3. Load Capacity and Life of Bearings

3. Load Capacity and Life of Rolling Contact Bearings

3.1 Basic Dynamic Load Rating and Rating Life

Although requirements of rolling contact bearings vary somewhat with the individual application the principal requirements are:

- High load capabilities
- Smooth and quiet rotation
- High rigidity
- Low friction
- High accuracy
- Reliability

The reliability or durability requirement sets the time frame over which all other requirements are to be maintained. The reliability requirement (life in the broad sense) includes grease and acoustic life, as well as fatigue life. Reliability is reduced by various types of damage and degradation.
Improper handling, mounting, lubrication, and fits are the major causes of problems leading to lower-than-calculated bearing life. Regardless of how well they are maintained or mounted or handled, dynamic bearings will eventually fail from rolling fatigue generated by the repetitive stress of bearing load.
The service life of a bearing can be examined from two perspectives: 1)If, from inspection, a trace of fatigue becomes noticeable, the bearing should be deemed not suitable for further use; or 2) length of bearing life in hours or revolutions can be predefined as a limit beyond which the bearing is automatically replaced.
Since calculated fatigue life will vary with the size and type of bearings used under identical load conditions, great care must be taken in the analysis of the load conditions and the final choice of bearings to satisfy the application requirements.
Fatigue lives of individual bearing are dispersed. When a group of identical bearings operate under the same conditions, the statistical phenomenon of dispersion will appear. Use of average life is not an adequate criterion for selecting rolling contact bearings. Instead, it is more appropriate to consider a limit (hours or numbers of revolutions) which a large percentage of the operating bearings can attain.
Accordingly, the rating life and basic dynamic load rating Cr or Ca are defined using the following definition:

- Basic rating life is defined as the total number of revolutions (or total operating hours at some given constant speed) that 90% of a group of identical bearings operated individually under equal conditions can complete without suffering material damage from rolling fatigue.
- Basic dynamic load rating (Cr or Ca) is defined as a bearing load of constant direction and size that ends the bearing life after a million revolutions.

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Constant-direction radial or thrust loads (for radial and thrust bearings, respectively) are used as the basis of the ratings.
The rating life of bearings is calculated by formulas (3.1) and (3.2):

$$L = \left(\frac{C}{P}\right)^{p} \quad \cdots\cdots (3.1)$$

$$Lh = \left(\frac{C}{P}\right)^{p} \cdot \frac{10^6}{60\ n} \quad \cdots\cdots (3.2)$$

Where:
- L : Basic rating life (10^6 rev.)
- Lh : Basic rating life in hours
- C : Basic dynamic load rating (N). (Cr for radial bearings and Ca for thrust bearings)
- P : Bearing load (dynamic equivalent load) (N) Pr for radial, and, Pa for thrust bearings
- p : 3 for ball, 10/3 for roller bearings
- n : Rotating speed (rpm)

The relationship of fh, the bearing life factor and fn, the speed factor, is outlined in Table 3.1.
Formula (3.3) may be used to determine the basic dynamic load rating, C, of bearings given the bearing equivalent load, P, and the operating speed, n, in revolutions-per-minute.
The lives of automobile wheel bearings may be defined in kilometers using the formula (3.4).
Table 3.2 shows values for the life factor, fh, by application and machine type.
If a bearing is used with vibrating or impact loads or low speed including no rotation, additional study with basic static load rating is required.

Table 3.1 Bearing Basic Rating Life; Life and Speed Factors

	Ball Bearings	Roller Bearings
Basic Rating Life	$Lh = 500fh^{3}$	$Lh = 500fh^{\frac{10}{3}}$
Life Factor	$fh = fn\dfrac{C}{P}$	$fh = fn\dfrac{C}{P}$
Speed Factor	$fn = \left(\dfrac{10^6}{500 \times 60n}\right)^{\frac{1}{3}}$	$fn = \left(\dfrac{10^6}{500 \times 60n}\right)^{\frac{3}{10}}$

$$C = \frac{P}{fn} \cdot \left(\frac{Lh}{500}\right)^{1/p} \quad \cdots\cdots (3.3)$$

$$Ls = \frac{\pi \cdot D}{1000} \cdot L \quad \cdots\cdots (3.4)$$

Where:
- Ls : Kilometer traveled (10^6 km)
- D : Outside diameter of wheel (m)
- L : Life in revolutions

Table 3.2 Life Factors (fh)

Table 3.2 Life Factors (fh)

Application conditions	Application example	Life Factor (fh)
Infrequent use	Hinges	to 1.5
Short period or Intermittent use	Hand tools Agricultural equipment Household apparatus Casting plant cranes	2 ~ 3
Intermittent, critical use	Power plant auxiliary machines Assembly line conveyers General crane applications Motors for home air conditioning	3 ~ 4
8 hour per day, intermittent	General gearing applications General industrial motors	3 ~ 5
8 hour per day, continuous	Cranes in continuous use Air blowers Mechanical power transmission General industrial machinery Industrdal wood-working machines	4 ~ 5
24 hour per day, continuous	Compressors Mine hoists Marine propeller shafts Rolling machine tables	5 ~ 8
24 hour per, critical	Paper manufacturing Power plants Water supply equipment Mine water pumps, air blowers	6-up

3.2 Basic Rating Life Calculation Guide

- Determine the bearing life normal to the application by using Table 3.2 to define the life factor, fh.
- Use rating life charts (nomograms) to calculate life. The nomogram for ball bearings is shown in Fig. 3.4. The nomogram for roller bearings is shown in Fig. 3.5. These nomograms are based on formulas (3.1) and (3.2).
- Where operating temperatures are to be in excess of 150°c, a correction factor must be applied to the bearing basic dynamic load rating. (See Item 3.3.1).
- If the bearings are to operate with vibration or impact loading, or where a bearing mounting or manufacturing error exists, the actual load may be greater than the calculated load. In this case, the calculated load must be multiplied by a safety coefficient to obtain an approximation of the actual load. For safety coefficients in actual application, refer to the machine and drive factors. (See Item 3.4.1 and 3.4.2)
- Bearings do not always operate under a constant load. When the bearing operates with a fluctuation load, the load must be converted to a constant size reflecting the effect of the fluctuating load. Conversion may be done using weighted average mean loading (See Item 3.4.4).
- By definition, bearing load Pr (net radial load) or Pa (net axial load) is a load with constant direction and size. When a composite load of radial and axial loads occurs on a radial bearing, these loads must be converted to a radial load reflecting the effect of the composite load. This effective load is called the DYNAMIC EQUIVALENT LOAD. (See Item 3.5).
- When calculating bearing load using the loads on a position on the shaft, it is necessary to calculate center distance between the load application point of the bearings. Many bearing types have load center points at the center line of the width as shown in Fig. 3.1. Single-row Angular Contact ball bearings and single-row Tapered roller bearings, have load center points off-center to the center line of the bearing width (See Fig. 3.2 and 3.3 respectively). Refer to the dimension tables for the value of the off-set.
- The axial load limit for Cylindrical roller bearings is a function of the lubrication conditions and speed of rotation. This limit differs from a rating load as determined by fatigue life. (See Item 3.7).

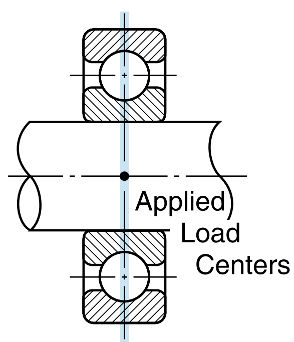

Fig 3.1

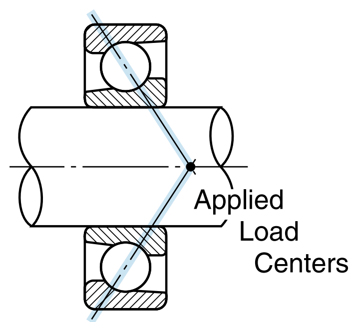

Fig 3.2

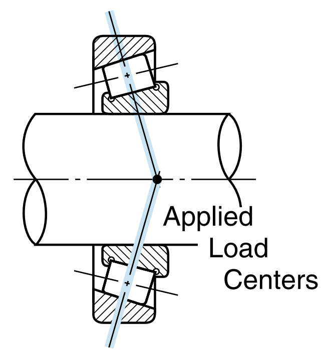

Fig 3.3

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Calculation example : 1

Suppose that an application has selection parameters as follows :
Bore : 50 mm or smaller
Outside diameter : 100 mm or smaller
Width : 20 mm or smaller
Radial load (Fr) : 4000 N (Newtons)
Rotating speed (n) : 1800rpm
Life factor (fh) : 2 or greater
Bearing type : Single-row deep groove ball bearing

From Table 3.1 the speed factor, fn is obtained as follows:

$$fn = \left(\frac{10^6}{500 \times 60 \times 1800} \right)^{1/3} = 0.265$$

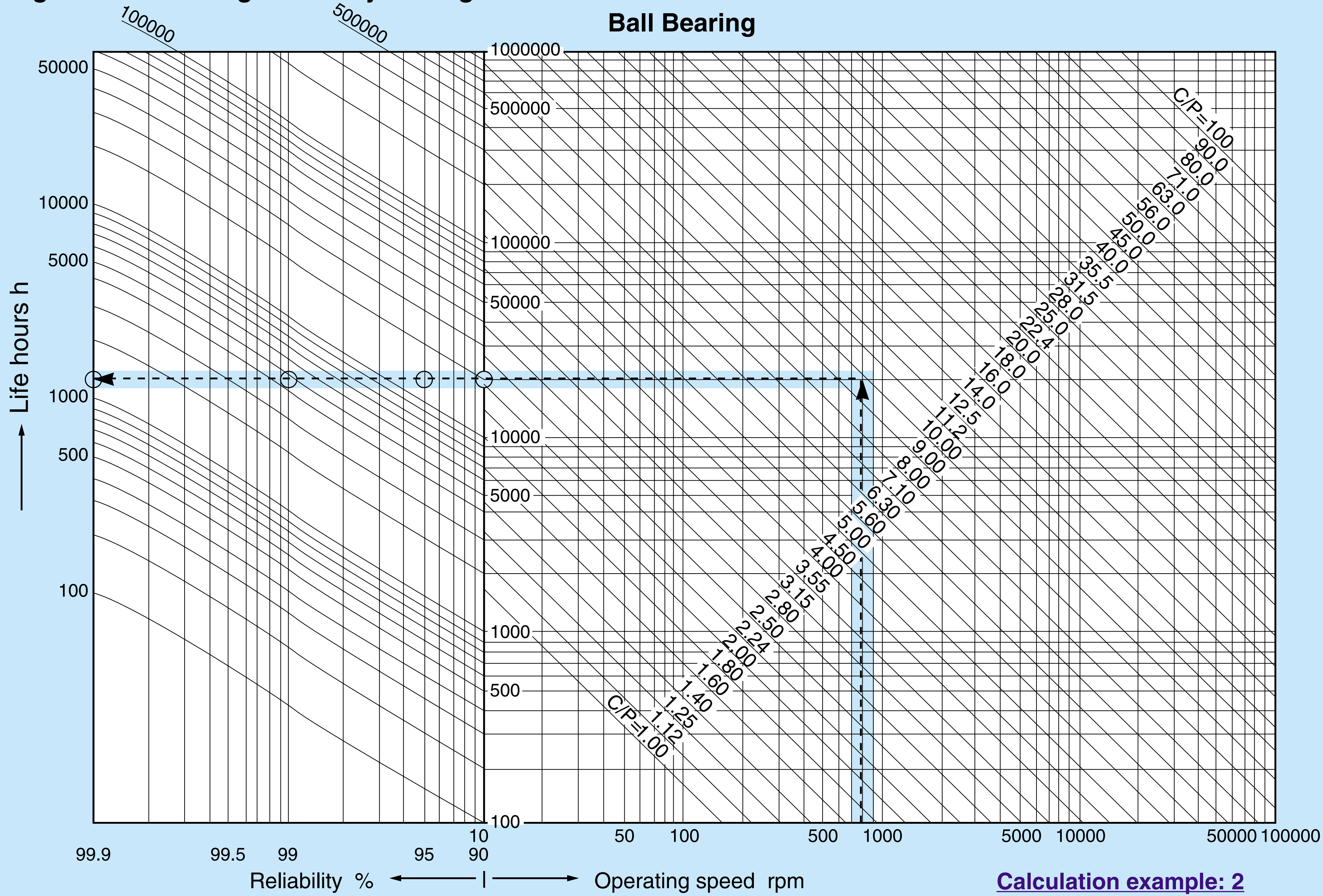
From Table 3.1,

$$Cr = \frac{fh \cdot P}{fn} = \frac{2 \times 4000}{0.265} = 30188N$$

Bearings having the required basic dynamic load rating are selected from the bearing dimension table(s). Of the two sizes meeting the load and diameter constraints, only bearing 6209 will satisfy the width constraint. Given the above parameters, bearing part 6209 would be the selection.

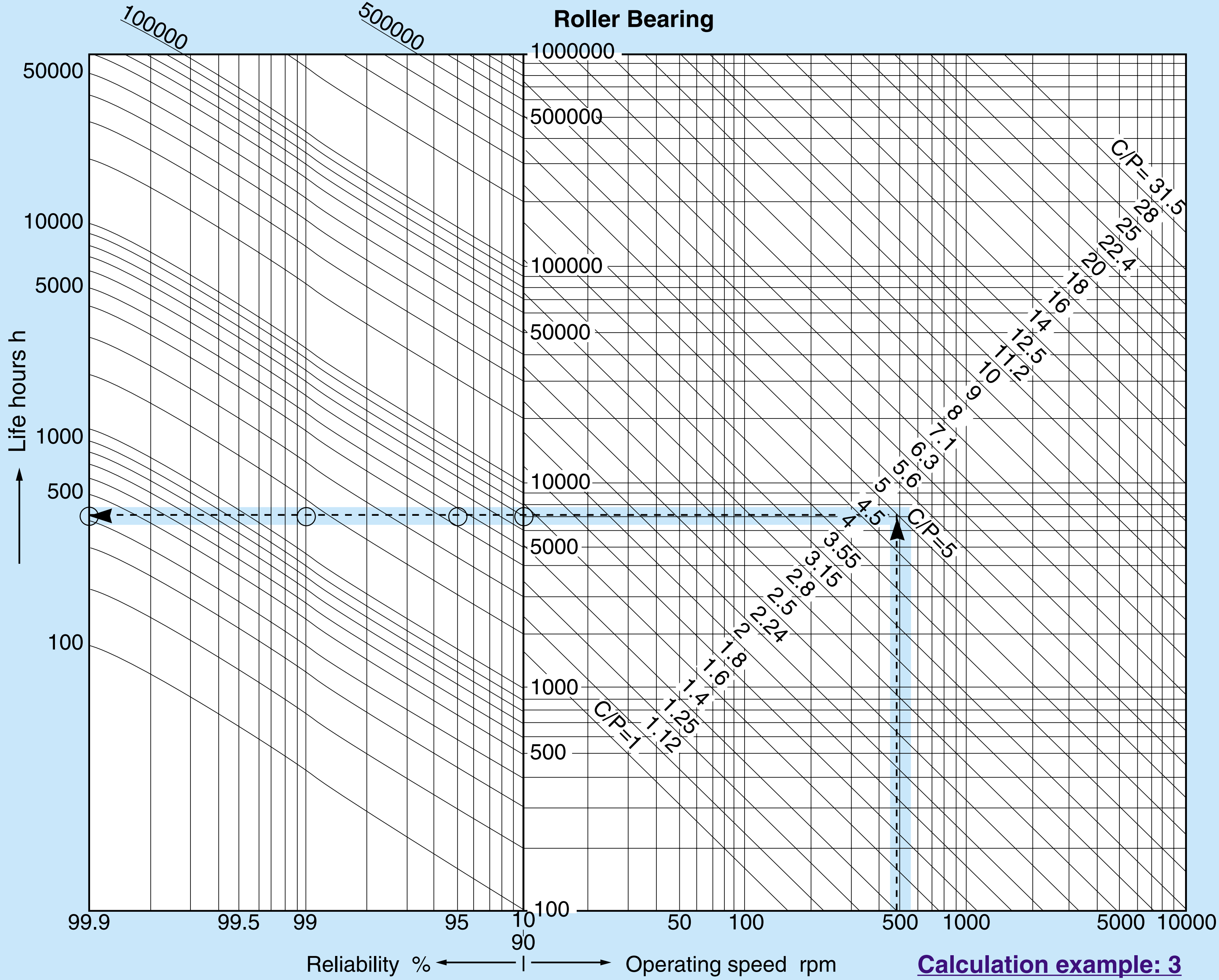
Bearing No.	Bore Dia. (mm)	Outside Dia. (mm)	Width (mm)	Basic Dynamic Load Rating (N)
6209	45	85	19	32500
6307	35	80	21	33500

Fig 3.4 Ball Bearing Reliability Nomogram



Calculation example: 2

Fig 3.5
Roller Bearing
Reliability Nomogram



Calculation example: 3

Fig 3.4 Ball Bearing Reliability Nomogram

Calculation example: 2

Bearing Number 6012 is loaded with an dynamic equivalent radial load Pr = 2950N.
Object is to obtain the life at various levels of reliability when the bearing is rotated at n = 800 rpm.
The basic dynamic load rating Cr is taken form the dimension table.

Cr = 29400N
Cr/Pr = 10

(*) For reliabilities, see Item 3. 3. 2.

By tracing the dotted lines, rating lives are obtained as follows:

Reliability (*)	Life hours
90%	20000
95%	15000
99%	4500
99.9%	1200

Fig 3.5 Roller Bearing Reliability Nomogram

Calculation example: 3

Bearing Number 22222EX is loaded with dynamic equivalent radial load Pr = 98000N.
Object is to obtain the life at various levels of reliability when the bearing is rotated at n = 500 rpm.
The basic dynamic load rating Cr is taken from the dimension table.

Cr = 490000N
Cr/Pr = 5

By tracing the dotted lines, rating lives are obtained as follows:

Reliability (*)	Life hours
90%	7000
95%	4400
99%	1500
99.9%	400

(*) For reliabilities, see Item 3. 3. 2.

3.3 Rating Life and Operating Temperature

3.3.1 Temperature-Related Decrease in Basic Dynamic Load Rating

Bearing ring diameters grow slightly with an increase in temperature. If the operating temperature does not exceed about 120°c , the bearing rings will return to their original dimensions at normal temperature. If the operating temperature exceeds this level (approximately 120°c), the bearing rings and rolling elements can undergo small, permanent changes in size.
To prevent these permanent changes in size, special heat-stabilization treatment can be used (see Table 3.3).

Table 3.3 Heat - Stabilization Treatment

Max. Operating temperature	Heat stabilization treatment symbol
~ 150°C	S26
~ 200°C	S28

The S26 heat-treated bearings will resist dimensional change through a maximum temperature of 150°c. Bearings with the S26 heat-treated steel will suffer decreases to their rating life and will have dimensional changes if they are used at temperatures in excess of 150°c.
The S28 heat-treated bearings will resist dimensional change and have a temperature factor of 0.90 through a maximum temperature of 200°c.
Bearings with the S28 heat-treated steel will suffer further decreases to their rating life and will have dimensional changes if they are used at temperatures in excess of 200°c.
Operation at temperatures exceed the limit of the heat-stabilization should be avoided to prevent bad effects of these dimensional changes.

If bearings are operated at temperatures exceeding the limit of the heat-stabilization, hardness of the bearing steel will be reduced. In calculating the life of such bearings, the basic dynamic load rating must be multiplied by the temperature factor as shown in Table 3.4. The temperature factor for standard bearings operating at a temperature under 120°c is 1 and these bearings will show no dimensional change. Standard bearings run at an operating temperature exceeding 120°c, will experience dimensional changes and are subject to the basic dynamic load rating decreases as shown in Table 3.4.

Table 3.4 Temperature Factor

Bearing Temperature	~ 150ºC	175ºC	200ºC
Temperature Factor	1	0.95	0.90

3.3.2 Life Calculation Factors

Rating Life Formula, L=(C/P)p....... (3.1), is used when applying rolling contact bearings for normal use.
To provide for utilization of lubrication theory, and advances in bearing material and bearing manufacturing technology, the ISO and JIS have adopted the following life calculation formula.

$$Lna = a_1 \cdot a_2 \cdot a_3 \cdot \left(\frac{C}{P}\right)^p \cdots\cdots\cdots (3.5)$$

where:

Lna	: Adjusted rating life (106 rev.)
a_1	: Reliability factor
a_2	: Material factor
a_3	: Application conditions factor

Formula (3.5) is applicable only when all bearing loads are considered and operating conditions are clearly defined.
Generally, reliability of 90% is used, and material and operating conditions may be considered as a_1, a_2, a_3=1, coinciding with formula (3.1).

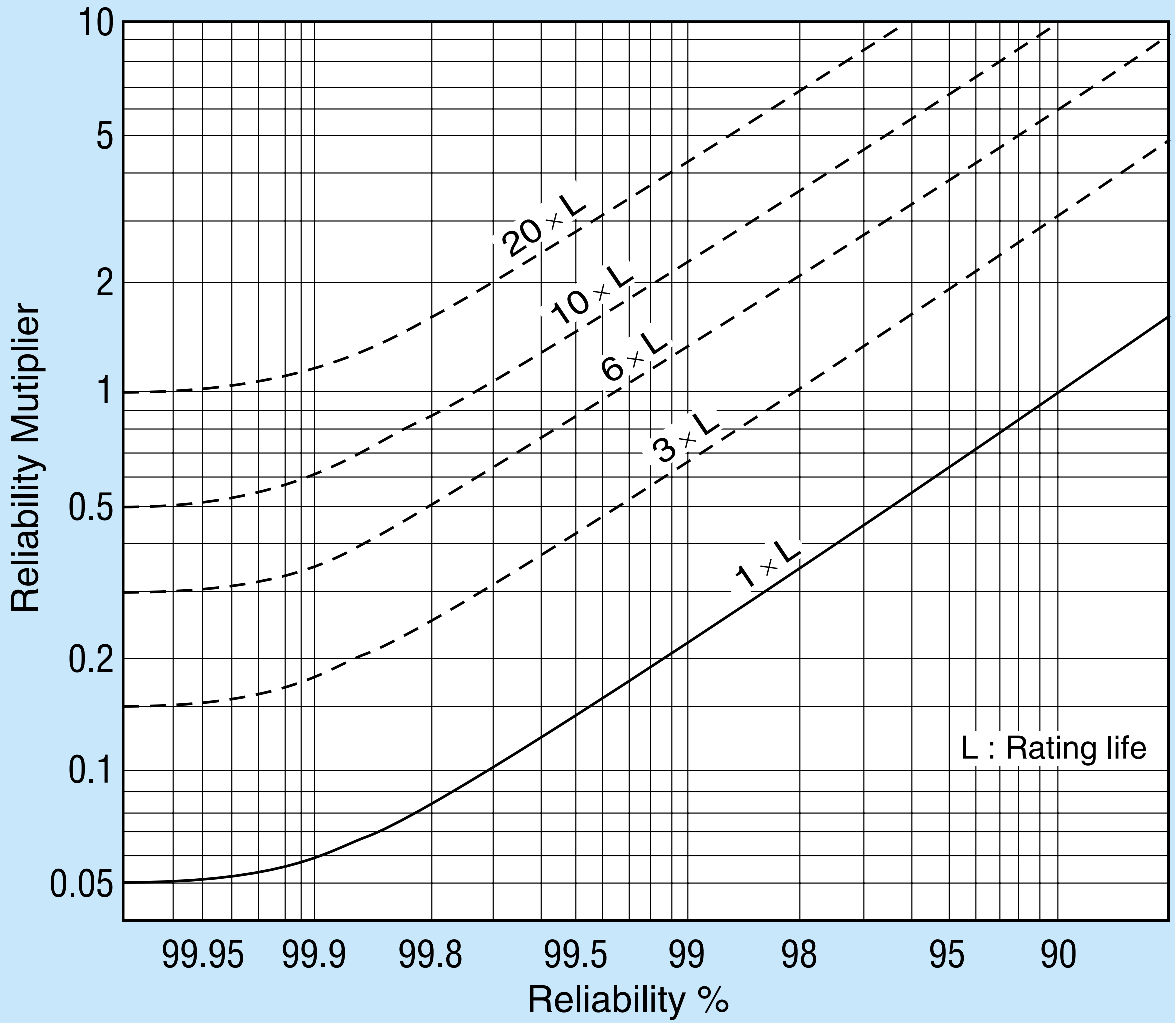
1) Reliability Factor, a_1

Reliability Factor, a1, becomes 1 if 90% of a group of identical bearings operated individually under the same conditions can complete the calculated life without exhibiting material damage from rolling fatigue. Reliability is then set as 90 %, and for reliability over 90%; a1 takes a value from Table 3.5.
As observed from Table 3.5, the calculated bearing life decreases in proportion to a higher level of bearing reliabilities.
Fig. 3.6 shows the improved reliabilities when bearings having rating lives of 3, 6, 10 and 20 times are used in comparison with the 90% reliability (life-multiplying factor being 1) of a bearing having a given rating life.

Table 3.5 Reliability Factor a_1

Reliability %	99	98	97	96	95	90
a_1 factor	0.21	0.33	0.44	0.53	0.62	1

Fig 3.6 Reliability Multiplier



Calculation example: 4

Bearing Number 6209 is used to support a radial load of 3160N. Object is to define the life and select a bearing which will have a reliability of 99.4%.

The life corresponding to the reliability of 90% is obtained as follows by reading the basic dynamic load rating, Cr=32500N from the dimension table and using formula (3.1):

$$\left(\frac{32500}{3160}\right)^3 \times 10^6 = 1088 \times 10^6 \text{rev.}$$

Reading Fig. 3.6, it can be seen that a bearing having a life -multiplying factor of 6 is required to attain 99.4% reliability. Applying this multiplier to the basic dynamic load rating, Cr as obtained from formula (3.1), will calculate as:

$$\left(\frac{Cr}{3160}\right)^3 \times 10^6 = 6 \times 1088 \times 10^6 \text{rev.}$$

From the above equation, obtain;

$$Cr = (6)^{1/3} \times 32500 = 1.817 \times 32500$$
$$= 59000\text{N}$$

The bearing meeting this basic dynamic load rating (in the same diameter series) is bearing number 6214.

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2) Material factor, a_2

Material factor, a_2, is the adjustment factor applied as an increase to rating life for type and quality of material, special manufacturing process and/or special design.

The basic dynamic load rating, Cr (or Ca), listed in the bearing dimension tables reflects both the use of vacuum-degassed, high-carbon chrome bearing steel for all NACHI rolling contact bearings as well as improvements in manufacturing technology. The a_2-factor has a base value of 1 for NACHI standard parts.

Unless specialty steels are utilized, a_2 is defined as 1 when calculating the life using the formula (3.5).

3) Application condition factor, a_3

The application condition factor, a_3, is used to consider bearing load conditions, lubricating conditions, and temperature conditions.

Factor a_3 is set as 1 if the rolling elements and raceway surfaces are separated (good lubricating condition). When lubricating conditions are poor (as in the following cases), a_3 is less than 1:

- When the operating speed is <dm · n of 10,000. (Where dm · n=rolling element pitch diameter in millimeters times the speed in revolutions-per-minute).
- When lubricant will tend to deteriorate rapidly.

At present, it is difficult to quantify the application condition factor because of the many variables involved.

Because factors a_2 and a_3 have interactive effects on each other, these two factors are treated as one value (a_2) (a_3). When lubrication and application conditions are good, the value (a_2) (a_3) can be set as equal to 1.

In case of poor lubrication such as when lubricant viscosity is considerably low, please consult NACHI.

3.4 Calculation of Bearing Load

Generally, the load that is applied to the bearings is composed of loads generated by machine operation, drive components, and dead weight of the shaft and components mounted to and on the shaft. These loads can be precisely calculated. The above loads are usually accompanied by vibration and impact. With the exception of very special cases, it is impractical to calculate and add the specific effects of vibration and impact loading on each component in a machine. To facilitate the calculation and analysis of loading in a machine system, loading factors (based on empirical experience) have been developed as multipliers to the driving and static loads.

$$F = f_s \cdot F_c \qquad \cdots\cdots\cdots (3.6)$$

where:

F : Bearing load (N)
fs : Machine factor (Table 3.6)
Fc : Calculated load (N)

When a load fluctuates in size, an average load must be calculated which reflects the effects of the fluctuating load.
When a composite load of radial and axial load occurs on a radial bearing, the loads must be converted to an effective radial load by use of the dynamic equivalent load formula for the specific bearing type. This value, P, is used in the basic rating life formula (3.1).

Table 3.6 Machine Factors, (f_s)

Type of Machine	f_s
Smooth running machinery (no impact) ; motors, conveyors, turbo compressor, paper manufacturing machinery	1 ~ 1.2
Machine with low impact; reciprocating pumps, internal combustion engine, hoists, cranes	1.2 ~ 1.5
Machines with high impact; shears, crushers, rolling mill equipment	1.5 ~ 3.0

Table 3.7 Belt Drive Factors, (f_1)

Type of drive	f_1
Flat leather belt (with tension pulley)	1.75 ~ 2.5
Flat leather belt (without tension pulley) Silk Rubber Balata	2.25 ~ 3.5
V-belt	1.5 ~ 2
Steel strip belt	4 ~ 6
Cotton belt / Hemp belt	2 ~ 6

Notes : 1. For low speed, use top value

Table 3.8 Gear Precisiion Factors, (f_z)

Type of gear	f_z
Precision (Pitch and form errors ≤ 0.02mm)	1 ~ 1.1
Normal (Pitch and form errors 0.02 ~ 0.1mm)	1.1 ~ 1.3

3.4.1 Belt Drives

Transferring of power through belt drives requires on initial belt tension. Radial load, K, that occurs from this tension can be calculated as follows:

$$M = 955000 \cdot \frac{H}{n} \quad \cdots\cdots (3.7)$$

$$Kt = \frac{M}{r} \quad \cdots\cdots (3.8)$$

where:
- M : Rotating moment of pulley (N • cm)
- Kt : Effective transfer power of belt (N) (tension side minus slack side)
- H : Transfer power (kW)
- n : Rotating speed of pulley (rpm)
- r : Radius of pulley (cm)

Load that works on the shaft through the pulley is calculated by multiplying the effective transfer power, Kt, by the belt drive factors, f_1, from Table 3.7.
Generally,

$$K = f_1 \cdot Kt \quad \cdots\cdots (3.9)$$

where:
- K : Radial load (N) applied to the pulley transferred by the belt
- f_1 : Belt drive factor (Table 3.7)

3.4.2 Gear Drives

Shaft load from gear drives are calculated using the transfer power and type of gear.
Helical, bevel and worm gears transmit radial loads and create an axial load component, while spur gears transmit only radial loads.
Gear load formulas described below refer to spur gears.

$$M = 955000 \cdot \frac{H}{n} \quad \cdots\cdots (3.10)$$

$$Kt = \frac{M}{r} \quad \cdots\cdots (3.11)$$

$$Ks = Kt \cdot \tan\alpha \quad \cdots\cdots (3.12)$$

$$Kg = \sqrt{Kt^2 + Ks^2} = Kt \cdot \sec\alpha \quad \cdots (3.13)$$

where:
- M : Rotating gear moment (N • cm)
- Kt : Tangential component of force (N)
- Ks : Radial component of force (N)
- Kg : Total gear load (N)
- H : Transfer power (kW)
- n : Rotating speed (rpm)
- r : Drive gear pitch radius (cm)
- α : Pressure angle of gear (°)

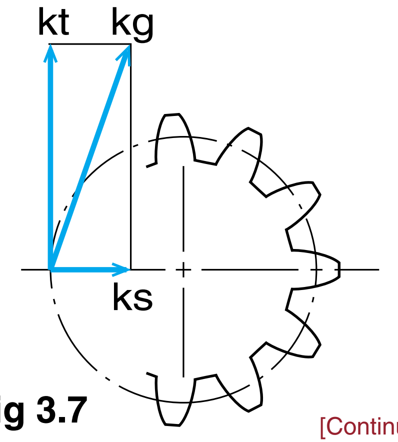

Fig 3.7

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Kg, the total theoretical gear load, must be multiplied by both the gear precision factor and the machine factor (the latter of which takes into account impact and other forces dependent on machinery type).

$$K = fz \cdot fs \cdot Kg \qquad \cdots\cdots (3.14)$$

where:
K : Gear load transmitted to shaft (N)
fz : Gear precision factor (Table 3.8)
fs : Machine factor (Table 3.6)

3.4.3 Load Distribution to Bearings

Load applied to a point on the shaft is distributed to the bearings supporting the shaft.
Reference Fig. 3.8,

$$Fr_{\mathrm{I}} = \frac{\ell + m}{\ell} K + \frac{x}{x+y} W \qquad \cdots\cdots (3.15)$$

$$Fr_{\mathrm{II}} = \frac{m}{\ell} K - \frac{y}{x+y} W \qquad \cdots\cdots (3.16)$$

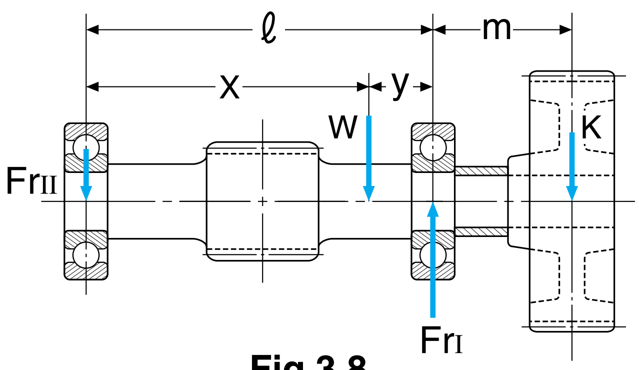

Fig 3.8

where:
Fr_{I} : Load working on bearing I (N)
Fr_{II} : Load working on bearing II (N)
K : Gear load transmitted to shaft (N)
W : Shaft Weight (N)
ℓ,m,x,y : Relative positions of the points of applied force

3.4.4 Averaging Fluctuating Loads

A large load will have an emphasized effect on bearing life even if it is applied only for a very short duration of the total life-span of the bearing.
When the size of bearing load fluctuates with a defined cycle, bearing life may be calculated by deriving an average load simulating the affects of the fluctuating load.

(1) Step Type Load Fluctuation

$$F_m = \sqrt[p]{\frac{F_1^p n_1 + F_2^p n_2 \cdots + F_n^p n_n}{n_1 + n_2 + \cdots\cdots + n_n}} \quad \cdots\cdots (3.17)$$

where:
F_m : Average of fluctuating load (N)
n_1 : Total number of revolutions at load F_1 (rev.)
n_2 : Total number of revolutions at load F_2 (rev.)
n_n : Total number of revolutions at load F_n (rev.)
p : 3 for ball; 10/3 for roller bearings

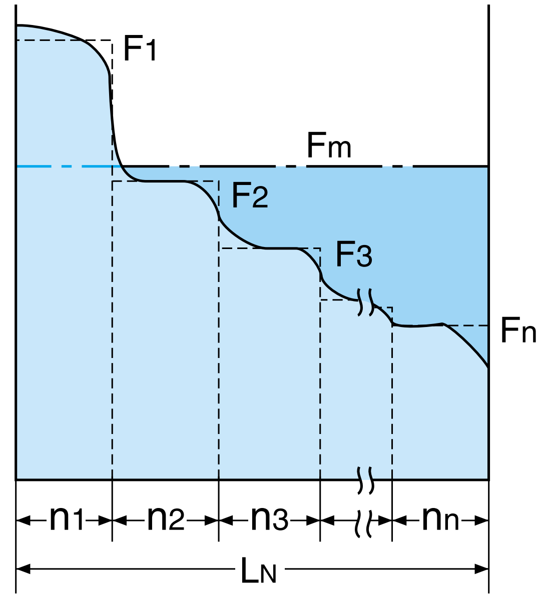

Fig 3.9

In formula (3.17), if rotating speed is constant, and (n_1 + n_2 + ... + n_n) is referenced as applied time, then n_1, n_2 and n_n , can be replaced by time periods t_1, t_2, ... t_n respectively, in the formula.

(2) Linear Load Fluctuation
When the load fluctuates almost linearly (see Fig. 3.10), the following formula is used to obtain the average load.

$$F_m \fallingdotseq \frac{1}{3} F_{min} + \frac{2}{3} F_{max} \quad \cdots\cdots (3.18)$$

where:
F_m : Average load (N)
F_{min} : Minimum load (N)
F_{max} : Maximum load (N)

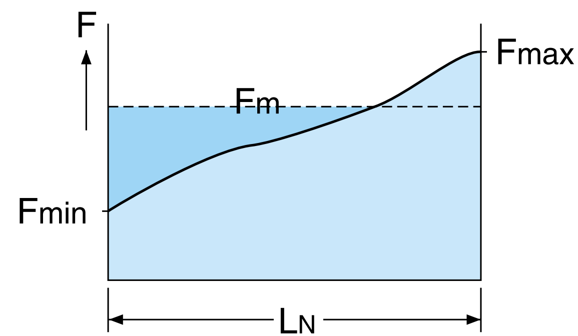

Fig 3.10

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(3) Dynamic plus static load fluctuation

Where load F_1 of a constant size and direction, is combined with a constantly revolving load F_2 caused by an unbalanced load on the bearing (see Fig. 3.11), the average load is calculated using formula 3.19.

$$Fm \fallingdotseq AF_1 + F_2 \quad \cdots\cdots\cdot (3.19)$$

Value of A is taken from Fig. 3.12.

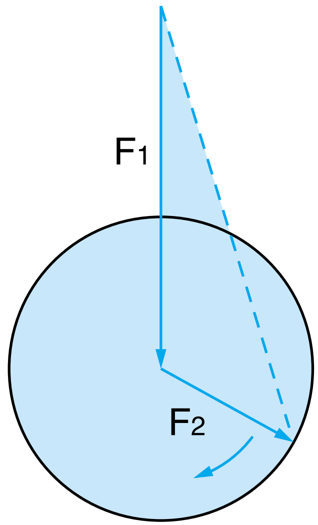

Fig 3.11

Calculation example: 5

A Single-row Deep-groove ball bearing is loaded with the fluctuating radial loads shown below.

Object: to obtain an average radial load on the bearing.

F_1=100N: 800 rpm for 6 sec

F_2= 50N: 1800 rpm for 20 sec

F_3=200N: 3600 rpm for 12 sec

Numbers of revolution for the individual loads F_1, F_2 and F_3 are derived for the formula as follows.

$$n_1 = \frac{6}{60} \times 800 = 80 \text{ rev.} \quad n_2 = \frac{20}{60} \times 1800 = 600 \text{ rev.} \quad n_3 = \frac{12}{60} \times 3600 = 720 \text{ rev.}$$

Therefore,

$$n = n_1 + n_2 + n_3 = 1400 \text{ rev.}$$

From formula (3.17),

$$Fm = \sqrt[3]{\frac{100^3 \times 80 + 50^3 \times 600 + 200^3 \times 720}{1400}}$$

$$= 162 \text{ N}$$

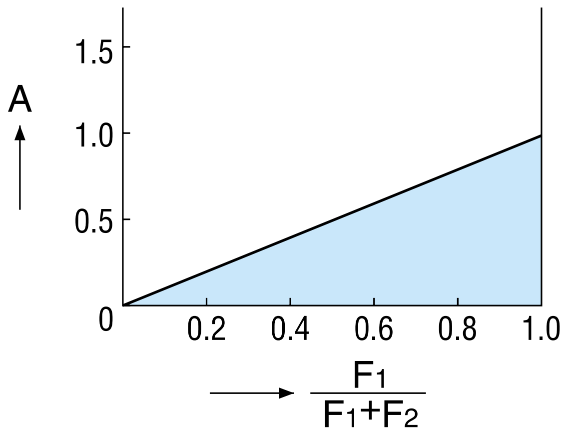

Fig 3.12

3.5 Dynamic Equivalent Load

Dynamic equivalent load refers to a load having constant direction and size such that theoretical calculations of bearing life using this load will simulate actual bearing life. This load is called dynamic equivalent radial load when calculated for radial bearings and dynamic equivalent axial load when calculated for thrust bearings.
In formula (3.1) expressing the relation between the bearing load and bearing life, bearing load, P, is either radial or axial load. Since radial and axial loads often occur simultaneously, the radial and axial loads must be converted to composite load within the dynamic equivalent load formula.

3.5.1 Dynamic Equivalent Radial Load

Dynamic equivalent radial load for radial bearings is calculated using the formula:

$$Pr=XFr+YFa \qquad \cdots\cdots (3.20)$$

where:
- Pr : Dynamic equivalent radial load (N)
- Fr : Radial load (N)
- Fa : Axial load (N)
- X : Radial load factor (from dimensional tables)
- Y : Axial load factor (from dimensional tables)

In the above formula, if the axial load to radial load ratio, Fa/Fr, is less than or equal to e (a value determined by the bearing size and load as shown in the dimension tables), X, Y, and Pr will be as follows:

X = 1
Y = 0
Pr = Fr

3.5.2 Dynamic Equivalent Axial Load

While most thrust bearings are incapable of supporting any radial load, Spherical roller thrust bearings will support some radial load. For Spherical roller thrust bearings, the dynamic equivalent axial load is derived using the formula:

$$Pa=Fa+1.2Fr \qquad \cdots\cdots (3.21)$$

where:
- Pa : Dynamic equivalent axial load (N)
- Fa : Axial load (N)
- Fr : Radial load (N)

Fr / Fa must be ≤ 0.55

3.5.3 Dynamic Equivalent Load for Oscillating Loads

The dynamic equivalent load of radial bearings sustaining oscillating movements is derived using the formula:

$$Pr = \left(\frac{\Psi}{90°}\right)^{1/p} (XFr + YFa) \quad \cdots\cdots (3.22)$$

where:

- Pr : Dynamic equivalent load (N)
- Ψ : Angle of oscillation (Ψ must be ≥ 90°/Z)
- p : 3 for ball, 10/3 for roller bearings
- Fr : Radial load (N)
- Fa : Axial load (N)
- X : Radial load factor (from dimensional tables)
- Y : Axial load factor (from dimensional tables)
- Z : Number of rolling elements in row

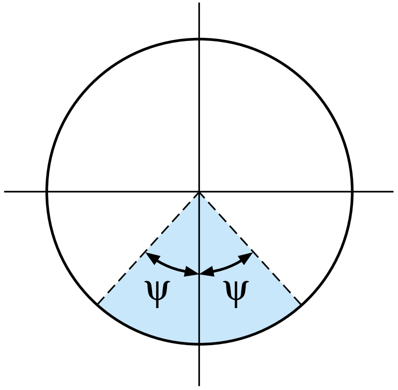

Fig 3.13

If the value of Ψ<90°/Z, the above formula may not accurately predict bearing life since localized wear may be generated in the raceways. (Oil lubrication may be tried to prevent the wear (false brinelling) associated with low-amplitude operation in this type application).

3.5.4 Angular Contact Ball; Tapered Roller Bearing Loads

For single-row Angular Contact ball and single-row Tapered roller bearings, the load center dimensions from the bearing tables must be used when determining the relative load positions. The load-center positions of these bearings are off-set from the midpoint of the width of these bearings as shown in Fig 3.14 and 3.15).

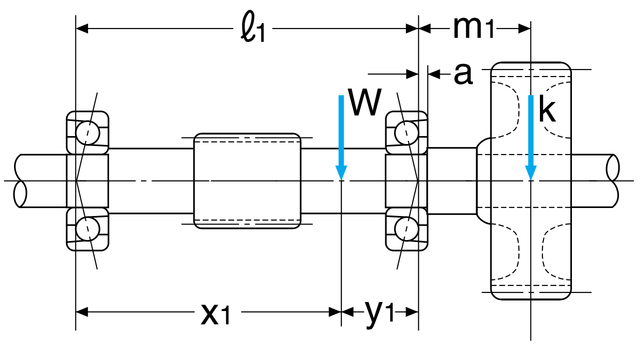

Fig 3.14

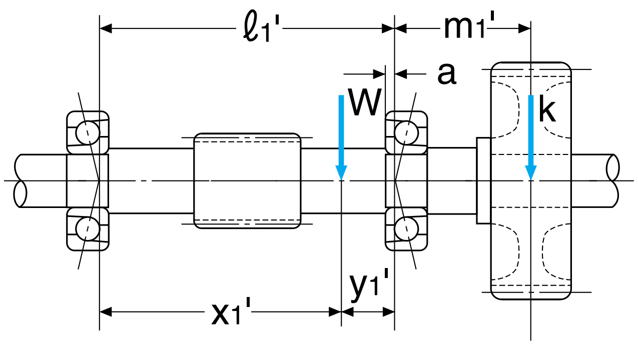

Fig 3.15

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The off-set dimension for Angular Contact ball and Tapered roller bearings is shown as the value "a" in the dimensional tables to indicate the load-center position. If moment loading is to be considered in a bearing system, location of load-center is of particular importance.
Where ℓ_1 , m_1 , x_1 or ℓ'_1 , m'_1 , x'_1 , and y'_1 are applied to formulas (3.15) and (3.16) as effective intervals instead of r, m, x, and y previously used in formulas (3.15) and (3.16). If the radial load is applied to two units of Tapered roller bearings used in pairs and induced axial load will be produced. The magnitude of this induced axial force Fa' is calculated using the formula:

$$Fa' = \frac{Fr}{2Y_1} \quad \cdots\cdots\cdots (3.23)$$

where:

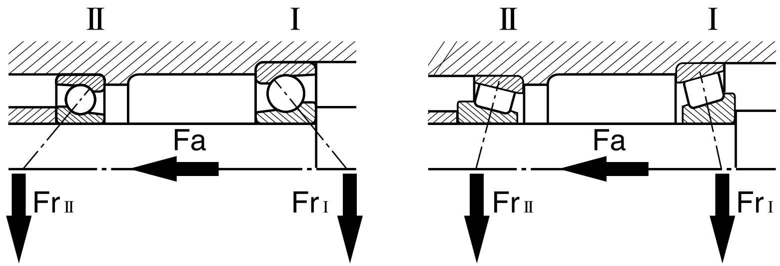
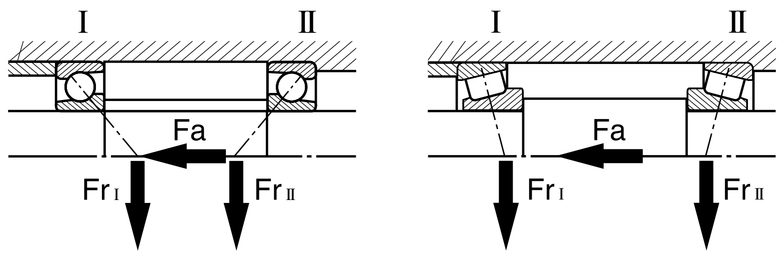
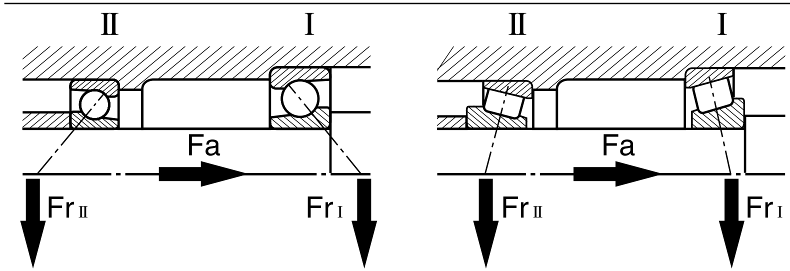
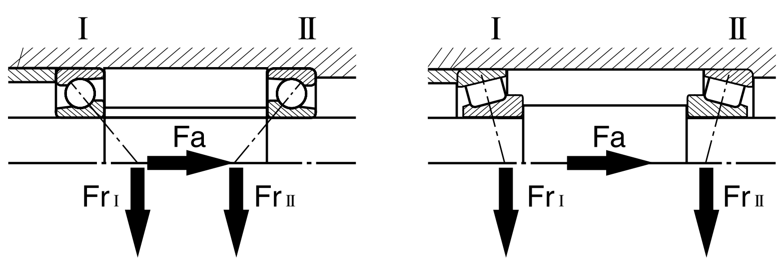
Y_1	: Axial load factor (from dimension tables)
Fa'	: External axial load (N)
Fr	: Radial load (N)

Axial and equivalent radial load on bearing calculated using formulas in Tables 3.9.

Table 3.9 Axial and Equivalent Load of Angular Contact Ball and Tapered Roller Bearings

Fr1,Fr2 : Radial load applied to bearings1and2(N)
Fa : External axial load (N) direction shown by Table 3.9
Fa1,Fa2 : Axial load on bearings1and2(N)
Pr1,Pr2 : Dynamic equivalent radial load on bearings1and2(N)
X1,X2 : Radial Load Factor for bearings1and2from dimension tables
Y1,Y2 : Axial Load Factor for bearings1and2from dimension tables
(Use Y1 for Tapered roller bearings)

Table 3.9 Axial and Equivalent Load of Angular Contact Ball and Tapered Roller Bearings

Bearing arrangement	Load conditions	Axial load	Dynamic equivalent radial load
	$Fa \geq 0.5\left(\frac{Fr_{I}}{Y_{I}} - \frac{Fr_{II}}{Y_{II}}\right)$	$Fa_{I}=Fa_{II}+Fa$ $Fa_{II}=0.5\,\frac{Fr_{II}}{Y_{II}}$	$Pr_{I}=X_{I}Fr_{I}+Y_{I}(Fa_{II}+Fa)$ $Pr_{II}=Fr_{II}$
	$Fa < 0.5\left(\frac{Fr_{I}}{Y_{I}} - \frac{Fr_{II}}{Y_{II}}\right)$	$Fa_{I}=0.5\,\frac{Fr_{I}}{Y_{I}}$ $Fa_{II}=Fa_{I}-Fa$	$Pr_{I}=Fr_{I}$ $Pr_{II}=X_{II}Fr_{II}+Y_{II}(Fa_{I}-Fa)$
	$Fa \geq 0.5\left(\frac{Fr_{II}}{Y_{II}} - \frac{Fr_{I}}{Y_{I}}\right)$	$Fa_{I}=0.5\,\frac{Fr_{I}}{Y_{I}}$ $Fa_{II}=Fa_{I}+Fa$	$Pr_{I}=Fr_{I}$ $Pr_{II}=X_{II}Fr_{II}+Y_{II}(Fa_{I}+Fa)$
	$Fa < 0.5\left(\frac{Fr_{II}}{Y_{II}} - \frac{Fr_{I}}{Y_{I}}\right)$	$Fa_{I}=Fa_{II}-Fa$ $Fa_{II}=0.5\,\frac{Fr_{II}}{Y_{II}}$	$Pr_{I}=X_{I}Fr_{I}+Y_{I}(Fa_{II}-Fa)$ $Pr_{II}=Fr_{II}$

Notes : 1. Equalities apply when the bearing clearance and preload are 0.
2. Radial load applied in reverse direction to the arrows above will be also treated as positive values.

3.6 Basic Static Load Rating and Static Equivalent Load

3.6.1 Basic Static Load Rating

Load applied to stationary bearings can create permanent indentations in the load surfaces. While some level of deformation can be tolerated, a level of deformation will be reached where noise and vibration during operation of the bearing, will make the bearing unusable.
The term "Basic static load rating" refers to the maximum contact stress value of the static load where the rolling element and raceways contact. The ratings are:

- Self-aligning ball bearing • • • 4600MPa
- Other ball bearings • • • 4200MPa
- Roller bearings • • • 4000 MPa

With these contact stresses, the sum of deformations (ball/roller and raceway) is approximately 1/10000 of the diameter of the rolling element.
Basic static load ratings are shown in the dimension tables for each bearing number. The symbol Cor is for radial bearings and the symbol Coa is for thrust bearings.

3.6.2 Static Equivalent Load

Static equivalent load is the static load that reflects the actual load conditions to the contact section of the rolling elements and raceway receiving the maximum stress. For radial bearings, radial load of a constant direction and size is called the static equivalent radial load, and for thrust bearings, axial load of a constant direction and size is called the static equivalent axial load.

1) Static equivalent radial load
To calculate the static equivalent radial load of a radial bearing supporting simultaneous radial and axial loads, the larger of the values obtained from formulas (3.24) and (3.25) are to be used

$$Por=XoFr+YoFa \quad \cdots\cdots \ (3.24)$$

$$Por=Fr \quad \cdots\cdots \ (3.25)$$

where:
Por : Static equivalent radial load (N)
Fr : Radial load (N)
Fa : Axial load (N)
Xo & Yo: Static radial and axial load factors from dimension tables

[Continue→]

[→Continue]

2) Static equivalent axial load
Static equivalent axial load for Spherical Thrust bearings is calculated using formula (3.26)

$$Poa=Fa+2.7Fr \quad \cdots\cdots \quad (3.26)$$

where:
Poa : Static equivalent axial load (N)
Fa : Axial load (N)
Fr : Radial load (N)
Fr/Fa must be ≤ 0.55

3.6.3 Safety Factor

The basic static load rating is considered as the limiting load for general applications. In terms of a safety factor, this means that, by definition, a safety factor, So, is set as a base of 1. An application may require a larger or allow a smaller safety factor. Table 3.10 provides a guide for selection of the safety factor, So, to be used with formula (3.27) for calculation of the maximum (weighted) static equivalent load.

$$Co=So \cdot Pomax \quad \cdots\cdots \quad (3.27)$$

where:
Co: Basic static load rating (N)
(Cor for radial; Coa for thrust bearings)
So: Safety factor
(select from Table 3.10)
Pomax: Static equivalent load (N)

Table 3.10 Static Safety Factor (So)

Application condition	So	
	Ball Bearings	Roller Bearings
High rotating accuracy is needed	2	3
Vibration and/or impact present	1.5	2
Normal operating conditions	1	1.5
Small amount of permanent deformation is tolerable	0.7	1

Note : So > 4 for spherical roller thrust bearings

3.7 Axial Load Capacity of Cylindrical Roller Bearings

Cylindrical roller bearings are generally used for supporting radial loads only. Bearings having flanges or loose ribs on both the inner and outer rings (such as on configurations NJ, NF, and NUP), however, are capable of supporting some amount of axial load. Since any axial loading on a cylindrical roller bearing is supported by a "sliding" action between the roller ends and flanges, allowable axial load is based on the limiting values of heat, seizure, and wear caused by this "sliding" contact.

Permissible axial loading (no consideration of bearing life as a radial bearing) on Cylindrical roller bearings is calculated using the following formula.

$$F_a = (pv)\frac{\lambda}{n} \quad \cdots\cdots \text{Allowable axial load (N)}$$

pv : Application factor from Table 3.11.1
λ : Bearing type factor from Table 3.11.2
n : Rotating speed (rpm)

However, there is another limits shown by the following formula because Fa exceeding the limits cause abnormal roller movement

Allowable axial load $\leq$ K1 · Fr

Bearing series	K_1
1000, 200, 200E 300, 300E, 400	0.2
2200, 2200E, 2300, 2300E	0.4

When cylindrical roller bearings are applied axial load, additional considerations are required as follows;

- Apply sufficient radial load to overcome axial load
- Supply sufficient lubricant between roller ends and flanges
- Use lubricant which has good film strength (pressure resistant) properties
- Practice good bearing mounting accuracy (see section 8.3)
- Allow sufficient running-in
- Minimize radial bearing clearance

Table 3.11.1 Application Factor (pv)

Operating conditions (Load and lubrication)	(pv)
Intermittent axial load, Good heat conduction and Good cooling or Very large amount of lubricant	5400 ~ 6900
Intermittent axial load, Good heat conduction and Large amount of lubricant	2600 ~ 3200
Oil lubrication, Good heat conduction or Good cooling	1900 ~ 2200
Continuous axial load and Oil lubrication or Intermittent axial load and Grease lubrication	1300 ~ 1600
Continuous axial load and Grease lubrication	690 ~ 780

Table 3.11.2 Bearing Type Factor λ

Diameter Series	λ
0	19d
2	32d
3	45d
4	60d

d=Bearing bore (mm)