5. SELECTION OF BEARING SIZE

5.1 Bearing Life

The various functions required of rolling bearings vary according to the bearing application. These functions must be performed for a prolonged period. Even if bearings are properly mounted and correctly operated, they will eventually fail to perform satisfactorily due to an increase in noise and vibration, loss of running accuracy, deterioration of grease, or fatigue flaking of the rolling surfaces.
Bearing life, in the broad sense of the term, is the period during which bearings continue to operate and to satisfy their required functions. This bearing life may be defined as noise life, abrasion life, grease life, or rolling fatigue life, depending on which one causes loss of bearing service.

Aside from the failure of bearings to function due to natural deterioration, bearings may fail when conditions such as heat-seizure, fracture, scoring of the rings, damage of the seals or the cage, or other damage occurs.
Conditions such as these should not be interpreted as normal bearing failure since they often occur as a result of errors in bearing selection, improper design or manufacture of the bearing surroundings, incorrect mounting, or insufficient maintenance.

5.1.1 Rolling Fatigue Life and Basic Rating Life

When rolling bearings are operated under load, the raceways of their inner and outer rings and rolling elements are subjected to repeated cyclic stress. Because of metal fatigue of the rolling contact surfaces of the raceways and rolling elements, scaly particles may separate from the bearing material (Fig. 5.1). This phenomenon is called "flaking". Rolling fatigue life is represented by the total number of revolutions at which time the bearing surface will start flaking due to stress. This is called fatigue life. As shown in Fig. 5.2, even for seemingly identical bearings, which are of the same type, size, and material and receive the same heat treatment and other processing, the rolling fatigue life varies greatly even under identical operating conditions. This is because the flaking of materials due to fatigue is subject to many other variables. Consequently, "basic rating life", in which rolling fatigue life is treated as a statistical phenomenon, is used in preference to actual rolling fatigue life.

Suppose a number of bearings of the same type are operated individually under the same conditions. After a certain period of time, 10 % of them fail as a result of flaking caused by rolling fatigue. The total number of revolutions at this point is defined as the basic rating life or, if the speed is constant, the basic rating life is often expressed by the total number of operating hours completed when 10 % of the bearings become inoperable due to flaking.

In determining bearing life, basic rating life is often the only factor considered. However, other factors must also be taken into account. For example, the grease life of grease-prelubricated bearings (refer to Section 12, Lubrication, Page A107) can be estimated. Since noise life and abrasion life are judged according to individual standards for different applications, specific values for noise or abrasion life must be determined empirically.

5.2 Basic Load Rating and Fatigue Life

5.2.1 Basic Load Rating

The basic load rating is defined as the constant load applied on bearings with stationary outer rings that the inner rings can endure for a rating life of one million revolutions (10^6 rev). The basic load rating of radial bearings is defined as a central radial load of constant direction and magnitude, while the basic load rating of thrust bearings is defined as an axial load of constant magnitude in the same direction as the central axis. The load ratings are listed under C_r for radial bearings and C_a for thrust bearings in the dimension tables.

5.2.2 Machinery in which Bearings are Used and Projected Life

It is not advisable to select bearings with unnecessarily high load ratings, for such bearings may be too large and uneconomical. In addition, the bearing life alone should not be the deciding factor in the selection of bearings. The strength, rigidity, and design of the shaft on which the bearings are to be mounted should also be considered. Bearings are used in a wide range of applications and the design life varies with specific applications and operating conditions. Table 5.1 gives an empirical fatigue life factor derived from customary operating experience for various machines. Also refer to Table 5.2.

Fig. 5.1 Example of Flaking

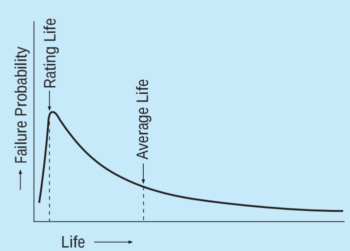

Fig. 5.2 Failure Probability and Bearing Life

Table 5. 1 Fatigue Life Factor f_h for Various Bearing Applications

Operating Periods	Fatigue Life Factor f_h ~3	2~4	3~5	4~7	6~
Infrequently used or only for short periods	·Small motors for home appliances like vacuum cleaners and washing machines ·Hand power tools	·Agricultural equipment			
Used only occasionally but reliability is important		·Motors for home heaters and air conditioners ·Construction equipment	·Conveyors ·Elevator cable sheaves		
Used intermittently for relatively long periods	·Rolling mill roll necks	·Small motors ·Deck cranes ·General cargo cranes ·Pinion stands ·Passenger cars	·Factory motors ·Machine tools ·Transmissions ·Vibrating screens ·Crushers	·Crane sheaves ·Compressors ·Specialized transmissions	
Used intermittently for more than eight hours daily		·Escalators	·Centrifugal separators ·Air conditioning equipment ·Blowers ·Woodworking machines ·Large motors ·Axle boxes on railway rolling stock	·Mine hoists ·Press flywheels ·Railway traction motors ·Locomotive axle boxes	·Paper making machines
Used continuously and high reliability is important					·Waterworks pumps ·Electric power stations ·Mine draining pumps

5.2.3 Selection of Bearing Size Based on Basic Load Rating

The following relation exists between bearing load and basic rating life:

For ball bearings $$L=\left(\frac{C}{P}\right)^{3} \quad \text{................} \ \mathbf{(5.1)}$$

For roller bearings $$L=\left(\frac{C}{P}\right)^{\frac{10}{3}} \quad \text{..............} \ \mathbf{(5.2)}$$

where L : Basic rating life (10^6 rev)
P : Bearing load (equivalent load) (N), {kgf}
..........(Refer to Page A30)
C : Basic load rating (N), {kgf}
For radial bearings, C is written C_r
For thrust bearings, C is written C_a

In the case of bearings that run at a constant speed, it is convenient to express the fatigue life in terms of hours. In general, the fatigue life of bearings used in automobiles and other vehicles is given in terms of mileage.

By designating the basic rating life as L_h (h), bearing speed as n (min^{-1}), fatigue life factor as f_h, and speed factor as f_n, the relations shown in Table 5.2 are obtained:

Table 5. 2 Basic Rating Life, Fatigue Life Factor and Speed Factor

Life Parameters	Ball Bearings	Roller Bearings
Basic Rating Life	$L_h=\frac{10^6}{60n}\left(\frac{C}{P}\right)^3=500f_h^3$	$L_h=\frac{10^6}{60n}\left(\frac{C}{P}\right)^{\frac{10}{3}}=500f_h^{\frac{10}{3}}$
Fatigue Life Factor	$f_h=f_n\frac{C}{P}$	$f_h=f_n\frac{C}{P}$
Speed Factor	$f_n=\left(\frac{10^6}{500\times60n}\right)^{\frac{1}{3}}=(0.03n)^{-\frac{1}{3}}$	$f_n=\left(\frac{10^6}{500\times60n}\right)^{\frac{3}{10}}=(0.03n)^{-\frac{3}{10}}$

n, f_n······Fig. 5.3 (See Page A26), Appendix Table 12 (See Page C24)
L_h, f_h····Fig. 5.4 (See Page A26), Appendix Table 13 (See Page C25)

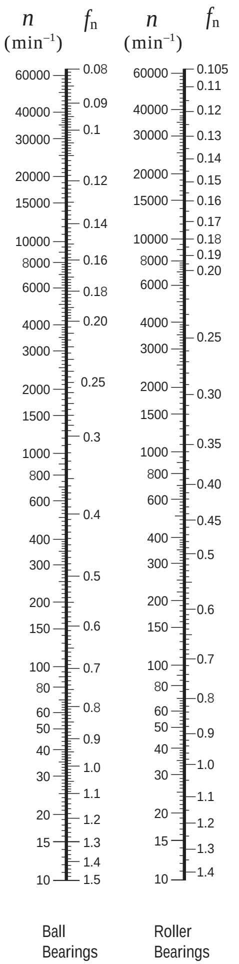

Fig. 5.3 Bearing Speed and Speed Factor

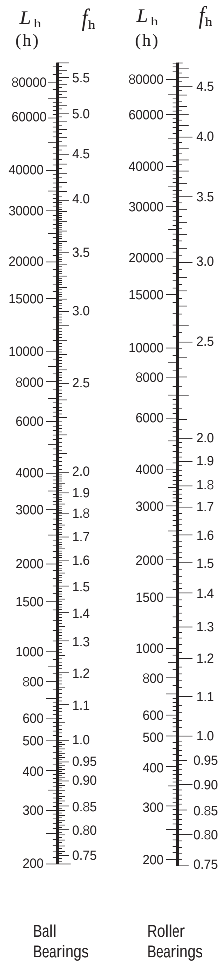

Fig. 5.4 Fatigue Life Factor and Fatigue Life

If the bearing load P and speed n are known, determine a fatigue life factor f_h appropriate for the projected life of the machine and then calculate the basic load rating C by means of the following equation.

$$C=\frac{f_h \cdot P}{f_n} \quad \text{.......................}(5.3)$$

A bearing which satisfies this value of C should then be selected from the bearing tables.

5.2.4 Temperature Adjustment for Basic Load Rating

If rolling bearings are used at high temperature, the hardness of the bearing steel decreases. Consequently, the basic load rating, which depends on the physical properties of the material, also decreases. Therefore, the basic load rating should be adjusted for the higher temperature using the following equation:

$$C_t=f_t \cdot C \quad \text{.......................}(5.4)$$

where C_t : Basic load rating after temperature correction (N), {kgf}
f_t : Temperature factor (See Table 5.3.)
C : Basic load rating before temperature adjustment (N), {kgf}

If large bearings are used at higher than 120°C, they must be given special dimensional stability heat treatment to prevent excessive dimensional changes. The basic load rating of bearings given such special dimensional stability heat treatment may become lower than the basic load rating listed in the bearing tables.

Table 5.3 Temperature Factor f_t

Bearing Temperature °C	125	150	175	200	250
Temperature Factor f_t	1.00	1.00	0.95	0.90	0.75

5.2.5 Correction of Basic Rating Life

As described previously, the basic equations for calculating the basic rating life are as follows:

For ball bearings $$L_{10}=\left(\frac{C}{P}\right)^3 \quad \text{................}(5.5)$$

For roller bearings $$L_{10}=\left(\frac{C}{P}\right)^{\frac{10}{3}} \quad \text{..............}(5.6)$$

The L_{10} life is defined as the basic rating life with a statistical reliability of 90%. Depending on the machines in which the bearings are used, sometimes a reliability higher than 90% may be required. However, recent improvements in bearing material have greatly extended the fatigue life. In addition, the developent of the Elasto-Hydrodynamic Theory of Lubrication proves that the thickness of the lubricating film in the contact zone between rings and rolling elements greatly influences bearing life. To reflect such improvements in the calculation of fatigue life, the basic rating life is adjusted using the following adjustment factors:

$$L_{na}=a_1\, a_2\, a_3\, L_{10} \quad \text{..........................}(5.7)$$

where L_{na} : Adjusted rating life in which reliability, material improvements, lubricating conditions, etc. are considered
L_{10} : Basic rating life with a reliability of 90%
a_1 : Life adjustment factor for reliability
a_2 : Life adjustment factor for special bearing properties
a_3 : Life adjustment factor for operating conditions

The life adjustment factor for reliability, a_1, is listed in Table 5.4 for reliabilities higher than 90%.
The life adjustment factor for special bearing properties, a_2, is used to reflect improvements in bearing steel.
NSK now uses vacuum degassed bearing steel, and the results of tests by NSK show that life is greatly improved when compared with earlier materials. The basic load ratings C_r and C_a listed in the bearing tables were calculated considering the extended life achieved by improvements in materials and manufacturing techniques. Consequently, when estimating life using Equation (5.7), it is sufficient to assume that is greater than one.

Table 5.4 Reliability Factor a_1

Reliability (%)	90	95	96	97	98	99
a_1	1.00	0.62	0.53	0.44	0.33	0.21

The life adjustment factor for operating conditions a_3 is used to adjust for various factors, particularly lubrication. If there is no misalignment between the inner and outer rings and the thickness of the lubricating film in the contact zones of the bearing is sufficient, it is possible for a_3 to be greater than one; however, a_3 is less than one in the following cases:

- When the viscosity of the lubricant in the contact zones between the raceways and rolling elements is low.
- When the circumferential speed of the rolling elements is very slow.
- When the bearing temperature is high.
- When the lubricant is contaminated by water or foreign matter.
- When misalignment of the inner and outer rings is excessive.

It is difficult to determine the proper value for a_3 for specific operating conditions because there are still many unknowns. Since the special bearing property factor a_2 is also influenced by the operating conditions, there is a proposal to combine a_2 and a_3 into one quantity($a_2 \times a_3$), and not consider them independently. In this case, under normal lubricating and operating conditions, the product ($a_2 \times a_3$) should be assumed equal to one. However, if the viscosity of the lubricant is too low, the value drops to as low as 0.2.
If there is no misalignment and a lubricant with high viscosity is used so sufficient fluid-film thickness is secured, the product of ($a_2 \times a_3$) may be about two.

When selecting a bearing based on the basic load rating, it is best to choose an a_1 reliability factor appropriate for the projected use and an empirically determined C/P or f_h value derived from past results for lubrication, temperature, mounting conditions, etc. in similar machines.

The basic rating life equations (5.1), (5.2), (5.5), and (5.6) give satisfactory results for a broad range of bearing loads. However, extra heavy loads may cause detrimental plastic deformation at ball/raceway contact points. When P_r exceeds C_{0r} (Basic static load rating) or 0.5 C_r, whichever is smaller, for radial bearings or P_a exceeds 0.5 C_a for thrust bearings, please consult NSK to establish the applicablity of the rating fatigue life equations.

5.3 Calculation of Bearing Loads

The loads applied on bearings generally include the weight of the body to be supported by the bearings, the weight of the revolving elements themselves, the transmission power of gears and belting, the load produced by the operation of the machine in which the bearings are used, etc. These loads can be theoretically calculated, but some of them are difficult to estimate. Therefore, it becomes necessary to correct the estimated using empirically derived data.

5.3.1 Load Factor

When a radial or axial load has been mathematically calculated, the actual load on the bearing may be greater than the calculated load because of vibration and shock present during operation of the machine. The actual load may be calculated using the following equation:

$$\left.\begin{aligned} F_r &= f_w \cdot F_{rc} \\ F_a &= f_w \cdot F_{ac} \end{aligned}\right\} \quad \cdots\cdots (5.8)$$

where F_r, F_a : Loads applied on bearing (N), {kgf}

F_{rc}, F_{ac} : Theoretically calculated load (N), {kgf}

f_w : Load factor

The values given in Table 5.5 are usually used for the load factor f_w.

Table 5. 5 Values of Load Factor f_w

Operating Conditions	Typical Applications	f_w
Smooth operation free from shocks	Electric motors, Machine tools, Air conditioners	1 to 1.2
Normal operation	Air blowers, Compressors, Elevators, Cranes, Paper making machines	1.2 to 1.5
Operation accompanied by shock and vibration	Construction equipment, Crushers, Vibrating screens, Rolling mills	1.5 to 3

5.3.2 Bearing Loads in Belt or Chain Transmission Applications

The force acting on the pulley or sprocket wheel when power is transmitted by a belt or chain is calculated using the following equations.

$$\left.\begin{aligned} M &= 9\,550\,000H / n \ldots (\mathrm{N \cdot mm}) \\ &= 974\,000H / n \ldots \{\mathrm{kgf \cdot mm}\} \end{aligned}\right\} \quad \cdots\cdots (5.9)$$

$$P_k = M / r \quad \cdots\cdots (5.10)$$

where M : Torque acting on pulley or sprocket wheel (N·mm), {kgf·mm}

P_k : Effective force transmitted by belt or chain (N), {kgf}

H : Power transmitted(kW)

n : Speed (min^{-1})

r : Effective radius of pulley or sprocket wheel (mm)

When calculating the load on a pulley shaft, the belt tension must be included. Thus, to calculate the actual load K_b in the case of a belt transmission, the effective transmitting power is multiplied by the belt factor f_b, which represents the belt tension. The values of the belt factor f_b for different types of belts are shown in Table 5.6.

$$K_b = f_b \cdot P_k \quad \cdots\cdots (5.11)$$

In the case of a chain transmission, the values corresponding to f_b should be 1.25 to 1.5.

Table 5. 6 Belt Factor f_b

Type of Belt	f_b
Toothed belts	1.3 to 2
V belts	2 to 2.5
Flat belts with tension pulley	2.5 to 3
Flat belts	4 to 5

5.3.3 Bearing Loads in Gear Transmission Applications

The loads imposed on gears in gear transmissions vary according to the type of gears used. In the simplest case of spur gears, the load is calculated as follows:

$$\left.\begin{aligned} M &= 9\,550\,000H / n \ldots (\mathrm{N \cdot mm}) \\ &= 974\,000H / n \ldots \{\mathrm{kgf \cdot mm}\} \end{aligned}\right\} \quad \cdots\cdots (5.12)$$

$$P_k = M / r \quad \cdots\cdots (5.13)$$

$$S_k = P_k \tan\theta \quad \cdots\cdots (5.14)$$

$$K_c = \sqrt{P_k^2 + S_k^2} = P_k \sec\theta \quad \cdots\cdots (5.15)$$

where M : Torque applied to gear (N · mm),{kgf · mm}

P_k : Tangential force on gear (N), {kgf}

S_k : Radial force on gear (N), {kgf}

K_c : Combined force imposed on gear (N), {kgf}

H : Power transmitted (kW)

n : Speed (min^{-1})

r : Pitch circle radius of drive gear (mm)

θ : Pressure angle

In addition to the theoretical load calculated above, vibration and shock (which depend on how accurately the gear is finished) should be included using the gear factor f_g by multiplying the theoretically calculated load by this factor.
The values of f_g should generally be those in Table 5.7. When vibration from other sources accompanies gear operation, the actual load is obtained by multiplying the load factor by this gear factor.

Table 5. 7 Values of Gear Factor f_g

Gear Finish Accuracy	f_g
Precision ground gears	1 ～1.1
Ordinary machined gears	1.1～1.3

5.3.4 Load Distribution on Bearings

In the simple examples shown in Figs. 5.5 and 5.6. The radial loads on bearings Ⅰ and Ⅱ can be calculated using the following equations:

$$F_{C\,\mathrm{I}} = \frac{b}{c} K \quad \cdots\cdots (5.16)$$

$$F_{C\,\mathrm{II}} = \frac{a}{c} K \quad \cdots\cdots (5.17)$$

where $F_{C\,\mathrm{I}}$: Radial load applied on bearing Ⅰ (N), {kgf}

$F_{C\,\mathrm{II}}$: Radial load applied on bearing Ⅱ (N), {kgf}

K : Shaft load (N), {kgf}

When these loads are applied simultaneously, first the radial load for each should be obtained, and then, the sum of the vectors may be calculated according to the load direction.

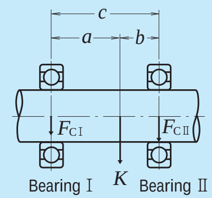

Fig. 5.5 Radial Load Distribution (1)

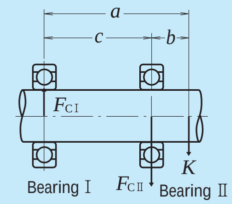

Fig. 5.6 Radial Load Distribution (2)

5.3.5 Average of Fluctuating Load

When the load applied on bearings fluctuates, an average load which will yield the same bearing life as the fluctuating load should be calculated.

(1) When the relation between load and rotating speed is divided into the following steps (Fig. 5.7)

Load F_1 : Speed n_1 ; Operating time t_1
Load F_2 : Speed n_2 ; Operating time t_2
⋮
Load F_n : Speed n_n ; Operating time t_n

Then, the average load F_m may be calculated using the following equation:

$$F_m = \sqrt[p]{\frac{F_1^{\,p} n_1 t_1 + F_2^{\,p} n_2 t_2 + \cdots + F_n^{\,p} n_n t_n}{n_1 t_1 + n_2 t_2 + \cdots\cdots + n_n t_n}} \quad \cdots\cdots (5.18)$$

where F_m : Average fluctuating load (N), {kgf}

p = 3 for ball bearings

p = 10/3 for roller bearings

The average speed n_m may be calculated as follows:

$$n_m = \frac{n_1t_1+n_2t_2+\cdots+n_nt_n}{t_1+t_2+\cdots+t_n} \quad \text{..........(5.19)}$$

(2) When the load fluctuates almost linearly (Fig. 5.8), the average load may be calculated as follows:

$$F_m \fallingdotseq \frac{1}{3}(F_{min}+2F_{max}) \quad \text{..........(5.20)}$$

where F_{min} : Minimum value of fluctuating load (N), {kgf}

F_{max} : Maximum value of fluctuating load (N), {kgf}

(3) When the load fluctuation is similar to a sine wave (Fig. 5.9), an approximate value for the average load F_m may be calculated from the following equation:

In the case of Fig. 5.9 (a)

$$F_m \fallingdotseq 0.65\,F_{max} \quad \text{..........(5.21)}$$

In the case of Fig. 5.9 (b)

$$F_m \fallingdotseq 0.75\,F_{max} \quad \text{..........(5.22)}$$

(4) When both a rotating load and a stationary load are applied (Fig. 5.10).

F_R : Rotating load (N), {kgf}

F_S : Stationary load (N), {kgf}

An approximate value for the average load F_m may be calculated as follows:

a) Where $F_R \geqq F_S$

$$F_m \fallingdotseq F_R + 0.3F_S + 0.2\frac{F_S^2}{F_R} \quad \text{..........(5.23)}$$

b) Where $F_R < F_S$

$$F_m \fallingdotseq F_S + 0.3F_R + 0.2\frac{F_R^2}{F_S} \quad \text{..........(5.24)}$$

5.4 Equivalent Load

In some cases, the loads applied on bearings are purely radial or axial loads; however, in most cases, the loads are a combination of both. In addition, such loads usually fluctuate in both magnitude and direction. In such cases, the loads actually applied on bearings cannot be used for bearing life calculations; therefore, a hypothetical load that has a constant magnitude and passes through the center of the bearing, and will give the same bearing life that the bearing would attain under actual conditions of load and rotation should be estimated. Such a hypothetical load is called the equivalent load.

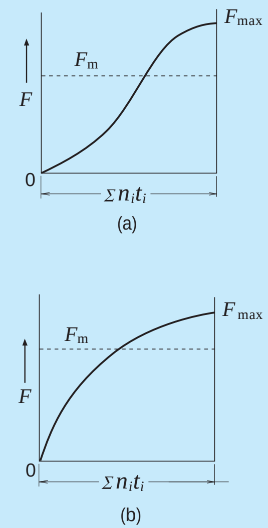

Fig. 5.9 Sinusoidal Load Variation

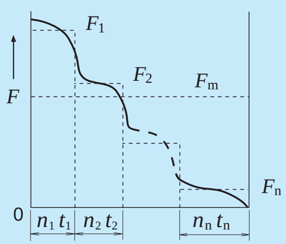

Fig. 5.7 Incremental Load Variation

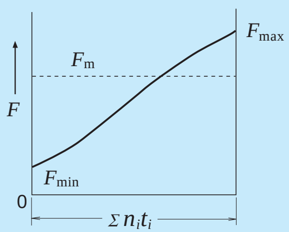

Fig. 5.8 Simple Load Fluctuation

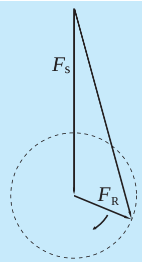

Fig. 5.10 Rotating Load and Stationary Load

5.4.1 Calculation of Equivalent Loads

The equivalent load on radial bearings may be calculated using the following equation:

$$P = XF_r + YF_a \quad \text{..........(5.25)}$$

where P : Equivalent Load (N), {kgf}

F_r : Radial load (N), {kgf}

F_a : Axial load (N), {kgf}

X : Radial load factor

Y : Axial load factor

The values of X and Y are listed in the bearing tables. The equivalent radial load for radial roller bearings with $\alpha = 0°$ is

$$P = F_r$$

In general, thrust ball bearings cannot take radial loads, but spherical thrust roller bearings can take some radial loads. In this case, the equivalent load may be calculated using the following equation:

$$P = F_a + 1.2F_r \quad \text{..........(5.26)}$$

where $\frac{F_r}{F_a} \leqq 0.55$

5.4.2 Axial Load Components in Angular Contact Ball Bearings and Tapered Roller Bearings

The effective load center of both angular contact ball bearings and tapered roller bearings is at the point of intersection of the shaft center line and a line representing the load applied on the rolling element by the outer ring as shown in Fig. 5.11. This effective load center for each bearing is listed in the bearing tables.

When radial loads are applied to these types of bearings, a component of load is produced in the axial direction. In order to balance this component load, bearings of the same type are used in pairs, placed face to face or back to back. These axial loads can be calculated using the following equation:

$$F_{a\,i} = \frac{0.6}{Y}F_r \quad \text{..........(5.27)}$$

where $F_{a\,i}$: Component load in the axial direction (N), {kgf}

F_r : Radial load (N), {kgf}

Y : Axial load factor

Assume that radial loads $F_{r\mathrm{I}}$ and $F_{r\mathrm{II}}$ are applied on bearings I and II (Fig. 5.12) respectively, and an external axial load F_{ae} is applied as shown. If the axial load factors are Y_{I}, Y_{II} and the radial load factor is X, then the equivalent loads P_{I}, P_{II} may be calculated as follows:

where $F_{ae} + \frac{0.6}{Y_{\mathrm{II}}}F_{r\mathrm{II}} \geqq \frac{0.6}{Y_{\mathrm{I}}}F_{r\mathrm{I}}$

$$\left.\begin{aligned} P_{\mathrm{I}} &= XF_{r\mathrm{I}} + Y_{\mathrm{I}}\left(F_{ae} + \frac{0.6}{Y_{\mathrm{II}}}F_{r\mathrm{II}}\right) \\ P_{\mathrm{II}} &= F_{r\mathrm{II}} \end{aligned}\right\} \quad \text{..........(5.28)}$$

where $F_{ae} + \frac{0.6}{Y_{\mathrm{II}}}F_{r\mathrm{II}} < \frac{0.6}{Y_{\mathrm{I}}}F_{r\mathrm{I}}$

$$\left.\begin{aligned} P_{\mathrm{I}} &= F_{r\mathrm{I}} \\ P_{\mathrm{II}} &= XF_{r\mathrm{II}} + Y_{\mathrm{II}}\left(\frac{0.6}{Y_{\mathrm{I}}}F_{r\mathrm{I}} - F_{ae}\right) \end{aligned}\right\} \quad \text{..........(5.29)}$$

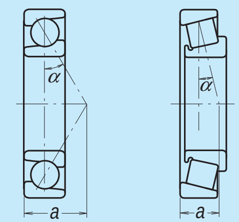

Fig. 5.11 Effective Load Centers

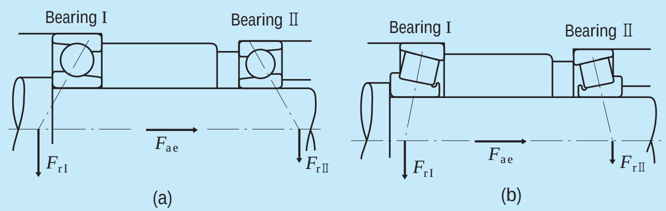

Fig. 5.12 Loads in Opposed Duplex Arrangement

5.5 Static Load Ratings and Static Equivalent Loads

5.5.1 Static Load Ratings

When subjected to an excessive load or a strong shock load, rolling bearings may incur a local permanent deformation of the rolling elements and permanent deformation of the rolling elements and raceway surface if the elastic limit is exceeded. The nonelastic deformation increases in area and depth as the load increases, and when the load exceeds a certain limit, the smooth running of the bearing is impeded.
The basic static load rating is defined as that static load which produces the following calculated contact stress at the center of the contact area between the rolling element subjected to the maximum stress and the raceway surface.

For self-aligning ball bearings	4 600MPa {469kgf/mm²}
For other ball bearings	4 200MPa {428kgf/mm²}
For roller bearings	4 000MPa {408kgf/mm²}

In this most heavily stressed contact area, the sum of the permanent deformation of the rolling element and that of the raceway is nearly 0.0001 times the rolling element's diameter. The basic static load rating C_o is written C_{or} for radial bearings and C_{oa} for thrust bearings in the bearing tables.
In addition, following the modification of the criteria for basic static load rating by ISO, the new C_o values for NSK's ball bearings became about 0.8 to 1.3 times the past values and those for roller bearings about 1.5 to 1.9 times. Consequently, the values of permissible static load factor f_s have also changed, so please pay attention to this.

5.5.2 Static Equivalent Loads

The static equivalent load is a hypothetical load that produces a contact stress equal to the above maximum stress under actual conditions, while the bearing is stationary (including very slow rotation or oscillation), in the area of contact between the most heavily stressed rolling element and bearing raceway. The static radial load passing through the bearing center is taken as the static equivalent load for radial bearings, while the static axial load in the direction coinciding with the central axis is taken as the static equivalent load for thrust bearings.

(a) Static equivalent load on radial bearings

The greater of the two values calculated from the following equations should be adopted as the static equivalent load on radial bearings.

$$P_o = X_o F_r + Y_o F_a \quad \text{.................................}(5.30)$$

$$P_o = F_r \quad \text{..}(5.31)$$

where
P_o : Static equivalent load (N), {kgf}
F_r : Radial load (N), {kgf}
F_a : Axial load (N), {kgf}
X_o : Static radial load factor
Y_o : Static axial load factor

(b)Static equivalent load on thrust bearings

$$P_o = X_o F_r + F_a \qquad \alpha \neq 90° \quad \text{......................}(5.32)$$

where
P_o : Static equivalent load (N), {kgf}
α : Contact angle

When $F_a < X_o F_r$, this equation becomes less accurate. The values of X_o and Y_o for Equations (5.30) and (5.32) are listed in the bearing tables.
The static equivalent load for thrust roller bearings with

$\alpha = 90°$ is $P_o = F_a$

5.5.3 Permissible Static Load Factor

The permissible static equivalent load on bearings varies depending on the basic static load rating and also their application and operating conditions.
The permissible static load factor f_s is a safety factor that is applied to the basic static load rating, and it is defined by the ratio in Equation (5.33). The generally recommended values of f_s are listed in Table 5.8. Conforming to the modification of the static load rating, the values of f_s were revised, especially for bearings for which the values of C_o were increased, please keep this in mind when selecting bearings.

$$f_s = \frac{C_o}{P_o} \quad \text{..}(5.33)$$

where
C_o : Basic static load rating (N), {kgf}
P_o : Static equivalent load (N), {kgf}

For spherical thrust roller bearings, the values of f_s should be greater than 4.

Table 5. 8 Values of Permissible Static Load Factor f_s

Operating Conditions	Lower Limit of f_s	
	Ball Bearings	Roller Bearings
Low-noise applications	2	3
Bearings subjected to vibration and shock loads	1.5	2
Standard operating conditions	1	1.5

5.6 Maximum Permissible Axial Loads for Cylindrical Roller Bearings

Cylindrical roller bearings having inner and outer rings with ribs, loose ribs or thrust collars are capable of sustaining radial loads and limited axial loads simultaneously. The maximum permissible axial load is limited by an abnormal temperature rise or heat seizure due to sliding friction between the end faces of rollers and the rib face, or the rib strength.
The maximum permissible axial load (the load considered the heat generation between the end face of rollers and the rib face) for bearings of diameter series 3 that are continuously loaded and lubricated with grease or oil is shown in Fig. 5.13.

Grease lubrication (Empirical equation)

$$\left.\begin{aligned} C_A &= 9.8f\left\{\frac{900\,(k\cdot d)^2}{n+1\,500} - 0.023\times(k\cdot d)^{2.5}\right\}\text{...(N)} \\ &= f\left\{\frac{900\,(k\cdot d)^2}{n+1\,500} - 0.023\times(k\cdot d)^{2.5}\right\}\text{.....\{kgf\}} \end{aligned}\right\}\text{.......}(5.34)$$

Oil lubrication (Empirical equation)

$$\left.\begin{aligned} C_A &= 9.8f\left\{\frac{490\,(k\cdot d)^2}{n+1\,000} - 0.000135\times(k\cdot d)^{3.4}\right\}\text{...(N)} \\ &= f\left\{\frac{490\,(k\cdot d)^2}{n+1\,000} - 0.000135\times(k\cdot d)^{3.4}\right\}\text{.....\{kgf\}} \end{aligned}\right\}\text{...}(5.35)$$

where
C_A : Permissible axial load (N), {kgf}
d : Bearing bore diameter (mm)
n : Speed (min⁻¹)

f : Load Factor

Loading Interval	Value of f
Continuous	1
Intermittent	2
Short time only	3

k : Size Factor

Diameter series	Value of k
2	0.75
3	1
4	1.2

In the equations (5.34) and (5.35), the examination for the rib strength is excluded. Concerning the rib strength, please consult with NSK.
In addition, for cylindrical roller bearings to have a stable axial-load carrying capacity, the following precautions are required for the bearings and their surroundings:

- Radial load must be applied and the magnitude of radial load should be larger than that of axial load by 2.5 times or more.
- Sufficient lubricant must exist between the roller end faces and ribs.
- Superior extreme-pressure grease must be used.
- Sufficient running-in should be done.
- The mounting accuracy must be good.
- The radial clearance should not be more than necessary.

In cases where the bearing speed is extremely slow, the speed exceeds the limiting speed by more than 50%, or the bore diameter is more than 200mm, careful study is necessary for each case regarding lubrication, cooling, etc. In such a case, please consult with NSK.

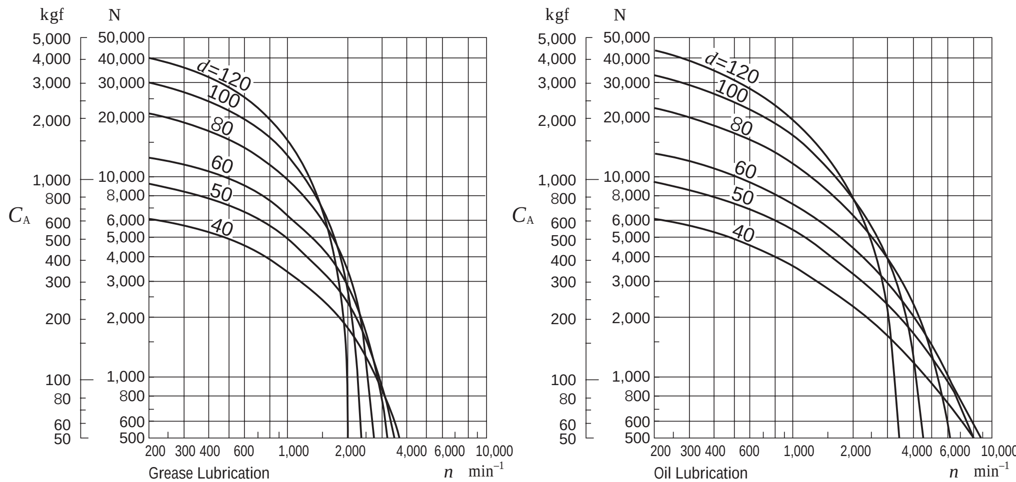

Fig. 5.13 Permissible Axial Load for Cylindrical Roller Bearings
For Diameter Series 3 bearings (k=1.0) operating under a continuous load and lubricated with grease or oil.

5.7 Examples of Bearing Calculations

(Example1)
Obtain the fatigue life factor f_h of single-row deep groove ball bearing **6208** when it is used under a radial load F_r=2 500 N, {255kgf} and speed n =900 min^{-1}.

The basic load rating C_r of **6208** is 29 100N, {2 970kgf} (Bearing Table, Page B10). Since only a radial load is applied, the equivalent load P may be obtained as follows:

$P = F_r = 2\,500\text{N}$, {255kgf}

Since the speed is n = 900 min^{-1}, the speed factor f_n can be obtained from the equation in Table 5.2 (Page A25) or Fig. 5.3(Page A26).

$f_n = 0.333$

The fatigue life factor f_h, under these conditions, can be calculated as follows:

$$f_h = f_n \frac{C_r}{P} = 0.333 \times \frac{29\,100}{2\,500} = 3.88$$

This value is suitable for industrial applications, air conditioners being regularly used, etc., and according to the equation in Table 5.2 or Fig. 5.4 (Page A26), it corresponds approximately to 29 000 hours of service life.

(Example 2)
Select a single-row deep groove ball bearing with a bore diameter of 50 mm and outside diameter under 100 mm that satisfies the following conditions:

Radial load F_r = 3 000N, {306kgf}

Speed n =1 900 min^{-1}

Basic rating life $L_h \geqq 10\,000$h

The fatigue life factor f_h of ball bearings with a rating fatigue life longer than 10 000 hours is $f_h \geqq 2.72$.
Because f_n = 0.26, $P = F_r$ = 3 000N. {306kgf}

$$f_h = f_n \frac{C_r}{P} = 0.26 \times \frac{C_r}{3\,000} \geqq 2.72$$

therefore, $C_r \geqq 2.72 \times \dfrac{3\,000}{0.26} = 31\,380\text{N}$, {3 200kgf}

Among the data listed in the bearing table on Page B12, **6210** should be selected as one that satisfies the above conditions.

(Example3)
Obtain C_r / P or fatigue life factor f_h when an axial load F_a=1 000N, {102kgf} is added to the conditions of (Example 1)

When the radial load F_r and axial load F_a are applied on single-row deep groove ball bearing **6208**, the dynamic equivalent load P should be calculated in accordance with the following procedure.
Obtain the radial load factor X, axial load factor Y and constant e obtainable, depending on the magnitude of $f_0 F_a / C_{or}$, from the table above the single-row deep groove ball bearing table.
The basic static load rating C_{or} of ball bearing **6208** is 17 900N, {1 820kgf} (Page B10)

$f_0 F_a / C_{or} = 14.0 \times 1\,000/17\,900 = 0.782$

$e \fallingdotseq 0.26$

and $F_a / F_r = 1\,000/2\,500 = 0.4 > e$

$X = 0.56$

$Y = 1.67$ (the value of Y is obtained by linear interpolation)

Therefore, the dynamic equivalent load P is

$P = XF_r + YF_a$
$= 0.56 \times 2\,500 + 1.67 \times 1\,000$
$= 3\,070\text{N}$, {313kgf}

$$\frac{C_r}{P} = \frac{29\,100}{3\,070} = 9.48$$

$$f_h = f_n \frac{C_r}{P} = 0.333 \times \frac{29\,100}{3\,070} = 3.16$$

This value of f_h corresponds approximately to 15 800 hours for ball bearings.

(Example 4)
Select a spherical roller bearing of series 231 satisfying the following conditions:

Radial load F_r = 45 000N, {4 950kgf}

Axial load F_a = 8 000N, {816kgf}

Speed n = 500min^{-1}

Basic rating life $L_h \geqq 30\,000$h

The value of the fatigue life factor f_h which makes $L_h \geqq 30\,000$h is bigger than 3.45 from Fig. 5.4 (Page A26).

The dynamic equivalent load P of spherical roller bearings is given by:

when $F_a / F_r \leqq e$

$P = XF_r + YX_a = F_r + Y_3 F_a$

when $F_a / F_r > e$

$P = XF_r + YF_a = 0.67\,F_r + Y_2 F_a$

$F_a / F_r = 8\,000/45\,000 = 0.18$

We can see in the bearing table that the value of e is about 0.3 and that of Y_3 is about 2.2 for bearings of series 231:

Therefore, $P = XF_r + YF_a = F_r + Y_3 F_a$
$= 45\,000 + 2.2 \times 8\,000$
$= 62\,600\text{N}$, {6 380kgf}

From the fatigue life factor f_h, the basic load rating can be obtained as follows:

$$f_h = f_n \frac{C_r}{P} = 0.444 \times \frac{C_r}{62\,600} \geqq 3.45$$

consequently, $C_r \geqq 490\,000\text{N}$, {50 000kgf}
Among spherical roller bearings of series 231 satisfying this value of C_r, the smallest is **23126CE4**
(C_r = 505 000N, {51 500kgf})
Once the bearing is determined, substitute the value of Y_3 in the equation and obtain the value of P.

$P = F_r + Y_3 F_a = 45\,000 + 2.4 \times 8\,000$
$= 64\,200\text{N}$, {6 550kgf}

$$L_h = 500 \left(f_n \frac{C_r}{P}\right)^{\frac{10}{3}}$$

$$= 500 \left(0.444 \times \frac{505\,000}{64\,200}\right)^{\frac{10}{3}}$$

$$= 500 \times 3.49^{\frac{10}{3}} \fallingdotseq 32\,000 \text{ h}$$

(Example 5)
Assume that tapered roller bearings **HR30305DJ** and **HR30206J** are used in a back-to-back arrangement as shown in Fig. 5.14, and the distance between the cup back faces is 50 mm.
Calculate the basic rating life of each bearing when beside the radial load F_r = 5 500N, {561kgf},
axial load F_{ae}=2 000N,{204kgf} are applied to **HR30305DJ** as shown in Fig. 5.14. The speed is 600 min^{-1}.

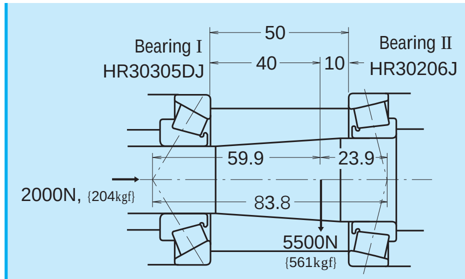

Fig. 5.14 Loads on Tapered Roller Bearings

To distribute the radial load F_r on bearings I and II, the effective load centers must be located for tapered roller bearings. Obtain the effective load center *a* for bearings I and II from the bearing table, then obtain the relative position of the radial load F_r and effective load centers. The result will be as shown in Fig. 5.14. Consequently, the radial load applied on bearings I (**HR30305DJ**) and II (**HR30206J**) can be obtained from the following equations:

$$F_{rI} = 5\,500 \times \frac{23.9}{83.8} = 1\,569\text{N}, \ \{160\text{kgf}\}$$

$$F_{rII} = 5\,500 \times \frac{59.9}{83.8} = 3\,931\text{N}, \ \{401\text{kgf}\}$$

From the data in the bearing table, the following values are obtained;

Bearings	Basic dynamic load rating C_r (N) {kgf}	Axial load factor Y_1	Constant e
Bearing I (**HR30305DJ**)	38 000 {3 900}	$Y_I = 0.73$	0.83
Bearing II (**HR30206J**)	43 000 {4 400}	$Y_{II} = 1.6$	0.38

When radial loads are applied on tapered roller bearings, an axial load component is produced, which must be considered to obtain the dynamic equivalent radial load (Refer to Paragraph 5.4.2, Page A31).

$$F_{ae}+\frac{0.6}{Y_{II}}F_{rII}=2\,000+\frac{0.6}{1.6}\times 3\,931$$
$$=3\,474\text{N},\ \{354\text{kgf}\}$$
$$\frac{0.6}{Y_I}F_{rI}=\frac{0.6}{0.73}\times 1\,569=1\,290\text{N},\ \{132\text{kgf}\}$$

Therefore, with this bearing arrangement, the axial load $F_{ae}+\frac{0.6}{Y_{II}}F_{rII}$ is applied on bearing Ⅰ but not on bearing Ⅱ.
For bearing Ⅰ

$$F_{rI}=1\,569\text{N},\ \{160\text{kgf}\}$$
$$F_{aI}=3\,474\text{N},\ \{354\text{kgf}\}$$

since $F_{aI}\,/\,F_{rI}=2.2>e=0.83$
the dynamic equivalent load $P_I=XF_{rI}+Y_IF_{aI}$

$$=0.4\times 1\,569+0.73\times 3\,474$$
$$=3\,164\text{N},\ \{323\text{kgf}\}$$

The fatigue life factor $f_h=f_n\frac{C_r}{P_I}$

$$=\frac{0.42\times 38\,000}{3\,164}=5.04$$

and the rating fatigue life $L_h=500\times 5.04^{\frac{10}{3}}=109\,750\text{h}$
For bearing Ⅱ
since $F_{rII}=3\,931\text{N}$, {401kgf}, $F_{aII}=0$
the dynamic equivalent load

$$P_{II}=F_{rII}=3\,931\text{N},\ \{401\text{kgf}\}$$

the fatigue life factor

$$f_h=f_n\frac{C_r}{P_{II}}=\frac{0.42\times 43\,000}{3\,931}=4.59$$

and the rating fatigue life $L_h=500\times 4.59^{\frac{10}{3}}=80\,400\text{h}$ are obtained.

Remarks For face-to-face arrangements (DF type), please contact NSK.

(Example 6)
Select a bearing for a speed reducer under the following conditions:
Operating conditions

Radial load $F_r=245\,000\text{N}$, {25 000kgf}
Axial load $F_a=49\,000\text{N}$, {5 000kgf}
Speed $n=500\text{min}^{-1}$

Size limitation
Shaft diameter: 300mm
Bore of housing: Less than 500mm

In this application, heavy loads, shocks, and shaft deflection are expected; therefore, spherical roller bearings are appropriate.
The following spherical roller bearings satisfy the above size limitation (refer to Page B196)

d	D	B	Bearing No.	Basic dynamic load rating C_r (N)	{kgf}	Constant e	Factor Y_3
300	420	90	**23960 CAE4**	1 230 000	125 000	0.19	3.5
	460	118	**23060 CAE4**	1 920 000	196 000	0.24	2.8
	460	160	**24060 CAE4**	2 310 000	235 000	0.32	2.1
	500	160	**23160 CAE4**	2 670 000	273 000	0.31	2.2
	500	200	**24160 CAE4**	3 100 000	315 000	0.38	1.8

since $F_a\,/\,F_r=0.20<e$
the dynamic equivalent load P is

$$P=F_r+Y_3F_a$$

Judging from the fatigue life factor f_h in Table 5.1 and examples of applications (refer to Page A25), a value of f_h, between 3 and 5 seems appropriate.

$$f_h=f_n\frac{C_r}{P}=\frac{0.444\,C_r}{F_r+Y_3F_a}=3\text{ to }5$$

Assuming that $Y_3=2.1$, then the necessary basic load rating C_r can be obtained

$$C_r=\frac{(F_r+Y_3F_a)\times(3\text{ to }5)}{0.444}$$
$$=\frac{(245\,000+2.1\times 49\,000)\times(3\text{ to }5)}{0.444}$$
$$=2\,350\,000\text{ to }3\,900\,000\text{ N},$$
$$\{240\,000\text{ to }400\,000\text{ kgf}\}$$

The bearings which satisfy this range are **23160CAE4**, and **24160CAE4**.

6. LIMITING SPEED

The speed of rolling bearings is subject to certain limits. When bearings are operating, the higher the speed, the higher the bearing temperature due to friction. The limiting speed is the empirically obtained value for the maximum speed at which bearings can be continuously operated without failing from seizure or generation of excessive heat. Consequently, the limiting speed of bearings varies depending on such factors as bearing type and size, cage form and material, load, lubricating method, and heat dissipating method including the design of the bearing's surroundings.
The limiting speeds for bearings lubricated by grease and oil are listed in the bearing tables. The limiting speeds in the tables are applicable to bearings of standard design and subjected to normal loads, i. e. $C/P\geqq 12$ and $F_a/F_r\leqq 0.2$ approximately. The limiting speeds for oil lubrication listed in the bearing tables are for conventional oil bath lubrication.
Some types of lubricants are not suitable for high speed, even though they may be markedly superior in other respects. When speeds are more than 70 percent of the listed limiting speed, it is necessary to select an oil or grease which has good high speed characteristics.

(Refer to)
Table 12.2 Grease Properties (Pages A110 and 111)
Table 12.5 Example of Selection of Lubricant for Bearing Operating Conditions (Page A113)
Table 15.8 Brands and Properties of Lubricating Grease (Pages A138 to A141)

6.1 Correction of Limiting Speed

When the bearing load P exceeds 8 % of the basic load rating C, or when the axial load F_a exceeds 20 % of the radial load F_r, the limiting speed must be corrected by multiplying the limiting speed found in the bearing tables by the correction factor shown in Figs. 6.1 and 6.2.
When the required speed exceeds the limiting speed of the desired bearing; then the accuracy grade, internal clearance, cage type and material, lubrication, etc., must be carefully studied in order to select a bearing capable of the required speed. In such a case, forced-circulation oil lubrication, jet lubrication, oil mist lubrication, or oil-air lubrication must be used.
If all these conditions are considered. The maximum permissible speed may be corrected by multiplying the limiting speed found in the bearing tables by the correction factor shown in Table 6.1. It is recommended that NSK be consulted regarding high speed applications.

6.2 Limiting Speed for Rubber Contact Seals for Ball Bearings

The maximum permissible speed for contact rubber sealed bearings (DDU type) is determined mainly by the sliding surface speed of the inner circumference of the seal. Values for the limiting speed are listed in the bearing tables.

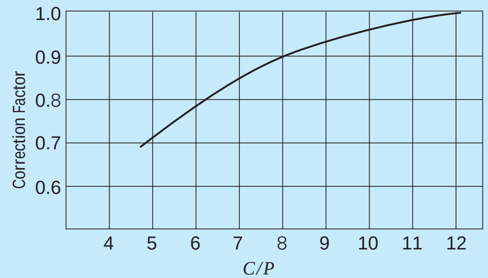

Fig. 6.1 Limiting Speed Correction Factor Variation with Load Ratio

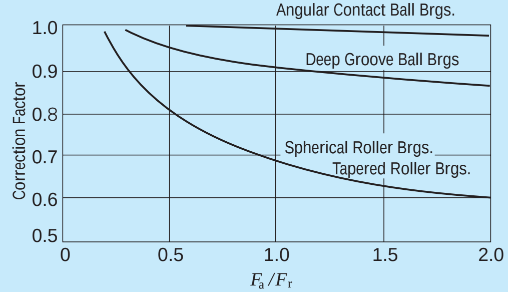

Fig. 6.2 Limiting Speed Correction Factor for Combined Radial and Axial Loads

Table 6.1 Limiting Speed Correction Factor for High-Speed Applications

Bearing Types	Correction Factor
Cylindrical Roller Brgs.(single row)	2
Needle Roller Brgs.(except broad width)	2
Tapered Roller Brgs.	2
Spherical Roller Brgs.	1.5
Deep Grooove Ball Brgs.	2.5
Angular Contact Ball Brgs.(except matched bearings)	1.5