



Large Bearings

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1. Load rating and life

1.1 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 cause **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, gnawing, rust, etc. However, these so-called "causes" of bearing failure are usually themselves caused by improper installation, insufficient or improper lubrication, faulty sealing or inaccurate bearing selection.

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

1.2 Basic rated life and basic dynamic load rating

A group of seemingly identical bearings when subjected to identical load 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 (1.1) and formula (1.2).

$$\text{For ball bearings : } L_{10} = \left(\frac{C}{P} \right)^3 \dots \dots \dots (1.1)$$

$$\text{For roller bearings: } L_{10} = \left(\frac{C}{P} \right)^{10/3} \dots \dots \dots (1.2)$$

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

n : Rotational speed, min^{-1}

Note: 1) For more details, please refer to the section "2. Bearing load calculation".

Table 1.1 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	$f_h \frac{C}{P}$	$f_h \frac{C}{P}$
Speed factor f_n	$\left(\frac{33.3}{n} \right)^{1/3}$	$\left(\frac{33.3}{n} \right)^{3/10}$

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 1.1** and **Fig. 1.1**.

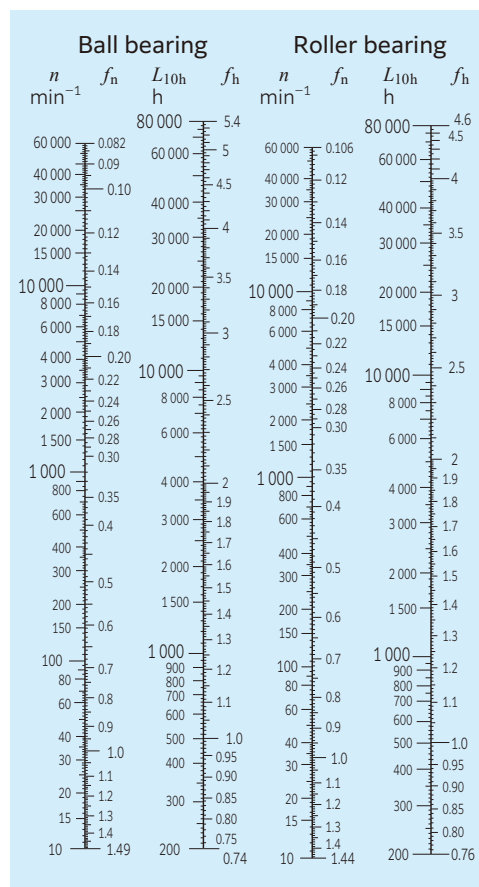


Fig. 1.1 Bearing life rating scale

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 1.3).

$$L = \frac{1}{\left(\frac{1}{L_1^e} + \frac{1}{L_2^e} + \dots + \frac{1}{L_n^e}\right)^{1/e}} \dots \dots \dots (1.3)$$

Where:

L : Total basic rating life of entire unit, h
 $L_1, L_2 \dots L_n$: Basic rating life of individual bearings
 1, 2, $\dots$ n, h

$e = 10/9 \dots \dots \dots$ For ball bearings
 $e = 9/8 \dots \dots \dots$ For roller bearings

When the load conditions vary at regular intervals, the life can be given by formula (1.4).

$$L_m = \left(\frac{\phi_1}{L_1} + \frac{\phi_2}{L_2} + \dots \dots \dots \frac{\phi_j}{L_j} \right)^{-1} \dots \dots (1.4)$$

Where:

L_m : Total life of bearing, h
 ϕ_j : Frequency of individual load conditions
 $(\sum \phi_j = 1)$
 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 1.1** and formula (1.5). Bearings that satisfy the required C can be selected from the bearing dimensions table provided in the catalog.

$$C = P \frac{f_h}{f_n} \dots \dots \dots (1.5)$$

1.3 Adjusted rating life

The basic bearing rating life can be calculated through the formulas mentioned earlier in Section 1.2. 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 (1.6).

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot L_{10} \dots \dots \dots (1.6)$$

Where:

L_{na} : Adjusted rating life 10^6 revolutions
 a_1 : Life adjustment factor for reliability
 a_2 : Life adjustment factor for special bearing properties
 a_3 : Life adjustment factor for operating conditions

1.3.1 Life adjustment factor for reliability a_1
 The value of life adjustment factor for reliability a_1 is provided in **Table 1.2** for reliability of 90 % or greater.

1.3.2 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 1.3**.

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

Table 1.2 Life adjustment factor for reliability a_1

Reliability %	L_n	Life adjustment factor for reliability a_1
90	L_{10}	1
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 1.3 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 with **NTN Engineering** for life adjustment factor for special bearing properties (a_2) when using dimensional stabilization treatment combined with any specialty bearing material.

1.3.3 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, the 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

For bearings used under special operating conditions, please consult with **NTN Engineering**.

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 (1.1, 1.2, and 1.6) 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.

1.4 Modified rating life

1.4.1 Background

Adjusted rating life L_{na} of bearings is as shown in formula (1.6). 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 (1.7).

$$L_{nm} = a_1 \cdot a_{ISO} \cdot L_{10} \quad (1.7)$$

1.4.2 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 (1.8).

$$a_{ISO} = f\left(\frac{e_c C_u}{P}, \kappa\right) \quad (1.8)$$

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) has 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 1.4**, 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 (1.9) by the ratio of dynamic viscosity ν in use with respect to reference dynamic viscosity ν_1 of the lubricant.

$$\kappa = \nu / \nu_1 \quad (1.9)$$

Reference dynamic viscosity ν_1 depends on rotational speed n and size (D_{pw}), and can be obtained by **Fig. 1.2** or formula (1.10) and formula (1.11).

Table 1.4 Value of contamination factor e_c

Level of contamination	e_c	
	$D_{pw} < 100 \text{ mm}$	$D_{pw} \geq 100 \text{ mm}$
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; coarse 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

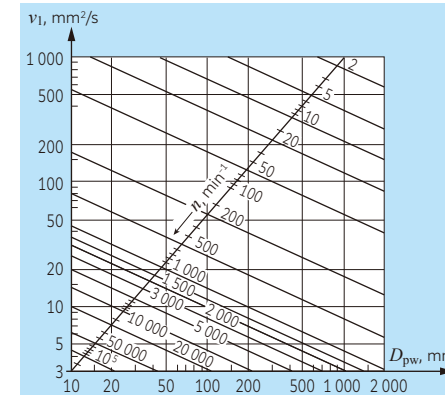


Fig. 1.2 Diagram for reference dynamic viscosity ν_1

In the case of $n < 1000 \text{ min}^{-1}$,
 $\nu_1 = 45\,000 n^{-0.83} D_{pw}^{-0.5} \quad (1.10)$

In the case of $n \geq 1000 \text{ min}^{-1}$,
 $\nu_1 = 4\,500 n^{-0.5} D_{pw}^{-0.5} \quad (1.11)$

Fig. 1.3 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 $a_{ISO} \leq 50$.
- 2) In the case of $\kappa > 4$, $\kappa = 4$. The same approach does not apply in the case of $\kappa < 0.1$.

Diagrams for radial roller bearings, thrust ball bearings, and thrust roller bearings have also been presented (see **Fig. 1.4** through **Fig. 1.6**). The diagrams can be applied regardless of lubrication types; however, for grease lubrication, special additives, and special rotating behaviors, consult with NTN Engineering.

1.4.3 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 heat treatment 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.

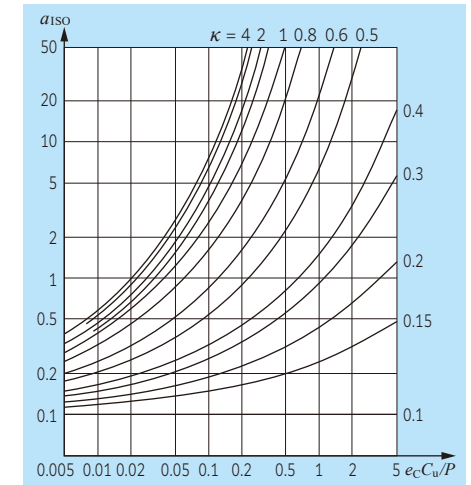


Fig. 1.3 Life modification factor a_{ISO} (radial ball bearing)

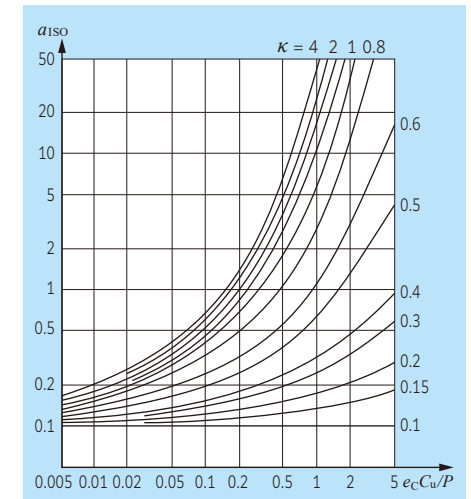


Fig. 1.4 Life modification factor a_{ISO} (radial roller bearing)

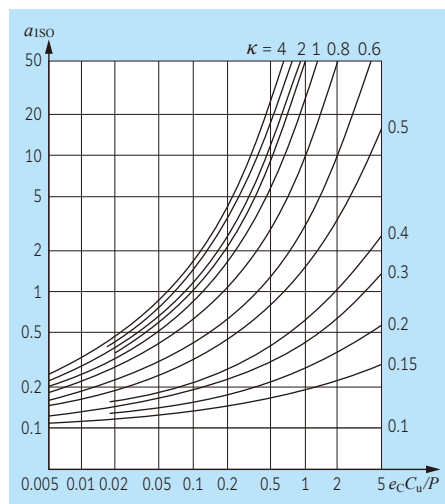


Fig. 1.5 Life modification factor a_{ISO} (thrust ball bearing)

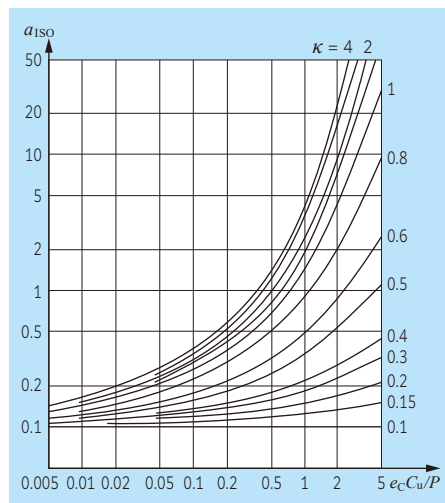


Fig. 1.6 Life modification factor a_{ISO} (thrust roller bearing)

1.5 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 impairment of bearing operation.

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

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.

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.

1.6 Allowable static equivalent load

Generally, the static equivalent load which can be permitted (refer to page A-14) is limited by the basic static load rating as stated in Section 1.5. 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 expressed by formula (1.12) and guidelines of **Table 1.5** into account.

$$S_0 = C_0 / P_0 \dots\dots\dots (1.12)$$

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 1.5 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. 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.
3. 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**.
4. When an As type raceway washer is used in a thrust bearing, min. S_0 value = 3.

1.7 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.

1.8 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](https://www.ntnglobal.com)).

2. Bearing load calculation

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

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

2.1 Load acting on shafts

2.1.1 Load factor

There are many instances where the actual operational shaft load is much greater than the theoretically calculated load, due to shock. This actual shaft load can be estimated by using formula (2.1).

$$K = f_w \cdot K_c \dots\dots\dots (2.1)$$

Where:

K : Actual shaft load, N

f_w : Load factor (see Table 2.1)

K_c : Theoretically calculated value, N

Table 2.1 Load factor f_w

Amount of shock	f_w	Machine application examples
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, printing machines, aircraft, textile machines, electrical units, office machines.
Heavy shock	1.5-3.0	Crushers, agricultural equipment, construction equipment, cranes.

2.2 Mean load

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

(1) Fluctuating stepped load (see Fig. 2.1)

The mean bearing load, F_m , for stepped loads is calculated from formula (2.2). $F_1, F_2, \dots, F_n$ are the loads acting on the bearing; $n_1, n_2, \dots, n_n$ and $t_1, t_2, \dots, t_n$ are the bearing speeds and operating times, respectively.

$$F_m = \left[\frac{\sum (F_i^p n_i t_i)}{\sum (n_i t_i)} \right]^{1/p} \dots\dots\dots (2.2)$$

Where:

$p = 3$ For ball bearings

$p = 10/3$ For roller bearings

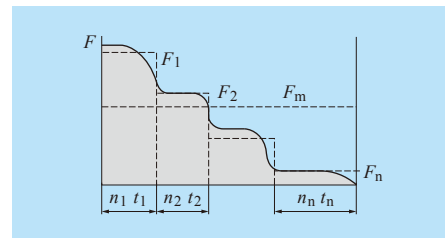


Fig. 2.1 Stepped load

(2) Continuously fluctuating load (see Fig. 2.2)

Where it is possible to express the function $F(t)$ in terms of load cycle t_0 and time t , the mean load is found by using formula (2.3).

$$F_m = \left[\frac{1}{t_0} \int_0^{t_0} F(t)^p dt \right]^{1/p} \dots\dots\dots (2.3)$$

Where:

$p = 3$ For ball bearings

$p = 10/3$ For roller bearings

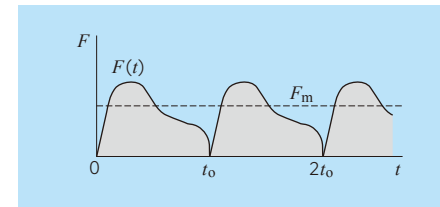


Fig. 2.2 Load that fluctuated as function of time

(3) Linear fluctuating load (see Fig. 2.3)

The mean load, F_m , can be approximated by formula (2.4).

$$F_m = \frac{F_{\min} + 2F_{\max}}{3} \dots\dots\dots (2.4)$$

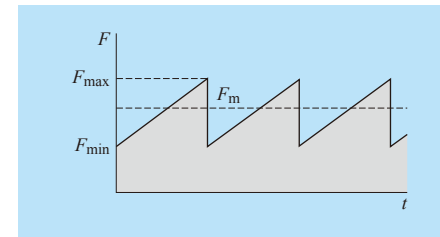


Fig. 2.3 Linear fluctuating load

(4) Sinusoidal fluctuating load (see Fig. 2.4)

The mean load, F_m , can be approximated by formula (2.5) and formula (2.6).

$$\text{Case (a) } F_m = 0.75F_{\max} \dots\dots\dots (2.5)$$

$$\text{Case (b) } F_m = 0.65F_{\max} \dots\dots\dots (2.6)$$

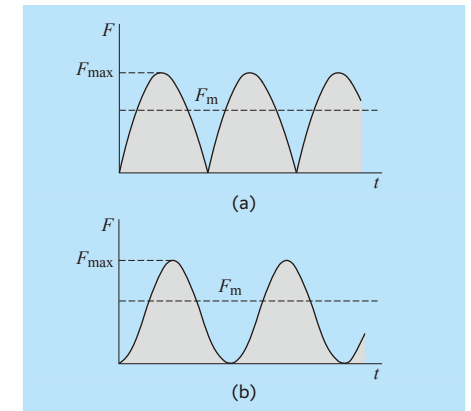


Fig. 2.4 Sinusoidal variable load

2.3 Equivalent load

2.3.1 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 (2.7).

$$P_r = XF_r + YF_a \quad (2.7)$$

Where:

P_r : Dynamic equivalent radial load, N

F_r : Actual radial load, N

F_a : Actual axial load, N

X : Radial load factor

Y : Axial load factor

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

(2) Dynamic equivalent axial load

As a rule, standard thrust bearings with α contact angle of 90° cannot carry radial loads.

However, thrust spherical roller bearings can accept some radial load. The dynamic equivalent axial load for these bearings is expressed by formula (2.8).

$$P_a = F_a + 1.2F_r \quad (2.8)$$

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.

2.3.2 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 (2.9) or formula (2.10). The greater of the two resultant values is always taken for P_{0r} .

$$P_{0r} = X_0 F_r + Y_0 F_a \quad (2.9)$$

$$P_{0r} = F_r \quad (2.10)$$

Where:

P_{0r} : Static equivalent radial load, N

F_r : Actual radial load, N

F_a : Actual axial load, N

X_0 : Static radial load factor

Y_0 : Static axial load factor

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

(2) Static equivalent axial load

For thrust spherical roller bearings the static equivalent axial load is expressed by formula (2.11).

$$P_{0a} = F_a + 2.7F_r \quad (2.11)$$

Where:

P_{0a} : Static equivalent axial load, N

F_a : Actual axial load, N

F_r : Actual radial load, N

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

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

For angular contact ball bearings and tapered roller bearings the pressure cone apex (load center) is located as shown in Fig. 2.5, and their values are listed in the bearing tables.

When radial loads act on these types of bearings a component force is induced in the axial direction. For this reason, these bearings

are used in pairs. For load calculation this component force must be taken into consideration and is expressed by formula (2.12).

$$F_a = \frac{0.5F_r}{Y} \quad (2.12)$$

Where:

F_a : Axial component force, N

F_r : Actual radial load, N

Y : Axial load factor

The axial loads for these bearing pairs are given in Table 2.2.

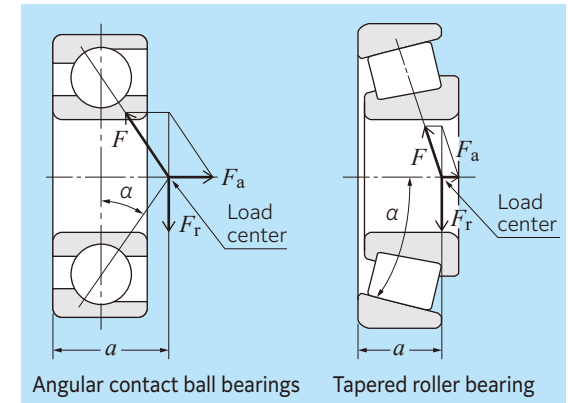


Fig. 2.5 Pressure cone apex and axial component force

Table 2.2 Bearing arrangement and equivalent load

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

Note: 1. Applies when preload is zero.

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

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

3. Bearing tolerances

3.1 Dimensional and rotational accuracy

Bearing “tolerances” or dimensional accuracy and running accuracy, are regulated by ISO and JIS standards, JIS B 1514 (rolling bearing tolerances) series. For **dimensional accuracy**, these standards prescribe the tolerances necessary when installing bearings on shafts or in housings. **Running accuracy** is defined as the allowable limits for bearing runout during operation.

Dimensional accuracy

Dimensional accuracy constitutes the acceptable values for bore diameter, outside diameter, assembled bearing width, and bore diameter uniformity as seen in chamfer dimensions, allowable inner ring tapered bore deviation and shape error. Also included are variation of mean bore diameter within a plane, outside diameter within a plane, mean outside diameter within a plane, as well as raceway thickness (for thrust bearings).

Running accuracy

Running accuracy constitutes the acceptable values for inner and outer ring radial runout and axial runout, inner ring side surface squareness, and outer ring outside diameter squareness.

Allowable rolling bearing tolerances have been established according to precision classes. Bearing precision is stipulated as JIS Class 6, Class 5, Class 4, or Class 2, with precision rising from ordinary precision indicated by JIS Class 0.

Table 3.1 indicates which standards and precision classes are applicable to the major bearing types. **Table 3.2** shows a relative comparison between JIS B 1514 precision class standards and other standards.

For details of allowable limitations and values, refer to **Table 3.3** through **Table 3.10**, which are described in the application table column of **Table 3.1**. Allowable values for chamfer dimensions are shown in **Table 3.11**. Allowable limitations and values for radial bearing inner ring tapered bores are shown in **Table 3.12**.

Table 3.1 Bearing types and applicable tolerance

Bearing type		Applicable standard	Accuracy class					Tolerance table
Deep groove ball bearings		JIS B 1514-1 (ISO 492)	Class 0	Class 6	Class 5	Class 4	Class 2	Table 3.3
Angular contact ball bearings			Class 0	Class 6	Class 5	Class 4	Class 2	
Self-aligning ball bearings			Class 0	—	—	—	—	
Cylindrical roller bearings			Class 0	Class 6	Class 5	Class 4	Class 2	
Spherical roller bearings			Class 0	—	—	—	—	
Tapered roller bearings	Metric series (single-row)	JIS B 1514-1	Class 0, 6X	Class 6 ¹⁾	Class 5	Class 4	—	Table 3.4
	Metric series (double-row/four-row)	—	Class 0 ¹⁾	—	—	—	—	Table 3.6
	Inch series	ANSI/ABMA Std.19	Class 4	Class 2	Class 3	Class 0	Class 00	Table 3.5
Thrust ball bearings		JIS B 1514-2 (ISO 199)	Class 0	Class 6	Class 5	Class 4	—	Table 3.7
Thrust cylindrical roller bearings			Class 0	Class 6	Class 5	Class 4	—	
Thrust spherical roller bearings				Class 0	—	—	—	—
Thrust tapered roller bearings	Metric series	NTN standard	Class 0	—	—	—	—	Table 3.9
	Inch series	ANSI/ABMA Std.23	Class 2	—	—	—	—	Table 3.10

1) The class is the NTN standard class.

Table 3.2 Comparison of tolerance classifications of national standards

Standard	Applicable standard	Accuracy class					Bearing type
Japanese industrial standards (JIS)	JIS B 1514-1	Class 0, 6	Class 6	Class 5	Class 4	Class 2	Radial bearings
	JIS B 1514-2	Class 0	Class 6	Class 5	Class 4	—	Thrust bearings
International Organization for Standardization (ISO)	ISO 492	Normal class Class 6X	Class 6	Class 5	Class 4	Class 2	Radial bearings
	ISO 199	Normal class	Class 6	Class 5	Class 4	—	Thrust bearings
	ISO 578	Class 4	—	Class 3	Class 0	Class 00	Tapered roller bearings (Inch series)
Deutsches Institut für Normung (DIN)	DIN 620	P0	P6	P5	P4	P2	All types
American National Standards Institute (ANSI)	ANSI/ABMA Std.20 ¹⁾	ABEC-1 RBEC-1	ABEC-3 RBEC-3	ABEC-5 RBEC-5	ABEC-7	ABEC-9	Radial bearings (excluding tapered roller bearings)
American Bearing Manufacturers Association (ABMA)	ANSI/ABMA Std.19	Class 4	Class 2	Class 3	Class 0	Class 00	Tapered roller bearings (Inch series)

1) “ABEC” is applied to ball bearings and “RBEC” to roller bearings.

Note: 1. JIS B 1514 series, ISO 492, 199, and DIN 620 have the same specification level.

2. The tolerance and allowance of JIS B 1514 series are slightly different from those of ABMA standards.

Table 3.3 Tolerance of radial bearings (except tapered roller bearings)

Table 3.3 (1) Inner rings

Nominal bearing bore diameter		Deviation of mean bore diameter in a single plane										Variation of bore diameter in a single plane														
d		Δ_{dmp}										V_{dsp}														
mm												Diameter series 9					Diameter series 0, 1					Diameter series 2, 3, 4				
Over	Incl.	Class 0	Class 6	Class 5	Class 4 ¹⁾	Class 2 ¹⁾	Class 0	Class 6	Class 5	Class 4 ¹⁾	Class 2 ¹⁾	Class 0	Class 6	Class 5	Class 4	Class 2	Class 0	Class 6	Class 5	Class 4	Class 2	Class 0	Class 6	Class 5	Class 4	Class 2
Over	Incl.	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.
80	120	0	-20	0	-15	0	-10	0	-8	0	-5	25	19	10	8	5	25	19	8	6	5	15	11	8	6	5
120	150	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
150	180	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
180	250	0	-30	0	-22	0	-15	0	-12	0	-8	38	28	15	12	8	38	28	12	9	8	23	17	12	9	8
250	315	0	-35	0	-25	0	-18	0	—	—	—	44	31	18	—	—	44	31	14	—	—	26	19	14	—	—
315	400	0	-40	0	-30	0	-23	—	—	—	—	50	38	23	—	—	50	38	18	—	—	30	23	18	—	—
400	500	0	-45	0	-35	—	—	—	—	—	—	56	44	—	—	—	56	44	—	—	—	34	26	—	—	—
500	630	0	-50	0	-40	—	—	—	—	—	—	63	50	—	—	—	63	50	—	—	—	38	30	—	—	—
630	800	0	-75	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
800	1 000	0	-100	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 000	1 250	0	-125	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 250	1 600	0	-160	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 600	2 000	0	-200	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

1) The dimensional difference Δd_s of the measured bore diameter applied to Classes 4 and 2 is the same as the tolerance of dimensional difference Δ_{dmp} of the mean bore diameter within a plane.
However, the dimensional difference is applied to diameter series 0, 1, 2, 3 and 4 for Class 4, and also to all the diameter series for Class 2.

Table 3.3 (2) Outer rings

Nominal bearing outside diameter		Deviation of mean outside diameter in a single plane										Variation of outside diameter in a single plane														
D		Δ_{Dmp}										V_{Dsp}														
mm												Diameter series 9					Diameter series 0, 1					Diameter series 2, 3, 4				
Over	Incl.	Class 0	Class 6	Class 5	Class 4 ⁴⁾	Class 2 ⁴⁾	Class 0	Class 6	Class 5	Class 4 ⁴⁾	Class 2 ⁴⁾	Class 0	Class 6	Class 5	Class 4	Class 2	Class 0	Class 6	Class 5	Class 4	Class 2	Class 0	Class 6	Class 5	Class 4	Class 2
Over	Incl.	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.
80	120	0	-15	0	-13	0	-10	0	-8	0	-5	19	16	10	8	5	19	16	8	6	5	11	10	8	6	5
120	150	0	-18	0	-15	0	-11	0	-9	0	-5	23	19	11	9	5	23	19	8	7	5	14	11	8	7	5
150	180	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
180	250	0	-30	0	-20	0	-15	0	-11	0	-8	38	25	15	11	8	38	25	11	8	8	23	15	11	8	8
250	315	0	-35	0	-25	0	-18	0	-13	0	-8	44	31	18	13	8	44	31	14	10	8	26	19	14	10	8
315	400	0	-40	0	-28	0	-20	0	-15	0	-10	50	35	20	15	10	50	35	15	11	10	30	21	15	11	10
400	500	0	-45	0	-33	0	-23	—	—	—	—	56	41	23	—	—	56	41	17	—	—	34	25	17	—	—
500	630	0	-50	0	-38	0	-28	—	—	—	—	63	48	28	—	—	63	48	21	—	—	38	29	21	—	—
630	800	0	-75	0	-45	0	-35	—	—	—	—	94	56	35	—	—	94	56	26	—	—	55	34	26	—	—
800	1 000	0	-100	0	-60	—	—	—	—	—	—	125	75	—	—	—	125	75	—	—	—	75	45	—	—	—
1 000	1 250	0	-125	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 250	1 600	0	-160	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 600	2 000	0	-200	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2 000	2 500	0	-250	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

4) The dimensional difference ΔD_s of the measured outside diameter applied to Classes 4 and 2 is the same as the tolerance of dimensional difference Δ_{Dmp} of the mean outside diameter within a plane.
However, the dimensional difference is applied to diameter series 0, 1, 2, 3 and 4 for Class 4, and also to all the diameter series for Class 2.

Unit: μm

Variation of mean bore diameter					Radial runout of inner ring of assembled bearing					Perpendicularity of inner ring face with respect to the bore			Axial runout of inner ring of assembled bearing			Deviation of a single inner ring width						Variation of inner ring width								
V_{dmp}					K_{ia}					S_d			$S_{ia}^{(2)}$			ΔB_s						V_{Bs}								
																Normal bearings			Duplex bearings ⁽³⁾											
																Class 0		Class 5	Class 0		Class 5									
Class 0	Class 6	Class 5	Class 4	Class 2	Class 0	Class 6	Class 5	Class 4	Class 2	Class 5	Class 4	Class 2	Class 5	Class 4	Class 2	Class 6	Class 4	Class 6	Class 4	Class 0	Class 5	Class 0	Class 5	Class 0	Class 6	Class 5	Class 4	Class 2		
Max.					Max.					Max.			Max.			Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	
15	11	5	4	2.5	25	13	6	5	2.5	9	5	2.5	9	5	2.5	0	-200	0	-200	0	-200	0	-380	0	-380	25	25	7	4	2.5
19	14	7	5	3.5	30	18	8	6	2.5	10	6	2.5	10	7	2.5	0	-250	0	-250	0	-250	0	-500	0	-380	30	30	8	5	2.5
19	14	7	5	3.5	30	18	8	6	5	10	6	4	10	6	4	0	-250	0	-250	0	-250	0	-500	0	-380	30	30	8	5	4
23	17	8	6	4	40	20	10	8	5	11	7	5	13	8	5	0	-300	0	-300	0	-300	0	-500	0	-500	30	30	10	6	5
26	19	9	—	—	50	25	13	—	—	13	—	—	15	—	—	0	-350	0	—	—	—	0	-500	0	—	35	35	13	—	—
30	23	12	—	—	60	30	15	—	—	15	—	—	20	—	—	0	-400	0	—	—	—	0	-630	0	—	40	40	15	—	—
34	26	—	—	—	65	35	—	—	—	—	—	—	—	—	—	0	-450	—	—	—	—	—	—	—	—	50	45	—	—	—
38	30	—	—	—	70	40	—	—	—	—	—	—	—	—	—	0	-500	—	—	—	—	—	—	—	—	60	50	—	—	—
55	—	—	—	—	80	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	70	—	—	—	—
75	—	—	—	—	90	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	80	—	—	—	—
94	—	—	—	—	100	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	100	—	—	—	—
120	—	—	—	—	120	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	120	—	—	—	—
150	—	—	—	—	140	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	140	—	—	—	—

Table 3.4 Tolerance of tapered roller bearings (metric series)

Table 3.4 (1) Inner rings

Nominal bearing bore diameter		Deviation of mean bore diameter in a single plane				Variation of bore diameter in a single plane				Variation of mean bore diameter				Radial runout of inner ring of assembled bearing				Perpendicularity of inner ring face with respect to the bore S_d	
d		Δ_{dmp}				V_{dsp}				V_{dmp}				K_{ia}				S_d	
mm		Class 0				Class 0				Class 0				Class 0				Class 5	
mm		Class 6X				Class 6X				Class 6X				Class 6X				Class 4	
Over	Incl.	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
80	120	0	-20	0	-15	0	-10	20	15	11	8	15	11	8	5	30	13	8	5
120	180	0	-25	0	-18	0	-13	25	18	14	10	19	14	9	7	35	18	11	6
180	250	0	-30	0	-22	0	-15	30	22	17	11	23	16	11	8	50	20	13	8
250	315	0	-35	—	—	—	—	35	—	—	—	26	—	—	—	60	—	—	—
315	400	0	-40	—	—	—	—	40	—	—	—	30	—	—	—	70	—	—	—
400	500	0	-45	—	—	—	—	45	—	—	—	34	—	—	—	80	—	—	—
500	630	0	-50	—	—	—	—	50	—	—	—	38	—	—	—	90	—	—	—
630	800	0	-75	—	—	—	—	75	—	—	—	56	—	—	—	105	—	—	—
800	1 000	0	-100	—	—	—	—	100	—	—	—	75	—	—	—	120	—	—	—
1 000	1 250	0	-125	—	—	—	—	125	—	—	—	94	—	—	—	140	—	—	—
1 250	1 600	0	-160	—	—	—	—	160	—	—	—	120	—	—	—	160	—	—	—

1) Class 6 is the NTN standard class.

2) The dimensional difference Δ_{ds} of the measured bore diameter applied to Class 4 is the same as the tolerance of dimensional difference Δ_{dmp} of the mean bore diameter within a plane.

Table 3.4 (2) Outer rings

Nominal bearing outside diameter		Deviation of mean outside diameter in a single plane				Variation of outside diameter in a single plane				Variation of mean outside diameter				Radial runout of outer ring of assembled bearing				Perpendicularity of outer ring outside surface with respect to the face $S_D^{3)}$	
D		Δ_{Dmp}				V_{Dsp}				V_{Dmp}				K_{ea}				$S_D^{3)}$	
mm		Class 0				Class 0				Class 0				Class 0				Class 5	
mm		Class 6X				Class 6X				Class 6X				Class 6X				Class 4	
Over	Incl.	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
80	120	0	-18	0	-13	0	-10	18	13	10	8	14	10	7	5	35	18	10	6
120	150	0	-20	0	-15	0	-11	20	15	11	8	15	11	8	6	40	20	11	7
150	180	0	-25	0	-18	0	-13	25	18	14	10	19	14	9	7	45	23	13	8
180	250	0	-30	0	-20	0	-15	30	20	15	11	23	15	10	8	50	25	15	10
250	315	0	-35	0	-25	0	-18	35	25	19	14	26	19	13	9	60	30	18	11
315	400	0	-40	0	-28	0	-20	40	28	22	15	30	21	14	10	70	35	20	13
400	500	0	-45	—	—	—	—	45	—	—	—	34	—	—	—	80	—	—	—
500	630	0	-50	—	—	—	—	60	—	—	—	38	—	—	—	100	—	—	—
630	800	0	-75	—	—	—	—	75	—	—	—	56	—	—	—	120	—	—	—
800	1 000	0	-100	—	—	—	—	100	—	—	—	75	—	—	—	140	—	—	—
1 000	1 250	0	-125	—	—	—	—	125	—	—	—	84	—	—	—	165	—	—	—
1 250	1 600	0	-160	—	—	—	—	160	—	—	—	120	—	—	—	190	—	—	—
1 600	2 000	0	-200	—	—	—	—	200	—	—	—	150	—	—	—	230	—	—	—

3) Does not apply to bearings with flange.

4) The dimensional difference Δ_{Ds} of the measured outside diameter applied to Class 4 is the same as the tolerance of dimensional difference Δ_{Dmp} of the mean outside diameter within a plane.

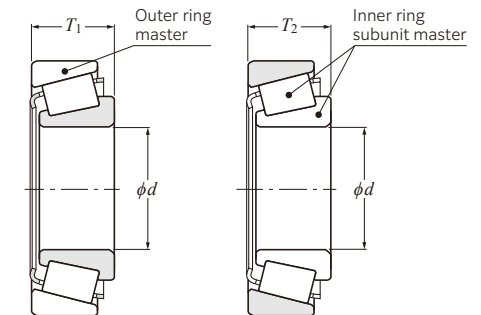
Unit: μm

Axial runout of inner ring of assembled bearing	Deviation of a single inner ring width						Deviation of the actual assembled bearing width					
	Δ_{Bs}						Δ_{Ts}					
	Class 0						Class 5					
	Class 6						Class 6X					
Class 4	Class 6	Class 6X	Class 4	Class 6	Class 6X	Class 4	Class 6	Class 6X	Class 4	Class 6	Class 6X	Class 4
Max.	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
5	0	-200	0	-50	0	-400	+200	-200	+100	0	+200	-200
7	0	-250	0	-50	0	-500	+350	-250	+150	0	+350	-250
8	0	-300	0	-50	0	-600	+350	-250	+150	0	+350	-250
—	0	-350	0	-50	—	—	+350	-250	+200	0	—	—
—	0	-400	0	-50	—	—	+400	-400	+200	0	—	—
—	0	-450	—	—	—	—	—	—	—	—	—	—
—	0	-500	—	—	—	—	—	—	—	—	—	—
—	0	-750	—	—	—	—	—	—	—	—	—	—
—	0	-1 000	—	—	—	—	—	—	—	—	—	—
—	0	-1 200	—	—	—	—	—	—	—	—	—	—
—	0	-1 500	—	—	—	—	—	—	—	—	—	—

Table 3.4 (3) Effective width of inner subunits and outer rings

Unit: μm

Nominal bearing bore diameter		Deviation of the actual effective width of inner subunit assembled with a master outer ring				Deviation of the actual effective width of outer ring assembled with a master inner subunit			
d		Δ_{T1s}				Δ_{T2s}			
mm		Class 0				Class 6X			
Over	Incl.	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
80	120	+100	-100	+50	0	+100	-100	+50	0
120	180	+150	-150	+50	0	+200	-100	+100	0
180	250	+150	-150	+50	0	+200	-100	+100	0
250	315	+150	-150	+100	0	+200	-100	+100	0
315	400	+200	-200	+100	0	+200	-200	+100	0



5) Applies to bearings with a nominal bore diameter d over 80 mm and 400 mm or less.

Table 3.5 Tolerance of tapered roller bearings (inch series)

Table 3.5 (1) Inner rings

Unit: μm

Nominal bearing bore diameter d mm(inch)		Deviation of a single bore diameter Δd_s									
Over	Incl.	Class 4		Class 2		Class 3		Class 0		Class 00	
		Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
76.2 (3)	266.7(10.5)	+25	0	+25	0	+13	0	+13	0	+8	0
266.7(10.5)	304.8(12)	+25	0	+25	0	+13	0	+13	0	—	—
304.8(12)	609.6(24)	+51	0	+51	0	+25	0	—	—	—	—
609.6(24)	914.4(36)	+76	0	—	—	+38	0	—	—	—	—
914.4(36)	1 219.2(48)	+102	0	—	—	+51	0	—	—	—	—
1 219.2(48)	—	+127	0	—	—	+76	0	—	—	—	—

Table 3.5 (2) Outer rings

Unit: μm

Nominal bearing outside diameter D mm(inch)		Deviation of a single outside diameter ΔD_s									
Over	Incl.	Class 4		Class 2		Class 3		Class 0		Class 00	
		Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
—	266.7(10.5)	+25	0	+25	0	+13	0	+13	0	+8	0
266.7(10.5)	304.8(12)	+25	0	+25	0	+13	0	+13	0	—	—
304.8(12)	609.6(24)	+51	0	+51	0	+25	0	—	—	—	—
609.6(24)	914.4(36)	+76	0	+76	0	+38	0	—	—	—	—
914.4(36)	1 219.2(48)	+102	0	—	—	+51	0	—	—	—	—
1 219.2(48)	—	+127	0	—	—	+76	0	—	—	—	—

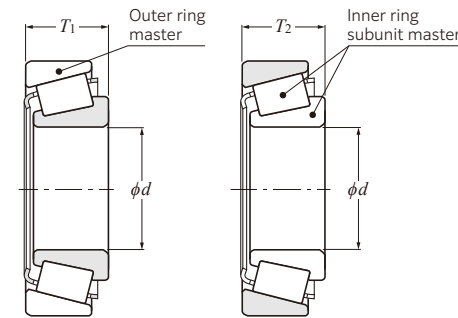
Table 3.5 (3) Assembly width of single-row bearings, combination width of 4-row bearings, effective width of inner ring subunits, effective width of outer rings

Nominal bearing bore diameter d mm(inch)		Nominal outside diameter D mm(inch)		Deviation of the actual assembled single row bearing width ΔT_s						Deviation of four-row bearing overall width $\Delta B_{2s}, \Delta C_{2s}$	
Over	Incl.	Over	Incl.	Class 4		Class 2		Class 3		Class 0, 00	
				Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
—	101.6 (4)	—	508.0 (20)	+203	0	+203	0	+203	-203	+1 524	-1 524
101.6 (4)	304.8 (12)	—	508.0 (20)	+356	-254	+203	0	+203	-203	+1 524	-1 524
304.8 (12)	609.6 (24)	—	508.0 (20)	+381	-381	+381	-381	+203	-203	+1 524	-1 524
304.8 (12)	609.6 (24)	508.0 (20)	—	+381	-381	+381	-381	+381	-381	+1 524	-1 524
609.6 (24)	—	—	—	+381	-381	—	—	+381	-381	+1 524	-1 524

Table 3.5 (4) Radial runout of inner and outer rings

Unit: μm

Nominal bearing outside diameter D mm(inch)		Radial runout of inner ring of assembled bearing K_{ia} Radial runout of outer ring of assembled bearing K_{ea}				
Over	Incl.	Class 4 Max.	Class 2 Max.	Class 3 Max.	Class 0 Max.	Class 00 Max.
—	304.8(14)	51	38	8	4	2
304.8(14)	609.6(24)	51	38	18	—	—
609.6(24)	914.4(36)	76	51	51	—	—
914.4(36)	—	76	—	76	—	—



Unit: μm

Deviation of the actual effective width of inner subunit assembled with a master outer ring ΔT_{1s}						Deviation of the actual effective width of outer ring assembled with a master inner subunit ΔT_{2s}					
Class 4		Class 2		Class 3		Class 4		Class 2		Class 3	
Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
+102	0	+102	0	+102	-102	+102	0	+102	0	+102	-102
+152	-152	+102	0	+102	-102	+203	-102	+102	0	+102	-102
—	—	+178	-178 ¹⁾	+102	-102 ¹⁾	—	—	+203	-203 ¹⁾	+102	-102 ¹⁾
—	—	—	—	—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—	—	—	—	—

1) Applies to nominal bearing bore diameters d of 406.400 mm (16 inch) or less.

Table 3.6 Tolerance of double-row and four-row tapered roller bearings (metric series)

Table 3.6 (1) Inner rings

Unit: μm

Nominal bearing bore diameter d mm		Deviation of mean bore diameter in a single plane Δd_{mp}		Variation of bore diameter in a single plane V_{dsp}	Variation of mean bore diameter V_{dmp}	Radial runout of inner ring of assembled bearing K_{ia}	Deviation of a single inner ring width ΔB_s		Deviation of bearing overall width			
									Double row bearing		Four-row bearing	
									ΔB_{1s}		ΔB_{2s}	
Over	Incl.	Upper	Lower	Max.	Max.	Max.	Upper	Lower	Upper	Lower	Upper	Lower
80	120	0	-20	20	15	30	0	-200	+400	-400	+500	-500
120	180	0	-25	25	19	35	0	-250	+500	-500	+600	-600
180	250	0	-30	30	23	50	0	-300	+600	-600	+750	-750
250	315	0	-35	35	26	60	0	-350	+700	-700	+900	-900
315	400	0	-40	40	30	70	0	-400	+800	-800	+1 000	-1 000
400	500	0	-45	45	34	80	0	-450	+900	-900	+1 200	-1 200
500	630	0	-60	60	40	90	0	-500	+1 000	-1 000	+1 200	-1 200
630	800	0	-75	75	45	100	0	-750	+1 500	-1 500	+1 500	-1 500
800	1 000	0	-100	100	55	115	0	-1 000	+1 500	-1 500	+1 500	-1 500
1 000	1 250	0	-125	125	65	130	0	-1 250	+1 500	-1 500	+1 500	-1 500
1 250	1 600	0	-160	160	80	150	0	-1 600	+1 500	-1 500	+1 500	-1 500
1 600	2 000	0	-200	200	100	170	0	-2 000	+1 500	-1 500	+1 500	-1 500

Note: This standard is the NTN standard.

Table 3.6 (2) Outer rings

Unit: μm

Nominal bearing outside diameter D mm		Deviation of mean outside diameter in a single plane ΔD_{mp}		Variation of outside diameter in a single plane V_{Dsp}	Variation of mean outside diameter V_{Dmp}	Radial runout of outer ring of assembled bearing K_{ea}	Deviation of a single outer ring width ΔC_s		Deviation of bearing overall width			
									Double row bearing		Four-row bearing	
									ΔC_{1s}		ΔC_{2s}	
Over	Incl.	Upper	Lower	Max.	Max.	Max.	Upper	Lower	Upper	Lower	Upper	Lower
80	120	0	-18	18	14	35	Depends on tolerance of ΔB_s in relation to d of the same bearing		Depends on tolerance of ΔB_{1s} in relation to d of the same bearing		Depends on tolerance of ΔB_{2s} in relation to d of the same bearing	
120	150	0	-20	20	15	40						
150	180	0	-25	25	19	45						
180	250	0	-30	30	23	50						
250	315	0	-35	35	26	60						
315	400	0	-40	40	30	70						
400	500	0	-45	45	34	80						
500	630	0	-50	60	38	100						
630	800	0	-75	80	55	120						
800	1 000	0	-100	100	75	140						
1 000	1 250	0	-125	130	90	160						
1 250	1 600	0	-160	170	100	180						
1 600	2 000	0	-200	210	110	200						
2 000	2 500	0	-250	265	120	220						

Note: For nominal bearing outside diameter D exceeding 1 600 mm, the NTN standard is followed.

Table 3.7 Tolerance of thrust ball bearings and thrust cylindrical roller bearings

Table 3.7 (1) Shaft raceway washer

Unit: μm

Nominal bearing bore diameter d mm		Deviation of mean bore diameter in a single plane Δd_{mp}				Variation of bore diameter in a single plane V_{dsp}		Variation in thickness between shaft washer raceway and back face S_i			
		Class 0, 6, 5		Class 4		Class 0, 6, 5	Class 4	Class 0	Class 6	Class 5	Class 4
		Upper	Lower	Upper	Lower	Max.	Max.	Max.	Max.	Max.	Max.
Over	Incl.	Upper	Lower	Upper	Lower	Max.	Max.	Max.	Max.	Max.	Max.
80	120	0	-20	0	-15	15	11	15	8	4	3
120	180	0	-25	0	-18	19	14	15	9	5	4
180	250	0	-30	0	-22	23	17	20	10	5	4
250	315	0	-35	0	-25	26	19	25	13	7	5
315	400	0	-40	0	-30	30	23	30	15	7	5
400	500	0	-45	0	-35	34	26	30	18	9	6
500	630	0	-50	0	-40	38	30	35	21	11	7
630	800	0	-75	0	-50	55	40	40	25	13	8
800	1 000	0	-100	—	—	75	—	45	30	15	—

Table 3.7 (2) Housing raceway washer

Unit: μm

Nominal bearing outside diameter		Deviation of mean outside diameter in a single plane				Variation of outside diameter in a single plane		Variation in thickness between shaft washer raceway and back face	
D		ΔD_{mp}				$V_{D\text{sp}}$		S_e	
mm		Class 0, 6, 5		Class 4		Class 0, 6, 5	Class 4	Class 0, 6, 5 4	
Over	Incl.	Upper	Lower	Upper	Lower	Max.	Max.	Max.	
80	120	0	-22	0	-13	17	10	Depends on tolerance of S_i against d of the same bearings	
120	180	0	-25	0	-15	19	11		
180	250	0	-30	0	-20	23	15		
250	315	0	-35	0	-25	26	19		
315	400	0	-40	0	-28	30	21		
400	500	0	-45	0	-33	34	25		
500	630	0	-50	0	-38	38	29		
630	800	0	-75	0	-45	55	34		
800	1 000	0	-100	0	-60	75	45		
1 000	1 250	0	-125	—	—	95	—		

Table 3.7 (3) Bearing height

Unit: μm

Nominal bearing bore diameter d mm		Deviation of the actual bearing height, single-direction bearing ¹⁾ ΔT_s	
		Upper	Lower
Over	Incl.	Upper	Lower
80	120	0	-150
120	180	0	-175
180	250	0	-200
250	315	0	-225
315	400	0	-300
400	500	0	-350
500	630	0	-400
630	800	0	-500
800	1 000	0	-600

1) Applies to flat back face bearing of Class 0.
The values are the NTN standard.

Table 3.8 Tolerance of thrust spherical roller bearings

Table 3.8 (1) Shaft raceway washer

Unit: μm

Nominal bearing bore diameter		Deviation of mean bore diameter in a single plane Δd_{mp}		Variation of bore diameter in a single plane V_{dsp}	Perpendicularity of shaft washer back face with respect to the bore ¹⁾ S_d	Deviation of the actual bearing height ¹⁾ ΔT_s	
Over	Incl.	Upper	Lower	Max.	Max.	Upper	Lower
80	120	0	-20	15	25	+200	-200
120	180	0	-25	19	30	+250	-250
180	250	0	-30	23	30	+300	-300
250	315	0	-35	26	35	+350	-350
315	400	0	-40	30	40	+400	-400
400	500	0	-45	34	45	+450	-450
500	630	0	-50	38	50	+500	-500
630	800	0	-75	55	60	+750	-750

1) The standard conforms to JIS B 1539.

Table 3.8 (2) Housing raceway washer Unit: μm

Nominal bearing outside diameter		Deviation of mean outside diameter in a single plane ΔD_{mp}	
Over	Incl.	Upper	Lower
120	180	0	-25
180	250	0	-30
250	315	0	-35
315	400	0	-40
400	500	0	-45
500	630	0	-50
630	800	0	-75
800	1 000	0	-100
1 000	1 250	0	-125
1 250	1 600	0	-160

Table 3.9 Tolerance of thrust tapered roller bearings of metric series

Table 3.9 (1) Shaft raceway washer

Unit: μm

Nominal bearing bore diameter		Deviation of mean bore diameter in a single plane Δd_{mp}		Deviation of the actual bearing height ΔT_s	
Over	Incl.	Upper	Lower	Upper	Lower
80	120	0	-20	0	-150
120	180	0	-25	0	-175
180	250	0	-30	0	-200
250	315	0	-35	0	-225
315	400	0	-40	0	-300
400	500	0	-45	0	-350
500	630	0	-50	0	-400
630	800	0	-75	0	-500
800	1 000	0	-100	0	-600

Table 3.9 (2) Housing raceway washer

Unit: μm

Nominal bearing outside diameter		Deviation of mean outside diameter in a single plane ΔD_{mp}	
Over	Incl.	Upper	Lower
180	250	0	-30
250	315	0	-35
315	400	0	-40
400	500	0	-45
500	630	0	-60
630	800	0	-75
800	1 000	0	-100
1 000	1 250	0	-125

Table 3.10 Tolerance of thrust tapered roller bearings of inch series

Table 3.10 (1) Shaft raceway washer

Unit: μm

Nominal bearing bore diameter		Deviation of mean bore diameter in a single plane Δd_{mp}		Deviation of the actual bearing height ΔT_s	
Over	Incl.	Upper	Lower	Upper	Lower
—	304.800	+25	0	+381	-381
304.800	609.600	+51	0	+381	-381
609.600	914.400	+76	0	+381	-381
914.400	1 219.200	+102	0	+381	-381

Table 3.10 (2) Housing raceway washer

Unit: μm

Nominal bearing outside diameter		Deviation of mean outside diameter in a single plane ΔD_{mp}	
Over	Incl.	Upper	Lower
—	304.800	+25	0
304.800	609.600	+51	0
609.600	914.400	+76	0
914.400	1 219.200	+102	0
1 219.200	—	+127	0

3.2 Chamfer measurements and tolerance or allowable values of tapered bore

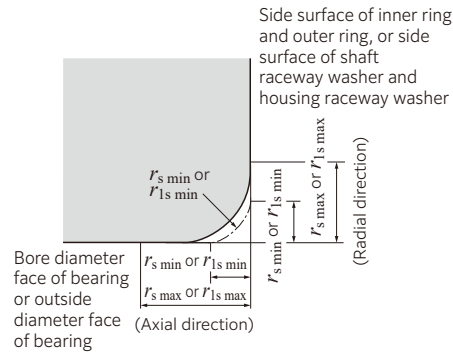


Table 3.11 Allowable critical-value of bearing chamfer

Table 3.11 (1) Radial bearings
(except tapered roller bearing)
Unit: mm

$r_s \min^{(1)}$ or $r_{ls} \min$	Nominal bearing bore diameter d		$r_s \max$ or $r_{ls} \max$	
	Over	Incl.	Radial direction	Axial direction
0.6	—	40	1	2
	40	—	1.3	2
1	—	50	1.5	3
	50	—	1.9	3
1.1	—	120	2	3.5
	120	—	2.5	4
1.5	—	120	2.3	4
	120	—	3	5
2	—	80	3	4.5
	80	220	3.5	5
	220	—	3.8	6
2.1	—	280	4	6.5
	280	—	4.5	7
2.5	—	100	3.8	6
	100	280	4.5	6
	280	—	5	7
3	—	280	5	8
	280	—	5.5	8
4	—	—	6.5	9
5	—	—	8	10
6	—	—	10	13
7.5	—	—	12.5	17
9.5	—	—	15	19
12	—	—	18	24
15	—	—	21	30
19	—	—	25	38

1) These are the allowable minimum dimensions of the chamfer dimension "r" or "r₁" and are described in the dimensional table.

Table 3.11 (2) Tapered roller bearings of metric series
Unit: mm

$r_s \min^{(2)}$ or $r_{ls} \min$	Nominal bearing bore diameter $d^{(3)}$ or nominal bearing outside diameter D		$r_s \max$ or $r_{ls} \max$	
	Over	Incl.	Radial direction	Axial direction
0.3	—	40	0.7	1.4
	40	—	0.9	1.6
0.6	—	40	1.1	1.7
	40	—	1.3	2
1	—	50	1.6	2.5
	50	—	1.9	3
1.5	—	120	2.3	3
	120	250	2.8	3.5
	250	—	3.5	4
2	—	120	2.8	4
	120	250	3.5	4.5
	250	—	4	5
2.5	—	120	3.5	5
	120	250	4	5.5
	250	—	4.5	6
3	—	120	4	5.5
	120	250	4.5	6.5
	250	400	5	7
	400	—	5.5	7.5
4	—	120	5	7
	120	250	5.5	7.5
	250	400	6	8
	400	—	6.5	8.5
5	—	180	6.5	8
	180	—	7.5	9
6	—	180	7.5	10
	180	—	9	11

2) These are the allowable minimum dimensions of the chamfer dimension "r" or "r₁" and are described in the dimensional table.

3) Inner rings shall be in accordance with the division of "d" and outer rings with that of "D".

Note: The standard applies to the bearings whose dimensional series (refer to the dimensional table) are specified in the standard ISO 355 or JIS B 1512-3. For further information concerning bearings outside of these standards or tapered roller bearings using a US customary unit, please contact NTN Engineering.

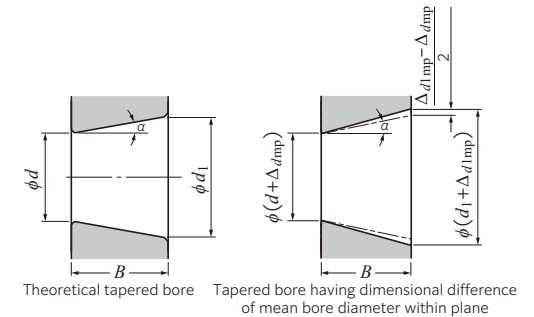


Table 3.12 (1) Tolerance of tapered bores of radial bearings and tapered bores with allowable standard taper ratio 1:12 (Class 0)
Unit: μm

d mm		Δd_{mp}		$\Delta d_{lmp} - \Delta d_{mp}$		$V_{dsp}^{(1)(2)}$
Over	Incl.	Upper	Lower	Upper	Lower	Max.
80	120	+54	0	+35	0	22
120	180	+63	0	+40	0	40
180	250	+72	0	+46	0	46
250	315	+81	0	+52	0	52
315	400	+89	0	+57	0	57
400	500	+97	0	+63	0	63
500	630	+110	0	+70	0	70
630	800	+125	0	+80	0	—
800	1 000	+140	0	+90	0	—
1 000	1 250	+165	0	+105	0	—
1 250	1 600	+195	0	+125	0	—

Table 3.12 (2) Tolerance of tapered bores of radial bearings and tapered bores with allowable standard taper ratio 1:30 (Class 0)
Unit: μm

d mm		Δd_{mp}		$\Delta d_{lmp} - \Delta d_{mp}$		$V_{dsp}^{(1)(2)}$
Over	Incl.	Upper	Lower	Upper	Lower	Max.
80	120	+20	0	+35	0	22
120	180	+25	0	+40	0	40
180	250	+30	0	+46	0	46
250	315	+35	0	+52	0	52
315	400	+40	0	+57	0	57
400	500	+45	0	+63	0	63
500	630	+50	0	+70	0	70

1) Applies to all radial flat planes of tapered bores.

2) Does not apply to diameter series 7 and 8.

Note: Quantifiers

For a standard taper ratio of $\frac{1}{12}$, $d_1 = d + \frac{1}{12} B$

For a standard taper ratio of $\frac{1}{30}$, $d_1 = d + \frac{1}{30} B$

Δd_{mp} : Dimensional difference of the mean bore diameter within the flat surface at the theoretical small end of the tapered bore

Δd_{lmp} : Dimensional difference of the mean bore diameter within the flat surface at the theoretical large end of the tapered bore

V_{dsp} : Unevenness of the bore diameter with the flat surface

B : Nominal width of inner ring

α : $\frac{1}{2}$ of the tapered bore's standard taper angle

For a standard taper ratio of $\frac{1}{12}$
 $\alpha = 2^\circ 23' 9.4''$
 $= 2.38594^\circ$
 $= 0.041643 \text{ rad}$

For a standard taper ratio of $\frac{1}{30}$
 $\alpha = 0^\circ 57' 17.4''$
 $= 0.95484^\circ$
 $= 0.016665 \text{ rad}$

4. Bearing fits

4.1 Resultant fits

For rolling bearings, it is necessary to fix inner and outer rings on the shaft or in the housing so that relative movement does not occur between fitting surfaces during operation or under load. This relative movement between the mating surfaces of the bearing and the shaft or housing can occur in a radial direction, an axial direction, or in the direction of rotation. Types of resultant fit include **tight**, **transition** and **loose fits**, which describe whether or not there is interference between the bearing and the shaft or housing. It is also necessary to select a reliable axial fixing method such as tightening nuts, bolts, retaining rings, etc. For more information on fixing of bearings, see section "8.1 Fixing of bearings".

The most effective way to fix the mating surfaces between a bearing and shaft or housing is to apply a "tight fit". The advantage of a tight fit for thin walled bearings is that it provides uniform load support over the entire ring circumference without any loss of load carrying capacity. However, with a tight fit, ease of installation and disassembly is lost; and when using a non-separable bearing as the floating-side bearing, axial displacement is not possible. For this reason, a tight fit cannot be recommended in all cases.

Table 4.1 Radial load and bearing fit

Design	Bearing rotation	Ring load	Fit
Static load 	 Inner ring: Rotating Outer ring: Stationary	Rotating inner ring load Static outer ring load	Inner ring: Tight fit Outer ring: Loose fit
Unbalanced load 	 Inner ring: Stationary Outer ring: Rotating		
Static load 	 Inner ring: Stationary Outer ring: Rotating	Static inner ring load Rotating outer ring load	Inner ring: Loose fit Outer ring: Tight fit
Unbalanced load 	 Inner ring: Rotating Outer ring: Stationary		

4.2 The necessity of a proper fit

In some cases, an improper fit may lead to damage and shorten bearing life. Therefore it is necessary to carefully select the proper fit. Some possible bearing failures caused by an improper fit are listed below.

- Raceway cracking, early spalling and displacement of raceway
- Raceway and shaft or housing abrasion caused by creeping and fretting corrosion
- Seizing caused by negative internal clearances
- Increased noise and deteriorated rotational accuracy due to raceway groove deformation

4.3 Fit selection

Selection of a proper fit is dependent upon thorough analysis of bearing operating conditions, including consideration of:

- Shaft and housing material, wall thickness, surface finish accuracy, etc.
- Machinery operating conditions (nature and magnitude of load, rotational speed, temperature, etc.)

4.3.1 "Tight fit" or "Loose fit"

- (1) For raceways under rotating loads, a **tight fit** is necessary (refer to Table 4.1). "Raceways under rotating loads" refers to raceways receiving loads rotating relative

to their radial direction.

For raceways under static loads, on the other hand, a **loose fit** is sufficient.

- (2) For non-separable bearings, such as deep groove ball bearings, it is generally recommended that either the inner ring or outer ring be given a **loose fit**.

4.3.2 Recommended fits

Bearing fit is governed by the tolerances selected for bearing shaft diameters and housing bore diameters.

Widely used fits for Class 0 tolerance bearings and various shaft and housing bore diameter tolerances are shown in Fig. 4.1.

Generally used, standard fits for most types of bearings and operating conditions are shown in Table 4.2 through Table 4.5.

Table 4.2: Fits for radial bearings

Table 4.3: Fits for thrust bearings

Table 4.4: Fits for electric motor bearings

Table 4.5: Inch series tapered roller bearings
Fits of (ANSI/ABMA CLASS 4)

Table 4.6: Inch series tapered roller bearings
Fits of (ANSI/ABMA CLASS 3,
CLASS 0)

4.3.3 Interference minimum and maximum values

The following points should be considered when it is necessary to calculate the interference for an application:

- Lower limit values
 - 1) Reduced interference due to radial loads
 - 2) Reduced interference due to differences between bearing temperature and ambient temperature
 - 3) Reduced interference due to variation in mating surface
 - 4) Reduced interference due to deformation
- Upper limit values

The upper limit value should not exceed 1/ 1 000 of the shaft diameter.

Required interference calculations are shown in the next page.

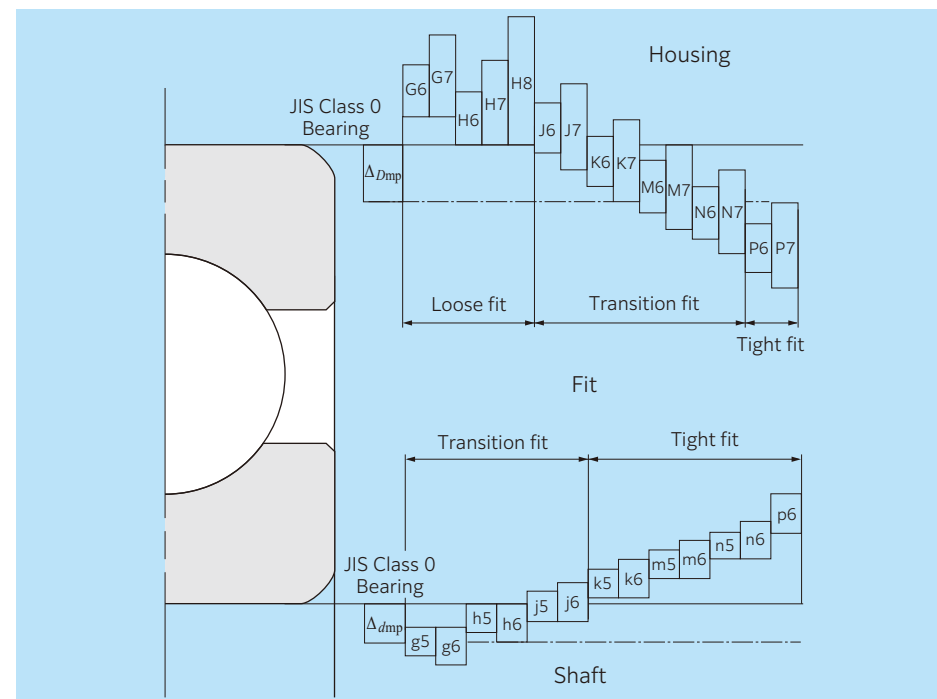


Fig 4.1 State of resultant fit

(1) Mating surface variation and interference

Interference decreases because the mating surface is smoothed by the resultant fit (surface roughness is reduced). The amount the interference decreases depends on the roughness of the mating surfaces. It is generally necessary to anticipate the following decrease in interference.

For ground surface finish: 1.0 to 2.5 μm

For turned surface finish: 5.0 to 7.0 μm

The interference including this decrease amount is called effective interference.

(2) Radial loads and required interference

Interference of the inner ring and shaft decreases when a radial load is applied to the bearing. The interference required for installation to solid shafts is expressed by formula (4.1) and formula (4.2) for each load condition.

General applications ($F_r \leq 0.3C_{0r}$)

$$\Delta d_F = 0.08(d \cdot F_r / B)^{1/2} \dots\dots\dots (4.1)$$

Under heavy load conditions ($F_r > 0.3C_{0r}$)

$$\Delta d_F = 0.02(F_r / B) \dots\dots\dots (4.2)$$

Where:

Δd_F : Required effective interference according to radial load, μm

d : Bearing bore diameter, mm

B : Inner ring width, mm

F_r : Actual radial load, N

C_{0r} : Basic static load rating, N

For hollow shafts, please contact **NTN Engineering**.

(3) Temperature difference and required interference

Interference between inner rings and steel shafts is reduced as a result of temperature increases (difference between bearing temperature and ambient temperature, ΔT) caused by bearing rotation. Calculation of the minimum required amount of interference in such cases is shown in formula (4.3).

$$\Delta d_T = 0.0015 \cdot d \cdot \Delta T \dots\dots\dots (4.3)$$

Where:

Δd_T : Required effective interference for temperature difference, μm

ΔT : Difference between bearing temperature and ambient temperature, °C

d : Bearing bore diameter, mm

(4) Maximum interference

When bearing rings are installed with an interference fit, tensile or compressive stress

may occur along their raceways. If interference is too great, this may cause damage to the rings and reduce bearing life. The maximum stress due to the resultant fit must not exceed approximately 127 MPa for safety. If the value is to be exceeded, consult **NTN Engineering**.

(5) Interference change amount when materials other than steel are used for shafts and housings

When materials other than steel are used for shafts and housings, the fits between the inner ring and the shaft and the outer ring and the housing change because of the difference in the linear expansion coefficient of each material as the temperature rises during the rotation of the bearing. Therefore, it is necessary to set the resultant fit with linear expansion coefficients in consideration. Calculation of the change in interference is shown in formula (4.4).

$$\Delta d_{TE} = (\alpha_1 - \alpha_2) \times d \times \Delta T \dots\dots\dots (4.4)$$

Where:

Δd_{TE} : Change in interference caused by the difference in the linear expansion coefficients, mm

α_1 : Bearing linear expansion coefficient, 1/°C

α_2 : Shaft and housing linear expansion coefficient, 1/°C

d : Reference dimension of resultant fit, mm

ΔT : Temperature increase by bearing rotation, °C

4.3.4 Other details

- (1) Large interference fits are recommended for,
 - Operating conditions with large vibrations or shock loads
 - Applications using hollow shafts or housings with thin walls
 - Applications using housings made of light alloys or plastic
- (2) Small interference fits are preferable for,
 - Applications requiring high running accuracy
 - Applications using small-sized bearings or thin-walled bearings
- (3) A particular type of fit is recommended for SL-type cylindrical roller bearings (refer to page C-73).
- (4) Bearing dimensions are measured and managed at a temperature of 20 °C.

Table 4.2 General standards for radial bearing fits (JIS Class 0, 6X, 6)

Table 4.2 (1) Tolerance class of shafts commonly used for radial bearings (Classes 0, 6X and 6)

Condition		Ball bearing		Cylindrical roller bearing Tapered roller bearing		Spherical roller bearing		Shaft tolerance class	Remarks	
		Shaft diameter (mm)								
		Over	Incl.	Over	Incl.	Over	Incl.			
Cylindrical bore bearing (Classes 0, 6X and 6)										
Inner ring rotational load or load of undetermined direction	Light load ¹⁾ or Fluctuating load	18 100 —	100 200 —	— 40 140	40 140 200	— — —	— — —	js6 k6 m6	When greater accuracy is required js5, k5, and m5 may be substituted for js6, k6, and m6.	
	Normal load ¹⁾	18 100 140 200 — —	100 140 200 280 — —	— 40 100 140 200 —	40 100 140 200 400 —	— 40 65 100 140 280 500	40 65 100 140 280 500	k5 m5 m6 n6 p6 r6	Alteration of inner clearances to accommodate fit is not a consideration with single-row angular contact bearings and tapered roller bearings. Therefore, k5 and m5 may be substituted for k6 and m6.	
		Heavy load ¹⁾ or Impact load	— — —	— — —	50 140 200	140 200 —	50 100 140	100 140 200	n6 p6 r6	Use bearings with larger internal clearances than CN clearance bearings.
		Static inner ring load	Inner ring must move easily over shaft	Overall shaft diameter						g6
Inner ring does not have to move easily over shaft			Overall shaft diameter						h6	When greater accuracy is required use h5.
Center axial load		Overall shaft diameter						js6	Generally, shaft and inner rings are not fixed using resultant fits.	
Tapered bore bearing (Class 0) (with adapter or withdrawal sleeve)										
Full load		Overall shaft diameter						h9/IT5 ²⁾	h10/IT7 ²⁾ will suffice for power transmitting shafts.	

Table 4.2 (2) Fit with shaft [fits for tapered bore bearings (Class 0) with adapter assembly/withdrawal sleeve]

Full load	All bearing types	Tolerance class	h9 /IT5 ²⁾ h10/IT7 ²⁾	General applications Transmission shafts, etc.
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1) Standards for light loads, normal loads, and heavy loads

- Light loads: dynamic equivalent radial load $\leq 0.05C_r$
- Normal loads: $0.05C_r < \text{dynamic equivalent radial load} \leq 0.10C_r$
- Heavy loads: $0.10C_r < \text{dynamic equivalent radial load}$

2) IT5 and IT7 show shaft roundness tolerances, cylindricity tolerances, and related values.

Note: 1. All values and fits listed in the above tables are for solid steel shafts.

2. For ULTAGE™ series spherical roller bearings, refer to **Table 2** (page B-181) in Section "Spherical Roller Bearings".

Table 4.2 (3) Tolerance class of housing bores commonly used for radial bearings
(Classes 0, 6X and 6)

Conditions			Housing bore tolerance class	Remarks
Housing	Load type, etc.	Outer ring axial direction movement ³⁾		
Single housing or Split housing	Static outer ring load	All types of loads	H7	G7 can be used for large bearings or bearings with large temperature differential between the outer ring and housing.
		Light load ¹⁾ or normal load ¹⁾	H8	—
		Shaft and inner ring become hot	G7	F7 can be used for large bearings or bearings with large temperature differential between the outer ring and housing.
		Requires precise rotation under light or normal loads	K6	Primarily applies to roller bearings.
Single housing	Indeterminate load	As a rule, it cannot move.	JS6	Primarily applies to ball bearings.
		Yes	JS6	Primarily applies to ball bearings.
		Requires low noise operation	H6	—
	Indeterminate load	Light load or normal load	JS7	If high accuracy is required, JS6 and K6 are used in place of JS7 and K7.
		Normal load or heavy load ¹⁾	K7	
		High impact load	M7	—
	Rotating outer ring load	Light load or fluctuating load	M7	—
		Normal load or heavy load	N7	Primarily applies to ball bearings.
		Heavy load or large impact load with thin wall housing ²⁾	P7	Primarily applies to roller bearings.

1) Standards for light loads, normal loads, and heavy loads

- Light loads: dynamic equivalent radial load $\leq 0.05C_r$
- Normal loads: $0.05C_r < \text{dynamic equivalent radial load} \leq 0.10C_r$
- Heavy loads: $0.10C_r < \text{dynamic equivalent radial load}$

2) The axial direction needs to be secured because the outer ring may move in the shaft direction, causing problems, depending on the use (Example: planetary gear, etc.).

3) Indicates whether or not outer ring axial movement is possible with non-separable type bearings.

Note: 1. All values and fits listed in the above tables are for cast iron or steel housings.

- If only a center axial load is applied to the bearing, select a tolerance class that provides clearance for the outer ring in the radial direction.

Table 4.3 Standard fits for thrust bearings (JIS Class 0 and 6)
Table 4.3 (1) Shaft fits

Bearing type	Load conditions		Fit	Shaft diameter mm Over Incl.	Tolerance class
All thrust bearings	Centered axial load only		Transition fit	Overall shaft diameter	js6 or h6
Thrust spherical roller bearings	Combined load	Static inner ring load	Transition fit	Overall shaft diameter	js6
		Rotating inner ring load or Indeterminate load	Transition fit	- 200 200 - 400	k6 or js6 m6 or k6
			Tight fit	400 -	n6 or m6

Table 4.3 (2) Housing fits

Bearing type	Load conditions		Fit	Tolerance class	Remarks
All thrust bearings	Centered axial load only		Loose fit	Select a tolerance class that will provide clearance between outer ring and housing.	
				H8	Greater accuracy required with thrust ball bearings
Thrust spherical roller bearings	Combined load	Static outer ring load		H7	——
		Indeterminate load or Rotating outer ring load	K7	Normal operating conditions	
			M7	For relatively large radial loads	

Note: All values and fits listed in the above tables are for cast iron or steel housings.

Table 4.4: Fits for electric motor bearings

Bearing type	Shaft fits		Housing fits	
	Shaft diameter mm Over Incl.	Tolerance class	Housing bore diameter	Tolerance class
Deep groove ball bearings	18 - 100 100 - 160	k5 m5	All sizes	H6 or J6
Cylindrical roller bearings	40 - 160 160 - 200	m5 n6	All sizes	H6 or J6

Table 4.5 General fit standards for tapered roller bearings using US customary unit (ANSI Class 4)

Table 4.5 (1) Fit with shaft

Unit: μm

Operating conditions	Nominal bearing bore diameter d mm		Bore diameter tolerance Δd_s		Shaft diameter tolerance		Fit ¹⁾	Remarks
	Over	Incl.	Upper	Lower	Upper	Lower		
Rotating inner ring load	Normal load	76.2 304.8 304.8 609.6 609.6 914.4	+25 0 +51 0 +76 0	0	+ 64 + 38 +127 + 76 +191 +114		64T - 13T 127T - 25T 191T - 38T	Applicable when a slight impact load is applied as well.
	Heavy load Impact load	76.2 304.8 304.8 609.6 609.6 914.4	+25 0 +51 0 +76 0	0	0.5 μm mean interference per 1 mm of inner ring bore diameter. Minimum interference is 25 μm . Tolerance for the shaft is adjusted to match tolerance of bearing bore diameter. +457 +381 457T - 305T			
Rotating outer ring load	Inner ring does not have to move easily over shaft with an ordinary load.	76.2 304.8 304.8 609.6 609.6 914.4	+25 0 +51 0 +76 0	0	+ 25 0 + 51 0 + 76 0		25T - 25L 51T - 51L 76T - 76L	Not applicable when impact load is applied.
	Inner ring must move easily over shaft with an ordinary load.	76.2 304.8 304.8 609.6 609.6 914.4	+25 0 +51 0 +76 0	0	0 - 25 0 - 51 0 - 76		0 - 51L 0 - 102L 0 - 152L	

Table 4.5 (2) Fit with housing

Unit: μm

Operating conditions	Nominal bearing outside diameter D mm		Outside diameter dimensional tolerance ΔD_s		Housing bore diameter tolerance		Fit ¹⁾	Types of fits
	Over	Incl.	Upper	Lower	Upper	Lower		
Rotating inner ring load	When used on floating or fixed side	76.2 127.0 127.0 304.8 304.8 609.6 609.6 914.4	+25 0 +25 0 +51 0 +76 0	0	+ 76 + 51 + 76 + 51 +152 +102 +229 +152		25L - 76L 25L - 76L 51L - 152L 76L - 229L	Loose fit
	When outer ring is adjusted in axial direction	76.2 127.0 127.0 304.8 304.8 609.6 609.6 914.4	+25 0 +25 0 +51 0 +76 0	0	+ 25 0 + 51 0 + 76 + 25 +127 + 51		25T - 25L 25T - 51L 25T - 76L 25T - 127L	Transition fit
Rotating outer ring load	When outer ring is not adjusted in axial direction	76.2 127.0 127.0 304.8 304.8 609.6 609.6 914.4	+25 0 +25 0 +51 0 +76 0	0	- 25 - 51 - 25 - 51 - 25 - 76 - 25 -102		76T - 25T 76T - 25T 127T - 25T 178T - 25T	Tight fit
	When outer ring is not adjusted in axial direction	76.2 127.0 127.0 304.8 304.8 609.6 609.6 914.4	+25 0 +25 0 +51 0 +76 0	0	- 25 - 51 - 25 - 76 - 25 -102		76T - 25T 76T - 25T 127T - 25T 178T - 25T	

1) Fit symbol "L" indicates clearance and "T" indicates interference.

Table 4.6 General fit standards for tapered roller bearings using US customary unit (ANSI Classes 3 and 0)

Table 4.6 (1) Fit with shaft

Unit: μm

Operating conditions	Nominal bearing bore diameter d mm		Bore diameter tolerance Δd_s		Shaft diameter tolerance		Fit ¹⁾
	Over	Incl.	Upper	Lower	Upper	Lower	
Rotating inner ring load	Precision machine tool spindles	— 304.8 304.8 609.6 609.6 914.4	+13 0 +25 0 +38 0	0	+ 30 +18 + 64 +38 +102 +64		30T - 5T 64T - 13T 102T - 25T
	Heavy load Impact load High-speed rotation	— 304.8 304.8 609.6 609.6 914.4	+13 0 +25 0 +38 0	0	Minimum interference is 0.25 μm per 1 mm of inner ring bore diameter		
Rotating outer ring load	Precision machine tool spindles	— 304.8 304.8 609.6 609.6 914.4	+13 0 +25 0 +38 0	0	+ 30 +18 + 64 +38 +102 +64		30T - 5T 64T - 13T 102T - 25T

Note: For Class 0, nominal bore diameter d applies to 304.8 mm or less.

Table 4.6 (2) Fit with housing

Unit: μm

Operating conditions	Nominal bearing outside diameter D mm		Outside diameter dimensional tolerance ΔD_s		Housing bore diameter tolerance		Fit ¹⁾	Types of fits
	Over	Incl.	Upper	Lower	Upper	Lower		
Rotating inner ring load	When used for floating-side	— 152.4 152.4 304.8 304.8 609.6 609.6 914.4	+13 0 +13 0 +25 0 +38 0	0	+38 +25 +38 +25 +64 +38 +89 +51		13L - 38L 13L - 38L 13L - 64L 13L - 89L	Loose fit
	When used for fixed side	— 152.4 152.4 304.8 304.8 609.6 609.6 914.4	+13 0 +13 0 +25 0 +38 0	0	+25 +13 +25 +13 +51 +25 +76 +38		0 - 25L 0 - 25L 0 - 51L 0 - 76L	
Rotating outer ring load	When outer ring is adjusted in axial direction	— 152.4 152.4 304.8 304.8 609.6 609.6 914.4	+13 0 +13 0 +13 0 +38 0	0	+13 0 +25 0 +25 0 +38 0		13T - 13L 13T - 25L 25T - 25L 38T - 38L	Transition fit
	When outer ring is not adjusted in axial direction	— 152.4 152.4 304.8 304.8 609.6 609.6 914.4	+13 0 +13 0 +25 0 +38 0	0	0 -13 0 -25 0 -25 0 -38		25T - 0 38T - 0 51T - 0 76T - 0	Tight fit
Rotating outer ring load	Normal load When outer ring is not adjusted in axial direction	— 152.4 152.4 304.8 304.8 609.6 609.6 914.4	+13 0 +13 0 +25 0 +38 0	0	-13 -25 -13 -38 -13 -38 -13 -51		38T - 13T 51T - 13T 64T - 13T 89T - 13T	

1) Fit symbol "L" indicates clearance and "T" indicates interference.

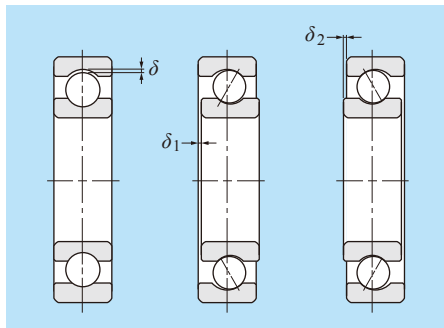
Note: For Class 0, nominal bearing outside diameter D applies to 304.8 mm or less.

5. Bearing internal clearance

5.1 Bearing internal clearance

Bearing internal clearance is the amount of internal free movement before mounting. As shown in **Fig. 5.1**, when either the inner ring or the outer ring is fixed and the other ring is free to move, displacement can take place in either an axial or radial direction. This amount of displacement (radially or axially) is termed the internal clearance and, depending on the direction, is called **the radial internal clearance** or **the axial internal clearance**.

The internal clearance values for each bearing class are shown in **Table 5.2** through **Table 5.9**.



Radial internal clearance = δ Axial internal clearance = $\delta_1 + \delta_2$

Fig. 5.1 Internal clearance

5.2 Selection of internal clearance

The **internal clearance** of a bearing **under operating conditions** (effective clearance) is **usually smaller** than the initial clearance before being installed and operated. This is due to several factors including bearing fit, the difference in temperature between the inner and outer rings, etc. As a bearing's operating clearance has an effect on bearing life, heat generation, vibration, noise, etc.; care must be taken in selecting the most suitable operating clearance.

5.2.1 Criteria for selecting bearing internal clearance

A bearing's life is theoretically at its maximum when operating clearance is slightly negative in steady operation. However, in reality it is difficult to constantly maintain this optimal condition. If the negative clearance becomes larger by fluctuating operating conditions, heat will be produced and the life will decrease severely. Under normal circumstances, a **study should be performed to select an operating clearance slightly larger than zero**.

For ordinary operating conditions, use fitting for ordinary loads. If rotational speed and operating temperature are ordinary, selecting normal clearance enables you to obtain the proper operating clearance.

Table 5.1 gives examples applying internal clearances other than CN (normal) clearance.

Table 5.1 Examples of applications where bearing clearances other than CN (normal) clearance are used

Operating conditions	Applications	Selected clearance
With a heavy or shock load, high fit.	Railway vehicle axles	C3
	Vibrating screens	C3, C4
With an indeterminate load, both inner and outer rings are tight fit.	Railway vehicle traction motors	C4
	Tractors and final reduction gear	C4
Shaft or inner ring is heated.	Paper making machines and driers	C3, C4
	Table rollers for rolling mill	C3
Adjustment of clearance to minimize shaft runout.	Main spindles of lathes (Double-row cylindrical roller bearings)	C9NA, C0NA

5.2.2 Calculation of operating clearance

Operating clearance of a bearing can be calculated from initial bearing internal clearance and considering the decrease in clearance due to fitting and the difference in temperature of the inner and outer rings using formula (5.1).

$$\Delta_e = \Delta_0 - (\delta_f + \delta_t) = \Delta_f - \delta_t \dots\dots\dots (5.1)$$

Where:

- Δ_e : Effective internal clearance, mm
- Δ_0 : Bearing internal clearance (initial), mm
- Δ_f : Residual clearance (clearance after preloading), mm
- δ_f : Reduced amount of clearance due to fitting, mm
- δ_t : Reduced amount of clearance due to temperature differential of inner and outer rings, mm

(1) Reduced internal clearance due to fitting

When bearings are installed with interference fits on shafts and in housings, the inner ring will expand and the outer ring will contract; **thus reducing the bearings' internal clearance**. The amount of expansion or contraction varies depending on the shape of the bearing, the shape of the shaft or housing, dimensions of the respective parts, and the type of materials used. The differential can range from **approximately 70 to 90 % of the effective interference**. Therefore, the reduced amount of clearance due to fitting is expressed by formula (5.2).

$$\delta_f = (0.70 \text{ to } 0.90) \Delta d_{\text{eff}} \dots\dots\dots (5.2)$$

Where:

- δ_f : Reduced amount of clearance due to interference, mm
- Δd_{eff} : Effective interference, mm

(2) Reduced internal clearance due to inner/outer ring temperature difference

During operation, normally the outer ring will range from 5 to 10 °C cooler than the inner ring or rotating parts. However, if the cooling effect of the housing is large, the shaft is connected to a heat source, or a heated substance is conducted through the hollow shaft; the temperature difference between the two rings can be even greater. **The amount of internal clearance is thus further reduced** by the differential expansion of the two rings. The reduced amount is expressed by formula (5.3).

$$\delta_t = \alpha \cdot \Delta T \cdot D_o \dots\dots\dots (5.3)$$

Where:

- δ_t : Reduced amount of clearance due to temperature differential of inner and outer rings, mm
- α : Linear thermal expansion coefficient of bearing material, $12.5 \times 10^{-6}/^\circ\text{C}$
- ΔT : Inner/outer ring temperature differential, °C
- D_o : Outer ring raceway diameter, mm

Outer ring raceway diameter, D_o , values can be approximated by using formula (5.4) or formula (5.5).

For ball bearings and spherical roller bearings,

$$D_o = 0.20(d + 4.0D) \dots\dots\dots (5.4)$$

For roller bearings (except spherical roller bearings),

$$D_o = 0.25(d + 3.0D) \dots\dots\dots (5.5)$$

d : Bearing bore diameter, mm

D : Bearing outside diameter, mm

For the ULTAGE™ series bearings, consult **NTN Engineering**.

Note that the formula in item 5.2.2 only applies to steel bearings, shafts and housings.

Table 5.2 Radial internal clearance of deep groove ball bearings

Unit: μm

Nominal bearing bore diameter d mm		C2		CN		C3		C4		C5	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
80	100	1	18	12	36	30	58	53	84	75	120
100	120	2	20	15	41	36	66	61	97	90	140
120	140	2	23	18	48	41	81	71	114	105	160
140	160	2	23	18	53	46	91	81	130	120	180
160	180	2	25	20	61	53	102	91	147	135	200
180	200	2	30	25	71	63	117	107	163	150	230
200	225	2	35	25	85	75	140	125	195	175	265
225	250	2	40	30	95	85	160	145	225	205	300
250	280	2	45	35	105	90	170	155	245	225	340
280	315	2	55	40	115	100	190	175	270	245	370
315	355	3	60	45	125	110	210	195	300	275	410
355	400	3	70	55	145	130	240	225	340	315	460
400	450	3	80	60	170	150	270	250	380	350	520
450	500	3	90	70	190	170	300	280	420	390	570
500	560	10	100	80	210	190	330	310	470	440	630
560	630	10	110	90	230	210	360	340	520	490	700
630	710	20	130	110	260	240	400	380	570	540	780
710	800	20	140	120	290	270	450	430	630	600	860
800	900	20	160	140	320	300	500	480	700	670	960
900	1 000	20	170	150	350	330	550	530	770	740	1 040
1 000	1 120	20	180	160	380	360	600	580	850	820	1 150
1 120	1 250	20	190	170	410	390	650	630	920	890	1 260

Table 5.3 Radial internal clearance of double row and duplex angular contact ball bearings

Unit: μm

Nominal bearing bore diameter d mm		C1		C2		CN		C3		C4	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
80	100	3	13	13	22	22	40	40	60	95	120
100	120	3	15	15	30	30	50	50	75	110	140
120	150	3	16	16	33	35	55	55	80	130	170
150	180	3	18	18	35	35	60	60	90	150	200
180	200	3	20	20	40	40	65	65	100	180	240
200	225	3	25	25	50	50	75	75	115	210	270
225	250	3	25	25	50	50	75	80	130	230	300
250	280	3	30	30	55	55	85	90	150	260	340
280	315	3	30	30	55	55	85	100	170	300	380
315	400	—	—	40	65	60	85	110	180	—	—
400	500	—	—	40	65	60	85	110	180	—	—

Note: 1. The clearance group in the table is applied only to contact angles in the table below.
2. This table shows **NTN** standard clearances.

Contact angle code	Nominal contact angle	Applicable clearance ²⁾
C	15°	C1, C2
A ¹⁾	30°	C2, CN, C3
B	40°	CN, C3, C4

1) Usually not to be indicated

2) For information concerning clearance other than applicable clearance, please contact **NTN** Engineering.

Table 5.4 Radial internal clearance of bearings for electric motor

Unit: μm

Nominal bearing bore diameter d mm		CM			
Over	Incl.	Deep groove ball bearings		Cylindrical roller bearings	
		Min.	Max.	Min.	Max.
80	100	18	30	35	55
100	120	18	30	35	60
120	140	24	38	40	65
140	160	24	38	50	80
160	180	—	—	60	90
180	200	—	—	65	100

Note: 1. Suffix CM is added to bearing numbers.

Example: 6220 CM

2. Clearance not interchangeable for cylindrical roller bearings.

Table 5.5 Interchangeable radial internal clearance for cylindrical roller bearing (cylindrical bore)

Unit: μm

Nominal bearing bore diameter d mm		C2		CN		C3		C4		C5	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
80	100	15	50	50	85	75	110	105	140	155	190
100	120	15	55	50	90	85	125	125	165	180	220
120	140	15	60	60	105	100	145	145	190	200	245
140	160	20	70	70	120	115	165	165	215	225	275
160	180	25	75	75	125	120	170	170	220	250	300
180	200	35	90	90	145	140	195	195	250	275	330
200	225	45	105	105	165	160	220	220	280	305	365
225	250	45	110	110	175	170	235	235	300	330	395
250	280	55	125	125	195	190	260	260	330	370	440
280	315	55	130	130	205	200	275	275	350	410	485
315	355	65	145	145	225	225	305	305	385	455	535
355	400	100	190	190	280	280	370	370	460	510	600
400	450	110	210	210	310	310	410	410	510	565	665
450	500	110	220	220	330	330	440	440	550	625	735
500	560	120	240	240	360	360	480	480	600	—	—
560	630	140	260	260	380	380	500	500	620	—	—
630	710	145	285	285	425	425	565	565	705	—	—
710	800	150	310	310	470	470	630	630	790	—	—
800	900	180	350	350	520	520	690	690	860	—	—
900	1 000	200	390	390	580	580	770	770	960	—	—
1 000	1 120	220	430	430	640	640	850	850	1 060	—	—
1 120	1 250	230	470	470	710	710	950	950	1 190	—	—

Note: For nominal bearing bore diameter d exceeding 500 mm, the **NTN** standard is followed.

Table 5.6 Non-interchangeable radial internal clearance for cylindrical roller bearings

Nominal bearing bore diameter <i>d</i> mm		Cylindrical bore bearing											
		C1NA		C2NA		NA ¹⁾		C3NA		C4NA		C5NA	
		Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
Over	Incl.												
80	100	10	25	25	45	45	70	80	105	105	125	155	180
100	120	10	25	25	50	50	80	95	120	120	145	180	205
120	140	15	30	30	60	60	90	105	135	135	160	200	230
140	160	15	35	35	65	65	100	115	150	150	180	225	260
160	180	15	35	35	75	75	110	125	165	165	200	250	285
180	200	20	40	40	80	80	120	140	180	180	220	275	315
200	225	20	45	45	90	90	135	155	200	200	240	305	350
225	250	25	50	50	100	100	150	170	215	215	265	330	380
250	280	25	55	55	110	110	165	185	240	240	295	370	420
280	315	30	60	60	120	120	180	205	265	265	325	410	470
315	355	30	65	65	135	135	200	225	295	295	360	455	520
355	400	35	75	75	150	150	225	255	330	330	405	510	585
400	450	45	85	85	170	170	255	285	370	370	455	565	650
450	500	50	95	95	190	190	285	315	410	410	505	625	720
500	560	—	—	100	210	210	320	350	450	450	550	720	815
560	630	—	—	110	230	230	350	380	500	500	615	800	910
630	710	—	—	130	260	260	400	435	570	570	695	900	1 030
710	800	—	—	140	290	290	450	485	635	635	780	1 000	1 140
800	900	—	—	160	330	330	500	540	700	700	860	1 130	1 290
900	1 000	—	—	180	360	360	560	600	780	780	970	1 270	1 440
1 000	1 120	—	—	200	400	400	620	670	900	900	1 100	1 410	1 620
1 120	1 250	—	—	220	440	440	690	750	1 000	1 000	1 220	1 580	1 820

1) For bearings with normal clearance, only NA is added to bearing numbers. Example: NU320NA

Table 5.7 Axial internal clearance for double row and duplex tapered roller bearings (metric series)

Table 5.7 (1) Contact angle $\alpha \leq 27^\circ$

Nominal bearing bore diameter <i>d</i> mm		Contact angle $\alpha \leq 27^\circ$ ($e \leq 0.76$)							
		C2		CN		C3		C4	
		Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
Over	Incl.								
80	100	45	150	150	260	280	390	390	500
100	120	45	175	175	305	350	480	455	585
120	140	45	175	175	305	390	520	500	630
140	160	60	200	200	340	400	540	520	660
160	180	80	220	240	380	440	580	600	740
180	200	100	260	260	420	500	660	660	820
200	225	120	300	300	480	560	740	720	900
225	250	160	360	360	560	620	820	820	1 020
250	280	180	400	400	620	700	920	920	1 140
280	315	200	440	440	680	780	1 020	1 020	1 260
315	355	220	480	500	760	860	1 120	1 120	1 380
355	400	260	560	560	860	980	1 280	1 280	1 580
400	500	300	600	620	920	1 100	1 400	1 440	1 740
500	560	350	650	750	1 050	1 250	1 550	1 650	1 950
560	630	400	700	850	1 150	1 400	1 700	1 850	2 150
630	710	500	850	1 000	1 350	1 650	2 000	2 100	2 450
710	800	550	950	1 100	1 500	1 800	2 200	2 300	2 700
800	900	650	1 050	1 250	1 650	2 000	2 400	2 550	2 950
900	1 000	700	1 100	1 400	1 800	2 200	2 600	2 900	3 300
1 000	1 120	750	1 250	1 500	2 000	2 500	3 000	3 250	3 750
1 120	1 250	850	1 350	1 700	2 200	2 850	3 350	3 700	4 200
1 250	1 400	1 000	1 500	2 000	2 500	3 000	3 500	4 000	4 500

Note: 1. This table applies to bearings contained in the catalog. For information concerning other bearings or bearings using US customary unit, please contact NTN Engineering.

2. The correlation of axial internal clearance (Δ_a) and radial internal clearance (Δ_r) is expressed as $\Delta_r = 0.667 \times e \times \Delta_a$.
e: Constant (see dimension table)

3. The table does not apply to the bearing series 329X, 330, 322C, and 323C, 303C, and T4CB.

Unit: μm

Tapered bore bearing												Nominal bearing bore diameter	
C9NA 2)		C0NA 2)		C1NA		C2NA		NA 1)		C3NA		d mm	
Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Over	Incl.
10	25	20	35	25	45	45	70	80	105	105	125	80	100
10	25	20	35	25	50	50	80	95	120	120	145	100	120
15	30	25	40	30	60	60	90	105	135	135	160	120	140
15	35	30	45	35	65	65	100	115	150	150	180	140	160
15	35	30	45	35	75	75	110	125	165	165	200	160	180
20	40	30	50	40	80	80	120	140	180	180	220	180	200
20	45	35	55	45	90	90	135	155	200	200	240	200	225
25	50	40	65	50	100	100	150	170	215	215	265	225	250
25	55	40	65	55	110	110	165	185	240	240	295	250	280
30	60	45	75	60	120	120	180	205	265	265	325	280	315
30	65	45	75	65	135	135	200	225	295	295	360	315	355
35	75	50	90	75	150	150	225	255	330	330	405	355	400
45	85	60	100	85	170	170	255	285	370	370	455	400	450
50	95	70	115	95	190	190	285	315	410	410	505	450	500
—	—	—	—	100	210	210	320	350	450	450	550	500	560
—	—	—	—	110	230	230	350	380	500	500	615	560	630
—	—	—	—	130	260	260	400	435	570	570	695	630	710
—	—	—	—	140	290	290	450	485	635	635	780	710	800
—	—	—	—	160	330	330	500	540	700	700	860	800	900
—	—	—	—	180	360	360	560	600	780	780	970	900	1 000
—	—	—	—	200	400	400	620	670	900	900	1 100	1 000	1 120
—	—	—	—	220	440	440	690	750	1 000	1 000	1 220	1 120	1 250

2) C9NA, C0NA and C1NA clearances are applied only to precision bearings of JIS Class 5 and higher.

Table 5.7 (2) Contact angle $\alpha > 27^\circ$

Unit: μm

Nominal bearing bore diameter <i>d</i> mm		Contact angle $\alpha > 27^\circ$ ($e > 0.76$)							
		C2		CN		C3		C4	
		Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
Over	Incl.								
80	100	20	70	70	120	130	180	180	230
100	120	20	70	70	120	150	200	210	260
120	140	20	70	70	120	160	210	210	260
140	160	30	100	100	160	180	240	240	300
160	180	40	110	110	180	200	270	280	340
180	200	50	120	120	190	230	300	310	380
200	225	60	140	140	200	260	340	340	420
225	250	80	160	170	260	290	380	380	470
250	280	90	190	190	280	320	420	430	520
280	315	90	200	200	310	360	470	470	580
315	355	100	220	230	350	400	510	520	630
355	400	120	260	260	400	450	590	590	730
400	500	140	280	280	420	510	640	650	780
500	630	160	310	310	460	530	650	680	820
630	800	180	350	350	520	590	760	760	930

Note: 1. This table applies to bearings contained in the catalog. For information concerning other bearings or bearings using US customary unit, please contact NTN Engineering.

2. The correlation of axial internal clearance (Δ_a) and radial internal clearance (Δ_r) is expressed as $\Delta_r = 0.667 \times e \times \Delta_a$.
e: Constant (see dimension table)

3. The table does not apply to the bearing series 329X, 330, 322C, and 323C, 303C, and T4CB.

Table 5.8 Axial internal clearance of double row and duplex tapered roller bearings (inch series)
Table 5.8 (1) Contact angle $\alpha < 12^\circ$ Unit: μm

Nominal bearing bore diameter ¹⁾ d mm		Contact angle $\alpha < 12^\circ$ ($e < 0.32$)							
		C2		CN		C3		C4	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
63.5	127	55	165	290	400	400	510	510	620
127	203.2	85	230	320	470	470	620	620	770
203.2	304.8	140	320	370	550	550	730	730	910
304.8	406.4	200	420	660	880	880	1 100	1 100	1 320
406.4	508	260	520	710	970	970	1 230	1 230	1 490
508	609.6	340	640	790	1 090	1 090	1 390	1 390	1 690
609.6	711.2	430	780	1 120	1 470	1 470	1 820	1 820	2 170
711.2	762	—	—	—	—	—	—	—	—
762	914.4	—	—	—	—	—	—	—	—

1) Nominal bore diameter is the minimum size among the same series.
Note: This table shows **NTN** standard clearances.

Table 5.8 (2) $12^\circ \leq$ Contact angle $\alpha < 15^\circ$ Unit: μm

Nominal bearing bore diameter ¹⁾ d mm		$12^\circ \leq$ Contact angle $\alpha < 15^\circ$ ($0.32 \leq e < 0.40$)							
		C2		CN		C3		C4	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
63.5	127	45	135	240	330	330	420	420	510
127	203.2	70	190	270	390	390	510	510	630
203.2	304.8	120	270	310	460	460	610	610	760
304.8	406.4	160	340	550	730	730	910	910	1 090
406.4	508	210	420	590	800	800	1 010	1 010	1 220
508	609.6	280	530	650	900	900	1 150	1 150	1 400
609.6	711.2	350	640	930	1 220	1 220	1 510	1 510	1 800
711.2	762	420	750	990	1 320	1 320	1 650	1 650	1 980
762	914.4	520	890	1 070	1 440	1 440	1 810	1 810	2 180

1) Nominal bore diameter is the minimum size among the same series.
Note: This table shows **NTN** standard clearances.

Table 5.8 (3) $15^\circ \leq$ Contact angle $\alpha < 20^\circ$ Unit: μm

Nominal bearing bore diameter ¹⁾ d mm		$15^\circ \leq$ Contact angle $\alpha < 20^\circ$ ($0.40 \leq e < 0.55$)							
		C2		CN		C3		C4	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
63.5	127	35	105	190	260	260	330	330	400
127	203.2	55	155	210	310	310	410	410	510
203.2	304.8	90	210	240	360	360	480	480	600
304.8	406.4	130	270	440	580	580	720	720	860
406.4	508	170	340	470	640	640	810	810	980
508	609.6	220	420	520	720	720	920	920	1 120
609.6	711.2	280	510	740	970	970	1 200	1 200	1 430
711.2	762	340	600	780	1 040	1 040	1 300	1 300	1 560
762	914.4	410	700	850	1 140	1 140	1 430	1 430	1 720

1) Nominal bore diameter is the minimum size among the same series.
Note: This table shows **NTN** standard clearances.

Table 5.8 (4) $20^\circ \leq$ Contact angle $\alpha < 30^\circ$ Unit: μm

Nominal bearing bore diameter ¹⁾ d mm		$20^\circ \leq$ Contact angle $\alpha < 30^\circ$ ($0.55 \leq e < 0.87$)							
		C2		CN		C3		C4	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
63.5	127	30	80	140	190	190	240	240	290
127	203.2	40	110	160	230	230	300	300	370
203.2	304.8	70	160	180	270	270	360	360	450
304.8	406.4	95	195	320	420	420	520	520	620
406.4	508	120	240	350	470	470	590	590	710
508	609.6	160	310	380	530	530	680	680	830
609.6	711.2	210	380	540	710	710	880	880	1 050
711.2	762	250	440	580	770	770	960	960	1 150
762	914.4	300	520	630	850	850	1 070	1 070	1 290

1) Nominal bore diameter is the minimum size among the same series.
Note: This table shows **NTN** standard clearances.

Table 5.8 (5) $30^\circ \leq$ Contact angle α Unit: μm

Nominal bearing bore diameter ¹⁾ d mm		$30^\circ \leq$ Contact angle α ($0.87 \leq e$)							
		C2		CN		C3		C4	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
63.5	127	15	50	90	125	125	160	160	200
127	203.2	25	70	100	145	145	190	190	240
203.2	304.8	45	100	110	170	170	230	230	290
304.8	406.4	60	130	200	270	270	340	340	410
406.4	508	80	160	220	300	300	380	380	460
508	609.6	100	200	—	—	—	—	—	—
609.6	711.2	130	250	—	—	—	—	—	—
711.2	762	160	290	—	—	—	—	—	—
762	914.4	190	330	—	—	—	—	—	—

1) Nominal bore diameter is the minimum size among the same series.
Note: This table shows **NTN** standard clearances.

Table 5.9 Radial internal clearance of spherical roller bearings

Nominal bearing bore diameter <i>d</i> mm		Cylindrical bore bearing									
		C2		CN		C3		C4		C5	
		Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
80	100	35	60	60	100	100	135	135	180	180	225
100	120	40	75	75	120	120	160	160	210	210	260
120	140	50	95	95	145	145	190	190	240	240	300
140	160	60	110	110	170	170	220	220	280	280	350
160	180	65	120	120	180	180	240	240	310	310	390
180	200	70	130	130	200	200	260	260	340	340	430
200	225	80	140	140	220	220	290	290	380	380	470
225	250	90	150	150	240	240	320	320	420	420	520
250	280	100	170	170	260	260	350	350	460	460	570
280	315	110	190	190	280	280	370	370	500	500	630
315	355	120	200	200	310	310	410	410	550	550	690
355	400	130	220	220	340	340	450	450	600	600	750
400	450	140	240	240	370	370	500	500	660	660	820
450	500	140	260	260	410	410	550	550	720	720	900
500	560	150	280	280	440	440	600	600	780	780	1 000
560	630	170	310	310	480	480	650	650	850	850	1 100
630	710	190	350	350	530	530	700	700	920	920	1 190
710	800	210	390	390	580	580	770	770	1 010	1 010	1 300
800	900	230	430	430	650	650	860	860	1 120	1 120	1 440
900	1 000	260	480	480	710	710	930	930	1 220	1 220	1 570
1 000	1 120	290	530	530	780	780	1 020	1 020	1 330	1 330	1 720
1 120	1 250	320	580	580	860	860	1 120	1 120	1 460	1 460	1 870
1 250	1 400	350	640	640	950	950	1 240	1 240	1 620	1 620	2 080
1 400	1 600	400	720	720	1 060	1 060	1 380	1 380	1 800	—	—
1 600	1 800	450	810	810	1 180	1 180	1 550	1 550	2 000	—	—

Note: For nominal bearing bore diameter *d* exceeding 1 000 mm, the **NTN** standard is followed.

5.3 Necessary minimum load

In general, when a bearing is operated under no load or a very light load, slippage may occur between the rolling element and the raceway.

Therefore, it is necessary to apply a minimum load to prevent slippage during bearing operation. A rough standard for the necessary minimum radial loads for radial bearings is shown below.

Ball bearings

(except self-aligning ball bearings): $0.023C_{0r}$

Self-aligning ball bearings : $0.018C_{0r}$

Roller bearings : $0.040C_{0r}$

Where:

C_{0r} : Basic static load rating, N

* Consult with **NTN** Engineering for the necessary minimum axial loads for thrust bearings.

Unit: μm

Tapered bore bearing										Nominal bearing bore diameter	
C2		CN		C3		C4		C5		<i>d</i> mm	
Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Over	Incl.
55	80	80	110	110	140	140	180	180	230	80	100
65	100	100	135	135	170	170	220	220	280	100	120
80	120	120	160	160	200	200	260	260	330	120	140
90	130	130	180	180	230	230	300	300	380	140	160
100	140	140	200	200	260	260	340	340	430	160	180
110	160	160	220	220	290	290	370	370	470	180	200
120	180	180	250	250	320	320	410	410	520	200	225
140	200	200	270	270	350	350	450	450	570	225	250
150	220	220	300	300	390	390	490	490	620	250	280
170	240	240	330	330	430	430	540	540	680	280	315
190	270	270	360	360	470	470	590	590	740	315	355
210	300	300	400	400	520	520	650	650	820	355	400
230	330	330	440	440	570	570	720	720	910	400	450
260	370	370	490	490	630	630	790	790	1 000	450	500
290	410	410	540	540	680	680	870	870	1 100	500	560
320	460	460	600	600	760	760	980	980	1 230	560	630
350	510	510	670	670	850	850	1 090	1 090	1 360	630	710
390	570	570	750	750	960	960	1 220	1 220	1 500	710	800
440	640	640	840	840	1 070	1 070	1 370	1 370	1 690	800	900
490	710	710	930	930	1 190	1 190	1 520	1 520	1 860	900	1 000
530	770	770	1 030	1 030	1 300	1 300	1 670	1 670	2 050	1 000	1 120
570	830	830	1 120	1 120	1 420	1 420	1 830	1 830	2 250	1 120	1 250
620	910	910	1 230	1 230	1 560	1 560	2 000	2 000	2 470	1 250	1 400
680	1 000	1 000	1 350	1 350	1 720	1 720	2 200	—	—	1 400	1 600
750	1 110	1 110	1 500	1 500	1 920	1 920	2 400	—	—	1 600	1 800

6. Lubrication

6.1 Purpose of lubrication

The purpose of rolling bearing lubrication is to prevent direct metallic contact between the various rolling and sliding elements. This is accomplished through the formation of a thin oil (or grease) film on the contact surfaces. Lubricant is necessary for operating rolling bearings. For rolling bearings, lubrication has the following advantages:

(1) **Reduction of friction and wear**

It prevents direct metallic contact between the rolling and sliding elements of bearing components and reduces friction and wear.

(2) **Prolonged bearing life**

The rolling fatigue life is prolonged by forming an oil film on the rolling contact surface part.

(3) **Friction heat dissipation** and cooling
Circulating lubrication can dissipate heat generated from friction or conducted from the outside.

(4) **Others**

It **prevents foreign materials from entering** inside the bearing and suppresses corrosion (rust) by covering the bearing surface with oil.

In order to exhibit these effects, a lubrication method that matches service conditions is required. In addition to this, a quality lubricant must be selected, the proper amount of lubricant must be used and the bearing must be designed to prevent foreign matter from getting in or lubricant from leaking out. If lubrication is insufficient, friction is not reduced, causing excessive rise in bearing temperature or abnormal wear. Therefore, an appropriate lubrication and lubrication method should be selected.

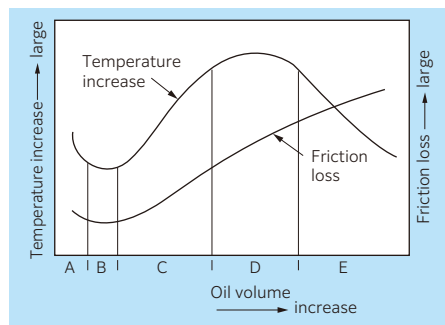


Fig. 6.1

Fig. 6.1 shows the relationship between oil volume, friction loss, and temperature rise.

Table 6.1 details the characteristics of this relationship.

Table 6.1 Oil volume, friction loss, and temperature increase (see Fig. 6.1)

Range	Characteristics	Lubrication method
A	When oil volume is extremely low, direct metallic contact occurs in places between the rolling elements and raceway surfaces. Bearing abrasion and seizing may occur.	—
B	A thin oil film develops over all surfaces, friction is minimal and bearing temperature is low.	Grease lubrication Oil mist Air-oil lubrication
C	As oil volume increases, heat buildup is balanced by cooling.	Circulating lubrication
D	Regardless of oil volume, temperature increases at a fixed rate.	Circulating lubrication
E	As oil volume increases, cooling dominates and bearing temperature decreases.	Forced circulation lubrication Oil jet lubrication

6.2 Lubrication methods and characteristics

Lubrication methods for bearings can be roughly divided into **grease** and **oil lubrication**. Each of these has its own features, so the lubrication method that best offers the required function must be selected.

Characteristics of each method are shown in Table 6.2.

Table 6.2 Comparison of grease lubrication and oil lubrication characteristics

Method Concern	Grease lubrication	Oil lubrication
Handling	◎	△
Reliability	○	◎
Cooling effect	×	○ (Circulation necessary)
Seal structure	○	△
Power loss	○	○
Environment contamination	○	△
High-speed rotation	×	○

◎ : Very good ○ : Good △ : Fair × : Poor

6.3 Grease lubrication

Grease lubricants are relatively easy to handle and require only the simplest sealing devices. For these reasons, grease is the most widely used lubricant for rolling bearings. It is used in a bearing that is pre-sealed with grease (sealed/shielded bearing), or if using an unsealed bearing, fill the bearing and housing with the proper amount of grease, and replenish or change the grease regularly.

With sealed bearings, the proper grease amount does not cause leakage; however, under use conditions including a lot of vibrations, which cause grease to flow easily, or under high speed outer ring rotation, in which a large centrifugal force is applied on the grease, the grease may purge (in rare cases). Please consult NTN Engineering.

6.3.1 Types and characteristics of grease

Lubricating grease is composed of either a mineral base oil or a synthetic base oil. **To this base a thickener and other additives are added.** The properties of all greases are mainly determined by the kind of base oil used and by the combination of thickening agent and various additives. Table 6.5 shows general grease varieties and characteristics, and Table 6.6 shows grease brand names and their characteristics (refer to page A-52 and page A-53). As performance characteristics of even the same type of grease will vary widely from brand to brand, **it is necessary to check the manufacturers' data when selecting grease.**

(1) Base oil

Mineral oil or synthetics such as **ester oil**, **synthetic hydrocarbon oil**, or **ether oil** are used as the base of greases.

Generally, greases with low viscosity base oils are best suited for low temperatures and high speeds; grease using high viscosity base oil has superior high temperature and heavy load characteristics.

(2) Thickening agents

Thickening agents are compounded with base oils to maintain the semi-solid state of the grease. Thickening agents consist of two types of bases: metallic soaps and non-soaps. Metallic soap thickeners include: **lithium**, **sodium**, **calcium**, etc. Non-soap base thickeners are divided into two groups: inorganic (**silica gel**, **bentonite**, etc.) and organic (**polyurea**, **fluorocarbon**, etc.). The

various special characteristics of a grease, such as **limiting temperature range**, **mechanical stability**, water resistance, etc., depend largely on the type of **thickening agent used**. For example, a sodium based grease is generally poor in water resistance properties, while greases with bentone, polyurea and other non-metallic soaps as the thickening agent are generally superior in high temperature properties.

(3) Additives

Various additives are added to grease depending on the purpose. Typical additives include **anti-oxidants**, **high-pressure additives** (EP additives), **rust preventives**, and **anti-corrosives**. For bearings subject to heavy loads and/ or shock loads, grease containing high-pressure additives should be used. Antioxidants are added to grease used in most types of rolling bearings.

(4) Consistency

Consistency is an index that indicates hardness and fluidity of grease. **The higher the NLGI number, the HARDER the grease is.** For the lubrication of rolling bearings, greases with the NLGI consistency numbers of 1, 2, and 3 are used. General relationships between consistency and application of grease are shown in Table 6.3.

Table 6.3 Consistency of grease

NLGI consistency No.	JIS [ASTM] 60 times blend consistency	Application
0	355 - 385	For centralized greasing use
1	310 - 340	For centralized greasing use
2	265 - 295	For general use and sealed bearing use
3	220 - 250	For general use, high temperature use, and sealed bearing use
4	175 - 205	For special use

(5) Mixing different types of greases

When greases of different kinds are mixed together, the consistency of the greases will change (usually softer), the operating temperature range will be lowered, and other changes in characteristics will occur. **As a rule, grease should not be mixed with grease of any other brand.** However, if different greases must be mixed, at least greases with the same base oil and thickening agent should be selected.

6.3.2 Amount of grease

The amount of grease used in any given situation will depend on many factors relating to the size and shape of the housing, space limitations, bearing's rotating speed and type of grease used. As a rule of thumb, **bearings should be filled to 30 to 40 % of their space and housing should be filled to 30 to 60 %**. Where speeds are high and temperature rises need to be kept to a minimum, a reduced amount of grease should be used. **Excessive amounts of grease cause temperature rises which in turn cause the grease to soften and may allow leakage. Oxidation and deterioration of excessive grease fills may cause the lubricating efficiency to be lowered.** Moreover, the standard bearing space can be found by formula (6.1).

$$V = K \cdot W \dots\dots\dots (6.1)$$

Where:

V : Quantity of bearing space open type (approx.), cm³

K : Bearing space factor (see **Table 6.4**)

W : Mass of bearing, kg

A predetermine amount of grease is filled in the bearing with a grease gun or a syringe. After sealing it is not possible to spread the grease by hand - only by rotating the bearing by hand.

Table 6.4 Bearing space factor K

Bearing type ¹⁾			Cage type	<i>K</i>
Deep groove ball bearings ²⁾			Pressed cage	61
Angular contact ball bearings			Pressed cage	54
			Machined cage	33
			Resin cage	33
Cylindrical roller bearings	NU type ³⁾		Pressed cage	50
			Machined cage	36
	N type ⁵⁾		Pressed cage	55
			Machined cage	37
	ULTAGE™ series (EA type) E type	NU type ⁴⁾	Machined cage	33
			Resin cage	33
		N type ⁴⁾	Machined cage	34
			Resin cage	35
Tapered roller bearings			Pressed cage	46
Spherical roller bearings	Type B Type 213		Machined cage	28
	ULTAGE™ series	Type EA	Pressed cage	33
		Type EM	Machined cage	31

1) Does not apply to model numbers that are not specified in the catalog.

2) Does not apply to 160 series bearings.

3) Does not apply to NU4 series.

4) Applies to G1 machined cages only.

5) Does not apply to N4 series.

6.4 Solid grease

"Solid grease" is a lubricant composed mainly of lubricating grease and ultra-high polymer polyethylene. Solid grease begins as grease that has the same viscosity as a more traditional grease. After being heated and cooled, a process known as "calcination", the grease hardens while maintaining a large quantity of lubricant within the polymer structure. The result of this solidification is that the grease does not easily leak from the bearing, even when the bearing is subjected to strong vibrations or centrifugal force.

Bearings with solid grease are available in two types: the spot-pack type in which solid grease is injected into the cage, and the full-pack type in which all free space around the rolling elements is completely filled with solid grease.

Spot-pack solid grease is available for deep groove ball bearings, small diameter ball bearings, and bearing units. Full-pack solid grease is available for self-aligning ball bearings and spherical roller bearings. Grease filling examples are shown in **Fig. 6.2** and **Fig. 6.3**.

Primary advantages:

- (1) Minimal grease leakage
- (2) Low bearing torque with spot-pack type solid grease

For details, see the special catalog "**Bearings with solid grease (CAT. No. 3022/E)**".

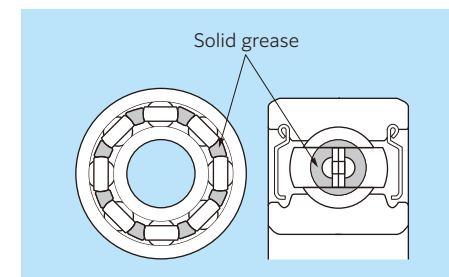


Fig. 6.2 Deep groove ball bearing with spot-pack solid grease (ZZ shield) (Available for deep groove ball bearings)

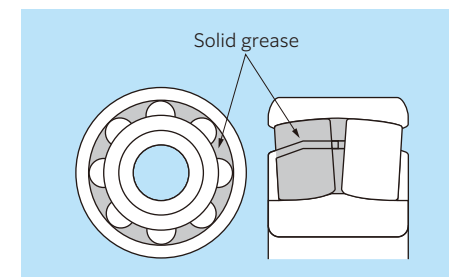


Fig. 6.3 Spherical roller bearing with full-pack solid grease (Available for spherical roller bearings)

Table 6.5 Grease varieties and characteristics ¹⁾

	Soap-based				
	Lithium (Li) grease				Calcium (Ca) grease
Thickening agent ²⁾	Li soap			Li complexed soap	Ca soap (cup grease)
Base oil ³⁾	Mineral oil	Ester oil	Silicone oil	Mineral oil	Mineral oil
Dropping point °C	170 to 190	170 to 190	200 to 210	>250	80 to 100
Operating temperature range °C	−30 to 120	−50 to 130	−50 to 160	−30 to 130	−20 to 70
Mechanical stability	Good	Good	Good	Good	OK
Pressure resistance	Good	Good	Poor	Good	OK
Water resistance	Good	Good	Good	Good	Good
Characteristics/ application	Balanced performance with less disadvantages All-purpose grease	Excellent low temperature and wear characteristics Suitable for miniature and small size ball bearings	Excellent characteristics at low and high temperatures Poor load resistance	Balanced performance with less disadvantages Usable for relatively high temperature	Used for low speed and light loads Unusable for high temperature

1) Use the grease performance as rough standards because it differs depending on the manufacturer's additive formation.

2) Na soap-based grease may be emulsified by water and high humidity conditions.

Urea-based grease may deteriorate polyfluorocarbons and rubber.

Table 6.6 Grease brands and their nature

Brand	Code	Thickener	Base oil	Base oil viscosity mm ² /s	
				40 °C	100 °C
Alvania Grease S2	2AS	Li soap	Mineral oil	131	12.2
Alvania Grease S3	3AS	Li soap	Mineral oil	131	12.2
Alvania EP Grease 2	8A	Li soap	Mineral oil	220	15.9
Multemp PS No. 2	1K	Li soap	Ester + PAO	15.9	—
Multemp SRL	5K	Li soap	Ester	24.1	—
SH44M	4M	Li soap	Silicone	80	19
ISOFLX NBU15	15K	Ba complexed soap	Diester + mineral oil	23	5
SHC POLYREE 462	L791	Urea	PAO	460	40
SE-1	L749	Urea	PAO + ester	22	5
ME-1	L700	Urea	Ester + PAO	61.3	9.3
EP-1	L542	Urea	PAO	46.8	—
NA103A	L756	Urea	PAO + ether	53.5	—
MP-1	L448	Urea	Synthetic oil	40.6	7.1
Grease J	L353	Urea	Ester	75	10
Cosmo Wide Grease WR3	2M	Na terephthalate	Diester + mineral oil	31.6	6
Mobilgrease 28	9B	Bentonite	PAO	30	5.7
Aeroshell Grease 7	5S	Microgel	Diester	10.3	3.1

Note: 1. Representative values are shown for the base oil viscosity, consistency, and dropping point.

2. The upper and lower limits of the operating temperature range differ depending on the usage environment and requirement specifications. Please consult with **NTN** Engineering.

Soap-based		Non-soap-based			
Calcium (Ca) grease	Sodium (Na) grease	Organic			Inorganic
Ca complexed soap	Na soap	Urea	Urea	PTFE	Silica gel
Mineral oil	Mineral oil	Mineral oil	Synthetic oil	Fluorinated oil	Ester oil
200 to 280	170 to 200	>260	>260	None	>260
−20 to 130	−20 to 130	−30 to 140	−40 to 180	−40 to 250	−70 to 150
Good	Good	Good to Excellent	Good to Excellent	OK to Good	Good
Good to Excellent	Good	Good to Excellent	Good to Excellent	Good	Good
Good	Poor	Good to Excellent	Good to Excellent	Good	Good
Excellent pressure resistance	Some emulsification when water is introduced Usable for relatively high temperature	Excellent water resistance and oxidation stability	Excellent water resistance and oxidation stability Used for high temperature and high speed applications	Excellent chemical resistance Used for high temperature applications	Excellent characteristics at low temperature

3) Ester oil-based grease may swell acrylic materials, and silicone-based grease may swell silicone materials.

Some silicone-based greases and fluorine-based greases have poor noise performance and rustproofing performance.

60 times blend consistency		Dropping point °C	Operating temperature range °C	Characteristics
Representative value	NLGI No.			
283	2	181	−25 to 120	All-purpose (standard grease for deep groove ball bearings)
242	3	182	−20 to 135	All-purpose (standard grease for ball bearings of bearing units)
284	2	184	−20 to 110	Heavy load all-purpose
270	2	190	−50 to 130	For low temperature and low torque
250	2 to 3	192	−40 to 150	For low temperature to high temperature, all-purpose (standard grease for miniature and small size ball bearings)
260	2 to 3	204	−40 to 160	For high temperature
280	2	220 or above	−40 to 130	For high speed
280	2	270	−20 to 170	For food machinery
265	2	220 or above	−50 to 120	For high speed
231	3	250 or above	−30 to 160	For high temperature and high speed
220	3	260 or above	−40 to 160	For high temperature and high speed
270	2	260 or above	−40 to 180	Brittle separation
243	3	250 or above	−40 to 150	For high temperature and high speed
305	1 to 2	280 or above	−20 to 180	For high temperature
238	3	230 or above	−40 to 150	For low temperature to high temperature, all-purpose
293	1 to 2	307	−54 to 177	MIL-PRF-81322 For low temperature to high temperature
296	1 to 2	260 or above	−73 to 149	MIL-PRF-23827C

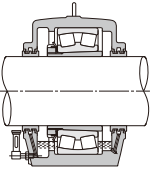
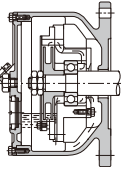
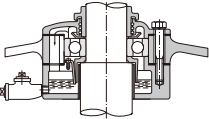
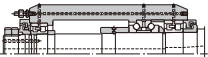
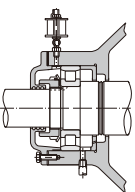
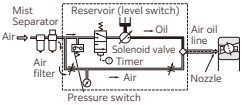
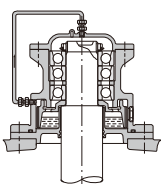
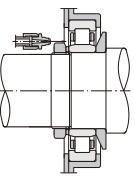
6.5 Oil lubrication

Oil lubrication is suitable for applications requiring that bearing-generated heat or heat applied to the bearing from other sources be carried away from the bearing and dissipated

to the outside.

Table 6.7 shows the main methods of oil lubrication.

Table 6.7 Oil lubrication methods

Lubrication method	Example	Lubrication method	Example
(Oil bath lubrication) - Oil bath lubrication is the most generally used method of lubrication, and is widely used for low to moderate rotational speed applications. - For horizontal shaft applications, oil level should be maintained at approximately the center of the lowest rolling element, according to the oil gauge, when the bearing is at rest. For vertical shafts at low speeds, the oil level should be maintained at 50 to 80% submergence of the rolling elements.		(Disc lubrication) In this method, a partially submerged disc rotates and pulls oil up into a reservoir from which it then drains down through the bearing, lubricating it.	
(Oil spray lubrication) In this method, an impeller or similar device mounted on the shaft draws up oil and sprays it onto the bearing. This method can be used at considerably high speeds.		(Oil mist lubrication) - Using pressurized air, lubricating oil is atomized before passing through the bearing. - Due to the low lubricant resistance, this method is well suited to high speed applications.	
(Drip lubrication) - In this method, oil is collected above the bearing and allowed to drip down into the housing where it becomes a lubricating mist as it strikes the rolling elements. Another version allows only slight amounts of oil to pass through the bearing. - Used at relatively high speeds for light to normal load applications. - In most cases, oil volume is a few drops per minute.		(Air-oil lubrication) - In this method, the required minimum amount of lubricating oil is measured and fed to each bearing at ideal intervals using compressed air. - Fresh lubricating oil is constantly fed. - Because the required oil quantity is very small, the working environment can be kept clean.	
(Circulating lubrication) - Used for bearing cooling or for automatic oil supply systems in which the oil supply is centrally located. - One of the advantages of this method is that oil cooling devices and filters to maintain oil purity can be installed within the system. - In order for oil to thoroughly lubricate the bearing, oil inlets and outlets must be provided on opposite sides of the bearing.		(Oil jet lubrication) - This method lubricates by injecting oil under high pressure directly into the side of the bearing. This is a reliable system for high speed, high temperature or otherwise severe conditions. - Used for lubricating the bearings in jet engines, gas turbines, and other high speed equipment. - Under-race lubrication is one example of this type of lubrication.	

6.5.1 Selection of lubricating oil

Under normal operating conditions, **machine oil**, **turbine oil**, and other mineral oils are widely used for the lubrication of rolling bearings. However, for temperatures **below -30 °C** or **above 150 °C**, synthetic oils such as **ester oil**, **silicone oil**, and **fluorinated oil** are used.

For lubricating oils, viscosity is one of the most important properties and determines an oil's lubricating efficiency. If viscosity is too low, formation of the oil film will be insufficient, and damage to the rolling surface will occur. If viscosity is too high, viscous resistance will also be great, resulting in temperature increase and friction loss. In general, **for higher speed applications, a lower viscosity oil** should be used; **for heavier load applications, a higher viscosity oil** should be used.

Lubrication of rolling bearings requires dynamic viscosity shown in **Table 6.8**, which is dependent on the use conditions. **Fig 6.4** shows the relation between lubricating oil dynamic viscosity and temperature. This is used to select a lubrication oil with viscosity characteristics appropriate for the operating temperature.

For reference, **Table 6.9** lists the selection standards for lubricating oil viscosity based on bearing operating conditions.

Table 6.8 Required lubricating oil dynamic viscosity for bearings

Bearing type	Dynamic viscosity mm ² /s
Ball bearings, Cylindrical roller bearings	13 or above
Spherical roller bearings, Tapered roller bearings	20 or above
Thrust spherical roller bearings	30 or above

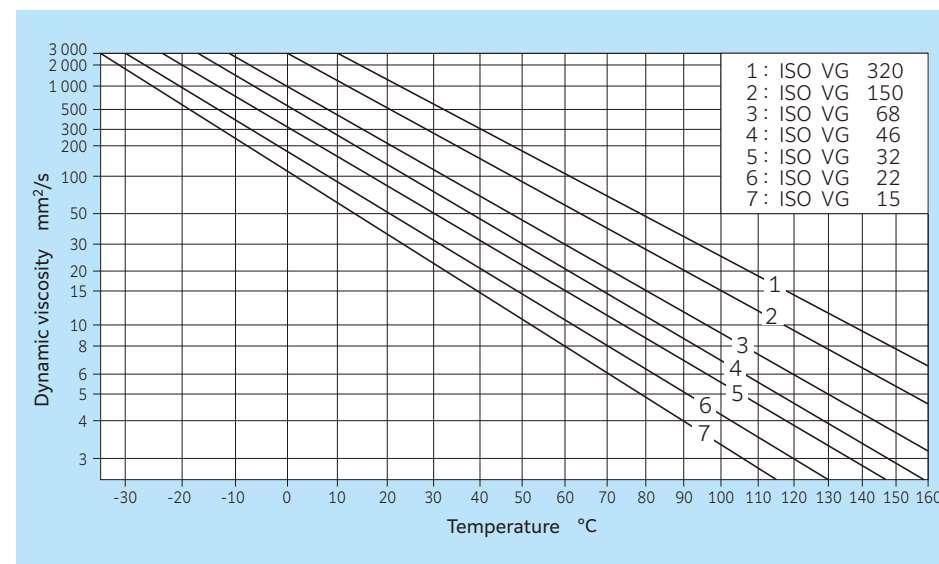


Fig 6.4 Relation between lubricating oil dynamic viscosity and temperature

Table 6.9 Standards for lubricating oil viscosity

Bearing operating temperature °C	dn-value ¹⁾	Lubricating oil ISO viscosity grade (VG)		Suitable bearing
		Normal load	Heavy load or shock load	
-30 - 0	Up to allowable rotational speed	22, 32	46	All types
0 - 60	Up to 15 000	46, 68	100	All types
	15 000 - 80 000	32, 46	68	All types
	80 000 - 150 000	22, 32	32	All types but thrust ball bearings
	150 000 - 500 000	10	22, 32	Single row radial ball bearings, cylindrical roller bearings
60 - 100	Up to 15 000	150	220	All types
	15 000 - 80 000	100	150	All types
	80 000 - 150 000	68	100, 150	All types but thrust ball bearings
	150 000 - 500 000	32	68	Single row radial ball bearings, cylindrical roller bearings
100 - 150	Up to allowable rotational speed	320		All types
0 - 60	Up to allowable rotational speed	46, 68		Spherical roller bearings
60 - 100	Up to allowable rotational speed	150		

1) dn value: [dn = bearing bore diameter *d* (mm) × rotational speed *n* (min⁻¹)]

Note: 1. Applied when lubrication method is either oil bath or circulating lubrication.

2. Please consult with NTN Engineering in cases where operating conditions fall outside the range covered by this table.

6.5.2 Oiling amount

When a bearing is to be supplied with oil forcibly, the amount of heat generated from the bearing is equal to the sum of the amount of heat dissipated from the housing and the amount of heat carried away by the oil.

The oiling amount that serves as a rough indication when a standard housing is used can be obtained by formula (6.2).

$$Q = K \cdot q \quad (6.2)$$

Where:

Q: oiling amount per bearing (cm³/min)

K: coefficient determined by allowable temperature rise of oil (see Table 6.10)

q: oiling amount obtained by diagram (cm³/min) (see Fig. 6.5)

The heat dissipation amount differs depending on the housing type. Therefore, in the actual operation, it is desirable to obtain the oiling amount suitable for the actual machine by adjusting the amount obtained by formula (6.2) to 1.5 to 2 times.

In addition, when calculating the oiling amount assuming that no heat is dissipated from the housing and the generated heat amount is completely carried away by the oil, use the shaft diameter in the diagram as *d* = 0.

Table 6.10 Value of *K*

Expelled oil temp minus supplied oil temp °C	<i>K</i>
10	1.5
15	1
20	0.75
25	0.6

(Example) For tapered roller bearing 30220U mounted on a flywheel shaft with a radial load of 9.5 kN, operating at 1 800 min⁻¹, what is the amount of lubricating oil *Q* required to keep the bearing temperature rise below 15 °C?

$$d = 100 \text{ mm,}$$

$$dn = 100 \times 1\,800 = 18 \times 10^4$$

From Fig. 6.5 *q* = 180 cm³/min

Assume the bearing temperature is approximately equal to the expelled oil temperature,

from Table 6.10, since *K* = 1

$$Q = K \times q = 1 \times 180 = 180 \text{ cm}^3/\text{min}$$

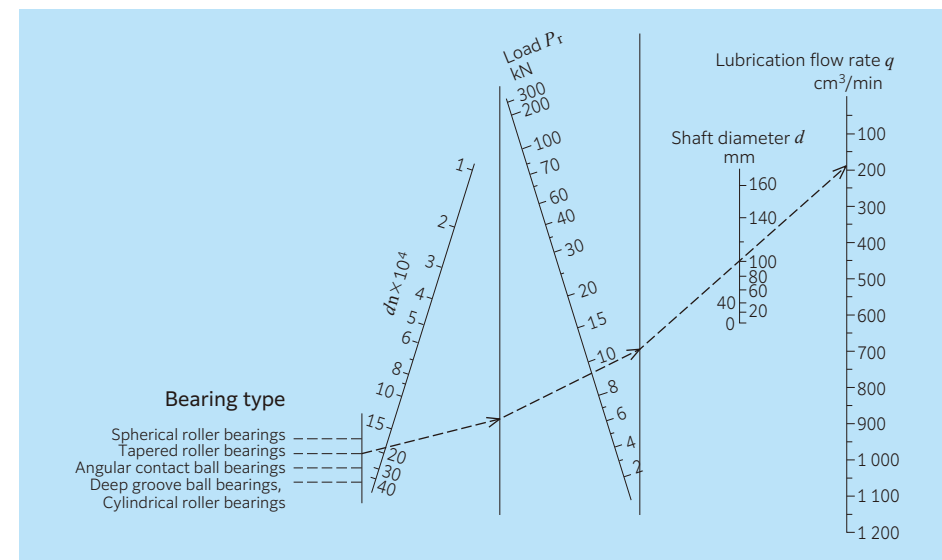


Fig. 6.5 Oil quantity guidelines

6.5.3 Relubrication intervals

The intervals at which lubricating oil should be changed varies depending upon operating conditions, oil quantity, and type of oil used. In general, for oil bath lubrication where the operating temperature is 50 °C or less, oil should be replaced once a year. When the operating temperature is between 80 to 100 °C, oil should be replaced at least once every three months. For important equipment, it is advisable that lubricating efficiency and oil purity deterioration be checked regularly to determine when oil replacement is necessary.

7. Bearing materials

7.1 Raceway and rolling element

While the contact surfaces of a bearing's raceways and rolling elements are subjected to repeated heavy stress, they must also maintain high precision and rotational accuracy.

To accomplish this, the raceways and rolling elements must be made of a material that has high hardness, is resistant to rolling fatigue, is wear resistant, and has good dimensional stability. The most common cause of fatigue in bearings is the inclusion of non-metallic inclusions in the steel. Nonmetallic inclusions contain hard oxides that can cause fatigue cracks. Clean steel with minimal non-metallic inclusions must therefore be used.

All **NTN** bearings use steel that is low in oxygen content and nonmetallic impurities, refined by a vacuum degassing process and outside hearth smelting. For bearings requiring especially high reliability and long life, steels of even higher in purity, such as vacuum melted steel (VIM / VAR) and electro-slag melted steel (ESR), are used.

7.1.1 Raceway and rolling element materials

1) High/mid carbon alloy steel

In general, steel types capable of being "through hardened" below the material surface are employed for raceways and rolling elements. Foremost among these is **high carbon chromium bearing steel**, which is widely used. For large type bearings and bearings with large cross sectional dimensions, induction hardened bearing steel is used, which incorporates manganese (Mn) or molybdenum (Mo). Mid-carbon steel incorporating silicon (Si) and manganese may also be used, which gives it hardening properties comparable to high carbon chromium steel.

SUJ2 is frequently used. SUJ3, with enhanced hardening characteristics containing a large quantity of Mn, is used for large bearings. SUJ5 is SUJ3 to which Mo has been added to further enhance hardening characteristics, and is used for oversized bearings or bearings with thick walls.

2) Carburizing (case hardened) steel

Carburizing hardens the steel from the surface to the proper depth, leaving a relatively soft core. This provides **hardness and toughness**, making the material **suitable for impact loads**. **NTN** uses carburizing (case hardened) steel for most of its tapered roller bearings.

In terms of case hardened steel for **NTN's** other bearings, chromium steel and chrome

molybdenum steel are used for small to medium sized bearings, and nickel chrome molybdenum steel is used for large sized bearings.

3) High temperature capable bearing steel

When bearings made of ordinary high carbon chromium steel which have undergone standard heat treatment are used for long durations at high temperatures, unacceptably large dimensional changes can occur. For this reason, a **dimension stabilizing treatment** (TS treatment) has been devised for very high temperature applications. This treatment however reduces the hardness of the material, thereby reducing rolling fatigue life (See section "1.3.2 Life adjustment factor for special bearing properties a_2 " on page A-7). Note that dimensional changes can occur in normal use too.

Standard high temperature bearings for use at temperatures from **150 to 200 °C**, add silicon to the steel to improve heat resistance. This results in a bearing with excellent rolling fatigue life with minimal dimensional change or softening at high temperatures.

A variety of heat resistant steels are also incorporated in bearings to minimize softening and dimensional changes when used at high temperatures. Two of these are high speed molybdenum steel and high speed tungsten steel. For bearings requiring heat resistance in high speed applications, there is also heat resistant case hardening molybdenum steel.

4) Corrosion resistant bearing steel

For applications requiring high corrosion resistance, **stainless steel** is used. To achieve this corrosion resistance a large proportion of the alloying element chrome is added to martensite stainless steel.

5) Induction hardened steel

Besides the use of surface hardening steel, induction hardening is also utilized for bearing raceway surfaces, and for this purpose **mid-carbon steel** is mainly used for its lower carbon content instead of through hardening steel.

For deep hardened layers required for **larger bearings and bearings with large surface dimensions**, mid-carbon steel is fortified with chromium and molybdenum.

6) Other bearing materials

For ultra-high-speed applications and applications requiring very high level corrosion resistance, ceramic bearing materials such as Si_3N_4 are also available.

7.1.2 Characteristics of bearing materials

1) Dimensional change of bearings

Dimensions of bearings used for a long time may change depending on the use condition. This phenomenon is called dimensional change.

<Mechanism of dimensional change>

A standard bearing steel structure contains a small amount of austenite in the matrix of hard martensite. This austenite is partially retained austenite without being transformed into martensite in the cooling process of the bearing steel quenching process, and is called residual austenite.

Since the residual austenite is an unstable structure, it is transformed into a stable structure (martensite) when the bearing is being used. This structure transformation is the cause of the dimensional change of bearings.

Fig. 7.1 shows measured values of dimensional change of a standard bearing held at 120 °C over an extended period of time.

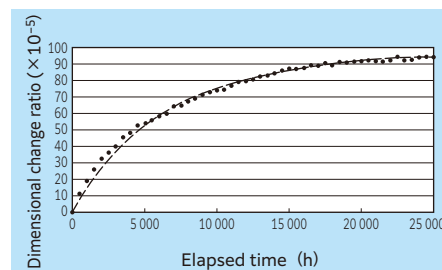


Fig. 7.1 Example of dimensional change rate of standard bearings that are held at 120 °C for a long time (measured values)

The dimensional change rate becomes larger as the elapsed time or the temperature of exposure increases.

Depending on the use condition, dimensional change may occur with bearings made of general bearing steel that did not reach 100 °C, which is the normal limit.

Bearings that underwent dimension stabilization treatment (TS treatment) have a significantly lower dimensional change. For details, please contact **NTN** Engineering.

<Dimensional change problems and countermeasures>

Among dimensional change, particular attention should be paid to inner ring expansion. When the inner ring expands by dimensional change, the interference between the inner ring and the shaft decreases, and the bearing may be heavily damaged by creeping or axial movement. Therefore, **when a bearing is to be used for a long time, the bearing specifications and fixing method must be determined with the interference decrease due to dimensional change taken into consideration**. For example, the interference can be increased (see section "4. Bearing fits") or fixing in the axial direction can be reinforced (see section "8. Shaft and housing design").

<Situations to monitor dimensional change>

The dimensional change of bearings is expressed by the bearing dimension x dimensional change rate. Therefore, under a given temperature and elapsed time, larger bearings show greater dimensional change. Pay particular attention to the amount of dimensional change when large bearings are to be used with fits with small interference.

In addition, dimensional change does not occur during the rotation inspection immediately following bearing installation. It is observed after a long-period operation. Therefore, for machines and parts used for a long time, periodic inspection is effective for preventing problems. For detailed consideration, please consult **NTN** Engineering beforehand.

7.2 Cage

Bearing cage materials must have the strength to withstand rotational vibrations and shock loads. These materials must also have a low friction coefficient, be lightweight, and be able to withstand bearing operating temperatures.

7.2.1 Metal materials

For **small and medium-sized bearings**, pressed steel cages of **cold or hot rolled material** with a low carbon content of approx. 0.1 % are used. However, depending on the application, **austenitic stainless steel** is also used. Machined cages are generally used for **large bearings**. **Carbon steel for machine structures** or **high-strength cast brass** is frequently used for the cages.

7.2.2 Resin materials

Recently resin cages are used in place of metals because the material is lightweight and easy to mold into complicated shapes. On the other hand, resins have disadvantages such as lower strength and heat resistance. Therefore, it is important to select resin materials that take advantage of their characteristics.

The resin materials are rarely used without being filled, and are usually reinforced with glass fiber (GF) or carbon fiber (CF).

[Characteristics of resin materials]

(Advantages)	(Disadvantages)
• Lightweight • High corrosion resistance • High self-lubricating performance with less abrasion powder • Low noise • Can easily be molded into complicated shapes and various designs • High productivity	• Lower strength compared with metal • Lower heat resistance compared with metal • The strength and elastic modulus largely vary widely with temperature. • The physical properties (strength) may change when resins are exposed to high temperatures for a long period. • The strength may deteriorate when resins are exposed to certain types of chemical or oils. • The thermal expansion coefficient is high, and the dimensional change is larger compared with metal.

<<Polyamide (PA): 66, 46>>

Polyamide is suitable for general cage materials because it is low cost and has high strength, heat resistance, wear resistance, and formability. This material has disadvantages such as high water absorbency, physical property deterioration and dimensional change due to water absorption. On the other hand, water absorption increases flexibility and toughness, enhancing the ease of assembly and shock resistance of cages.

However, the physical property (strength) may deteriorate rapidly at high temperatures when polyamide is exposed to lubricating oil containing an S (sulfur) type or P (phosphorus) type extreme pressure additive.

Polyamide 66 reinforced with glass fibers is the most used material because it has excellent performance as a cage material.

<<Polyphenylene sulfide (PPS)>>

Polyphenylene sulfide has high heat resistance (continuous operating temperature: 220 to 240 °C), chemical resistance, melt fluidity, and formability.

<<Polyetheretherketone (PEEK)>>

Polyetheretherketone has the highest heat resistance among thermoplastic resins (continuous operating temperature: 240 to 260 °C). It has excellent self-lubricating performance, shock resistance, and chemical resistance, but it is very expensive.

<<Fabric reinforced phenolic resin>>

Phenolic resin is a thermosetting resin. It overcomes the disadvantages of hard and brittle phenolic resin having low shock resistance using fabric reinforcement. It is lightweight and has high lubricity and good mechanical properties. Injection molding cannot be performed because of the thermosetting property, so cages are made by machining.

7.3 Rubber seal materials

Synthetic rubbers with high heat resistance and oil resistance are used as materials for seals. Different rubber is used depending on the degree of heat resistance.

<<Nitrile rubber (NBR)>>

Nitrile rubber has high oil resistance, heat resistance, and wear resistance, and is widely used as a general material for seals. The allowable temperature range is -20 to 120 °C.

<<Acrylic rubber (ACM)>>

Acrylic rubber has high heat resistance and can be used above the application temperature of NBR. It has excellent oil resistance but swells in ester oil. An ester oil resistant grade is also available. The allowable temperature range is -15 to 150 °C.

<<Fluorinated rubber (FKM)>>

Fluorinated rubber has excellent heat resistance, oil resistance, and chemical resistance. It is deteriorated by amine, so attention needs to be paid when combining fluorinated rubber with urea grease that precipitates amine at high temperatures. The allowable temperature range is -20 to 230 °C.

8. Shaft and housing design

Depending upon the design of a shaft or housing, the shaft may be influenced by an unbalanced load or other factors which can then cause large fluctuations in bearing efficiency. For example, depending on the dimensional accuracy and shape accuracy of the shaft and housing, there could be insufficient interference fit with the bearing, leading to material creep during operation. When the machining accuracy of the shaft or the housing is insufficient or when there is an error in the installation, the inner ring or the outer ring of the bearing can become misaligned. Operation under this condition may cause excessive loading at the edges of the inner and outer rings as well as rolling elements, deteriorating the fatigue life. Furthermore, chipping damage may occur on the rib face of roller bearings due to heavy contact with the rolling element end surface while operating under misalignment. A speed differential between the rolling elements and cage may apply abnormal force to the cage, causing damage. For this reason, it is necessary to pay attention to the following when designing shaft and housing:

- (1) Bearing arrangement; most effective fixing method for bearing arrangement.
- (2) Selection of shoulder height and fillet radius of housing and shaft.
- (3) Shape precision and fitting dimensions; runout tolerance of shoulder area.
- (4) Machining precision and mounting error of housing and shaft suitable for allowable alignment angle and permissible misalignment of bearing.

When the housing rigidity is insufficient, excessive deformation of the inner and outer rings may lead to poor distribution of loading among rolling elements, causing abnormal noise and deterioration of fatigue life. Therefore, the housing requires sufficient rigidity.

When two or more bearings are to be attached to a shaft, one typically serves as a fixed end bearing and the other serves as a floating end bearing to compensate for axial mounting error and allow for thermal expansion. In addition, when two or more bearings are to be attached to a housing, the design must allow through hole machining to improve the accuracy of the housing.

8.1 Fixing of bearings

When a bearing that receives axial loads and preloads is to be attached to a shaft or a housing, an axial fixing method that is sufficient to withstand the axial loading such as a tightening nut, bolts, or snap rings should be selected because a serious problem may be caused when the raceway moves in the axial direction.

In addition, **cylindrical roller bearings (NU and N types) that are to be mainly used as floating side bearings also need to be fixed in the axial direction because the raceway may move in the axial direction when the shaft is bent by a moment load, damaging the bearing.**

Table 8.1 shows general bearing fixing methods, and **Table 8.2** shows fixing methods for bearings with tapered bores.

Table 8.1 General bearing fixing methods

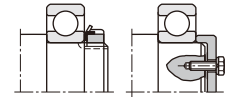
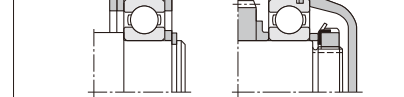
Inner ring clamp	Outer ring clamp	Snap ring
		
The most common method of fixing bearings in place is to use clamping nuts or bolts to hold the shaft or housing abutment against the ring end face. The tightening nuts and bolts must be fixed so that they will not be loosened by axial loads or vibration when the bearing is being used.		Use of snap rings regulated under JIS B 2804, B 2805, and B 2806, makes for a very simple construction. However, interference with chamfers, bearing installation dimensions, and other related specifications must be considered carefully. Snap rings are not suitable for applications requiring high accuracy or where the snap ring receives large axial loads.

Table 8.2 Fixing methods—bearings with tapered bores

Adapter sleeve mounting	Withdrawal sleeve mounting	Split ring mounting
When installing bearings on cylindrical shafts, adapter sleeves or withdrawal sleeves can be used to fix bearings in place axially. The adapter sleeve is fastened in place by frictional force between the shaft and bore diameter of the sleeve.		For installation of tapered bore bearings directly on tapered shafts, the bearing is held in place by a split ring inserted into a groove on the shaft, and is fixed in place with a split ring nut or screw.

8.2 Bearing fitting dimensions

8.2.1 Abutment height and fillet radius

The shaft and housing **abutment height** (h) should be **larger than the bearings' maximum allowable chamfer dimensions** ($r_{s \max}$), such that the abutment directly contacts the flat part of the bearing end face. The **fillet radius** (r_a) must be **smaller than the bearing's minimum allowable chamfer dimension** ($r_{s \min}$) so that it does not interfere with bearing seating.

Table 8.3 lists abutment height (h) and fillet radius (r_a). For bearings to be subjected to very large axial loads, shaft abutments (h) should be higher than the values in the table.

Table 8.3 Fillet radius and abutment height
Unit: mm

$r_{s \min}$	$r_{as \max}$	h (Min.)	
		Normal use ¹⁾	Special use ²⁾
0.05	0.05	0.3	
0.08	0.08	0.3	
0.1	0.1	0.4	
0.15	0.15	0.6	
0.2	0.2	0.8	
0.3	0.3	1.25	1
0.6	0.6	2.25	2
1	1	2.75	2.5
1.1	1	3.5	3.25
1.5	1.5	4.25	4
2	2	5	4.5
2.1	2	6	5.5
2.5	2	6	5.5
3	2.5	7	6.5
4	3	9	8
5	4	11	10
6	5	14	12
7.5	6	18	16
9.5	8	22	20
12	10	27	24
15	12	32	29
19	15	42	38

- 1) If a bearing supports a large axial load, the height of the shoulder must exceed the value given here.
2) Used when an axial load is light. These values are not suitable for tapered roller bearings, angular contact ball bearings and spherical roller bearings.

Note: $r_{as \max}$ indicates maximum allowable fillet radius.

8.2.2 For spacer and ground undercut

In cases where a fillet radius ($r_{a \max}$) larger than the bearing chamfer dimension is required to strengthen the shaft or to relieve stress concentration [see Fig. 8.1(a)], or abutment height is too low to afford adequate contact surface with the bearing [see Fig. 8.1(b)], the use of a spacer may be beneficial.

Relief dimensions for ground shaft and housing fitting surfaces are given in Table 8.4.

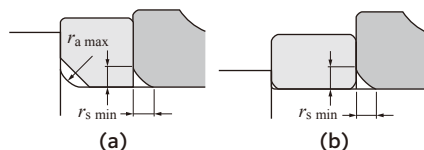
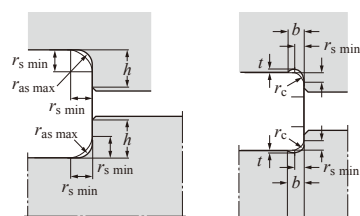


Fig. 8.1 Bearing mounting with spacer

Table 8.4 Relief dimensions for ground shaft
Unit: mm

$r_{s \min}$	Relief dimensions		
	b	t	r_c
1	2	0.2	1.3
1.1	2.4	0.3	1.5
1.5	3.2	0.4	2
2	4	0.5	2.5
2.1	4	0.5	2.5
2.5	4	0.5	2.5
3	4.7	0.5	3
4	5.9	0.5	4
5	7.4	0.6	5
6	8.6	0.6	6
7.5	10	0.6	7



8.2.3 Fitting dimensions for thrust bearings

For thrust bearings, it is necessary to make the raceway washer back face sufficiently wide in relation to load and rigidity. Consequently, fitting dimensions from the dimension tables should be adopted (see Fig. 8.2 and Fig. 8.3).

For this reason, **shaft and abutment heights will be larger than for radial bearings** (Refer to dimension tables for all thrust bearing fitting dimensions).

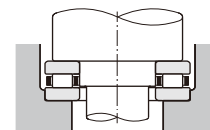


Fig. 8.2

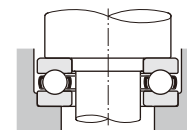


Fig. 8.3

8.3 Shaft and housing accuracy

Table 8.5 shows the required accuracies for shaft and housing fitting surface dimensions and configurations, as well as fitting surface roughness and abutment squareness for normal operating conditions.

Table 8.5 Shaft and housing accuracy

Concern		Shaft	Housing
Dimensional accuracy		IT6(IT5)	IT7(IT5)
Roundness (max.)		IT3	IT4
Cylindricity			
Abutment squareness		IT3	IT3
Fitting surface roughness R_a	Small size bearings	0.8	1.6
	Mid-large size bearings	1.6	3.2

Note: For precision bearings (P4, P5 accuracy), it is necessary to improve the circularity and cylindricity accuracies in this table to approximately 50 % of these values. For details, see the special catalog "Precision Rolling Bearings (CAT. No. 2260/E)".