ball bearings. Self-aligning ball or spherical roller bearings are more tolerant of misalignment. The bibliography of Ref. [18.4] is extensive on this subject.

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