Contents

4 Rolling Bearings

4.1 Generalities

A bearing is a connector that permits the connected parts to rotate or to move relative to one another. Often one of the parts is fixed, and the bearing acts as a support for the moving part. Most bearings support rotating shafts against either transverse (radial) or thrust (axial) forces. To minimize friction, the contacting surfaces in a bearing may be partially or completely separated by a film of liquid (usually oil) or gas. These are *sliding bearings*, and the part of the shaft that turns in the bearing is the journal. Under certain combinations of force, speed, fluid viscosity, and bearing geometry, a fluid film forms and separates the contacting surfaces in a sliding bearing, and this is known as a *hydrodynamic* film. The *hydrostatic film* is the oil film that can be developed with a separate pumping unit that supplies pressurized oil.

The surfaces in a bearing can also be separated by balls, rollers, or needles; these are known as *rolling bearings*. Because shaft speed is required for the development of a hydrodynamic film, the starting friction in hydrodynamic bearings is higher than in rolling bearings. To minimize friction for metal-to-metal contact, in hydrodynamic bearings, materials with low coefficient of friction have been developed (bronze alloys and babbitt metal).

The principal advantage of the rolling bearings is the ability to operate at friction levels considerably lower at start-up, the friction coefficient having the values $\mu = 0.001 - 0.003$. Other advantages over bearings with sliding contact are: accurate shaft alignment for long periods of time, easy lubrication, little attention, easy replacement in case of failure, and heavy momentary overloads without failure.

The rolling bearings have the following disadvantages: design and processing of the shaft and house are more complicated, higher cost, more noise for higher speeds, and lower resistance to impact forces.

4.2 Classification

The important parts of rolling bearings are illustrated in Fig. 4.1: these include the outer ring, inner ring, rolling element, and separator (retainer). The role of the separator is to maintain an equal distance between the rolling elements. The races are the outer ring or the inner ring of a bearing. The raceway is the path of the rolling element on either ring or the bearing.

Rolling bearings can be classified using the following criteria (Fig. 4.2):

• the rolling element shape: ball bearings [Figs. 4.2(a)-(f)], roller bearings [cylinder, Figs. 4.2(g) and (h), cone, Fig. 4.2(i), barrel, Fig. 4.2(j)], and needle bearings [Fig. 4.2(k)];

• the direction of the principal force: radial bearings [Figs. 4.2(a)(b)(g)(h)], thrust bearings [Figs. 4.2(d)(e)], radial-thrust bearings [Figs. 4.2(c)(i)], or thrust-radial bearings [Fig. 4.2(f)];

• the number rolling bearing rows: rolling bearings with one row [Figs. 4.2(a)(d)(g)(k)], with two rows [Figs. 4.2(b)(e)(h)], etc.

The radial bearing is primarily designed to support a force perpendicular to the shaft axis. The thrust bearing is primarily designed to support a force parallel to the shaft axis.

Single row rolling bearings are manufactured to take radial forces and some thrust forces. The angular contact bearings provide a greater thrust capacity. Double row bearings are made to carry heavier radial and thrust forces. The single row bearings will withstand a small misalignment or deflection of the shaft. The self-aligning bearings [Fig. 4.2(f)] are used for severe misalignments and deflections of the shaft.

Cylinder roller bearings provide a greater force than ball bearings of the same size because of the greater contact area. This type of bearing will not take thrust forces. Tapered (cone) roller bearings combine the advantages of ball and cylinder roller bearings, because they can take either radial or thrust forces, and they have high force capacity.

Needle bearings are used where the radial space is limited, and when the separators are used they have high force capacity. In many practical cases they are used without the separators.

4.3 Geometry

Figure 4.3 shows a ball bearing with the *pitch diameter* given by

$$d_m \approx \frac{d_0 + d}{2}, \tag{4.1}$$

where d_0 is the outer diameter of the ball bearing and d is the bore.

Exactly, the *pitch diameter* can be calculated as

$$d_m = \frac{D_i + D_e}{2}, \tag{4.2}$$

where D_i is the race diameter of the inner ring and D_e is the race diameter of the outer ring.

In general the ball bearings are manufactured with a clearance between the balls and the raceways. The clearance measured in the radial plane is the *diametral clearance*, s_d, and is computed with the relation (Fig. 4.3):

$$s_d = D_e - D_i - 2D, \tag{4.3}$$

where D is the ball diameter.

Because a radial ball bearing has a diametral clearance in the no load state, the bearing also has an axial clearance. Removal of this axial clearance causes the ball raceway contact to assume an oblique angle with the radial plane. Angular contact ball bearings are designed to operate under thrust force and the clearance built into the unloaded bearing along with the raceway groove curvatures determines the bearing free contact angle. Due to the diametral clearance for a radial ball bearing there is what is known as *free endplay*, s_a, (Fig. 4.4). In Fig. 4.4 the center of the outer ring raceway circle is O_e, the center of the inner ring raceway circle is O_i, and the center of the ball is O.

The distance between the centers O_e and O_i is

$$A = r_e + r_i - D, \tag{4.4}$$

where r_e is the radius of the outer ring raceway and r_i is the radius of the inner ring raceway.

If the raceway groove curvature radius is $r = f\,D$, where f is a dimensionless coefficient, then

$$A = (f_e + f_i - 1)\,D = B\,D, \tag{4.5}$$

where $B = f_e + f_i - 1$ is defined as the *total curvature of the bearing*. In the previous formula $r_e = f_e\,D$ and $r_i = f_i\,D$, where f_e and f_i are dimensionless coefficients.

The *free contact angle*, α_0, is the angle made by the line passing through the points of contact of the ball and both raceways and a plane perpendicular to the bearing axis of rotation (Fig. 4.4). The magnitude of the free contact angle can be written as

$$\sin\alpha_0 = 0.5\,s_a/A. \tag{4.6}$$

The diametral clearance can allow the ball bearing to misalign slightly under no load. The *free angle of misalignment*, θ, is defined as the maximum angle through which the axis of the inner ring can be rotated with respect to the axis of the outer ring before stressing bearing components

$$\theta = \theta_i + \theta_e \,, \tag{4.7}$$

where θ_i [Fig. 4.5(a)] is the misalignment angle for the inner ring

$$\cos\theta_i = 1 - \frac{s_d\left[\left(2f_i - 1\right)D - s_d/4\right]}{2d_m\left[d_m + \left(2f_i - 1\right)D + s_d/2\right]} \,, \tag{4.8}$$

and θ_e [Fig. 4.5(b)] is the misalignment angle for the outer ring

$$\cos\theta_e = 1 - \frac{s_d\left[\left(2f_e - 1\right)D - s_d/4\right]}{2d_m\left[d_m - \left(2f_e - 1\right)D + s_d/2\right]} \,. \tag{4.9}$$

With the following trigonometric identity:

$$\cos\theta_i + \cos\theta_e = 2\cos\left[\left(\theta_i + \theta_e\right)/2\right]\cos\left[\left(\theta_i - \theta_e\right)/2\right] \,, \tag{4.10}$$

and with the approximation $\theta_i - \theta_e \approx 0$, the free angle of misalignment becomes

$$\theta = 2\arccos\left[\left(\cos\theta_i + \cos\theta_e\right)/2\right] \,, \tag{4.11}$$

or

$$\theta = 2\arccos\left\{1 - \frac{s_d}{4d_m}\left[\frac{\left(2f_i - 1\right)D - s_d/4}{d_m + \left(2f_i - 1\right)D + s_d/2} + \frac{\left(2f_e - 1\right)D - s_d/4}{d_m - \left(2f_e - 1\right)D + s_d/2}\right]\right\} . \tag{4.12}$$

4.4 Static Loading

In Fig. 4.6(a) a single row radial thrust (angular contact) ball bearing is shown. The *contact angle*, α, is the angle of the axis of contact between balls and races. For a single row radial ball bearing the angle α is zero. If F_r is the radial force applied to the ball, then the normal force to be supported by the ball is

$$F = \frac{F_r}{\cos\alpha}, \tag{4.13}$$

and the axial force, F_a (or F_t), is

$$F_a = F\sin\alpha. \tag{4.14}$$

For self-aligning roller bearings [Fig. 4.6(b)] the above relations are valid for each roller, and the total axial force is zero.

For taper roller bearings [Fig. 4.6(c)] there are three contact angles: α_i the contact angle for the inner ring, α_e the contact angle for the outer ring, and α_f the contact angle for the frontal face.

The normal and axial forces for the inner ring are

$$F_i = \frac{F_{ri}}{\cos\alpha_i} \text{ and } F_{ai} = F_{ri} \tan\alpha_i, \tag{4.15}$$

where F_{ri} is the radial force acting on the inner ring. The normal and axial forces for the outer ring are

$$F_e = \frac{F_{re}}{\cos\alpha_e} \text{ and } F_{ae} = F_{re} \tan\alpha_e, \tag{4.16}$$

where F_{re} is the radial force acting on the outer ring. The normal and axial forces for the frontal face are

$$F_f = \frac{F_{rf}}{\cos\alpha_f} \text{ and } F_{af} = F_{rf} \tan\alpha_f, \tag{4.17}$$

where F_{rf} is the radial force acting on the frontal face.

The equilibrium equations for radial and axial directions are

$$F_{ri} - F_{rf} - F_{re} = 0 \text{ or } F_{ri} - F_f \cos\alpha_f - F_e \cos\alpha_e = 0, \tag{4.18}$$

$$F_{ai} + F_{af} - F_{ae} = 0 \text{ or } F_{ri} \tan\alpha_i + F_f \sin\alpha_f - F_e \sin\alpha_e = 0. \tag{4.19}$$

From Eqs. (4.18) and (4.19) the forces F_e and F_f are obtained:

$$F_e = \frac{F_{ri}\left(\sin\alpha_f + \tan\alpha_i \cos\alpha_f\right)}{\sin\left(\alpha_e + \alpha_f\right)}, \tag{4.20}$$

$$F_f = \frac{F_{ri}\left(\sin\alpha_e - \tan\alpha_i \cos\alpha_e\right)}{\sin\left(\alpha_e + \alpha_f\right)}. \tag{4.21}$$

4.5 Standard Dimensions

The Annular Bearing Engineers Committee (ABEC) of the Anti-Friction Bearing Manufacturers Association (AFBMA) has established four primary grades of precision, designated ABEC 1, 5, 7, and 9 for ball bearings. The standard grade is ABEC 1 and is adequate for most normal applications. The other grades have progressively finer tolerances. The AFBMA Roller Bearing Engineers Committee has established RBEC standards 1 and 5 for cylindrical roller bearings.

The bearing manufacturers have established standard dimensions (Fig. 4.7 and Table 4.1) for ball and straight roller bearings in the metric sizes, which define the bearing bore d, the outside diameter d_0, the width w, the fillet sizes on the shaft and housing shoulders r, the shaft diameter d_S, and the housing diameter d_H.

For a given bore, there is an assortment of widths and outside diameters. Furthermore, the outside diameters selected are such that, for a particular outside diameter, one can usually find a variety of bearings having different bores. That is why the bearings are made in various proportions for different series (Fig. 4.8): extra-extra-light series (LL00), extra-light series (L00), light series (200) and medium series (300).

4.6 Bearing Selection

Bearing manufacturers' catalogues identify bearings by number, give complete dimensional information, list rated load capacities, and furnish details concerning mounting, lubrication, and operation. Lubrication (fine oil mist or spray) is important in high-speed bearing applications. For ball bearings nonmetallic separators permit highest speeds.

The size of bearing selected for an application is usually influenced by the size of shaft required (for strength and rigidity considerations) and by the available space. In addition, the bearing must have a high load rating to provide a good combination of life and reliability.

Juvinall and Marshek [7] proposed the following expressions for bearing selection.

Life requirement

The *life* of an individual ball or roller bearing is the number of revolutions (or hours at some constant speed) that the bearings run before the first evidence of fatigue develops in the material of either the rings or of any of the rolling elements. Bearing applications usually require lives different from that used for the catalogue rating. Palmgren determined that ball bearing life varies inversely with approximately the third power of the force. Later studies have indicated that this exponent ranges between 3 and 4 for various rolling-element bearings. Many manufacturers retain Palmgren's exponent of 3 for ball bearings and use 10/3 for roller bearings. Following the recommendation of other manufacturers, the exponent 10/3 will be used for both bearing types. Thus, the *life required* by the application is

$$L = L_R\,(C/F_r)^{10/3}\,, \tag{4.22}$$

where C is the *rated capacity*, from Table 4.2, L_R is the life corresponding to rated capacity (i.e., $L_R = 9 \times 10^7$ revolutions), and F_r is radial force involved in the application.

The values of the rated capacity in Table 4.2 correspond to a constant radial load that 90 of a group of identical bearings can endure a rating life of $L_R = 9 \times 10^7$ revolutions without surface fatigue failure.

The required value of the rated capacity for the application is

$$C_{req} = F_r\,(L/L_R)^{3/10}\,. \tag{4.23}$$

For a group of apparently identical bearings the *rating life*, L_R, is the life in revolutions (at a given constant speed and force) that 90 percent of the group tested bearings will exceed before the first evidence of fatigue develops. For example, the Timken Company rates the bearings for 3000 h at a speed of 500 rpm operation. The corresponding life is

$$L = (3000\,\mathrm{h})\left(\frac{60\,\mathrm{min}}{\mathrm{h}}\right)\left(\frac{500\,\mathrm{rev}}{\mathrm{min}}\right) = 9\left(10^7\right)\,\mathrm{rev}.$$

Different manufacturers' catalogues use different values of L_R (some use 10^6 revolutions).

Reliability requirement

Tests show that the *median life* of rolling-element bearings is about five times the standard 10 percent failure fatigue life. The *standard life* is commonly designated as the L_{10} life (sometimes as the B_{10} life) and this life corresponds to 10 percent failures. It means that this is the life for which 90 percent have not failed, and corresponds to 90 percent reliability ($r = 90\%$). Using the general Weibull equation together with extensive experimental data, a life adjustment *reliability factor*, K_r, is recommended. The life adjustment reliability factor K_r is plotted in Fig. 4.9. This factor is applicable to both ball and roller bearings. The rated bearing life for any given reliability (greater than 90 percent) is thus the product $K_r\,L_R$. Incorporating this factor into Eqs. (4.22) and (4.23) gives

$$L = K_r L_R \, (C/F_r)^{3.33} \, ,$$
$$C_{req} = F_r \left(\frac{L}{K_r L_R} \right)^{0.3} \, . \tag{4.24}$$

Influence of axial force

For ball bearings (load angle $\alpha = 0°$) any combination of radial force (F_r) and thrust force (F_a) results in approximately the same life as does a pure *radial equivalent force*, F_e, calculated as

- for $0.00 < F_a/Fr < 0.35 \implies F_e = F_r$;
- for $0.35 < F_a/F_r < 10.0 \implies F_e = F_r\,[1 + 1.115(F_a/F_r - 0.35)]$;
- for $F_a/Fr > 10.0 \implies F_e = 1.176\,F_a$. Standard values of load angle α for angular ball bearings are 15°, 25°, and 35°. Only the 25° angular ball bearings will be discussed here. The radial equivalent force, F_e, for angular ball bearings with $\alpha = 25°$ is
- for $0.00 < F_a/Fr < 0.68 \implies F_e = F_r$;
- for $0.68 < F_a/F_r < 10.0 \implies F_e = F_r\,[1 + 0.87(F_a/F_r - 0.68)]$;
- for $F_a/Fr > 10.0 \implies F_e = 0.911\,F_a$.

Shock force

The standard bearing rated capacity is for the condition of uniform force without shock, which is a desirable condition. In many applications there are various degrees of shock loading. This has the effect of increasing the nominal force by an *application factor*, K_a. In Table 4.3 some representative sample values of K_a are given. The force application factor in Table 4.3 serves the same purpose as factors of safety.

Substituting F_e for F_r and adding K_a, Eq. (4.24) gives

$$L = K_r L_R \left(\frac{C}{K_a F_e} \right)^{3.33} ,$$
$$C_{req} = K_a F_e \left(\frac{L}{K_r L_R} \right)^{0.3} . \tag{4.25}$$

When more specific information is not available, Table 4.4 can be used as a guide for the life of a bearing in industrial applications. Table 4.4 contains recommendations on bearing life for some classes of machinery. The information has been accumulated by experience.

Shigley and Mischke [20] proposed the following expressions for bearing selection.

The minimum basic load rating (load for which 90% of the bearings from a given group will survive 1 million revolutions is defined as

$$C_s = P\,L^{1/a},$$

where P is the design load, L is the design life in millions of revolutions, and $a = 3$ for ball bearings and $a = 10/3$ for roller bearings. The equivalent radial load is

$$P_e = X\,V\,F_r + Y\,F_a,$$

where F_r is the radial force and F_a is the thrust force. For a rotating inner ring V=1, and for a rotating outer ring V=1.2. The AFBMA recommendations are based on the ratio of the thrust force F_a to the basic static load rating C_0, and a variable reference value,

$$e = 0.513\left(\frac{F_a}{C_0}\right)^{0.236}.$$

The static load rating C_0 is tabulated in bearing catalogs. The X and Y factors have the values

- for $F_a/(V\,F_r) > e \implies X$=0.56 and $Y = 0.840\left(\frac{F_a}{C_0}\right)^{-0.247}$,
- for $F_a/(V\,F_r) \leq e \implies X$=1 and $Y = 0$.

4.7 Examples

Example 4.1. Select a light series (200) radial ball bearing for a machine for continuous 24 hour service. The machine rotates at the angular speed of 1000 rpm. The radial force is F_r=1.5 kN, and the thrust force is F_a=1.8 kN, with light impact (Fig. 4.10).

Solution.

For $F_a/F_r = 1.8/1.5$=1.2, the equivalent radial force (for radial ball bearings with $0.35 < F_a/F_r < 10.0$) is

$F_e = F_r\,[1 + 1.115(F_a/F_r - 0.35)]$=2.921 kN.
From Table 4.3 choose (conservatively) $K_a = 1.5$ for light impact.
From Table 4.4 choose (conservatively) 60 000 hour life.
The life in revolutions is

$L = 1000$ rpm $\times$ 60 000 h $\times$ 60 min/h = 3 600 $\times$ 10^6 rev.
For standard 90 percent reliability (K_r=1, Fig. 4.9), and for $L_R = 90 \times 10^6$ rev (for use with Table. 4.2), Eq. (4.25) gives

$$C_{req} = K_a F_e \left(\frac{L}{K_r L_R}\right)^{0.3} = (1.5)(2.921)(3\,600/90)^{0.3} = 13.253\,\text{kN}$$

From Table 4.2 with 13.253 kN for 200 series $\Longrightarrow$ C=13.6 kN and d=60 mm bore. From Table 4.1 with 60 mm bore and 200 series the bearing number is 212.

Example 4.2. A no. 305 radial contact ball bearing carries a radial load of 4 kN, 5 kN, and 6 kN for, respectively, 50%, 40%, and 10% of the time. The loads are uniform, so that $K_a = 1$. The bearing supports a shaft that rotates with 2000 rpm. Determine the B_{10} life and the median life of the bearing.

Solution.

For no. 305 radial contact bearing from Table 4.2, the rated capacity is $C = 5.9$ kN with $L_R = 90 \times 10^6$ and standard 90% reliability ($K_r = 1$). Equation (4.25) gives

$$L = K_r\ L_R \left(\frac{C}{K_a\ F_e}\right)^{3.33},$$

where $K_a = 1$ and $F_e = F_r$.

The corresponding life is calculated from the previous relation:

$$N_1 = (1)(90 \times 10^6)\left[\frac{5.9}{(1)(4)}\right]^{3.33} = 3.283 \times 10^8 \text{ rev, for } F_r = 4 \text{ kN};$$

$$N_2 = (1)(90 \times 10^6)\left[\frac{5.9}{(1)(5)}\right]^{3.33} = 1.561 \times 10^8 \text{ rev, for } F_r = 5 \text{ kN};$$

$$N_3 = (1)(90 \times 10^6)\left[\frac{5.9}{(1)(6)}\right]^{3.33} = 8.510 \times 10^7 \text{ rev, for } F_r = 6 \text{ kN}.$$

The Miner rule is

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} = 1, \tag{4.26}$$

where

$n_1 = (50\%)\, n\, X = (0.5)(2000 \text{ rpm})\, X = 1000\, X$ rev, for 50% of the time;
$n_2 = (40\%)\, n\, X = (0.4)(2000 \text{ rpm})\, X = 800\, X$ rev, for 40% of the time;
$n_3 = (10\%)\, n\, X = (0.1)(2000 \text{ rpm})\, X = 200\, X$ rev, for 10% of the time.

The minutes of operation are $X = B_{10}$ and the shaft rotates with $n =$ 2000 rpm.

Equation (4.26) gives

$$\frac{1000\, X}{3.283 \times 10^8} + \frac{800\, X}{1.561 \times 10^8} + \frac{200\, X}{8.510 \times 10^7} = 1,$$

or $X = B_{10} = 95072.6$ minutes (=1584.54 hours).

The median life is approximately five times the B_{10} life or 475363 minutes (7922.72 hours).

4.8 Problems

4.1 A number 208 radial ball bearing has a 4000 hour B_{10} life at 1200 rpm. Find the bearing radial capacity.

4.2 A radial ball bearing has a given radial load F and a given life L. Find the radial load F_{new} if the life of the bearing is tripled, $L_{new} = 3\,L$.

4.3 A number 207 angular ball bearing is selected to carry a radial load of 250 lb and a thrust load of 150 lb at 1000 rpm. Determine the bearing life B_{10} for steady loading.

4.4 A ball bearing can withstand a radial load of 4 kN and a thrust load of 6 kN at a speed of 600 rpm. The bearing is intended for an aircraft engine with heavy impacts. Select an angular ball bearing for this application.

4.5 The life of a bearing for 90% reliability is 12000 h. Determine the lives of the bearing for respectively 60% and 98% reliability.

4.6 A ball bearing carries a radial load of 2.5 kN and a thrust load of 1.5 kN at 900 rpm. The application is considered to be light to moderate with respect to shock loading. The required life is 6000 hours with only 4% probability of failure. Select a suitable ball bearing: a) radial ball bearing and b) angular ball bearing.

4.7 Repeat the previous problem for 3 kN radial load, 2 kN thrust load at 1200 rpm, and 2% probability of failure.

4.8 A number 211 radial ball bearing is intended for a continuous one-shift (8 h per day) operation at 1000 rpm. The radial load varies in such a way that 60% of the time the load is 5 kN and 40% of the time the load is 10 kN. The application factor $K_a = 1.5$ (light to moderate impact). Estimate the B_{10} life and the median life of the radial ball bearing.

4.9 A no. 207 radial-contact ball bearing supports a shaft that rotates 1500 rpm. A radial load varies in such a way that 30%, 30%, and 40% of the time the load is 5, 2, and 10 kN. The loads are uniform, so that $K_a = 1$. Estimate the B_{10} life and the median life of the bearing.

4.10 The shaft shown in Fig. 4.11 rotates at 900 rpm and is supported by radial ball bearings at points A and B. The length dimension of the shaft is $l = 300$ mm. The radial force acting on the shaft at R is on the yz plane and has the magnitude $F_R = 900$ N. The angle of the force F_R with the z-axis is $\alpha = 45°$. The thrust load (along x-axis) on the bearing at A is $F_{aA} = 500$ N and at B is $F_{aB} = 500$ N. The bearings are subjected to steady loading with 98% reliability and 30 000 hours of life. Select radial ball bearings for A and B.

4.11 Figure 4.12 shows two bearings at A and B supporting a shaft that rotates at 1000 rpm. The loads acting on the shaft are at point P, $\mathbf{F}_P = F_{Py}\mathbf{J} + F_{Py}\mathbf{k} = -500\,\mathbf{J} + 600\,\mathbf{k}$ N, and at point R, $\mathbf{F}_R = F_{Ry}\mathbf{J} = -1000\,\mathbf{J}$ N. The loading is light to moderate impact. The length dimensions are $s = 100$ mm and $l = 200$ mm. The required life is 5000 hours with only 2% probability of failure. Select identical ball bearings for A and B.

4.12 Figure 4.13 shows a countershaft a with two rigidly connected gears 1 and 2. The angular speed of the countershaft is 300 rpm. The force on the countershaft gear 1 at P is $\mathbf{F}_P = F_{Py}\mathbf{J} + F_{Pz}\mathbf{k} = -200\mathbf{J} + 500\mathbf{k}$ N, and the force on the gear 2 at R is $\mathbf{F}_R = F_{Ry}\mathbf{J} + F_{Rz}\mathbf{k} = -600\mathbf{J} - 1500\mathbf{k}$ N. The radius of the gear 1 is $OP = 0.15$ m and the radius of the gear 2 is $QP = 0.05$ m. The distance between the bearings is $s = 100$ mm and the other distance is $l = 25$ mm. The gear reducer is a part of an industrial machine intended for continuous one-shift (8 hours per day). Select identical extra-light series (L00) ball bearings for A and B.

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Figure captions

Fig. 4.1. Rolling bearing nomenclature.
Fig. 4.2. Rolling bearing classification.
Fig. 4.3. Ball bearing geometry.
Fig. 4.4. Clearance for radial ball bearing.
Fig. 4.5. Misalignment angle for the rings.
Fig. 4.6. Static loading for rolling bearing
Fig. 4.7. Standard dimensions for rolling bearing, shaft, and housing shoulder.
Fig. 4.8. Different series for rolling bearing.
Fig. 4.9 Life adjustment reliability factor
Fig. 4.10. Radial ball bearing for Example 4.1.
Fig. 4.11. Sketch of the shaft for Problem 4.10.
Fig. 4.12. Sketch of the shaft for Problem 4.11.
Fig. 4.13. Sketch of the shaft for Problem 4.12.