6. Bearing Load Calculation

6.1 Loads acting on shafts

To compute bearing loads, the forces which act on the shaft being supported by the bearing must be determined. These forces include the inherent dead weight of the rotating body (the weight of the shafts and components themselves), loads generated by the working forces of the machine, and loads arising from transmitted power.

It is possible to calculate theoretical values for these loads; however, there are many instances where the load acting on the bearing is usually determined by the nature of the load acting on the main power transmission shaft.

6.1.1. 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 load calculation methods given herein are for two general-use gear and shaft arrangements: parallel shaft gears, and cross shaft gears. For load calculation methods regarding other types of gear and shaft arrangements, please consult NTN.

(1) Loads acting on parallel shaft gears

The forces acting on spur and helical parallel shaft gears are depicted in Figs. 6.1, 6.2, and 6.3. The load magnitude can be found by using formulas (6.1), through (6.4).

$$K_t = \frac{19.1 \times 10^6 \bullet HP}{D_p \bullet n} \cdots\cdots (6.1)$$

$$K_s = K_t \bullet \tan\alpha \text{(Spur gear)} \cdots\cdots (6.2a)$$

$$= K_t \bullet \frac{\tan\alpha}{\cos\beta} \text{(Helical gear)} \cdots\cdots (6.2b)$$

$$K_r = \sqrt{K_t^2 + K_s^2} \cdots\cdots (6.3)$$

$$K_a = K_t \bullet \tan\beta \text{(Helical gear)} \cdots\cdots (6.4)$$

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
HP : Transmission force kW
n : Rotational speed, r/min
D_p : Gear pitch circle diameter mm
α : Gear pressure angle
β : Gear helix angle

Because the actual gear load also contains vibrations and shock loads as well, the theoretical load obtained by the above formula should also be adjusted by the gear factor fz as shown in Table 6.1.

Table 6.1 Gear factor f_z

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

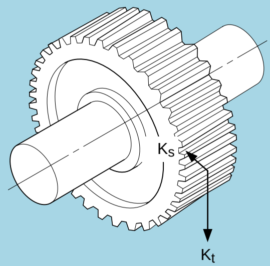


Fig. 6.1 Spur gear loads

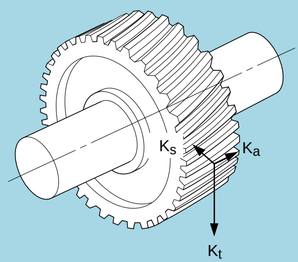


Fig. 6.2 Helical gear loads

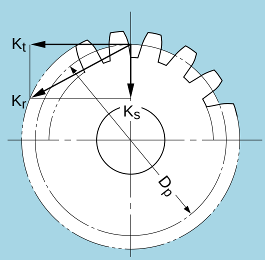


Fig. 6.3 Radial resultant forces

(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. 6.4 and 6.5. The calculation methods for these gear loads are shown in Table 6.2. Herein, to calculate gear loads for straight bevel gears, the helix angle β = 0. The symbols and units used in Table 6.2 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
HP : Transmission force kW
n : Speed in rpm
D_{pm} : Mean pitch circle diameter mm
α : Gear pressure angle
β : Helix angle
δ : Pitch cone angle

In general, the relationship between the gear load and the pinion gear load, due to the right angle intersection of the two shafts, is as follows:

$$K_{sp} = K_{ag} \quad (6.5)$$

$$K_{ap} = K_{sg} \quad (6.6)$$

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. 6.5 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. 6.5 is assumed to be clockwise (right).

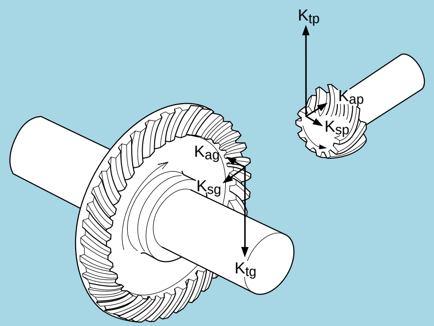


Fig. 6.4 Loads on bevel gears

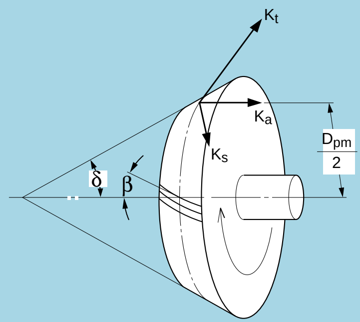


Fig. 6.5 Bevel gear diagram

Table 6.2 Loads acting on bevel gears — Unit N

Pinion	Rotation direction	Clockwise	Counter clockwise	Clockwise	Counter clockwise
	Helix direction	Right	Left	Right	Left
Tangential load K_t		$K_t = \frac{19.1\times10^6 \bullet HP}{D_{pm} \bullet n}$			
Separating 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	$K_s = K_t\left(\tan\alpha\frac{\cos\delta}{\cos\beta} - \tan\beta\sin\delta\right)$		$K_s = K_t\left(\tan\alpha\frac{\cos\delta}{\cos\beta} + \tan\beta\sin\delta\right)$	
Axial load K_a	Driving side	$K_s = K_t\left(\tan\alpha\frac{\sin\delta}{\cos\beta} - \tan\beta\cos\delta\right)$		$K_a = K_t\left(\tan\alpha\frac{\sin\delta}{\cos\beta} + \tan\beta\cos\delta\right)$	
	Driven side	$K_a = K_t\left(\tan\alpha\frac{\sin\delta}{\cos\beta} + \tan\beta\cos\delta\right)$		$K_a = K_t\left(\tan\alpha\frac{\sin\delta}{\cos\beta} - \tan\beta\cos\delta\right)$	

6.1.2. 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 (6.7).

$$K_t = \frac{19.1\times10^6 \bullet HP}{D_p \bullet n} \cdots\cdots (6.7)$$

where,

K_t : Sprocket/pulley tangential load N
HP : Transmitted force kW
D_p : Sprocket/pulley pitch diameter mm

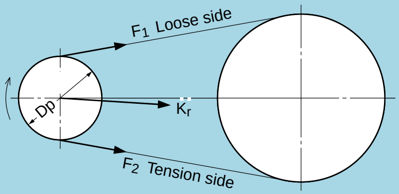


Fig. 6.6 Chain/belt loads

Table 6.3 Chain or belt factor f_b

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

For belt drives, and 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 (6.8). For chain drives, the same formula can also be used if vibrations and shock loads are taken into consideration.

$$K_r = f_b \bullet K_t \cdots\cdots (6.8)$$

where,

K_r : Sprocket or pulley radial load N
f_b : Chain or belt factor (Table 6.3)

6.1.3 Load factor

There are many instances where the actual operational shaft load is much greater than the theoretically calculated load, due to machine vibration and/or shock. This actual shaft load can be found by using formula (6.9).

where, $$K = f_w \bullet K_c \cdots\cdots (6.9)$$

K : Actual shaft load N
K_c : Theoretically calculated value N
f_w : Load factor (Table 6.4)

Table 6.4 Load factor fw

Amount of shock	fw	Application
Very little or no shock	1.0 ~ 1.2	Electric machines, machine tools, measuring instruments
Light shock	1.2 ~ 1.5	Railway vehicles, automobiles, rolling mills, metal working machines, paper making machines, rubber mixing machines, printing machines, aircraft, textile machines, electrical units, office machines
Heavy shock	1.5 ~ 3.0	Crushers, agricultural equipment, constuction equipment, cranes

6.2 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.

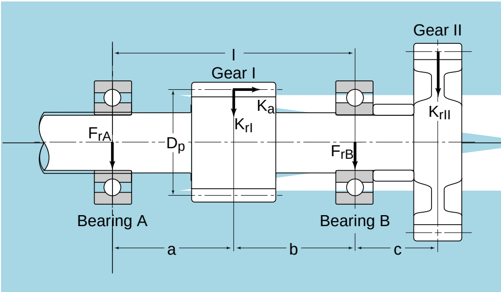
For example, in the gear shaft assembly depicted in Fig. 6.7, the applied bearing loads can be found by using formulas (6.10) and (6.11).

$$F_{rA} = K_{rI}\frac{b}{l} - K_{rII}\frac{c}{l} - K_a\frac{D_p}{2l} \cdots\cdots (6.10)$$

$$F_{rB} = K_{rI}\frac{a}{l} + K_{rII}\frac{a+b+c}{l} + K_a\frac{D_p}{2l} \cdots\cdots (6.11)$$

where,

F_{rA} : Radial load on bearing A N
F_{rB} : Radial load on bearings B N
K_{rI} : Radial load on gear I N
K_a : Axial load on gear I N
K_{rII} : Radial load on gear II N
D_p : Gear I pitch diameter mm
l : Distance between bearings mm



6.3 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

The mean bearing load, F_m, for stepped loads is calculated from formula (6.12). F_1, F_2 ··· F_n are the loads acting on the bearing; n_1, n_2···.n_n and t_1, t_2 ··· t_n are the bearing speeds and operating times respectively.

$$F_m = \left[\frac{\Sigma\left(F_i^p n_i t_i\right)}{\Sigma\left(n_i t_i\right)}\right]^{1/p} \cdots\cdots\cdots\cdots\cdots\cdots (6.12)$$

where,

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

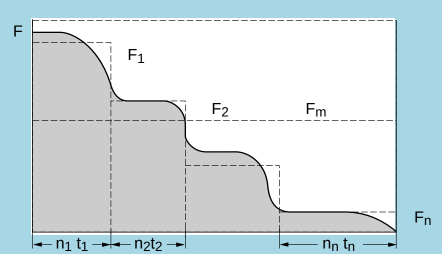


Fig. 6.8 Stepped load

(2) Consecutive series load

Where it is possible to express the function $F(t)$ in terms of load cycle to and time t, the mean load is found by using formula (6.13).

$$F_m = \left[\frac{1}{t_0}\int_0^{t_0} F(t)^p d_t\right]^{1/p} \cdots\cdots\cdots\cdots\cdots\cdots (6.13)$$

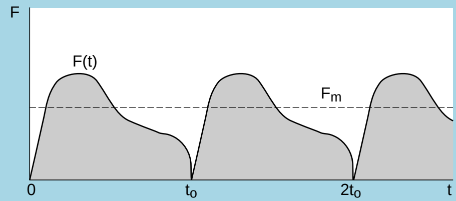


Fig. 6.9 Time function series load

(3) Linear fluctuating load

The mean load, F_m, can be approximated by formula (6.14).

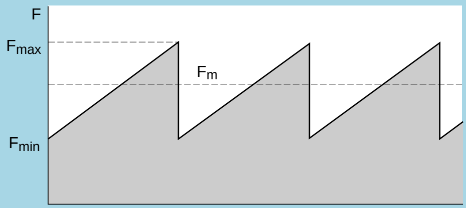


Fig. 6.10 Linear fluctuating load

$$F_m = \frac{F_{min} + 2F_{max}}{3} \cdots\cdots\cdots\cdots\cdots\cdots (6.14)$$

(4) Sinusoidal fluctuating load

The mean load, F_m, can be approximated by formula (6.15) and (6.16).

(a) $F_m = 0.75\ F_{max}$ .... (6.15)

(b) $F_m = 0.65\ F_{max}$ .... (6.16)

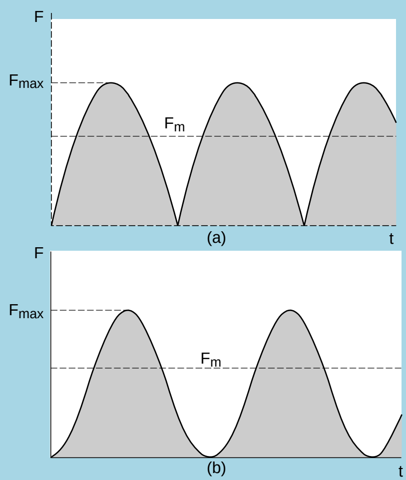


Fig. 6.11 Sinusoidal variable load

6.4 Equivalent load

6.4.1 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 (6.17).

$$P_r = XF_r + YF_a \cdots\cdots\cdots\cdots (6.17)$$

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 a contact angle of 90° 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 (6.18).

$$P_a = F_a + 1.2F_r \cdots\cdots\cdots\cdots (6.18)$$

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.

6.4.2. 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 (6.19) or (6.20). The greater of the two resultant values is always taken for P_{or}.

$$P_{or} = X_o F_r + Y_o F_a \cdots\cdots\cdots\cdots (6.19)$$

$$P_{or} = F_r \cdots\cdots\cdots\cdots (6.20)$$

where,

P_{or} : Static equivalent radial load N
X_o : Static radial load factor
Y_o : Static axial load factor
F_r : Actual radial load N
F_a : Actual axial load N

The values for X_o and Y_o 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 (6.21).

$$P_{oa} = F_a + 2.7F_r \cdots\cdots\cdots\cdots (6.21)$$

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.

6.4.3 Load calculation for angular ball bearings and tapered roller bearings

For angular ball bearings and tapered roller bearings the pressure cone apex (load center) is located as shown in Fig. 6.12, and their values are listed in the bearing tables.

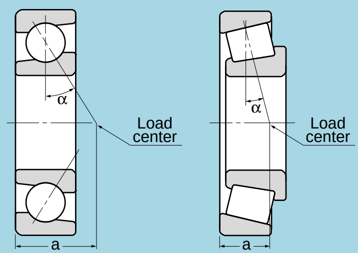


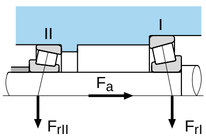
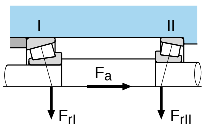
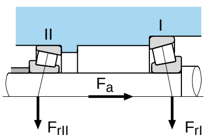
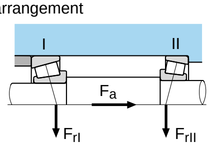
Fig. 6.12 Pressure cone apex

When radial loads act on these types of bearings the component force is induced in the axial direction. For this reason, these bearings are used in pairs (either DB or DF arrangements). For load calculation this component force must be taken into consideration and is expressed by formula (6.22).

$$F_a = \frac{0.5F_r}{Y} \cdots\cdots (6.22)$$

The equivalent radial loads for these bearing pairs are given in Table 6.5.

Table 6.5 Bearing arrangement and dynamic equivalent load

Bearing arrangement	Load condition	Axial load	Equivalent radial load
DB arrangement 	$\frac{0.5F_{rII}}{Y_{II}} \le \frac{0.5F_{rI}}{Y_I} + F_a$	$F_{aI} = \frac{0.5F_{rI}}{Y_I}$ $F_{aII} = \frac{0.5F_{rI}}{Y_I} + F_a$	$P_{rI} = F_{rI}$ $P_{rII} = XF_{rII} + Y_{II}F_{aII}$
DF arrangement 	$\frac{0.5F_{rII}}{Y_{II}} > \frac{0.5F_{rI}}{Y_I} + F_a$	$F_{aI} = \frac{0.5F_{rII}}{Y_{II}} - F_a$ $F_{aII} = \frac{0.5F_{rII}}{Y_{II}}$	$P_{rI} = XF_{rI} + Y_IF_{aI}$ $P_{rII} = F_{rII}$
DB arrangement 	$\frac{0.5F_{rI}}{Y_I} \le \frac{0.5F_{rII}}{Y_{II}} + F_a$	$F_{aI} = \frac{0.5F_{rII}}{Y_{II}} + F_a$ $F_{aII} = \frac{0.5F_{rII}}{Y_{II}}$	$P_{rI} = XF_{rI} + Y_IF_{aI}$ $P_{rII} = F_{rII}$
DF arrangement 	$\frac{0.5F_{rI}}{Y_I} > \frac{0.5F_{rII}}{Y_{II}} + F_a$	$F_{aI} = \frac{0.5F_{rI}}{Y_I}$ $F_{aII} = \frac{0.5F_{rI}}{Y_I} - F_a$	$P_{rI} = F_{rI}$ $P_{rII} = XF_{rII} + Y_{II}F_{aII}$

Note: 1) The above are valid when the bearing internal clearance and preload are zero.
2) Radial forces in the opposite direction to the arrow in the above illustration are also regarded as positive.

6.5 Bearing rated 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.

(Example 1)

What is the rating life in hours of operation (L_{10h}) for deep groove ball bearing 6208 operating at 650 r/min, with a radial load F_r of 3.2 kN?

For formula (6.17) the dynamic equivalent radial load *Pr* is:

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

The basic dynamic rated load for bearing 6208 (from bearing table) is 29.1 kN, and the speed factor (*fn*)for ball bearings at 650 r/min (*n*) from Fig. 5.1 is 0.37. The life factor, *fh*, from formula (5.3) is:

$$f_h = f_n \frac{C_r}{P_r} = 0.37 \times \frac{29.1}{3.2} = 3.36$$

Therefore, with f_h=3.36 from Fig. 5.1 the rated life, L_{10h}, is approximately 19,000 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 and axial load factor Y are used. The basic static load rating, C_{or}, for bearing 6208 is 17.8 kN.

$$\frac{F_a}{C_{or}} = \frac{1.8}{17.8} = 0.10$$

Therefore, from the bearing tables e=0.29.
For the operating radial load and axial load:

$$\frac{F_a}{F_r} = \frac{1.8}{3.2} = 0.56 > e = 0.29$$

From the bearing tables X=0.56 and Y=1.48, and from formula (6.17) the equivalent radial load, P_r, is:

$$P_r = XF_r + YF_a = 0.56 \times 3.2 + 1.48 \times 1.8 = 4.46 \text{ kN}$$

From Fig. 5.1 and formula (5.3) the life factor, f_h, is:

$$f_h = f_n \frac{C_r}{P_r} = 0.37 \times \frac{29.1}{4.46} = 2.41$$

Therefore, with life factor f_h=2.41, from Fig. 5.1 the rated life, L_{10h}, is approximately 7,000 hours.

(Example 3)

Determine the optimum model number for a cylindrical roller bearing operating at 450 r/min, with a radial load F_r of 200 kN, and which must have a life of over 20,000 hours.

From Fig. 5.1 the life factor f_h=3.02 (L_{10h} at 20,000), and the speed factor f_n=0.46 (n=450 r/min). To find the required basic dynamic load rating, C_r, formula (5.3) is used.

$$C_r = \frac{f_h}{f_n} P_r = \frac{3.02}{0.46} \times 200 = 1313 \text{ kN}$$

From the bearing table, the smallest bearing that fulfills all the requirements is NU2336 (C_r=1,380 kN).

(Example 4)

What are the rated lives of the two tapered roller bearings supporting the shaft shown in Fig. 6.13?

Bearing II is an ET-32206 with a C_r=54.5 kN, and bearing I is an ET-32205 with a C_r=42.0 kN. The spur gear shaft has a pitch circle diameter D_p of 150 mm, and a pressure angle α of 20°. The gear transmitted force HP=150 kW at 2,000 r/min (speed factor n).

The gear load from formula (6.1), (6.2a) and (6.3) is:

$$K_t = \frac{19.1 \times 10^6 \bullet HP}{D_p \bullet n} = \frac{19\,100 \times 150}{150 \times 2\,000} = 9.55 \text{ kN}$$

$$K_s = K_t \bullet \tan\alpha = 9.55 \times \tan 20° = 3.48 \text{ kN}$$

$$K_r = \sqrt{K_t^2 + K_s^2} = \sqrt{9.55^2 + 3.48^2} = 10.16 \text{ kN}$$

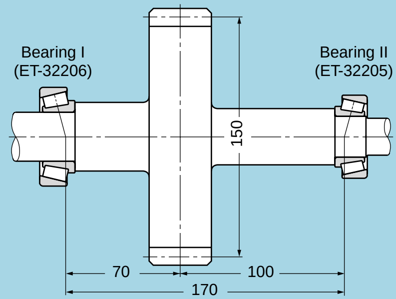


Fig. 6.13 Spur gear diagram

The radial loads for bearings I and II are:

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

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

$$\frac{0.5F_{rI}}{Y_I} = 1.87 > \frac{0.5F_{rII}}{Y_{II}} = 1.31$$

The equivalent radial load is:

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

$$P_{rII} = XF_{rII} + Y_{II} \bullet \frac{0.5F_{rI}}{Y_I} = 0.4 \times 4.18 + 1.60 \times 1.87$$
$$= 4.66 \text{ kN}$$

From formula (5.3) and Fig. 5.1 the life factor, f_h, for each bearing is:

$$f_{hI} = f_n \frac{C_{rI}}{P_{rI}} = 0.293 \times \frac{54.5}{5.98} = 2.67$$

$$f_{hII} = f_n \frac{C_{rII}}{P_{rII}} = 0.293 \times \frac{42.0}{4.66} = 2.64$$

Therefore,

L_{hI} =13,200 hours

L_{hII} =12,700 hours

The combined bearing life, L_h, from formula (5.6) is:

$$L_h = \frac{1}{\left(\frac{1}{L_{hI}{}^e} + \frac{1}{L_{hII}{}^e}\right)^{1/e}} = \frac{1}{\left(\frac{1}{13\ 200^{9/8}} + \frac{1}{12\ 700^{9/8}}\right)^{8/9}}$$

$$= 6\ 990 \text{ hours}$$

(Example 5)

Find the mean load for spherical roller bearing 23932 (C_r=320 kN) when operated under the fluctuating conditions shown in Table 6.6.

Table 6.6

Condition No. i	Operating time % ϕ_i	radial load F_{ri} kN	axial load F_{ai} kN	revolution n_i r/min
1	5	10	2	1200
2	10	12	4	1000
3	60	20	6	800
4	15	25	7	600
5	10	30	10	400

The equivalent radial load, P_r, for each operating condition is found by using formula (6.17) and shown in Table 6.7. Because all the values for F_{ri} and F_{ai} from the bearing tables are greater than $\frac{F_a}{F_r} > e = 0.18,\ X = 0.67$ and $Y_2 = 5.50.$

$$P_{ri} = XF_{ri} + Y_2F_{ai} = 0.67F_{ri} + 5.50F_{ai}$$

Table 6.7

Condition No. i	Equivalent radial load P_{ri} kN
1	17.7
2	30.0
3	46.4
4	55.3
5	75.1

From formula (6.12) the mean load, F_m, is:

$$F_m = \left[\frac{\Sigma\left(P_{ri}^{10/3} \bullet n_i \bullet \phi_i\right)}{\Sigma\left(n_i \bullet \phi_i\right)}\right]^{3/10} = 48.1 \text{ kN}$$