

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).

$$K = f_w \cdot K_C \dots\dots\dots (4.1)$$

Where:

- $K$  : Actual shaft load N
- $f_w$  : Load factor (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 to Fig. 4.3.

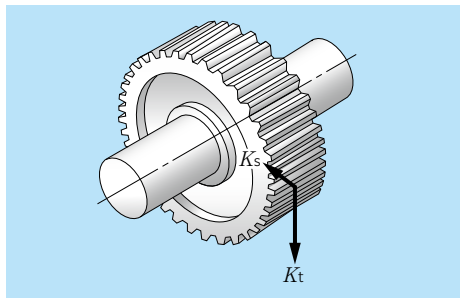


Fig. 4.1 Spur gear loads

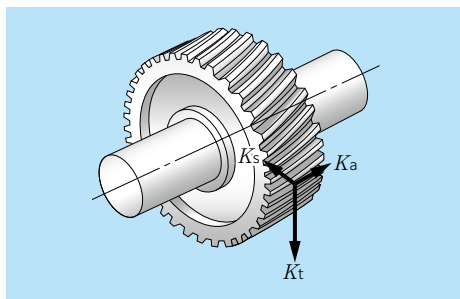


Fig. 4.2 Helical gear loads

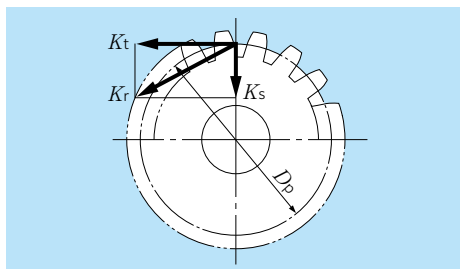


Fig. 4.3 Radial resultant forces

Loads acting on gears are obtained from formulas (4.2) to (4.6).

Equation 4.2 describes the gear load in the tangential direction when the shaft input torque is known

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

Equation 4.3 describes the gear load in the tangential direction when the transmitted power is known

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

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

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

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

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

Where:

- $K_t$  : Tangential gear load (tangential force), N
- $K_s$  : Radial gear load (separating force), N
- $K_r$  : Right angle shaft load (resultant force of tangential force and separating force), N
- $K_a$  : Parallel load on shaft, N
- $T$  : Input torque, N · mm
- $H$  : Transmitted force, kW
- $n$  : Rotational speed min<sup>-1</sup>
- $D_p$  : Gear pitch circle diameter, mm
- $\alpha$  : Gear pressure angle, deg
- $\beta$  : Helix angle, deg

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 0.02mm)	1.05 to 1.1
Ordinary machined gears (Pitch and tooth profile errors of less than 0.1mm)	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 Figs. 4.4 and 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<sup>-1</sup>
- $D_{pm}$  : Mean pitch circle diameter, mm
- $\alpha$  : Gear pressure angle, deg
- $\beta$  : Helix angle, deg
- $\delta$  : Pitch cone angle, deg

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

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

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

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).

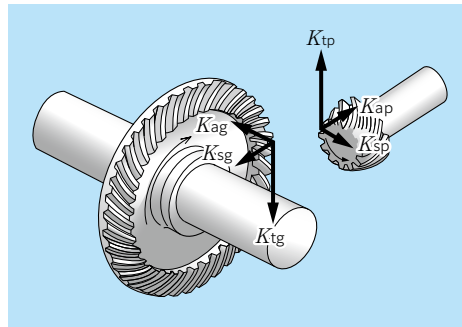


Fig. 4.4 Loads on bevel gears

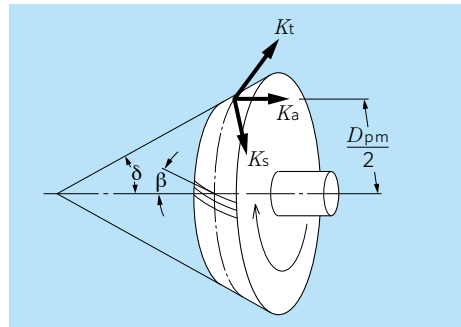


Fig. 4.5 Bevel gear diagram

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	$K_s = K_t \left[ \tan \alpha \frac{\cos \delta}{\cos \beta} + \tan \beta \sin \delta \right]$	$K_s = K_t \left[ \tan \alpha \frac{\cos \delta}{\cos \beta} - \tan \beta \sin \delta \right]$		
	Driven side	$K_s = K_t \left[ \tan \alpha \frac{\cos \delta}{\cos \beta} - \tan \beta \sin \delta \right]$	$K_s = K_t \left[ \tan \alpha \frac{\cos \delta}{\cos \beta} + \tan \beta \sin \delta \right]$		
Parallel load on gear shaft (axial load) $K_a$	Driving side	$K_a = K_t \left[ \tan \alpha \frac{\sin \delta}{\cos \beta} - \tan \beta \cos \delta \right]$	$K_a = K_t \left[ \tan \alpha \frac{\sin \delta}{\cos \beta} + \tan \beta \cos \delta \right]$		
	Driven side	$K_a = K_t \left[ \tan \alpha \frac{\sin \delta}{\cos \beta} + \tan \beta \cos \delta \right]$	$K_a = K_t \left[ \tan \alpha \frac{\sin \delta}{\cos \beta} - \tan \beta \cos \delta \right]$		

**(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.

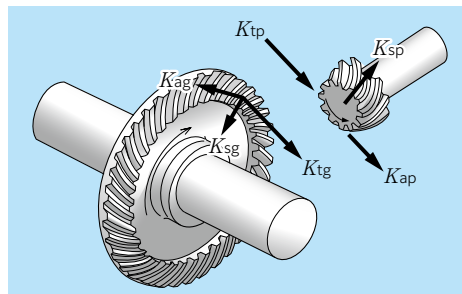


Fig. 4.6

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  $\text{min}^{-1}$
- $D_p$  : Gear mean pitch circle diameter, mm
- $\alpha$  : Gear pressure angle, deg
- $\beta$  : Helix angle, deg
- $\delta_1$  : Tooth tip cone angle, deg
- $\delta_2$  : Tooth bottom cone angle, deg
- \* 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)	

**(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.

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

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

$$K_{sp} = \frac{K_{tp}}{\cos \beta_p} (\tan \alpha_p \cos \delta_{p1} + \sin \beta_p \sin \delta_{p1}) \dots (4.11)$$

$$K_{sp} = \frac{K_{tp}}{\cos \beta_p} (\tan \alpha_p \cos \delta_{p1} - \sin \beta_p \sin \delta_{p1}) \dots (4.12)$$

$$K_{sg} = \frac{K_{tg}}{\cos \beta_g} (\tan \alpha_g \cos \delta_{g2} - \sin \beta_g \sin \delta_{g2}) \dots (4.13)$$

$$K_{sg} = \frac{K_{tg}}{\cos \beta_g} (\tan \alpha_g \cos \delta_{g2} + \sin \beta_g \sin \delta_{g2}) \dots (4.14)$$

$$K_{ap} = \frac{K_{tp}}{\cos \beta_p} (\tan \alpha_p \sin \delta_{p1} - \sin \beta_p \cos \delta_{p1}) \dots (4.15)$$

$$K_{ap} = \frac{K_{tp}}{\cos \beta_p} (\tan \alpha_p \sin \delta_{p1} + \sin \beta_p \cos \delta_{p1}) \dots (4.16)$$

$$K_{ag} = \frac{K_{tg}}{\cos \beta_g} (\tan \alpha_g \sin \delta_{g2} + \sin \beta_g \cos \delta_{g2}) \dots (4.17)$$

$$K_{ag} = \frac{K_{tg}}{\cos \beta_g} (\tan \alpha_g \sin \delta_{g2} - \sin \beta_g \cos \delta_{g2}) \dots (4.18)$$

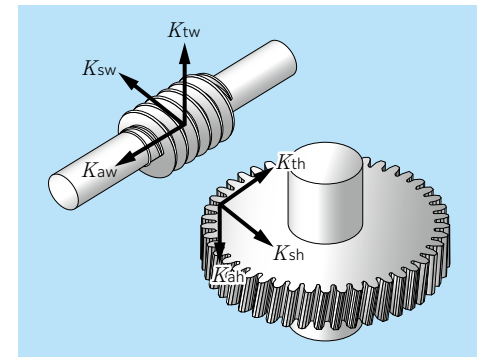


Fig. 4.7 Worm gears

- $K_t$  : 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  $\text{min}^{-1}$
- $D_p$  : Gear mean pitch circle diameter, mm
- $\alpha$  : Gear pressure angle, deg
- $\gamma$  : Worm lead angle, deg
- \* The worm shaft has a subscript  $w$ , and the worm gear has a subscript  $h$ .

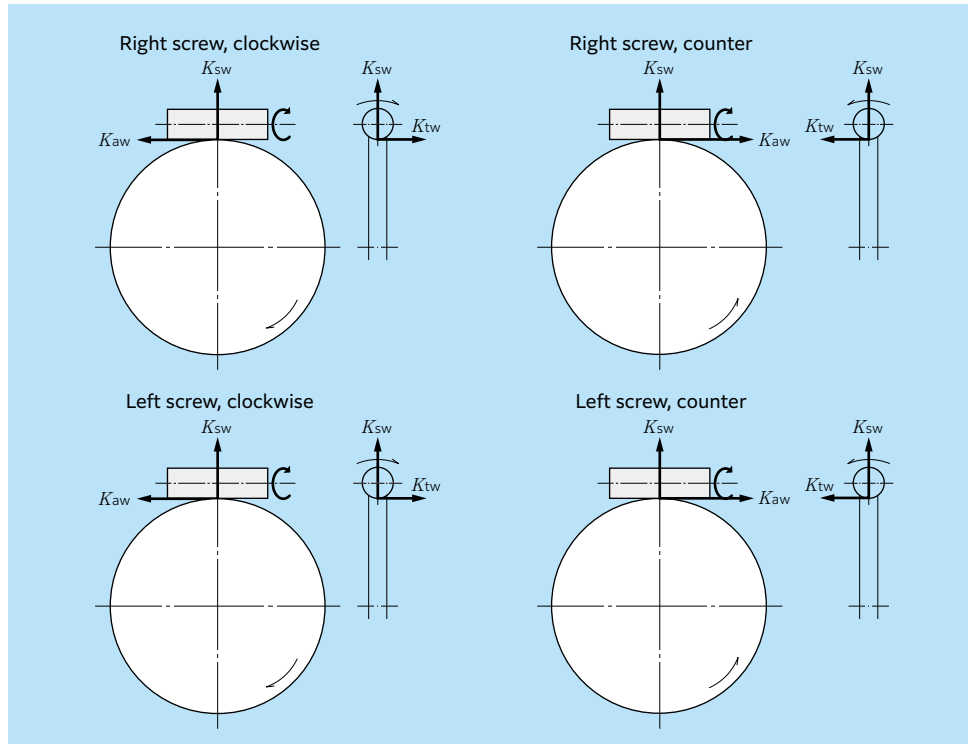


Fig. 4.8 Load acting on worm gears

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.

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

Where:

- $K_t$  : Sprocket/pulley tangential load, N
- $H$  : Transmitted power, kW
- $D_p$  : Sprocket/pulley pitch diameter, mm

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.

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

Where:

- $K_r$  : Sprocket or pulley radial load, N
- $f_b$  : Chain or belt factor (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

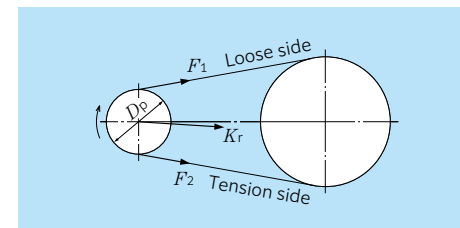


Fig. 4.9 Chain / belt loads

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 formulas (4.21) and (4.22).

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

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

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

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.

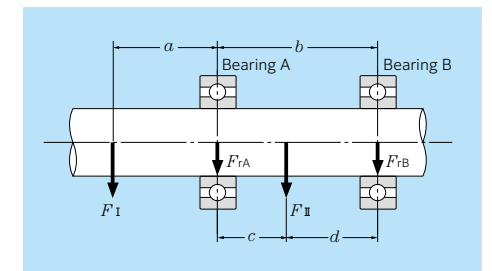


Fig. 4.10

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 (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.

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

Where:

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

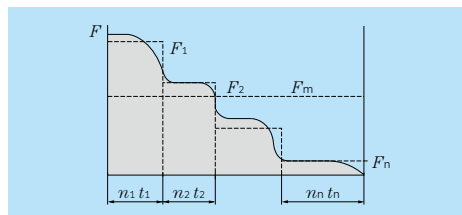


Fig. 4.11 Stepped load

(2) Continuously fluctuating load (Fig. 4.12)

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

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

Where:

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

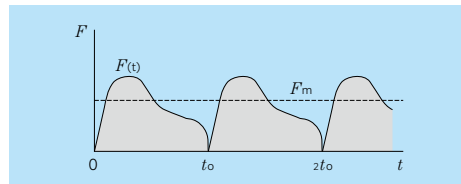


Fig. 4.12 Load that fluctuated as function of time

(3) Linear fluctuating load (Fig. 4.13)

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

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

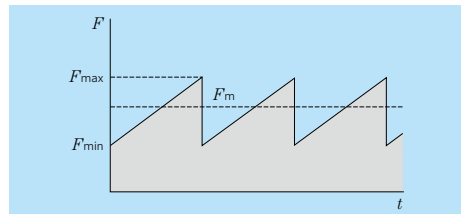


Fig. 4.13 Linear fluctuating load

(4) Sinusoidal fluctuating load (Fig. 4.14)

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

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

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

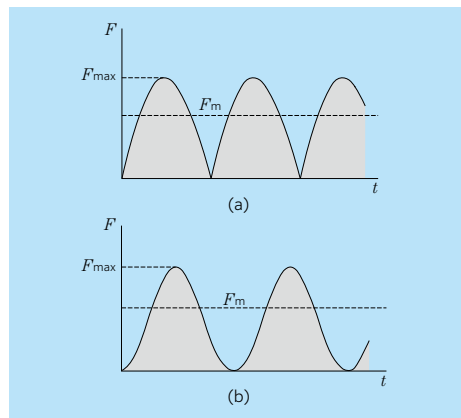


Fig. 4.14 Sinusoidal variable load

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).

$$P_r = XF_r + YF_a \dots \dots \dots (4.28)$$

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  $\alpha$  contact angle of  $90^\circ$  cannot carry radial loads. However, self-aligning thrust roller bearings can accept some radial load. The dynamic equivalent axial load for these bearings is given in formula (4.29).

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

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 (4.31). The greater of the two resultant values is always taken for  $P_{Or}$ .

$$P_{Or} = X_0 F_r + Y_0 F_a \dots \dots \dots (4.30)$$

$$P_{Or} = F_r \dots \dots \dots (4.31)$$

Where:

- $P_{Or}$  : 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 spherical thrust roller bearings the static equivalent axial load is expressed by formula (4.32).

$$P_{Oa} = F_a + 2.7F_r \dots \dots \dots (4.32)$$

Where:

- $P_{Oa}$  : 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 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).

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

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.

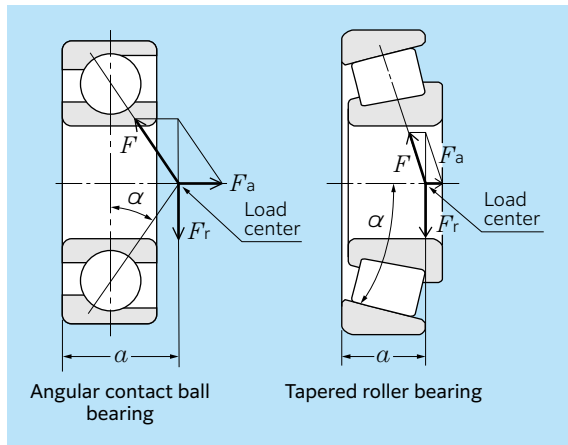


Fig. 4.15 Pressure cone apex and axial component force

Table 4.7 Bearing arrangement and equivalent load

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

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.

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:

$$P_r = XF_r + YF_a = 0.56 \times 3.2 + 1.43 \times 1.8 = 4.38 \text{ kN}$$

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<sup>-1</sup>.

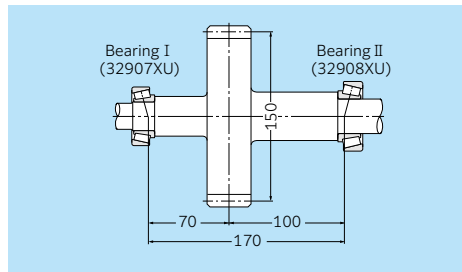


Fig. 4.16 Spur gear diagram

The gear load from formulas (4.3), (4.4a) and (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:

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

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

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

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:

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

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

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

$$L_{hI} = 500 f_{hI} = 1\,490 \text{ hours}$$

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

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

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

### (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 $\phi_i$ %	Radial load $F_{ri}$ kN	Axial load $F_{ai}$ kN	Rotational speed $n_i$ min <sup>-1</sup>
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.

Radial load  $F_r = 10$  kN

Rotational speed  $n = 2\,000$  min<sup>-1</sup>

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<sup>-1</sup>, 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.

In formula (3.13) on page A-32, based on **NUP312** from Table 3.7 on page A-33,  $k = 0.065$ .

In addition,  $D_{pw} = (60 + 130) / 2 = 95$  mm,  $n = 2\,000$  min<sup>-1</sup>.

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,  $D_{pw} \cdot n = 19 \times 10^4$ . In the case of the intermittent axial load, allowable surface pressure at the lip  $P_t = 40$  MPa.

Therefore the allowable axial load,  $P_t$ , becomes the following.

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

Based on Table 3.7 on page A-33, it is within the limits of  $F_{a \max} < 0.4 \times 10\,000 = 4\,000$  N.

Therefore  $P_t < 4\,000$  N.