

4. Bearing load calculation

To compute bearing loads, the forces which act on the shaft being supported by the bearing must be determined. Loads which act on the shaft and its related parts include weight of the rotating components, load produced when the machine performs work, and load produced by transmission of dynamic force. These can be mathematically calculated, but calculation is difficult in many cases.

A method of calculating loads that act upon shafts that convey dynamic force, which is the primary application of bearings, is provided herein.

4.1 Load acting on shafts

4.1.1 Load factor

There are many instances where the actual operational shaft load is much greater than the theoretically calculated load, due to shock. This actual shaft load can be estimated by using formula (4.1).

$$K = f_w \cdot K_c \quad (4.1)$$

Where:

K : Actual shaft load, N

f_w : Load factor (see Table 4.1)

K_c : Theoretically calculated value, N

Table 4.1 Load factor f_w

Amount of shock	f_w	Machine application examples
Very little or no shock	1.0 to 1.2	Electric machines, machine tools, measuring instruments.
Light shock	1.2 to 1.5	Railway vehicles, automobiles, rolling mills, metal working machines, paper making machines, printing machines, aircraft, textile machines, electrical units, office machines.
Heavy shock	1.5 to 3.0	Crushers, agricultural equipment, construction equipment, cranes.

4.1.2 Gear load

The loads operating on gears can be divided into three main types according to the direction in which the load is applied; i.e. tangential (K_t), radial (K_s), and axial (K_a). The magnitude and direction of these loads differ

according to the types of gears involved. The following refers to the calculation methods of loads acting on four types of gears.

(1) Loads acting on parallel shaft gears

The forces acting on spur gears and helical gears are depicted in Fig. 4.1 through Fig. 4.3.

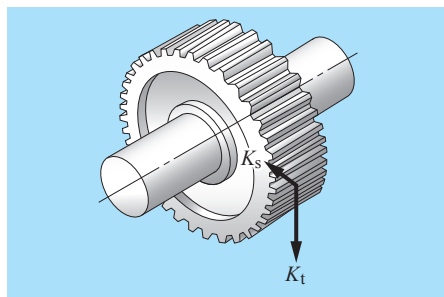


Fig. 4.1 Spur gear loads

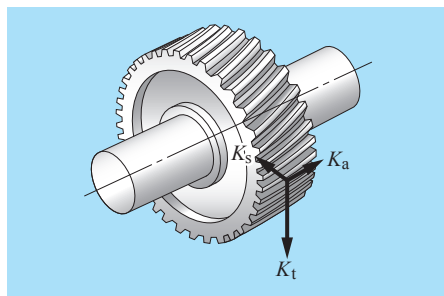


Fig. 4.2 Helical gear loads

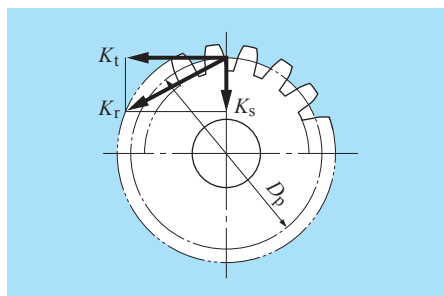


Fig. 4.3 Radial resultant forces

Loads acting on gears are obtained from formula (4.2) through formula (4.6).

Formula (4.2) describes the gear load in the tangential direction when the shaft input torque is known

$$K_t = \frac{2T}{D_p} \quad (4.2)$$

Formula (4.3) describes the gear load in the tangential direction when the transmitted power is known

$$K_t = \frac{19.1 \times 10^6 \cdot H}{D_p \cdot n} \quad (4.3)$$

$$K_s = K_t \cdot \tan \alpha \quad (\text{Spur gear}) \quad (4.4a)$$

$$= K_t \cdot \frac{\tan \alpha}{\cos \beta} \quad (\text{Helical gear}) \quad (4.4b)$$

$$K_r = \sqrt{K_t^2 + K_s^2} \quad (4.5)$$

$$K_a = K_t \cdot \tan \beta \quad (\text{Helical gear}) \quad (4.6)$$

Where:

K_t : Tangential gear load (tangential force), N

K_s : Radial gear load (separating force), N

K_r : Right angle shaft load (Combined radial load, resultant force of tangential force and separating force), N

K_a : Parallel shaft load (axial load), N

T : Input torque, N · mm

H : Transmitted force, kW

n : Rotational speed, min⁻¹

D_p : Gear pitch circle diameter, mm

α : Gear pressure angle, °

β : Helix angle, °

Because the actual gear load also contains vibrations and shock loads as well, the theoretical load obtained by the above formula can also be adjusted by the gear factor f_z as shown in Table 4.2.

Table 4.2 Gear factor f_z

Gear type	f_z
Precision ground gears (Pitch and tooth profile errors of less than 0.02 mm)	1.05 to 1.1
Ordinary machined gears (Pitch and tooth profile errors of less than 0.1 mm)	1.1 to 1.3

(2) Loads acting on cross shafts

Gear loads acting on straight tooth bevel gears and spiral bevel gears on cross shafts are shown in Fig. 4.4 and Fig. 4.5. The calculation methods for these gear loads are shown in Table 4.3.

Herein, to calculate gear loads for straight bevel gears, the helix angle $\beta = 0$.

The symbols and units used in Table 4.3 are as follows:

K_t : Tangential gear load (tangential force), N

K_s : Radial gear load (separating force), N

K_a : Parallel shaft load (axial load), N

H : Transmitted power, kW

n : Rotational speed, min⁻¹

D_{pm} : Mean pitch circle diameter, mm

α : Gear pressure angle, °

β : Helix angle, °

δ : Pitch cone angle, °

Because the two shafts intersect, pinions and gears have the relationship of formula (4.7) and formula (4.8).

$$K_{sp} = K_{ag} \quad (4.7)$$

$$K_{ap} = K_{sg} \quad (4.8)$$

Where:

K_{sp}, K_{sg} : Pinion and Gear separating force, N

K_{ap}, K_{ag} : Pinion and Gear axial load, N

For spiral bevel gears, the direction of the load varies depending on the direction of the helix angle, the direction of rotation, and which side is the driving side or the driven side. The directions for the separating force (K_s) and axial load (K_a) shown in Fig. 4.5 are positive directions. The direction of rotation and the helix angle direction are defined as viewed from the large end of the gear. The gear rotation direction in Fig. 4.5 is assumed to be clockwise (right).

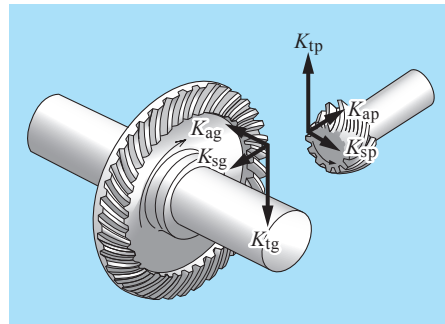


Fig. 4.4 Loads on bevel gears

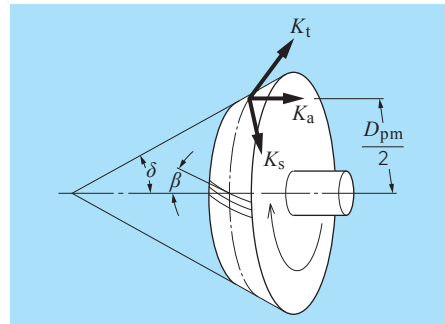


Fig. 4.5 Bevel gear diagram

Table 4.3 Loads acting on bevel gears

Types of load	Rotation direction	Clockwise	Counter clockwise	Clockwise	Counter clockwise
	Helix direction	Right	Left	Left	Right
Tangential load (tangential force) K_t		$K_t = \frac{19.1 \times 10^6 \cdot H}{D_{pm} \cdot n}$			
Radial load (separation force) K_s	Driving side	$K_s = K_t \left[\tan \alpha \frac{\cos \delta}{\cos \beta} + \tan \beta \sin \delta \right]$	$K_s = K_t \left[\tan \alpha \frac{\cos \delta}{\cos \beta} - \tan \beta \sin \delta \right]$	$K_s = K_t \left[\tan \alpha \frac{\cos \delta}{\cos \beta} - \tan \beta \sin \delta \right]$	$K_s = K_t \left[\tan \alpha \frac{\cos \delta}{\cos \beta} + \tan \beta \sin \delta \right]$
	Driven side	$K_s = K_t \left[\tan \alpha \frac{\cos \delta}{\cos \beta} - \tan \beta \sin \delta \right]$	$K_s = K_t \left[\tan \alpha \frac{\cos \delta}{\cos \beta} + \tan \beta \sin \delta \right]$	$K_s = K_t \left[\tan \alpha \frac{\cos \delta}{\cos \beta} + \tan \beta \sin \delta \right]$	$K_s = K_t \left[\tan \alpha \frac{\cos \delta}{\cos \beta} - \tan \beta \sin \delta \right]$
Parallel load on gear shaft (axial load) K_a	Driving side	$K_a = K_t \left[\tan \alpha \frac{\sin \delta}{\cos \beta} - \tan \beta \cos \delta \right]$	$K_a = K_t \left[\tan \alpha \frac{\sin \delta}{\cos \beta} + \tan \beta \cos \delta \right]$	$K_a = K_t \left[\tan \alpha \frac{\sin \delta}{\cos \beta} + \tan \beta \cos \delta \right]$	$K_a = K_t \left[\tan \alpha \frac{\sin \delta}{\cos \beta} - \tan \beta \cos \delta \right]$
	Driven side	$K_a = K_t \left[\tan \alpha \frac{\sin \delta}{\cos \beta} + \tan \beta \cos \delta \right]$	$K_a = K_t \left[\tan \alpha \frac{\sin \delta}{\cos \beta} - \tan \beta \cos \delta \right]$	$K_a = K_t \left[\tan \alpha \frac{\sin \delta}{\cos \beta} - \tan \beta \cos \delta \right]$	$K_a = K_t \left[\tan \alpha \frac{\sin \delta}{\cos \beta} + \tan \beta \cos \delta \right]$

(3) Load acting on hypoid gears

A hypoid gear is a spiral bevel gear that transmits power by offset shafts. Fig. 4.6 shows the load acting on a hypoid gear, and Table 4.4 shows the calculation method.

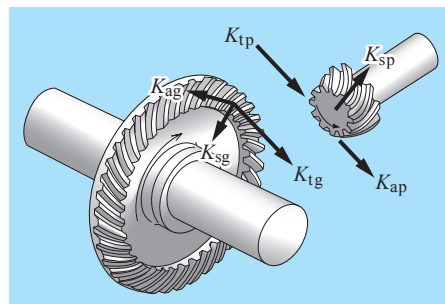


Fig. 4.6

Where:

- K_t : Tangential gear load (tangential force), N
- K_s : Radial gear load (separating force), N
- K_a : Parallel shaft load (axial load), N
- H : Transmitted force, kW
- n : Rotational speed, min⁻¹
- D_p : Gear mean pitch circle diameter, mm
- α : Gear pressure angle, °
- β : Helix angle, °
- δ_1 : Tooth tip cone angle, °
- δ_2 : Tooth bottom cone angle, °

* The driving shaft has a subscript p, and the driven shaft has a subscript g.

Table 4.4 Formula of load acting on hypoid gears

Types of load	Rotation direction	Clockwise	Counter clockwise	Clockwise	Counter clockwise
	Helix direction	Right	Left	Left	Right
Tangential load (tangential force) K_t	Driving shaft	Formula (4.9)		Formula (4.10)	
	Driven shaft				
Radial load (separation force) K_s	Driving shaft	Formula (4.11)		Formula (4.12)	
	Driven shaft	Formula (4.13)		Formula (4.14)	
Parallel load on gear shaft (axial load) K_a	Driving shaft	Formula (4.15)		Formula (4.16)	
	Driven shaft	Formula (4.17)		Formula (4.18)	

$$K_{tp} = \frac{19.1 \times 10^6 H}{D_{pmp} n_p} \dots\dots\dots (4.9)$$

$$K_{tg} = \frac{19.1 \times 10^6 H}{D_{pmg} n_g} = \frac{\cos \beta_g}{\cos \beta_p} K_{tp} \dots\dots\dots (4.10)$$

$$K_{sp} = \frac{K_{tp}}{\cos \beta_p} (\tan \alpha_p \cos \delta_{1p} + \sin \beta_p \sin \delta_{1p}) \dots\dots (4.11)$$

$$K_{sp} = \frac{K_{tp}}{\cos \beta_p} (\tan \alpha_p \cos \delta_{1p} - \sin \beta_p \sin \delta_{1p}) \dots\dots (4.12)$$

$$K_{sg} = \frac{K_{tg}}{\cos \beta_g} (\tan \alpha_g \cos \delta_{2g} - \sin \beta_g \sin \delta_{2g}) \dots\dots (4.13)$$

$$K_{sg} = \frac{K_{tg}}{\cos \beta_g} (\tan \alpha_g \cos \delta_{2g} + \sin \beta_g \sin \delta_{2g}) \dots\dots (4.14)$$

$$K_{ap} = \frac{K_{tp}}{\cos \beta_p} (\tan \alpha_p \sin \delta_{1p} - \sin \beta_p \cos \delta_{1p}) \dots\dots (4.15)$$

$$K_{ap} = \frac{K_{tp}}{\cos \beta_p} (\tan \alpha_p \sin \delta_{1p} + \sin \beta_p \cos \delta_{1p}) \dots\dots (4.16)$$

$$K_{ag} = \frac{K_{tg}}{\cos \beta_g} (\tan \alpha_g \sin \delta_{2g} + \sin \beta_g \cos \delta_{2g}) \dots\dots (4.17)$$

$$K_{ag} = \frac{K_{tg}}{\cos \beta_g} (\tan \alpha_g \sin \delta_{2g} - \sin \beta_g \cos \delta_{2g}) \dots\dots (4.18)$$

(4) Load acting on worm gears

A worm gear is a gear made by combining a worm (screw gear) and a worm wheel (helical gear). The gear direction differs depending on the rotation direction and the screw direction (right screw, left screw) of the worm shaft.

Fig. 4.8 shows the direction of loads acting on the gear, and Table 4.5 shows the calculation method of the loads.

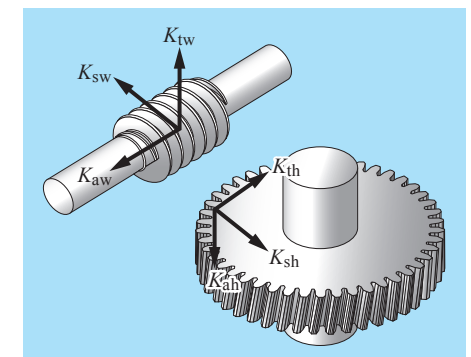


Fig. 4.7 Worm gears

- K_t : Tangential gear load (tangential force), N
 - K_s : Radial gear load (separating force), N
 - K_a : Parallel shaft load (axial load), N
 - H : Transmitted force, kW
 - n : Rotational speed, min⁻¹
 - D_p : Gear mean pitch circle diameter, mm
 - α : Gear pressure angle, °
 - γ : Worm lead angle, °
- * The worm shaft has a subscript w, and the worm gear has a subscript h.

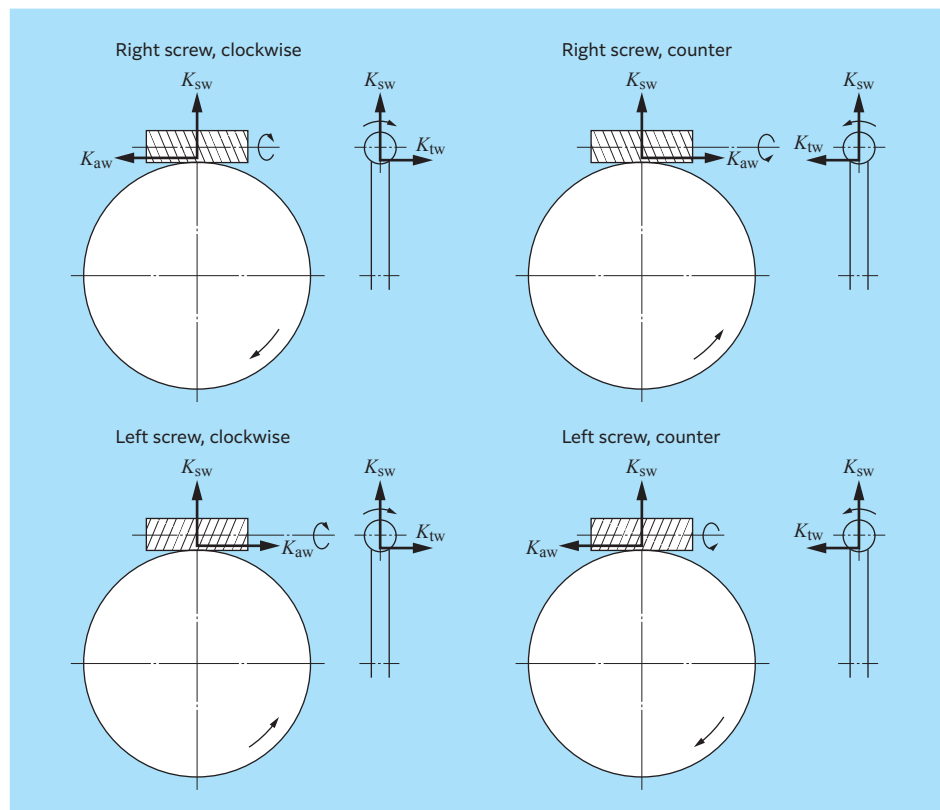


Fig. 4.8 Load acting on worm gears

Table 4.5 Calculation method of load acting on worm gears

Gear type	Worm shaft	Worm gear
Tangential load (tangential force) K_t	$K_{tw} = \frac{19.1 \times 10^6 H}{n D_{pw}}$	$K_{th} = \frac{K_{tw}}{\tan \gamma} = K_{aw}$
Radial load (separating force) K_s	$K_{sw} = \frac{K_{tw} \tan \alpha}{\tan \gamma}$	$K_{sh} = \frac{K_{tw} \tan \alpha}{\tan \gamma} = K_{sw}$
Parallel load on gear shaft (axial load) K_a	$K_{aw} = \frac{K_{tw}}{\tan \gamma}$	$K_{ah} = K_{tw}$

4.1.3 Chain / belt shaft load

The tangential loads on sprockets or pulleys when power (load) is transmitted by means of chains or belts can be calculated by formula (4.19) as shown in Fig. 4.9.

$$K_t = \frac{19.1 \times 10^6 \cdot H}{D_p \cdot n} \quad (4.19)$$

Where:

- K_t : Sprocket/pulley tangential load, N
- H : Transmitted power, kW
- D_p : Sprocket/pulley pitch diameter, mm
- n : Rotational speed, min^{-1}

For belt drives, an initial tension is applied to give sufficient constant operating tension on the belt and pulley.

Taking this tension into account, the radial loads acting on the pulley are expressed by formula (4.20). For chain drives, the same formula can also be used if vibrations and shock loads are taken into consideration.

$$K_r = f_b \cdot K_t \quad (4.20)$$

Where:

- K_r : Sprocket or pulley radial load, N
- f_b : Chain or belt factor (see Table 4.6)

Table 4.6 Chain or belt factor f_b

Chain or belt type	f_b
Chain (single)	1.2 to 1.5
V-belt	1.5 to 2.0
Timing belt	1.1 to 1.3
Flat belt (w / tension pulley)	2.5 to 3.0
Flat belt	3.0 to 4.0

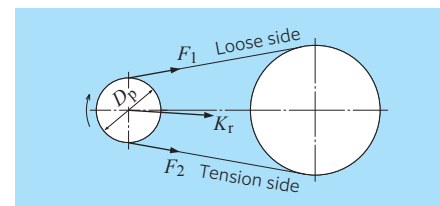


Fig. 4.9 Chain / belt loads

4.2 Bearing load distribution

For shafting, the static tension is considered to be supported by the bearings, and any loads acting on the shafts are distributed to the bearings. For example, in the gear shaft assembly depicted in Fig. 4.10, the applied bearing loads can be found by using formula (4.21) and formula (4.22).

This example is a simple case, but in reality, many of the calculations are quite complicated.

$$F_{rA} = \frac{a+b}{b} F_I + \frac{d}{c+d} F_{II} \quad (4.21)$$

$$F_{rB} = -\frac{a}{b} F_I + \frac{c}{c+d} F_{II} \quad (4.22)$$

Where:

- F_{rA} : Radial load on bearing A, N
- F_{rB} : Radial load on bearing B, N
- F_I, F_{II} : Radial load on shaft, N

If directions of radial load differ, the vector sum of each respective load must be determined.

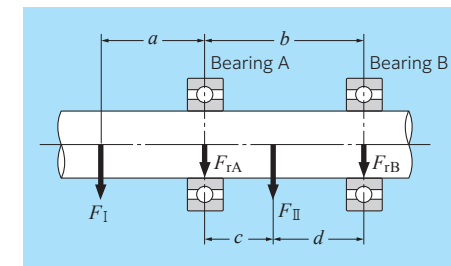


Fig. 4.10

4.3 Mean load

The load on bearings used in machines under normal circumstances will, in many cases, fluctuate according to a fixed time period or planned operation schedule. The load on bearings operating under such conditions can be converted to a mean load (F_m). This is a load which gives bearings the same life they would have under constant operating conditions.

(1) Fluctuating stepped load (see Fig. 4.11)

The mean bearing load, F_m , for stepped loads is calculated from formula (4.23). $F_1, F_2 \dots F_n$ are the loads acting on the bearing; $n_1, n_2 \dots n_n$ and $t_1, t_2 \dots t_n$ are the bearing speeds and operating times respectively.

$$F_m = \left[\frac{\sum (F_i^p n_i t_i)}{\sum (n_i t_i)} \right]^{1/p} \quad (4.23)$$

Where:

$p = 3$ For ball bearings
 $p = 10/3$ For roller bearings

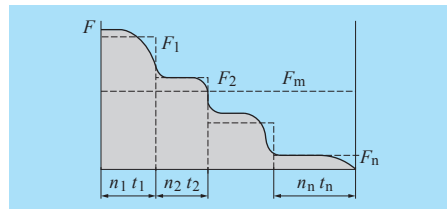


Fig. 4.11 Stepped load

(2) Continuously fluctuating load (see Fig. 4.12)

Where it is possible to express the function $F(t)$ in terms of load cycle t_0 and time t , the mean load is found by using formula (4.24).

$$F_m = \left[\frac{1}{t_0} \int_0^{t_0} F(t)^p dt \right]^{1/p} \quad (4.24)$$

Where:

$p = 3$ For ball bearings
 $p = 10/3$ For roller bearings

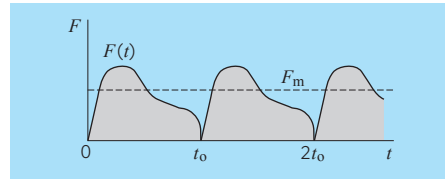


Fig. 4.12 Load that fluctuated as function of time

(3) Linear fluctuating load (see Fig. 4.13)

The mean load, F_m , can be approximated by formula (4.25).

$$F_m = \frac{F_{\min} + 2F_{\max}}{3} \quad (4.25)$$

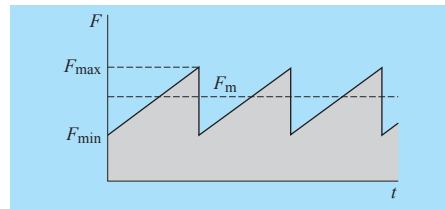


Fig. 4.13 Linear fluctuating load

(4) Sinusoidal fluctuating load (see Fig. 4.14)

The mean load, F_m , can be approximated by formula (4.26) and formula (4.27).

$$\text{Case (a) } F_m = 0.75 F_{\max} \quad (4.26)$$

$$\text{Case (b) } F_m = 0.65 F_{\max} \quad (4.27)$$

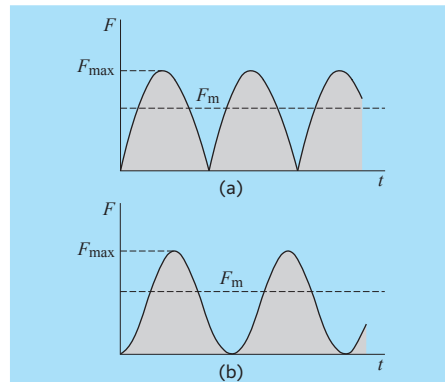


Fig. 4.14 Sinusoidal variable load

4.4 Equivalent load

4.4.1 Dynamic equivalent load

When both dynamic radial loads and dynamic axial loads act on a bearing at the same time, the hypothetical load acting on the center of the bearing which gives the bearings the same life as if they had only a radial load or only an axial load is called the dynamic equivalent load.

For radial bearings, this load is expressed as pure radial load and is called the dynamic equivalent radial load. For thrust bearings, it is expressed as pure axial load, and is called the dynamic equivalent axial load.

(1) Dynamic equivalent radial load

The dynamic equivalent radial load is expressed by formula (4.28).

$$P_r = X F_r + Y F_a \quad (4.28)$$

Where:

P_r : Dynamic equivalent radial load, N
 F_r : Actual radial load, N
 F_a : Actual axial load, N
 X : Radial load factor
 Y : Axial load factor

The values for X and Y are listed in the bearing tables.

(2) Dynamic equivalent axial load

As a rule, standard thrust bearings with α contact angle of 90° cannot carry radial loads. However, thrust spherical roller bearings can accept some radial load. The dynamic equivalent axial load for these bearings is given in formula (4.29).

$$P_a = F_a + 1.2 F_r \quad (4.29)$$

Where:

P_a : Dynamic equivalent axial load, N
 F_a : Actual axial load, N
 F_r : Actual radial load, N

Provided that $F_r / F_a \leq 0.55$ only.

4.4.2 Static equivalent load

The static equivalent load is a hypothetical

load which would cause the same total permanent deformation at the most heavily stressed contact point between the rolling elements and the raceway as under actual load conditions; that is when both static radial loads and static axial loads are simultaneously applied to the bearing.

For radial bearings this hypothetical load refers to pure radial loads, and for thrust bearings it refers to pure centric axial loads. These loads are designated static equivalent radial loads and static equivalent axial loads respectively.

(1) Static equivalent radial load

For radial bearings the static equivalent radial load can be found by using formula (4.30) or formula (4.31). The greater of the two resultant values is always taken for P_{0r} .

$$P_{0r} = X_0 F_r + Y_0 F_a \quad (4.30)$$

$$P_{0r} = F_r \quad (4.31)$$

Where:

P_{0r} : Static equivalent radial load, N
 F_r : Actual radial load, N
 F_a : Actual axial load, N
 X_0 : Static radial load factor
 Y_0 : Static axial load factor

The values for X_0 and Y_0 are given in the respective bearing tables.

(2) Static equivalent axial load

For thrust spherical roller bearings the static equivalent axial load is expressed by formula (4.32).

$$P_{0a} = F_a + 2.7 F_r \quad (4.32)$$

Where:

P_{0a} : Static equivalent axial load, N
 F_a : Actual axial load, N
 F_r : Actual radial load, N

Provided that $F_r / F_a \leq 0.55$ only.

4.4.3 Load calculation for angular contact ball bearings and tapered roller bearings

For angular contact ball bearings and tapered roller bearings the pressure cone apex (load

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center) is located as shown in **Fig. 4.15**, and their values are listed in the bearing tables.

When radial loads act on these types of bearings a component force is induced in the axial direction. For this reason, these bearings are used in pairs. For load calculation this component force must be taken into consideration and is expressed by formula (4.33).

$$F_a = \frac{0.5F_r}{Y} \quad (4.33)$$

Where:

F_a : Axial component force, N

F_r : Actual radial load, N

Y : Axial load factor

The axial loads for these bearing pairs are given in **Table 4.7**.

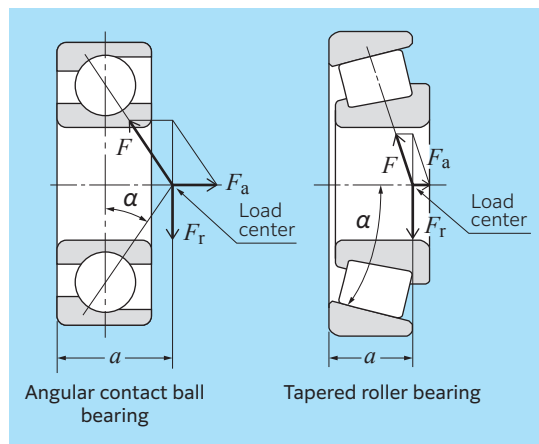


Fig. 4.15 Pressure cone apex and axial component force

Table 4.7 Bearing arrangement and equivalent load

Load center		Load conditions	Axial load
Rear	Brg I Brg II	$\frac{0.5F_{rI}}{Y_I} \leq \frac{0.5F_{rII}}{Y_{II}} + F_a$	$F_{aI} = \frac{0.5F_{rII}}{Y_{II}} + F_a$
	Brg II Brg I	$\frac{0.5F_{rI}}{Y_I} > \frac{0.5F_{rII}}{Y_{II}} + F_a$	$F_{aII} = \frac{0.5F_{rI}}{Y_I} - F_a$
Front	Brg I Brg II	$\frac{0.5F_{rII}}{Y_{II}} \leq \frac{0.5F_{rI}}{Y_I} + F_a$	$F_{aII} = \frac{0.5F_{rI}}{Y_I} + F_a$
	Brg II Brg I	$\frac{0.5F_{rII}}{Y_{II}} > \frac{0.5F_{rI}}{Y_I} + F_a$	$F_{aI} = \frac{0.5F_{rII}}{Y_{II}} - F_a$

Note: 1. Applies when preload is zero.

2. Radial forces in the opposite direction to the arrow in the above illustration are also regarded as positive.

3. Dynamic equivalent radial load is calculated by using the table on the right of the size table of the bearing after axial load is obtained for X and Y factor.

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4.5 Bearing rating life and load calculation examples

In the examples given in this section, for the purpose of calculation, all hypothetical load factors as well as all calculated load factors may be presumed to be included in the resultant load values.

Bearing loads and the basic rating life of bearings can be calculated using the bearing technical calculation tool on the **NTN** website (<https://www.ntnglobal.com>).

(Example 1)

What is the rating life in hours of operation L_{10h} for deep groove ball bearing **6208** operating at rotational speed $n = 650 \text{ min}^{-1}$, with a radial load F_r of 3.2 kN?

From formula (4.28) the dynamic equivalent radial load P_r :

$$P_r = F_r = 3.2 \text{ kN}$$

Basic dynamic load rating C_r for bearing **6208** given on page B-26 is 32.5 kN, ball bearing speed factor f_n relative to rotational speed $n = 650 \text{ min}^{-1}$ from **Fig. 3.1** is $f_n = 0.37$. Thus life factor f_h from formula (3.5) is:

$$f_h = f_n \frac{C_r}{P_r} = 0.37 \times \frac{32.5}{3.2} = 3.76$$

Therefore, with $f_h = 3.76$ from **Fig. 3.1** the rated life, L_{10h} , is approximately 27 000 hours.

(Example 2)

What is the life rating L_{10h} for the same bearing and conditions as in **Example 1**, but with an additional axial load F_a of 1.8 kN?

To find the dynamic equivalent radial load value for P_r , the radial load factor X , axial load factor Y , and Constant e are used.

Basic static load rating C_{0r} for bearing **6208** given on page B-26 is 17.8 kN and f_0 is 14.0.

Therefore:

$$\frac{f_0 \cdot F_a}{C_{0r}} = \frac{14 \times 1.8}{17.8} = 1.42$$

Calculated by the proportional interpolation method given on B-27, $e = 0.30$.

For the operating radial load and axial load:

$$\frac{F_a}{F_r} = \frac{1.8}{3.2} = 0.56 > e = 0.30$$

From B-27, $X = 0.56$ and $Y = 1.44$, and from formula (4.28) the dynamic equivalent radial load, P_r , is:

$$\begin{aligned} P_r &= XF_r + YF_a \\ &= 0.56 \times 3.2 + 1.44 \times 1.8 \\ &= 4.38 \text{ kN} \end{aligned}$$

From **Fig. 3.1** and **Table 3.1** the life factor, f_h , is:

$$f_h = f_n \frac{C_r}{P_r} = 0.37 \times \frac{32.5}{4.38} = 2.75$$

Therefore, with life factor $f_h = 2.75$, from **Fig. 3.1** the rated life, L_{10h} , is approximately 10 500 hours.

(Example 3)

Determine the optimum model number for a cylindrical roller bearing operating at the rotational speed $n = 450 \text{ min}^{-1}$, with a radial load F_r of 200 kN, and which must have a life (L_{10h}) of over 20 000 hours.

From **Fig. 3.1** the life factor $f_h = 3.02$ (L_{10h} at 20 000), and the speed factor $f_n = 0.46$ ($n = 450 \text{ min}^{-1}$). To find the required basic dynamic load rating, C_r , formula (3.5) is used.

$$C_r = \frac{f_h}{f_n} P_r = \frac{3.02}{0.46} \times 200 = 1\,313 \text{ kN}$$

From page B-108, the smallest bearing that fulfills all the requirements is **NU2332E** ($C_r = 1\,460 \text{ kN}$).

(Example 4)

The spur gear shown in Fig. 4.16 (pitch diameter $D_p = 150$ mm, pressure angle $\alpha = 20^\circ$) is supported by a pair of tapered roller bearings, **32907XU** ($C_r = 30.5$ kN) and **32908XU** ($C_r = 36.0$ kN). Find rating life for each bearing when gear transfer power $H = 150$ kW and rotational speed $n = 2\,000$ min⁻¹.

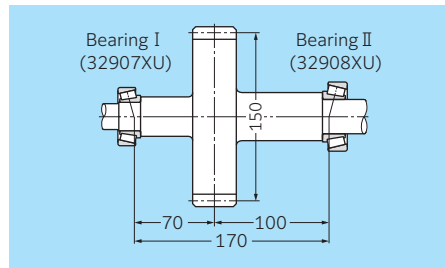


Fig. 4.16 Spur gear diagram

The gear load from formula (4.3), formula (4.4a) and formula (4.5) is:

$$K_t = \frac{19.1 \times 10^6 \cdot H}{D_p \cdot n} = \frac{19\,100\,000 \times 150}{150 \times 2\,000}$$

$$= 9.55 \text{ kN}$$

$$K_s = K_t \cdot \tan \alpha = 9.55 \times \tan 20^\circ$$

$$= 3.48 \text{ kN}$$

$$K_r = \sqrt{K_t^2 + K_s^2} = \sqrt{9.55^2 + 3.48^2}$$

$$= 10.16 \text{ kN}$$

The radial loads for bearings I and II are:

$$F_{rI} = \frac{100}{170} K_r = \frac{100}{170} \times 10.16 = 5.98 \text{ kN}$$

$$F_{rII} = \frac{70}{170} K_r = \frac{70}{170} \times 10.16 = 4.18 \text{ N}$$

$$\frac{0.5F_{rI}}{Y_I} = 1.45 > \frac{0.5F_{rII}}{Y_{II}} = 1.01 \text{ Therefore,}$$

The axial loads for bearing I and II are:

$$F_{aI} = 0 \text{ kN}$$

$$F_{aII} = \frac{0.5F_{rI}}{Y_I} = \frac{0.5 \times 5.98}{2.06} = 1.45 \text{ kN}$$

From page B-137, the dynamic equivalent radial load for bearing I is:

$$\frac{F_{aI}}{F_{rI}} = \frac{0}{5.98} = 0 < e = 0.29$$

$$P_{rI} = F_{rI} = 5.98 \text{ kN}$$

Equally, the dynamic equivalent radial load for bearing II is:

$$\frac{F_{aII}}{F_{rII}} = \frac{1.45}{4.18} = 0.35 > e = 0.29$$

$$\begin{aligned} P_{rII} &= XF_{rII} + Y_{II} F_{aII} \\ &= 0.4 \times 4.18 + 2.07 \times 1.45 \\ &= 4.67 \text{ kN} \end{aligned}$$

From formula (3.5) and Fig. 3.1 the life factor, f_h , for each bearing is

$$f_{hI} = f_n \frac{C_{rI}}{P_{rI}} = 0.293 \times 30.5 / 5.98 = 1.49$$

$$f_{hII} = f_n \frac{C_{rII}}{P_{rII}} = 0.293 \times 36.0 / 4.67 = 2.26$$

Therefore, from Table 3.1

$$L_{hI} = 500 f_{hI} = 1\,900 \text{ hours}$$

$$L_{hII} = 500 f_{hII} = 7\,550 \text{ hours}$$

The combined system bearing life, L_h , from formula (3.3) is:

$$\begin{aligned} L_h &= \frac{1}{\left[\frac{1}{L_{hI}^e} + \frac{1}{L_{hII}^e} \right]^{1/e}} \\ &= \frac{1}{\left[\frac{1}{1\,900^{9/8}} + \frac{1}{7\,550^{9/8}} \right]^{8/9}} \\ &= 1\,600 \text{ hours} \end{aligned}$$

(Example 5)

Find the mean load for spherical roller bearing **23932EMD1** ($C_r = 455$ kN) when operated under the fluctuating conditions shown in Table 4.8.

Table 4.8

Condition No. i	Operating time ϕ_i %	Radial load F_{ri} kN	Axial load F_{ai} kN	Rotational speed n_i min ⁻¹
1	5	10	2	1 200
2	10	12	4	1 000
3	60	20	6	800
4	15	25	7	600
5	10	30	10	400

The dynamic equivalent radial load, P_r , for each operating condition is found by using formula (4.28) and shown in Table 4.9.

Because all the values for F_{ri} and F_{ai} from the bearing tables are greater than $F_a / F_r > e = 0.17$, $X = 0.67$, $Y_2 = 5.81$

$$P_{ri} = XF_{ri} + Y_2 F_{ai} = 0.67F_{ri} + 5.81F_{ai}$$

From formula (4.23) the mean load, F_m , is:

$$F_m = \left[\frac{\sum (P_{ri}^{10/3} \cdot n_i \cdot \phi_i)}{\sum (n_i \cdot \phi_i)} \right]^{3/10} = 50.0 \text{ kN}$$

Table 4.9

Condition No. i	Dynamic equivalent radial load. P_{ri} (kN)
1	18.3
2	31.3
3	48.3
4	57.4
5	78.2

(Example 6)

Find the threshold values for rating life time and allowable axial load when cylindrical roller bearing **NUP312** is used under the following conditions:

Provided that intermittent axial load and oil lubricant.

$$\text{Radial load } F_r = 10 \text{ kN}$$

$$\text{Rotational speed } n = 2\,000 \text{ min}^{-1}$$

Radial load F_r is 10 kN, and

$$P_r = F_r = 10 \text{ kN}$$

The speed factor of cylindrical roller bearing, f_n , at $n = 2\,000$ min⁻¹, from Table 3.1

$$f_n = \left[\frac{33.3}{2\,000} \right]^{3/10} = 0.293$$

The life factor, f_h , from Table 3.1

$$f_h = 0.293 \times \frac{137}{10} = 4.01$$

Therefore the basic rated life, L_{10h} , from Table 3.1

$$L_{10h} = 500 \times 4.01^{10/3} \approx 51\,000 \text{ hours}$$

Next, the allowable axial load of the cylindrical roller bearing is shown on page A-32.

First, calculate allowable axial load P_t based on allowable surface pressure of rib. In formula (3.13) on page A-32, according to **NUP312** in Table 3.7 on page A-33, $k_1 = 0.065$, $D_{pw} = (60 + 130) / 2 = 95$ mm, rotational speed $n = 2\,000$ min⁻¹.

Thus, the formula below holds when the case of the intermittent axial load is taken into consideration.

$$D_{pw} \cdot n \times 10^4 = 19 \times 10^4.$$

In Fig. 3.16 on page A-33, $D_{pw} \cdot n \times 10^4 = 19 \times 10^4$. In the case of the intermittent axial load, allowable surface pressure of rib is $P_z = 40$ MPa.

Therefore, P_t is calculated as follows.

$$P_t = 0.065 \times 60^2 \times 40 = 9\,360 \text{ N}$$

Next, calculate allowable axial load F_{ar} based on the radial load. In formula (3.14) on page A-32, according to **NUP312** in Table 3.7 on page A-33, $k_2 = 0.4$.

Based on radial load $F_r = 10\,000$ N, F_{ar} is calculated as follows.

$$F_{ar} = 0.4 \times 10\,000 = 4\,000 \text{ N}$$

To actually calculate allowable axial load $F_{a \max}$, of P_t and F_{ar} calculated above, employ a smaller value.

Therefore, based on $P_t = 9\,360 \text{ N} > F_{ar} = 4\,000 \text{ N}$, $F_{a \max}$ is calculated as follows.

$$F_{a \max} = 4\,000 \text{ N}$$