3. Load rating and life

3.1 Bearing life

Even in bearings operating under normal conditions, the surfaces of the raceway and rolling elements are constantly being subjected to repeated compressive stresses which causes **spalling** (flaking) of these surfaces to occur. This **spalling** is due to material fatigue and will eventually cause bearings to fail.

The effective life of a bearing is usually defined in terms of **the total number of revolutions** a bearing can undergo before **spalling** of either the raceway surface or the rolling element surfaces occurs.

Other causes of bearing failure are often attributed to problems such as seizing, abrasions, cracking, chipping, scuffing, rust, etc. However, these so called "causes" of bearing failure are usually themselves caused by improper installation, insufficient or improper lubrication, faulty sealing or improper bearing selection.

Since the above mentioned "causes" of bearing failure can be avoided by taking proper precautions, and are not simply caused by material fatigue, they are considered separately from the **spalling** aspect.

3.2 Basic rating life and basic dynamic load rating

A group of seemingly identical bearings when subjected to identical loads and operating conditions will exhibit a wide diversity in their durability. This "life" disparity can be accounted for by the difference in the fatigue of the bearing material itself.

This disparity is considered statistically when calculating bearing life, and **the basic rating life** is defined as follows.

The basic rating life is based on a 90 % statistical model which is expressed as the total number of revolutions 90 % of the bearings in an identical group of bearings subjected to identical operating conditions will attain or surpass before spalling due to material fatigue. For bearings operating at fixed constant speeds, the basic rating life (90 % reliability) is expressed in the total number of hours of operation. Basic dynamic load rating expresses a rolling bearing's capacity to support a dynamic load.

The basic dynamic load rating is **the load which a bearing can theoretically endure for a basic rating life of one million revolutions**. This is expressed as pure radial load for radial bearings and pure axial load for thrust bearings. These are referred to as "**basic dynamic radial load rating (C_r)**" and "**basic dynamic axial load rating (C_a)**".

The basic dynamic load ratings given in the bearing tables of this catalog are for bearings constructed of **NTN** high quality bearing materials and of good manufacturing quality.

The relationship between the basic rating life, the basic dynamic load rating and the dynamic equivalent load is shown in formula (3.1) and formula (3.2).

For ball bearings : $L_{10} = \left(\frac{C}{P}\right)^3$ ········· (3.1)

For roller bearings: $L_{10} = \left(\frac{C}{P}\right)^{10/3}$ ······ (3.2)

Where:

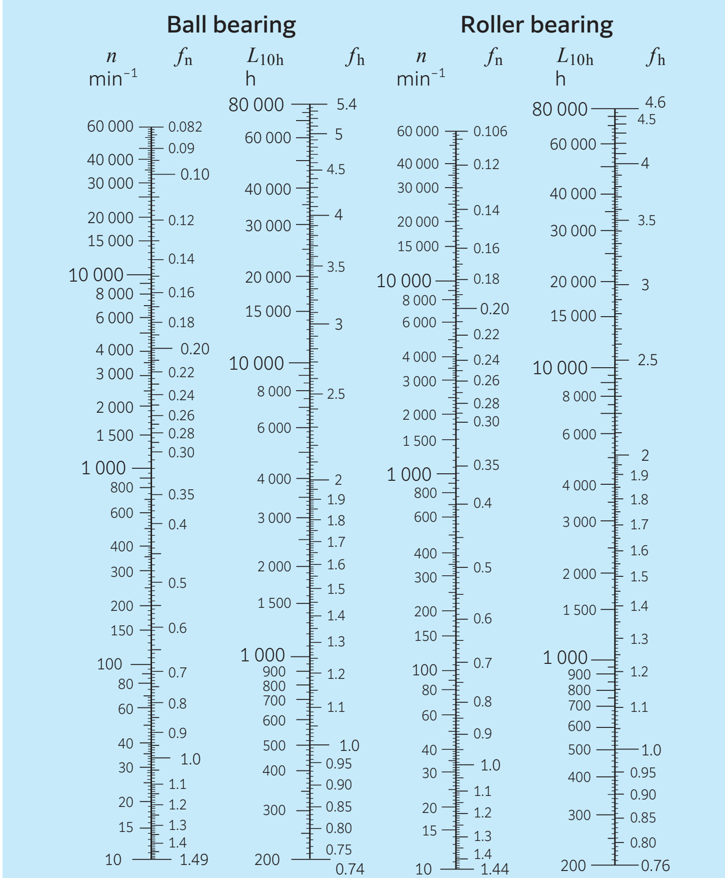
L_{10} : Basic rating life 10^6 revolutions
C : Basic dynamic load rating, N
Radial bearing C_r
Thrust bearing C_a
P : Dynamic equivalent load, N[1)]
Radial bearing P_r
Thrust bearing P_a

Note: 1) For more details, please refer to the section "4. Bearing load calculation".

The relationship between rotational speed n and speed factor f_n as well as the relationship between life factor f_h and basic rating life L_{10h} are shown in **Table 3.1** and **Fig. 3.1**.

Table 3.1 Bearing basic rating life, life factor, and speed factor

Division	Ball bearing	Roller bearing
Basic rating life L_{10h} h	$\frac{10^6}{60n}\left(\frac{C}{P}\right)^3 = 500 f_h^3$	$\frac{10^6}{60n}\left(\frac{C}{P}\right)^{10/3} = 500 f_h^{10/3}$
Life factor f_h	$f_n \frac{C}{P}$	$f_n \frac{C}{P}$
Speed factor f_n	$\left(\frac{33.3}{n}\right)^{1/3}$	$\left(\frac{33.3}{n}\right)^{3/10}$



Note n : Rotational speed, min^{-1}

Fig. 3.1 Bearing life rating scale

When several bearings are incorporated in machines or equipment as complete units, all the bearings in the unit are considered as a whole when computing bearing system life (see formula 3.3).

$$L = \frac{1}{\left(\frac{1}{L_1^{e}}+\frac{1}{L_2^{e}}+\cdots\frac{1}{L_n^{e}}\right)^{1/e}} \quad \cdots\cdots\cdots\cdots \quad (3.3)$$

Where:

L : Total basic rating life of entire unit, h

$L_1, L_2 \cdots L_n$: Basic rating life of individual bearings 1, 2, ⋯ n, h

e : $e = 10/9$ ⋯⋯⋯ For ball bearings

$e = 9/8$ ⋯⋯⋯⋯ For roller bearings

When the load conditions vary at regular intervals, the life can be given by formula (3.4).

$$L_m = \left(\frac{\phi_1}{L_1}+\frac{\phi_2}{L_2}+\cdots\cdots\frac{\phi_j}{L_j}\right)^{-1} \quad \cdots\cdots\cdots \quad (3.4)$$

Where:

L_m : Total life of bearing, h

ϕ_j : Frequency of individual load conditions ($\Sigma\phi_j = 1$)

L_j : Life under individual conditions, h

If dynamic equivalent load P and rotational speed n are operating conditions of the bearing, basic rated dynamic load C that satisfies required life of the bearing is determined using **Table 3.1** and formula (3.5). Bearings that satisfy the required C can be selected from the bearing dimensions table provided in the catalog.

$$C = P\frac{f_h}{f_n} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \quad (3.5)$$

3.3 Adjusted rating life

The basic bearing rating life can be calculated through the formulas mentioned earlier in Section 3.2. However, in some applications bearing reliability higher than 90 % may be required. In addition, bearing life may be enhanced by the use of specialty bearing materials or manufacturing processes. Bearing life is also sometimes affected by operating conditions such as lubrication, temperature and rotational speed.

Basic rating life adjusted to compensate for reliability, special bearing materials and enhancements, and specific operation conditions is called "**adjusted rating life**", and is determined using formula (3.6).

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot L_{10} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots \quad (3.6)$$

Where:

L_{na} : Adjusted rating life in millions of revolutions (10^6)

a_1 : Life adjustment factor for reliability

a_2 : Life adjustment factor for special bearing properties

a_3 : Life adjustment factor for operating conditions

3.3.1 Life adjustment factor for reliability a_1

The value of **life adjustment factor for reliability a_1** is provided in **Table 3.2** for reliability of 90 % or greater.

3.3.2 Life adjustment factor for special bearing properties a_2

Bearing characteristics concerning life vary according to bearing material, quality of material and if using a special manufacturing process. In this case, life is adjusted using **life adjustment factor for special bearing properties a_2**.

The basic dynamic load ratings listed in the catalog are based on **NTN**'s standard material and the adjustment factor used is $a_2 = 1$. However, an adjustment factor of a_2 other than 1 may be used for bearings with specially enhanced materials and manufacturing methods.

[NOTE: $a_2 < 1$ may occur for temperature stabilization]

$a_2 > 1$ may be used for bearings with specially improved materials and manufacturing methods.

Bearings made of high carbon chrome bearing steel, conventionally heat treated, may experience dimensional changes during operation if used at high temperatures for extended periods of time. Temperature stabilization treatment (**TS treatment**) can be used to provide increased dimensional stability of bearing materials at high operational temperatures. However, the dimensional stabilization treatment results in a lower overall hardness of heat treated bearing materials; therefore, the life is adjusted by multiplying by life adjustment factor for special bearing properties a_2 given in **Table 3.3**.

For further clarification please consult with **NTN** Engineering.

Table 3.2 Life adjustment factor for reliability a_1

Reliability %	L_n	Life adjustment factor for reliability a_1
90	L_{10}	1.00
95	L_5	0.64
96	L_4	0.55
97	L_3	0.47
98	L_2	0.37
99	L_1	0.25
99.2	$L_{0.8}$	0.22
99.4	$L_{0.6}$	0.19
99.6	$L_{0.4}$	0.16
99.8	$L_{0.2}$	0.12
99.9	$L_{0.1}$	0.093
99.92	$L_{0.08}$	0.087
99.94	$L_{0.06}$	0.080
99.95	$L_{0.05}$	0.077

Table 3.3 Treatment for dimensional stabilization

Code	Max. operating temperature °C	Life adjustment factor for special bearing properties a_2
TS2	160	1.00
TS3	200	0.73
TS4	250	0.48

Please consult **NTN** Engineering for life adjustment factor for special bearing properties (a_2) when using dimensional stabilization treatment combined with any specialty bearing material.

3.3.3 Life adjustment factor for operating conditions a_3

Life adjustment factor for operating conditions a_3 is used to compensate for when lubrication condition worsens due to a rise in temperature or rotational speed, lubricant deteriorates or it becomes contaminated with foreign matter.

Generally speaking, when lubricating conditions are satisfactory, the a_3 factor has a value of 1.0; and when lubricating conditions are exceptionally favorable, and all other operating conditions are normal, a_3 can have a value greater than 1.0. The factor a_3 may be less than 1.0 due to the following cases:

- Dynamic viscosity of lubrication is too low for bearing operating temperature (13 mm²/s or less for ball bearings, 20 mm²/s or less for roller bearings as a standard)
- Rotational speed is particularly low (when the product of pitch diameter D_{pw} mm and rotational speed n min^{-1} is $D_{pw} \cdot n < 10\,000$)
- Lubricant contaminated with foreign matter or moisture

If using a special operating condition, consult with **NTN** Engineering.

The operating life may be also shortened by misalignment and operating clearance but these operating conditions are not accounted for by the a_3 factor [See sections "3.7 Misalignment angle (installation error) and life" and "3.8 Clearance and life"].

Even if $a_2 > 1$ is used for specialty bearings made of enhanced materials or produced by special manufacturing methods, $a_2 \times a_3 < 1$ is used if lubricating conditions are not favorable.

When an excessively heavy load is applied, harmful plastic distortion may result at the contact surfaces between the rolling elements and raceways. The formulas for determining basic rating life (3.1, 3.2, and 3.6) do not apply if P_r exceeds either C_{0r} or $0.5C_r$ for radial bearings, or if P_a exceeds $0.5C_a$ for thrust bearings.

3.4 Modified rating life

3.4.1 Background

Adjusted rating life L_{na} of bearings is as shown in formula (3.6). System conditions corresponding to a_2 and a_3 are considered independently in that approach. However, it is desirable to consider the integrated system as a whole, resulting in adoption of ISO 281:2007. This approach considers **life modification factor a_{ISO}**, which provides a more practical method to consider the influence of lubrication, contamination and fatigue load on bearing life. Based on these decisions in ISO 281, JIS B 1518 was similarly revised in 2013.

Modified rating life L_{nm} using life modification factor a_{ISO} can be obtained by formula (3.7).

$$L_{nm} = a_1 \cdot a_{ISO} \cdot L_{10} \quad \cdots\cdots\cdots\cdots (3.7)$$

3.4.2 Life modification factor a_{ISO}

The life modification factor, a_{ISO}, is a function of lubrication, contamination, material characteristics, and load as shown in formula (3.8).

$$a_{ISO} = f\left(\frac{e_C\,C_u}{P}, \kappa\right) \quad \cdots\cdots\cdots\cdots (3.8)$$

Where:

C_u: Fatigue load limit
The fatigue load limit is a load applied on bearings that results in the fatigue limit stress at the maximum loaded contact within the raceway. This depends on the bearing type, internal specifications, quality, and material strength. In ISO 281:2007, 1.5 GPa is recommended as contact stress corresponding to C_u for the bearings made of commonly used high quality material and good manufacturing quality. The fatigue load limit values with respect to the **NTN** bearing numbers are specified in each specification table.

e_C : Contamination factor
The presence of hard particle contaminants in the lubricant (oil) have the potential to form indentations on the raceway surface, resulting in surface initiated damage and in reduction in bearing life. Contamination factor e_C considers this and depends on the level of contamination, bearing size, and lubricant viscosity (oil film thickness). As shown in **Table 3.4**, approximate values are determined by the bearing size (may be substituted by rolling element pitch diameter D_{pw}, average bearing diameter $(d + D)/2$), filtration and seal structures (including presence of pre-washing).

κ : Viscosity ratio
Bearings are used on the assumption that the rolling contact surface is separated by the lubricant. However, when the viscosity of the lubricant is low, separation becomes insufficient and metal to metal contact occurs, causing surface initiated damage. Viscosity ratio κ considers this effect and is represented by formula (3.9) by the ratio of dynamic viscosity ν in use with respect to reference dynamic viscosity ν_1 of the lubricant.

$$\kappa = \nu / \nu_1 \quad \cdots\cdots\cdots\cdots (3.9)$$

Reference dynamic viscosity ν_1 depends on rotational speed n and size (D_{pw}), and can be obtained by **Fig. 3.2** or formula (3.10) and formula (3.11).

Table 3.4 Value of contamination factor e_C

Level of contamination	e_C D_{pw} < 100 mm	e_C D_{pw} ≥ 100 mm
Extreme cleanliness Particle size of the order of lubricant film thickness; laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter; conditions typical of bearing greased for life and sealed	0.8–0.6	0.9–0.8
Normal cleanliness Oil filtered through fine filter; conditions typical of bearings greased for life and shielded	0.6–0.5	0.8–0.6
Slight contamination Slight contamination in lubricant	0.5–0.3	0.6–0.4
Typical contamination Conditions typical of bearings without integral seals; course filtering; wear particles and ingress from surroundings	0.3–0.1	0.4–0.2
Severe contamination Bearing environment heavily contaminated and bearing arrangement with inadequate sealing	0.1–0	0.1–0
Very severe contamination	0	0

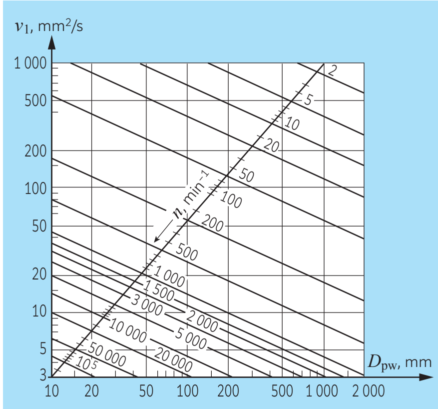


Fig. 3.2 Diagram for reference dynamic viscosity ν_1

In the case of n < 1 000 min^{-1},

$$\nu_1 = 45\,000\, n^{-0.83}\, D_{pw}^{-0.5} \quad \cdots\cdots\cdots (3.10)$$

In the case of n ≥ 1 000 min^{-1},

$$\nu_1 = 4\,500\, n^{-0.5}\, D_{pw}^{-0.5} \quad \cdots\cdots\cdots (3.11)$$

Fig. 3.3 shows the relationship among C_u/P, e_C, κ and, a_{ISO} of radial ball bearings. Using the figure has the following restrictions:

1) For practical use, the life modification factor shall be limited a_{ISO} ≤ 50.
2) In the case of κ > 4, κ = 4 shall be assumed. The same approach does not apply in the case of κ < 0.1.

Diagrams for radial roller bearings, thrust ball bearings, and thrust roller bearings have also been presented (see **Fig. 3.4** through **Fig. 3.6**). The diagrams can be applied regardless of lubrication types; however, for grease lubrication, special additives, and special rotating behaviors, consult with **NTN** Engineering.

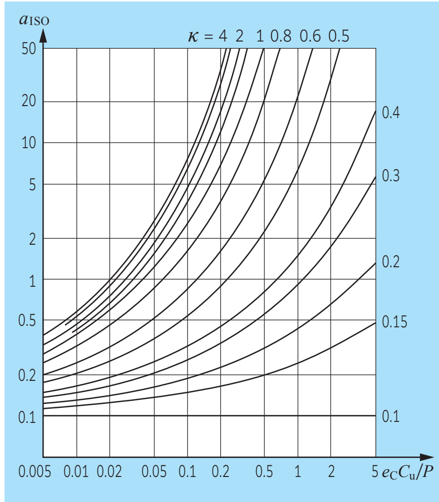


Fig. 3.3 Life modification factor a_{ISO} (radial ball bearing)

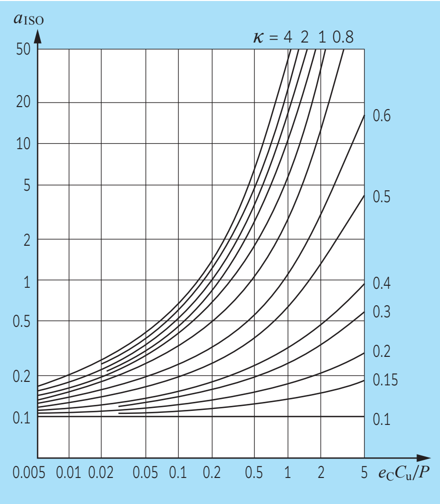


Fig. 3.4 Life modification factor a_{ISO} (radial roller bearing)

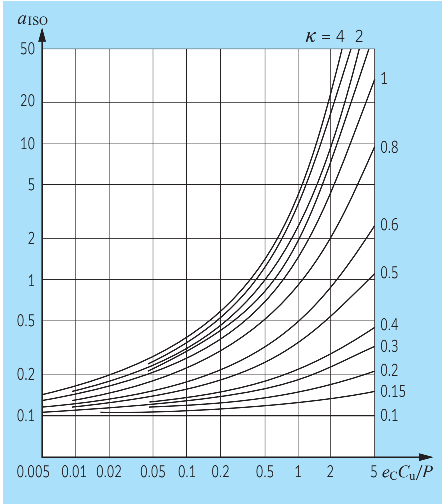


Fig. 3.6 Life modification factor a_{ISO} (thrust roller bearing)

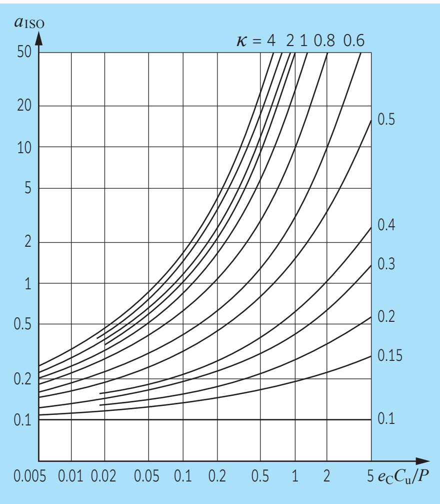


Fig. 3.5 Life modification factor a_{ISO} (thrust ball bearing)

3.4.3 Applicable bearings of modified rating life

Fatigue load limit C_u used for the calculation of life modification factor a_{ISO} depends on the bearing materials. **NTN** bearings that have undergone standard through hardening (immersion quenching) and is made of bearing steel, the fatigue load limit value with respect to each bearing number is specified in each dimension table, and a_{ISO} can be applied.

3.5 Machine applications and requisite life

When selecting a bearing, it is essential that the requisite life of the bearing be established in relation to the operating conditions. The requisite life of the bearing is usually determined by the type of machine in which the bearing will be used, and duration of service and operational reliability requirements. A general guide to these requisite life criteria is shown in **Table 3.5**.

When determining bearing size, the fatigue life of the bearing is an important factor; however, besides bearing life, the strength and rigidity of the shaft and housing must also be taken into consideration.

Table 3.5 Machine application and requisite life (reference)

Service classification	Machine application and requisite life L_{10h} ×10^3 hours				
	Up to 4	4 to 12	12 to 30	30 to 60	60 or more
Machines used for short periods or used only occasionally	Household appliances Electric hand tools	Farm machinery Office equipment			
Short period or intermittent use, but with high reliability requirements	Medical appliances Measuring instruments	Home air-conditioning motor Construction equipment Elevators Cranes	Crane (sheaves)		
Machines not in constant use, but used for long periods	Automobiles Two-wheeled vehicles	Small motors Buses/trucks General gear drives Woodworking machines	Machine spindles Industrial motors Crushers Vibrating screens	Main gear drives Rubber/plastic Calender rolls Printing machines	
Machines in constant use over 8 hours a day		Roll neck of steel mill Escalators Conveyors Centrifuges	Railway vehicle axles Air conditioners Large motors Compressor pumps	Locomotive axles Traction motors Mine hoists Pressed flywheels	Papermaking machines Propulsion equipment for marine vessels
24 hour continuous operation, non-interruptible					Water supply equipment Mine drain pumps/ventilators Power generating equipment

3.6 Weibull distribution and life adjustment factor for reliability

As described in “3.2 Basic rating life and basic dynamic load rating”, a group of seemingly identical bearings when subjected to an identical load and operating conditions may exhibit a wide variation in their durability. In general, this variation is known to follow the “Weibull distribution”, and the basic theory is constructed on the premise that the bearing operating life also follows the Weibull distribution while adhering to formula (3.1) and formula (3.2) for life calculation and the calculation formula for basic dynamic load rating C.

As an index representing the variation of the Weibull distribution, there is a coefficient called a Weibull slope. A value 10/9 for ball bearings and 9/8 for roller bearings are given in the basic life calculation theory of ISO and JIS. According to this, for example, for a deep groove ball bearing, a difference of 5 times or more is generated between the L_{10} life of 90 % reliability and the L_{50} life of 50 % reliability.

In some applications where a bearing is used, a life study with reliability exceeding 90 % may be required, and in such a case, a life adjustment factor for reliability a_1 is used. In the latest ISO (ISO 281:2007) and JIS (JIS B 1518:2013), a_1 values were updated based on measured test data (see **Fig. 3.7**). **Table 3.2** shows the latest a_1 values after review.

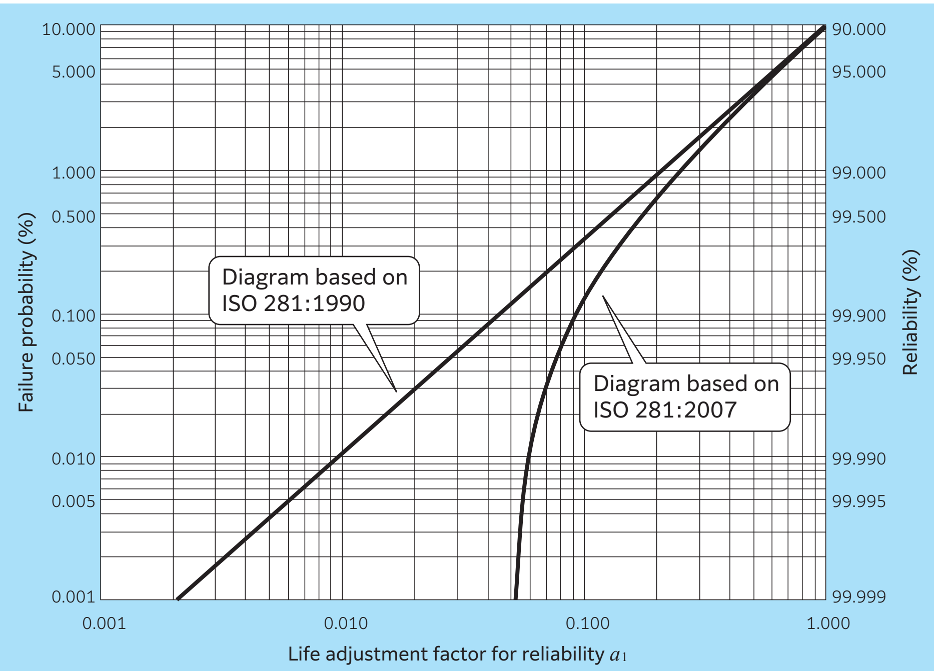


Fig. 3.7 Life adjustment factor for reliability a_1

3.7 Misalignment angle (installation error) and life

A lack of accuracy and/or rigidity of the shaft or housing can cause misalignment between the bearing inner and outer rings similar to an externally applied moment load.

The bearing operating life calculation in the case of receiving a moment load cannot be obtained by the commonly used $L = (C_r / P_r)^p$, which is generally used, and it is necessary to obtain it considering the internal design, clearance, etc. of each bearing.

Since the life decrease rate differs depending on the internal clearance, the load condition, and the internal design, it is necessary to calculate the ratio under individual conditions, and the rate cannot be given as a factor in general.

Fig. 3.8 and **Fig. 3.9** show the results of detailed calculation of the relationship between the misalignment angle (installation error) and the life of a deep groove ball bearing and a cylindrical roller bearing.

See **Table 14.6** in section “14. Shaft and housing design” for the rough standard of allowable misalignment and allowable misalignment of each bearing type.

For further clarification please consult with **NTN** Engineering.

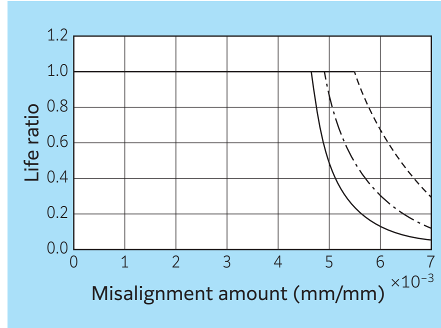


Fig. 3.8 Misalignment angle and life ratio of deep groove ball bearing

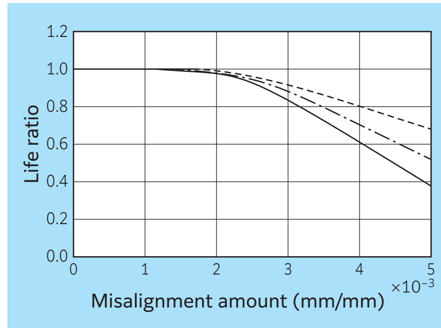


Fig. 3.9 Misalignment angle and life ratio of cylindrical roller bearing

————————	Light load
—·—·—·—·—	Normal load
- - - - - - - - -	Heavy load

3.8 Clearance and life

It is very difficult to accurately determine what the clearance of a rolling bearing should be in a normal operating state.

When a bearing is subjected to a simple load and full rotation slight clearance is preferable. However, too large of a clearance can cause life deterioration and vibration. In contrast, a negative clearance (preload) can extend the operating life and prevent shaft runout. However, too large of a preload increases friction, temperature rise, lubrication degradation and can cause seizures in extreme cases.

As a general guideline a target of zero operating clearance should be acceptable.

1) Clearance and rolling element load W

(1) In the case of bearing clearance larger than 0 [see **Fig. 3.11**], load distribution $\varepsilon < 0.5$ holds. The maximum rolling element load becomes larger than when the bearing clearance is zero [see **Fig. 3.10**].

[Load factor ε and conceptual diagram]

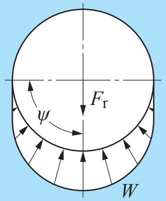


Fig.3.10
$\varepsilon = 0.5$ $\psi = \pm 90°$
Radial clearance 0

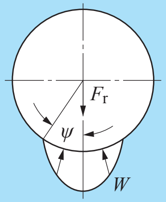


Fig.3.11
$0 < \varepsilon < 0.5$ $0 < \psi < 90°$
There is radial clearance

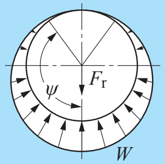


Fig.3.12
$0.5 < \varepsilon < 1$
$90° < \psi < 180°$
Radial preload state, or large axial load

(2) **Fig. 3.13** shows an ideal graph in which operating in a slightly preloaded condition results in maximum bearing life.

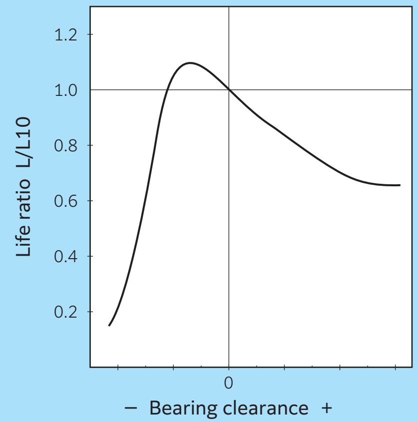


Fig. 3.13 Bearing clearance and life ratio

3.9 Basic static load rating

It has been found through experience that a permanent deformation of 0.0001 times the diameter of the rolling element, occurring at the most heavily stressed contact point between the raceway and the rolling elements, can be tolerated without any subsequent impariment of bearing operation.

Testing indicates the above level of permanent deformation corresponds to a calculated contact stress as shown below. The basic static load rating is defined as the static applied load which results in such a contact stress at the center of the contact patch between the raceway and the rolling element receiving the maximum load.

Roller bearings: 4 000 MPa
Ball bearings
(excluding self-aligning ball bearings): 4 200 MPa
Self-aligning ball bearings: 4 600 MPa

Referred to as "**basic static radial load rating**" for radial bearings and "**basic static axial load rating**" for thrust bearings, the basic static load rating is expressed as C_{0r} or C_{0a} respectively and is provided in the bearing dimensions table.

3.10 Allowable static equivalent load

Generally the static equivalent load which can be permitted (refer to page A-41) is limited by the basic static load rating as stated in Section 3.9. However, depending on application requirements regarding friction and smooth operation, these limits may be greater or lesser than the basic static load rating.

This is generally determined by taking the safety factor S_0 given in formula (3.12) and guidelines of **Table 3.6** into account.

$$S_0 = C_0 / P_0 \quad \cdots\cdots (3.12)$$

Where:

S_0 : Safety factor
C_0: Basic static load rating, N
Radial bearing: C_{0r}
Thrust bearing: C_{0a}
P_0: Static equivalent load, N
Radial bearing: P_{0r}
Thrust bearing: P_{0a}

Table 3.6 Minimum safety factor values S_0

Operating conditions	Ball bearing	Roller bearing
Applications that require quiet rotation	2	3
Applications subjected to impact loads	1.5	3
Normal rotation applications	1	1.5

Note: 1. For thrust spherical roller bearings, min. S_0 value = 4.
2. For drawn cup needle roller bearings, min. S_0 value = 3. However, for HK-F type[1)] min. S_0 value = 2.
3. When vibration and/or shock loads are present, a load factor based on the shock load needs to be included in the P_0 max value.
4. If a large axial load is applied to deep groove ball bearings or angular ball bearings, the contact ellipse may exceed the raceway surface. For more information, please contact **NTN** Engineering.
5. When an AS type raceway washer is used in a thrust bearing, min. S_0 value = 3.

1) For details, see the special catalog "**HK-F Type Drawn Cup Needle Roller Bearings (CAT.No.3029/JE)**".

3.11 Allowable axial load

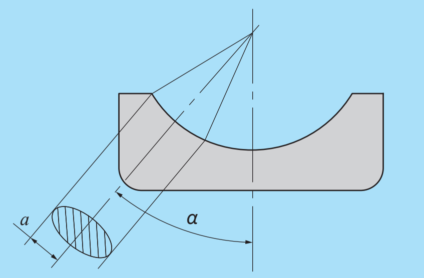
Radial bearings can also receive axial loads, but load is limited depending on the bearing type.

(1) Ball bearing

When an axial load acts on ball bearings, such as deep groove ball bearings and angular contact ball bearings, the contact angle changes with the load. The contact ellipse formed between the ball and the raceway surface may protrude from the groove when the load exceeds the allowable range.

This contact surface has an elliptical shape in which 1/2 the major diameter becomes a as shown in **Fig. 3.14**. The maximum allowable axial load is the maximum applied load in which the contact ellipse does not exceed the shoulder of the raceway groove. It is important to note that the axial load must result in P_{max} < 4 200 MPa even if the contact ellipse does not exceed the shoulder of the groove. The allowable axial load differs depending on the bearing internal clearance, groove curvature, and groove shoulder dimension.

When a combination radial and axial load is applied, verify truncation does not occur at the maximum loaded rolling element.



α : Contact angle
a : Contact ellipse 1/2 major diameter

Fig. 3.14 Contact ellipse

(2) Tapered roller bearing (see Fig. 3.15)

A tapered roller bearing supports axial load at the raceway surface and at the interface between the roller end face and large end rib. Therefore, the bearing can receive a larger axial force by increasing the contact angle α. However, there are different limits depending on the rotational speed and lubrication conditions because sliding contact occurs between the roller large end face and the large end rib inside face. Generally, the PV value, which is obtained by multiplying the sliding speed to the sliding surface pressure, is checked and calculated by a computer.

For further clarification please consult with **NTN** Engineering.

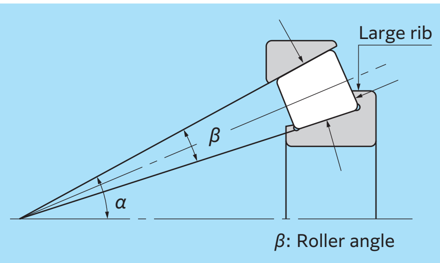


Fig. 3.15 Tapered roller bearings

(3) Cylindrical roller bearings

Cylindrical roller bearings with ribs on the inner and outer rings are capable of simultaneously bearing a radial load (F_r) and a certain degree of axial load (F_a). Unlike basic dynamic load ratings based on rolling fatigue, allowable axial load ($F_{a\ max}$) is defined by the following two methods. When determining the actual allowable axial load, the smaller value out of P_t and F_{ar} determined with formula (3.13) and formula (3.14) respectively, is used.

①Allowable axial load P_t based on allowable surface pressure of rib

This is the allowable axial load that is determined by factors such as the amount of heat produced on the sliding surface between the ends of the rollers and rib, seizure and wear. The surface pressure on rib defines the amount of axial load which can be applied to the bearing through the center line. The allowable axial load is approximately determined by formula (3.13), which is based upon experience and testing.

$$P_t = k_1 \cdot d^2 \cdot P_z \quad \cdots\cdots (3.13)$$

Where:

P_t : Allowable axial load based on allowable surface pressure of rib, N
k_1 : Factor determined by internal design of bearing (see **Table 3.7**)
d : Bearing bore diameter, mm
P_z : Allowable surface pressure of rib, MPa (see **Fig. 3.16**)

②Allowable axial load F_{ar} based on radial load

If the ratio of the axial load to the radial load is large, the rollers will not rotate properly. The allowable axial load F_{ar} based on the radial load is determined by formula (3.14).

$$F_{ar} = k_2 \cdot F_r \quad \cdots\cdots (3.14)$$

Where:

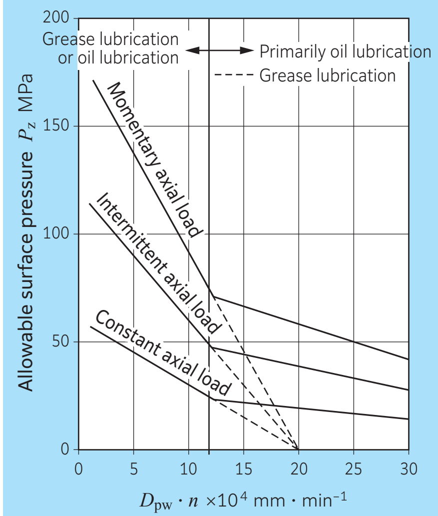
F_{ar} : Allowable axial load based on radial load, N
k_2 : Factor determined by internal design of bearing (see **Table 3.7**)
F_r : Radial load, N

The following are also important to operate the bearing smoothly under an axial load:

1) Do not make the internal radial clearance any larger than necessary because it may affect life and abrasion between the raceway surface and the roller.
2) Use lubricant with an extreme pressure additive to suppress heat generation, seizure, and abrasion between the roller end surface and the rib.
3) Make the shoulder of the housing and shaft high enough for the rib of the bearing to prevent it from being damaged.
4) If the bearing is to support an extreme axial load, mounting precision should be improved and the bearing should be rotated slowly before actual use.

If large cylindrical roller bearings (bore of 300 mm or more) are to support an axial load or moment load simultaneously, please contact **NTN** Engineering.

NTN Engineering also offers cylindrical roller bearings for high axial loads (HT type). For details, please contact **NTN** Engineering.



D_{pw}: Roller pitch diameter, mm
$D_{pw} \approx$ (Bearing bore diameter + bearing outside diameter)/2, mm
n : Rotational speed, min^{-1}

Fig. 3.16 Allowable surface pressure of rib

Table 3.7 Factors k_1 and k_2

Bearing series	k_1	k_2
NJ, NUP10 NJ, NUP, NF, NH2 NJ, NUP, NH22	0.040	0.4
NJ, NUP, NF, NH3 NJ, NUP, NH23	0.065	0.4
NJ, NUP, NH2EA (E) NJ, NUP, NH22EA (E)	0.050	0.4
NJ, NUP, NH3EA (E) NJ, NUP, NH23EA (E)	0.080	0.4
NJ, NUP, NH4	0.100	0.4
SL01-48	0.022	0.2
SL01-49	0.034	0.2
SL04-50	0.044	0.2

Note: EA type and E type have the same value.

3.12 Review of basic dynamic load ratings

As a result of continuous improvement related to material cleanliness, and production techniques, years of in-house durability testing has confirmed **NTN** bearings produced today have a longer operating life compared with past products. Based on this bearing life test data, the basic dynamic load ratings of ball and roller bearings were reviewed and updated to more accurately reflect true bearing performance.

The basic dynamic load ratings for many **NTN** products have been formally increased and can be found in the dimensional tables for each bearing type within this catalog.

* Some bearings use the same basic dynamic load rating as conventional products.

3.13 Bearing life calculation tool

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