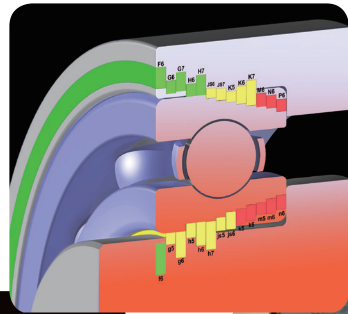
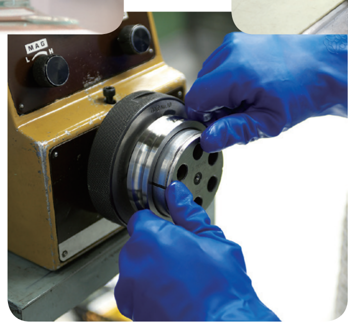
TECHNICAL INFORMATION






Part A

TECHNICAL INFORMATION

1. TYPES AND FEATURES OF ROLLING BEARINGS

1.TYPES AND FEATURES OF ROLLING BEARINGS

1.1 Design and Classification

Rolling bearings generally consist of rolling elements, two rings, and a cage. They are classified into radial bearings or thrust bearings depending on the direction of the main load. In addition, depending on the type of rolling elements, they are classified into ball bearings or roller bearings and further divided by differences in their design or specific purpose.

The most common bearing types and part names are shown in Fig.1.1, and a general classification of rolling bearings is shown in Fig. 1.2.

1.2 Characteristics of Rolling Bearings

Compared with plain bearings, rolling bearings have the following major advantages:

(1) Their starting torque or friction is low and the difference between the starting torque and running torque is small.
(2) With the advancement of worldwide standardization, rolling bearings are internationally available and interchangeable.
(3) Maintenance, replacement, and inspection are easy because of the simple structure surrounding rolling bearings.
(4) Many rolling bearings are capable of taking both radial and axial loads simultaneously or independently.
(5) Rolling bearings can be used under a wide range of temperatures.
(6) Rolling bearings can be preloaded to produce a negative clearance and achieve greater rigidity.

Furthermore, different types of rolling bearings have their own individual advantages. The features of the most common rolling bearings are described on Pages A010 to A013 and in Table 1.1 (Pages A014 and A015).

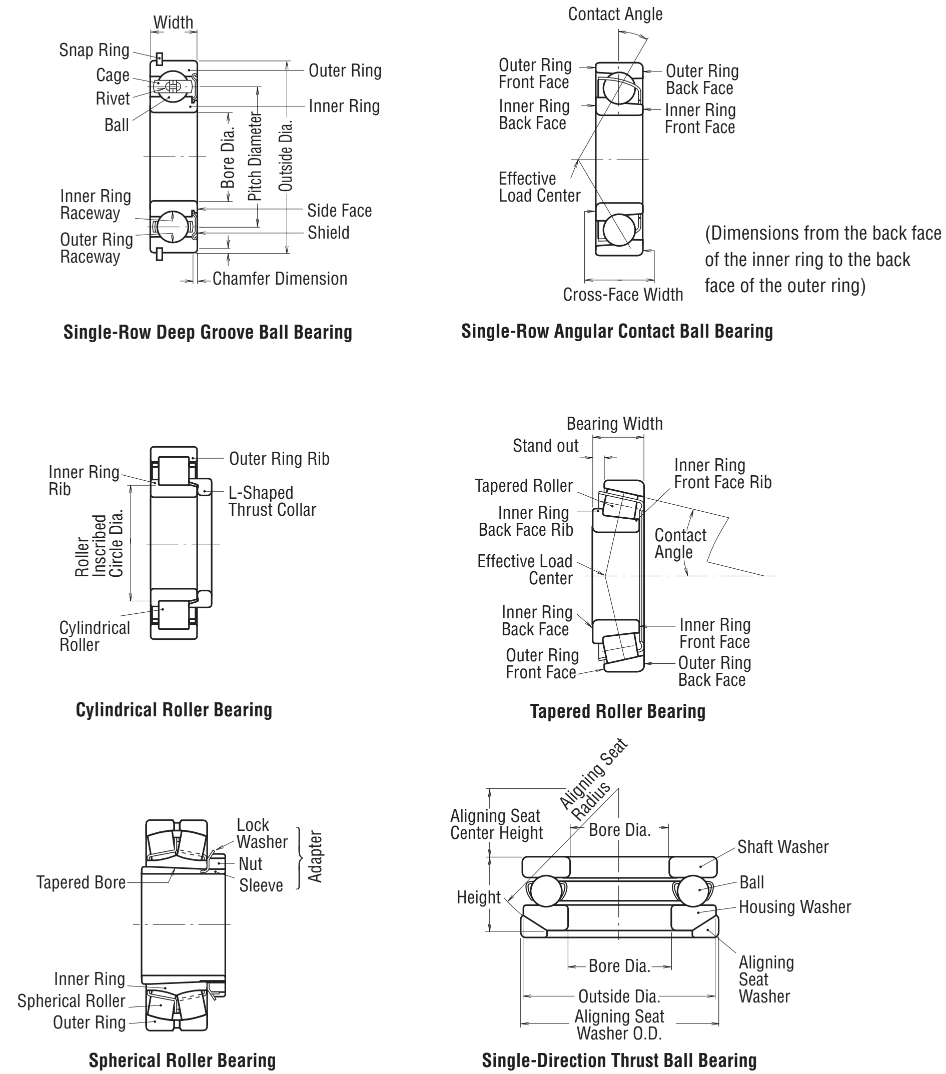


Fig. 1.1 Names of Bearing Parts

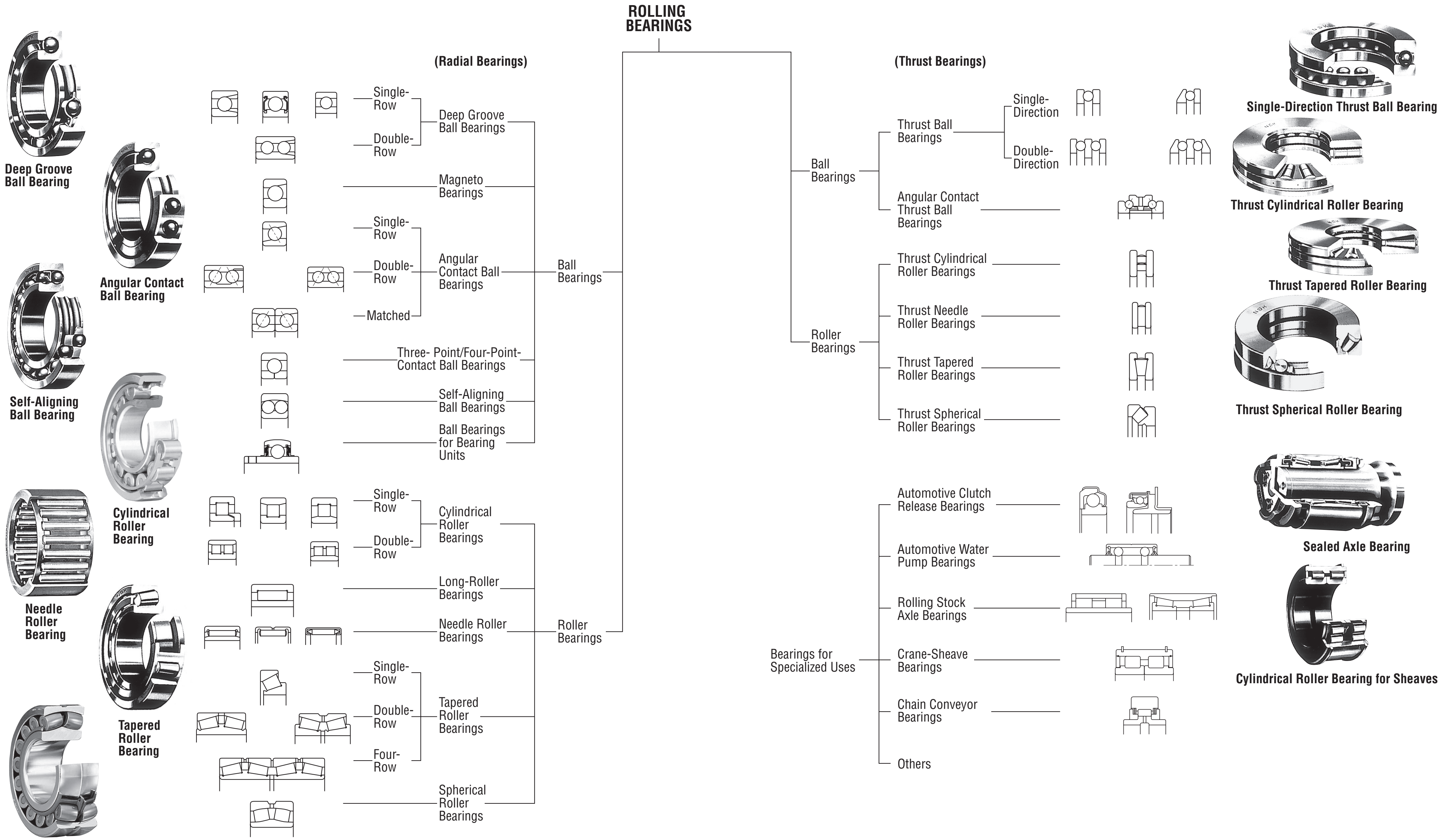
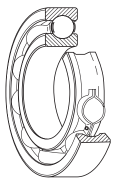


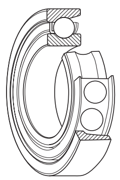
Fig. 1.2 Classification of Rolling Bearings

Single-Row Deep Groove Ball Bearings



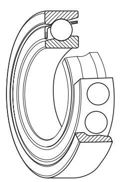
Single-row deep groove ball bearings are the most common type of rolling bearing and are in widespread use. The raceway grooves on both the inner and outer rings have circular arcs of slightly larger radii than those of the balls. They are capable of taking radial loads. In addition, axial loads can be applied in either direction. Because of their low torque, they are highly suitable for applications where high speeds and low power loss are required.
While they can be used as open bearings, single-row deep groove bearings often have steel shields or rubber seals installed on one or both sides and are prelubricated with grease. In addition, snap rings are sometimes used on the periphery. Pressed-steel cages are most commonly used.

Magneto Bearings



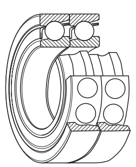
The inner groove of magneto bearings is slightly more shallow than that of deep groove bearings. Since the outer ring has a shoulder on only one side, the outer ring may be removed, which is often advantageous for mounting. In general, two such bearings are used in a paired mounting. Magneto bearings are small bearings with a bore diameter of 4 to 20 mm and are mainly used for small magnetos, gyroscopes, instruments, etc. Pressed-brass cages are generally used.

Single-Row Angular Contact Ball Bearings



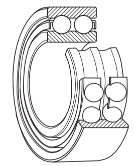
Individual bearings of this type are capable of taking axial loads in one direction and radial loads. Four contact angles of 15°, 25°, 30°, and 40° are available. The larger the contact angle, the higher the axial load capacity. For high-speed operation however, smaller contact angles are preferred. Usually, two bearings are used in a paired mounting, and the clearance between them must be adjusted properly.
Pressed-steel and machined-brass cages are commonly used; however, for high precision bearings with a contact angle less than 30°, polyamide resin cages are often used.

Paired Mounting



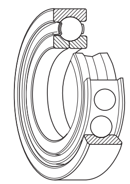
A combination of two radial bearings is called a paired mounting. Usually, they are formed using angular contact ball bearings or tapered roller bearings. Possible arrangements include: face-to-face (type DF), in which the outer ring faces are oriented towards each other; back-to-back (type DB); or same-direction (type DT), in which both front faces are oriented in the same direction. DF and DB arrangements are capable of taking radial loads and axial loads in both directions. Type DT is used when there is a strong axial load in one direction and it is necessary to divide the load equally across each bearing.

Double-Row Angular Contact Ball Bearings



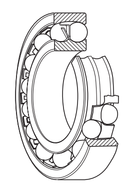
Double-row angular contact ball bearings are like two single-row angular contact ball bearings mounted back-to-back, except that they have only one inner ring and one outer ring. They have a narrower width than two single bearings, and can take thrust loads in both directions.

Four-Point-Contact Ball Bearings



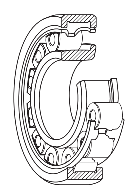
The inner and outer rings of four-point-contact ball bearings are separable because the inner ring is split in a radial plane. They can take axial loads from either direction, and the balls have a contact angle of 35° with each ring. Radial loads are not reccomended. Just one bearing of this type can replace a combination of face-to-face or back-to-back angular contact bearings.
Machined-brass cages are generally used.

Self-Aligning Ball Bearings



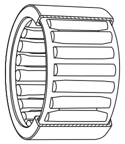
The inner ring of this type of bearing has two raceways, and the outer ring has a single spherical raceway with its center of curvature coincident with the bearing axis. Therefore, the axis of the inner ring, balls, and cage can deflect to some extent around the bearing center. Consequently, minor angular misalignment of the shaft and housing caused by machining or mounting error is automatically corrected.
This type of bearing often has a tapered bore for mounting using an adapter sleeve.

Cylindrical Roller Bearings



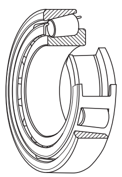
In bearings of this type, the cylindrical rollers are in linear contact with the raceways. They have a high radial load capacity and are suitable for high speeds.
NU, NJ, NUP, N, and NF are single-row bearing types, while NNU and NN are double-row bearing types, with designations depending on the design or absence of side ribs.
The outer and inner rings of all types are separable.
Some cylindrical roller bearings have no ribs on either the inner or outer ring, so that the rings can move axially relative to each other. These can be used as free-end bearings. Cylindrical roller bearings, in which either the inner or outer ring has two ribs and the other ring has one, are capable of taking some axial load in one direction. Double-row cylindrical roller bearings have high radial rigidity and are used primarily for precision machine tools.
Pressed steel or machined brass cages are generally used, but sometimes molded polyamide cages are employed.

Needle Roller Bearings



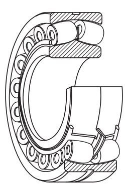
Needle roller bearings contain many slender rollers with a length 3 to 10 times their diameter. As a result, the ratio of the bearing outside diameter to the inscribed circle diameter is small, and they have a rather high radial load capacity.
There are numerous types available, and many have no inner ring. The drawn-cup type has a pressed-steel outer ring and the solid type has a machined outer ring. There are also cage and roller assemblies without rings. Most bearings have pressed-steel cages, but some do not use cages.

Tapered Roller Bearings



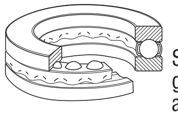
Bearings of this type use conical rollers guided by a back-face rib on the inner ring. These bearings are capable of taking high radial loads and also axial loads in one direction. The HR Series features a greater quantity of larger rollers, allowing for even higher load capacity.
They are generally mounted in pairs in a manner similar to single-row angular contact ball bearings. In this case, the proper internal clearance can be obtained by adjusting the axial distance between the inner or outer rings of the two opposed bearings. Since they are separable, the inner ring assemblies and outer rings can be mounted independently.
Tapered roller bearings are divided into three types depending on contact angle; these are normal angle, medium angle, and steep angle. Double-row and four-row tapered roller bearings are also available. Pressed-steel cages are generally used.

Spherical Roller Bearings

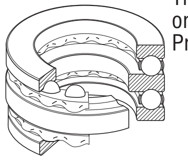


These bearings have barrel-shaped rollers between the inner ring, which has two raceways, and the outer ring, which has one spherical raceway. Since the center of curvature of the outer ring raceway surface coincides with the bearing axis, they are self-aligning in a manner similar to that of self-aligning ball bearings. Therefore, if there is deflection of the shaft or housing or misalignment of their axes, it is automatically corrected so that excessive force is not applied to the bearings.
Spherical roller bearings can take not only heavy radial loads, but also some axial loads in either direction. They have excellent radial load-carrying capacity and are suitable for use where there are heavy or impact loads.
Some bearings have tapered bores and may be mounted directly on tapered shafts or cylindrical shafts using adapters or withdrawal sleeves.
Pressed-steel and machined-brass cages are used.

Single-Direction Thrust Ball Bearings

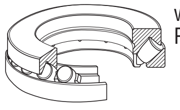


Double-Direction Thrust Ball Bearings



Single-direction thrust ball bearings are composed of washer-like bearing rings with raceway grooves. The ring attached to the shaft is called the shaft washer (or inner ring) while the ring attached to the housing is called the housing washer (or outer ring).
Double-direction thrust ball bearings have three rings with the middle ring (center ring) fixed to the shaft.
As their names imply, single-direction thrust bearings can take axial loads in one direction, while double-direction thrust bearings can take axial loads in both directions.
There are also thrust ball bearings with an aligning seat washer beneath the housing washer in order to compensate for shaft misalignment or mounting error.
Pressed-steel cages are usually used in smaller bearings and machined cages in larger bearings.

Thrust Spherical Roller Bearings



These bearings have a spherical raceway in the housing washer and barrel-shaped rollers obliquely arranged around it. Since the raceway in the housing washer is spherical, these bearings are self-aligning. They have a very high axial load capacity and are capable of taking moderate radial loads when an axial load is applied.
Pressed-steel cages or machined-brass cages are usually used.

1.3 Bearing Sizes

Rolling berings are classified into the following sizes based on their dimensions:

Size Classification	Nominal Bore Diameter	Nominal Outside Diameter
Miniature	–	$<$9 mm
Extra Small	$<$10 mm	$\geq$9 mm
Small	$\geq$10 mm	Up to ~80 mm
Medium	–	~80 to ~180 mm
Large	–	~180 to 800 mm
Extra Large	–	~800 mm and above

TYPES AND FEATURES OF ROLLING BEARINGS

Table 1. 1 Rolling Bearings: Types and Characteristics

Features \ Bearing Type	Deep Groove Ball Bearings	Magneto Bearings	Angular Contact Ball Bearings	Double-Row Angular Contact Ball Bearings	Duplex Angular Contact Ball Bearings	Four-Point-Contact Ball Bearings	Self-Aligning Ball Bearings	Cylindrical Roller Bearings	Double-Row Cylindrical Roller Bearings	Cylindrical Roller Bearings with Single Rib
Load Capacity: Radial Loads	◯	○	◎	◎	◎	○	◯	◎	⊚	◎
Load Capacity: Axial Loads	↔ ◯	← ○	← ◎	↔ ◎	↔ ◎	↔ ◎	↔ ○	×	×	← ◯
Load Capacity: Combined Loads	◯	○	◎	◎	◎	◯	○	×	×	◯
High Speeds	⊚	◎	⊚	◯	◎	◎	◎	⊚	◎	◎
High Accuracy	⊚		⊚		⊚	◎		⊚	⊚	
Low Noise and Torque	⊚							◎		
Rigidity					◎			◎	⊚	◎
Angular Misalignment	◎	○	○	○	○	○	⊚	◯	○	◯
Self-Aligning Capability							☆			
Ring Separability		☆				☆		☆	☆	☆
Fixed-End Bearing	☆			☆	☆	☆	☆			
Free-End Bearing	★			★	★	★	★	☆	☆	
Inner Ring Tapered Bore							☆		☆	
Remarks		Two bearings are usually mounted in opposition.	Contact angles of 15°, 25°, 30°, and 40°. Two bearings are usually mounted in opposition. Clearance adjustment is necessary.		Combination of DF and DT pairs is possible, but not for use on free-end.	Contact angle of 35°		Including N type	Including NNU type	Including NF type
Page No.	C005 C053	C005 C050	C072	C072 C106	C072	C072 C108	C114	C124	C124 C158	C124

Features \ Bearing Type	Cylindrical Roller Bearings with Thrust Collars	Needle Roller Bearings	Tapered Roller Bearings	Double-and Multi-Row Tapered Roller Bearings	Spherical Roller Bearings	Thrust Ball Bearings	Thrust Ball Bearings with Aligning Seat	Double-Direction Angular Contact Thrust Ball Bearings	Thrust Cylindrical Roller Bearings	Thrust Tapered Roller Bearings	Thrust Spherical Roller Bearings	Page No.
Load Capacity: Radial Loads	◎	◎	◎	⊚	⊚	×	×	×	×	×	○	—
Load Capacity: Axial Loads	↔ ◯	×	← ◎	↔ ◎	↔ ◯	← ◎	← ◎	↔ ◎	← ⊚	← ⊚	← ⊚	—
Load Capacity: Combined Loads	◯	×	◎	⊚	◎	×	×	×	×	×	○	—
High Speeds	◎	◎	◯	◯	◯	○	×	◯	○	○	○	A022 A098
High Accuracy			◎			◎		⊚				A023 A126 A151
Low Noise and Torque												A023
Rigidity	◎	◎	◎	⊚				◎	⊚	⊚		A023 A192
Angular Misalignment	◯	○	◯	○	⊚	×	⊚	×	×	×	⊚	A022 Blue pages of each brg. type
Self-Aligning Capability					☆		☆				☆	A022
Ring Separability	☆	☆	☆	☆		☆	☆	☆	☆	☆	☆	A023 A024
Fixed-End Bearing	☆			☆	☆							A026 to A029
Free-End Bearing		☆		★	★							A026 to A029
Inner Ring Tapered Bore					☆							A150 B008 B012
Remarks	Including NUP type		Two bearings are usually mounted in opposition. Clearance adjustment is necessary.	KH and KV types are also available but not for use on free-end.					Including needle roller thrust bearings		To be used with oil lubrication	
Page No.	C124	C341	C182	C182 C246	C258	C296	C296	—	C314	C322	C332	

⊚ Excellent ◎ Good ◯ Fair ○ Conditional × Unsuitable ← One direction only ↔ Two directions

☆ Applicable ★ Applicable, allow for shaft contraction/elongation at bearing fitting surfaces.

1.4 Contact Angle and Bearing Types

A bearing's contact angle (α) refers to the angle between a vertical plane of the rotation axis of the bearing and a straight line between the points where the rolling element comes in contact with the inner ring raceway and outer ring raceway.
Radial bearings and thrust bearings are classified depending on the size of the contact angle.

Figure 1.3 shows the relation between contact angle and load direction on the bearing.

$$\frac{\text{Radial bearing } \alpha\text{: Less than } 45°}{\text{(A primarily radial load is supported.)}}$$

$$\frac{\text{Thrust bearing } \alpha\text{: Over } 45°}{\text{(A primarily axial load is supported.)}}$$

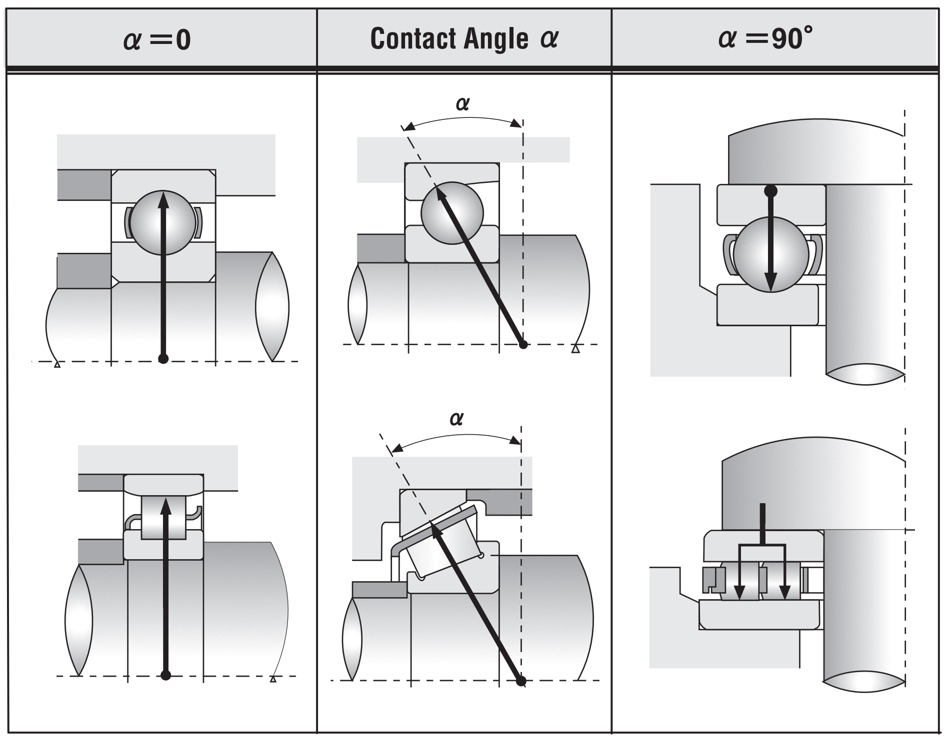


Fig. 1.3 Contact Angle α

1.5 Types of Load on Bearings

An example deep groove ball bearing is shown below in Figure 1.4 along with the types of load that may be applied to a rolling bearing. These are:

(a) Radial load
(b) Axial load
(c) Combined radial and axial load
(d) Moment load

It is important to select the optimum bearing type according to the type and magnitude of the load.

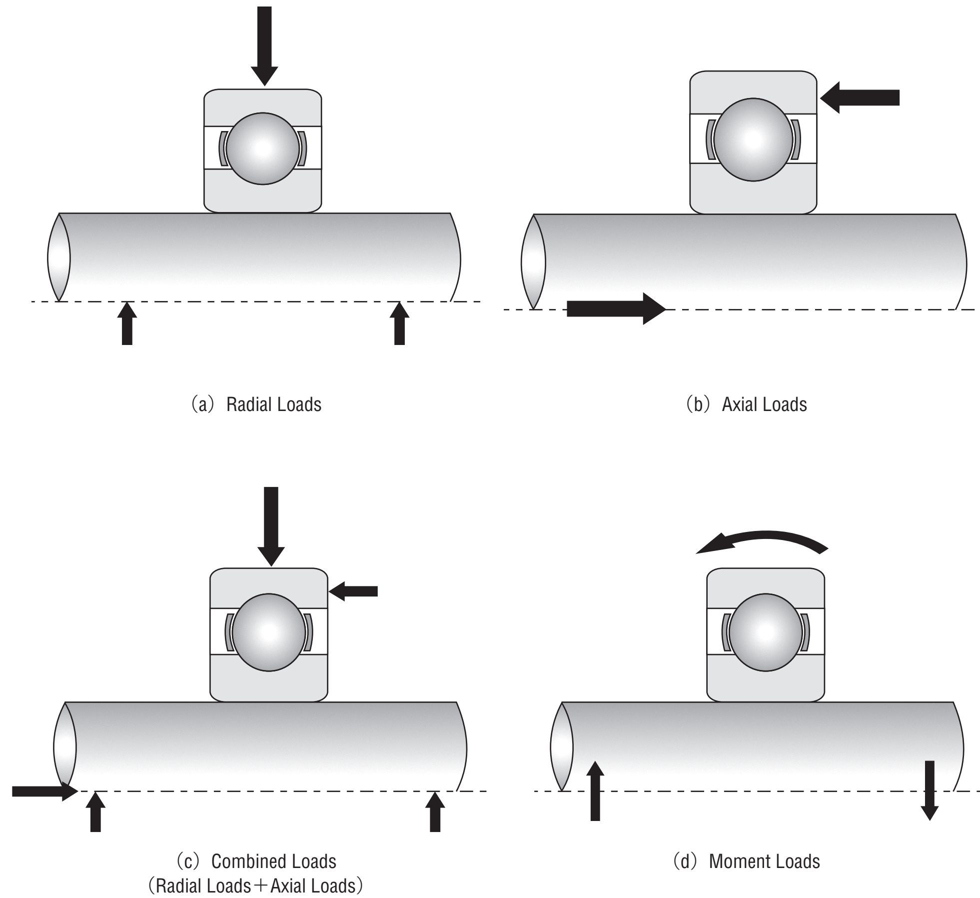


Fig. 1.4 Types of Load

2. SELECTION OF BEARING TYPE

2. SELECTION OF BEARING TYPE

2.1 Bearing Selection Procedure

There are nearly countless applications for rolling bearings; therefore, operating conditions and environments vary greatly. In addition, the diversity of operating conditions and bearing requirements continues to grow with the rapid advancement of technology. Bearings must be carefully studied from many angles to select the best choice from the thousands of types and sizes available.

Usually, a bearing type is provisionally chosen considering operating conditions, mounting arrangement, ease of mounting in the machine, allowable space, cost, availability, and other factors. Then the size of the bearing is chosen to satisfy the desired life requirement. When doing this, in addition to fatigue life, be sure to consider grease life, noise and vibration, wear, and other factors.

There is no fixed procedure for selecting bearings. It is good to investigate experience with similar applications and studies relevant to any special requirements for the specific application. When selecting bearings for new machines, unusual operating conditions, or harsh environments, please consult with NSK.

The following diagram (Fig.2.1) shows an example bearing selection procedure.

- Operating conditions and required performance
- Environmental conditions
- Dimensions of shaft and housing

Evaluation of bearing types

	Page number
Allowable space	A022, A104
Magnitude and direction of loads	A022
Vibration and shock	A044
Operating speed, maximum speed	A022, A098
Misalignment of inner and outer rings	A022
Fixing in axial direction and mounting arrangement	A026 to A029
Ease of bearing mounting and dismounting	A023
Sound and torque	A023
Required rigidity	A023, A192
Availability and cost	

→ Determination of bearing type and mounting arrangement

Determination of bearing size

	Page number
Expected life of machine	A032, A034
Dynamic and static equivalent loads	A050, A052
Speed	A098
Permissible static load factor	A052
Permissible axial loads (in the case of cylindrical roller bearings)	A064

→ Determination of bearing size

Evaluation of accuracy

	Page number
Running accuracy	A023
Rotational stability	A022, A098, A151
Torque fluctuation	A023

→ Selection of bearing accuracy class

Examination of fitting

	Page number
Operating conditions	A154
Magnitude and characteristics of loads	A154, A156
Temperature range	A156
Materials, size, accuracies of shaft and housing	A164, A270

→ Determination of fitting

Examination of internal clearance

	Page number
Fitting	
Difference in temperature between inner and outer rings	A175
Speed	
Misalignment of inner and outer rings	A022
Amount of preload	A194

→ Determination of internal clearance

Study of cage

- Speed
- Noise
- Operating temperature
- External vibration and shock
- Rapid acceleration and deceleration
- Moment load and misalignment

→ Selection of cage type and material

Examination of special specifications

	Page number
Operating temperature	A123
Environment (seawater, vacuum, gases, chemicals, etc.)	
Type of lubrication	

→ Selection of special material, heat treatment for dimensional stability

Examination of lubricating methods

	Page number
Operating temperature range	A229, A231, A236, A238
Speed	A098
Lubricating methods	A228
Type of seals	A272
Maintenance and inspection intervals	B013

→ Selection of lubricating method, lubricant, and type of seals

Examination of ease of mounting/dismounting

	Page number
Procedure for mounting and dismounting	B006, B011
Necessary equipment	B006, B011
Dimensions affecting mounting	A270

→ Determination of dimensions affecting mounting and procedure for mounting/dismounting

→ Final specifications for bearing and surrounding parts

Fig. 2.1 Flowchart for Selection of Rolling Bearings

2.2 Allowable Bearing Space

The allowable space for a rolling bearing and its adjacent parts is generally limited, so the type and size of the bearing must be selected within such limits. In most cases, the shaft diameter is fixed first by the machine design; therefore, the bearing is often selected based on bore size. Rolling bearings have numerous standardized Dimension Series and types from which to select the optimum bearing. Fig. 2.2 shows the Dimension Series of radial bearings and corresponding bearing types.

2.3 Load Capacity and Bearing Types

The axial load-carrying capacity of a bearing is closely related to radial load capacity based on the bearing design, as shown in Fig. 2.3. This figure shows that when bearings of the same Dimension Series are compared, roller bearings have a higher load capacity than ball bearings and are superior if shock loads exist.

2.4 Permissible Speed and Bearing Types

The maximum speed of rolling bearings varies not only with the type of bearing, but also with size, cage type, loads, lubricating method, heat dissipation, etc. Assuming the common oil bath lubrication method, the bearing types are roughly ranked from higher speed to lower speed as shown in Fig. 2.4.

2.5 Misalignment of Inner/Outer Rings and Bearing Types

Because of shaft deflection caused by applied loads, dimensional errors of the shaft and housing, and mounting errors, the inner and outer rings are slightly misaligned. The permissible misalignment varies depending on the bearing type and operating conditions, but usually it is less than 0.0012 radian (4').

When a large misalignment is expected, bearings with self-aligning capability, such as self-aligning ball bearings, spherical roller bearings, and certain bearing units should be selected (Figs. 2.5 and 2.6).

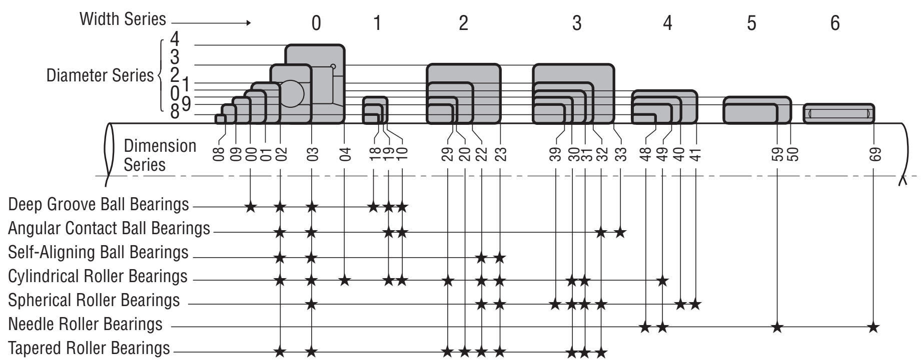
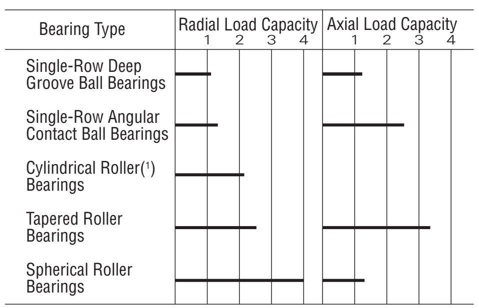


Fig. 2.2 Dimension Series of Radial Bearings



Note(1) Bearings with ribs can take some axial loads.

Fig. 2.3 Relative Load Capacities of Various Bearing Types

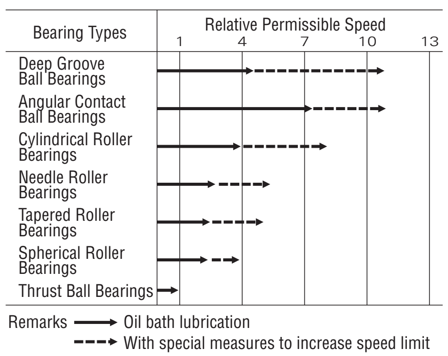


Fig. 2.4 Relative Permissible Speeds of Various Bearing Types

Permissible bearing misalignment is given at the beginning of the dimensional tables for each bearing type.

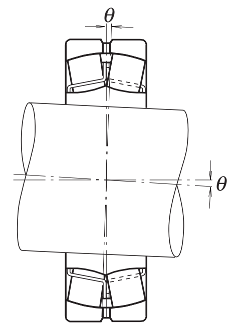


Fig. 2.5 Permissible Misalignment of Spherical Roller Bearings

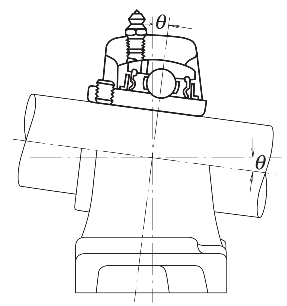


Fig. 2.6 Permissible Misalignment of Ball Bearing Units

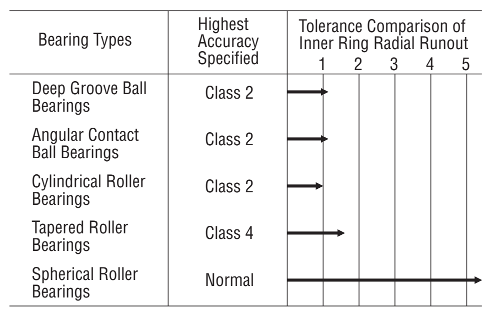


Fig. 2.7 Relative Inner Ring Radial Runout of Highest Accuracy Class for Various Bearing Types

2.6 Rigidity and Bearing Types

When loads are imposed on a rolling bearing, some elastic deformation occurs in the contact areas between the rolling elements and raceways. The rigidity of the bearing is determined by the ratio of bearing load to the amount of elastic deformation of the inner and outer rings and rolling elements. The main spindles of machine tools must have highly rigid bearings together with the rest of the spindle. Consequently, since roller bearings are deformed less by load, they are selected more often than ball bearings. When extra-high rigidity is required, bearings are given a preload, which means they have a negative clearance. Angular contact ball bearings and tapered roller bearings are often preloaded.

2.7 Noise and Torque of Various Bearing Types

Since rolling bearings are manufactured with very high precision, noise and torque are minimal. For deep groove ball bearings and cylindrical roller bearings in particular, the noise level is sometimes specified depending on their purpose. For high-precision miniature ball bearings, the starting torque can be specified. Deep groove ball bearings are recommended for applications in which low noise and torque are required, such as in motors or instruments.

2.8 Running Accuracy and Bearing Types

For the main spindles of machine tools that require high running accuracy or high-speed applications like superchargers, high-precision bearings of Accuracy Class 5, 4 or 2 are usually used.

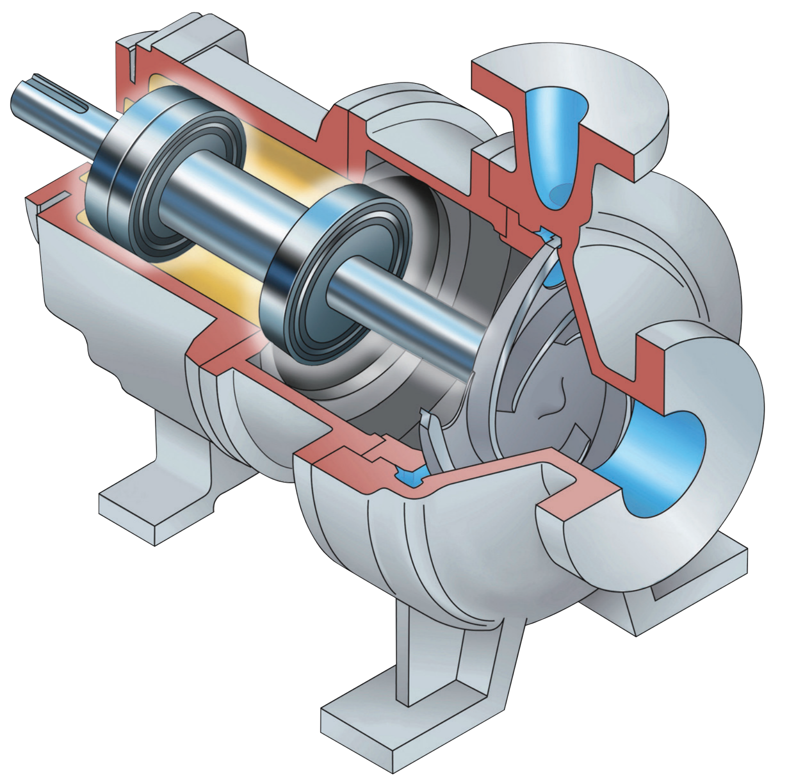
The running accuracy of rolling bearings is specified in various ways, and specified accuracy classes vary depending on the bearing type. A comparison of the inner ring radial runout for the highest running accuracy specified for each bearing type is shown in Fig. 2.7.

Deep groove ball bearings, angular contact ball bearings, and cylindrical roller bearings are most suitable for applications requiring high running accuracy.

2.9 Mounting and Dismounting of Various Bearing Types

Separable bearings, such as cylindrical roller bearings, needle roller bearings, and tapered roller bearings are convenient for mounting and dismounting. These types of bearings are recommended for machines in which bearings are mounted and dismounted rather often for periodic inspection. In addition, self-aligning ball bearings and spherical roller bearings (small-sized) with tapered bores can be mounted and dismounted relatively easily using sleeves.

3. SELECTION OF BEARING ARRANGEMENT



3. SELECTION OF BEARING ARRANGEMENT

In general, shafts are supported by only two rolling bearings. When considering the bearing mounting arrangement, the following items must be investigated:

(1) Expansion and contraction of the shaft caused by temperature variations
(2) Ease of bearing mounting and dismounting
(3) Misalignment of the inner and outer rings caused by deflection of the shaft or mounting error
(4) Rigidity of the entire system, including bearings and preloading method
(5) Capability to sustain loads at their proper positions and to transmit them

3.1 Fixed-End and Free-End Bearings

Usually only one "fixed-end" bearing on a shaft is used to fix the shaft axially. For this fixed-end bearing, a bearing type that can carry both radial and axial loads must be selected.

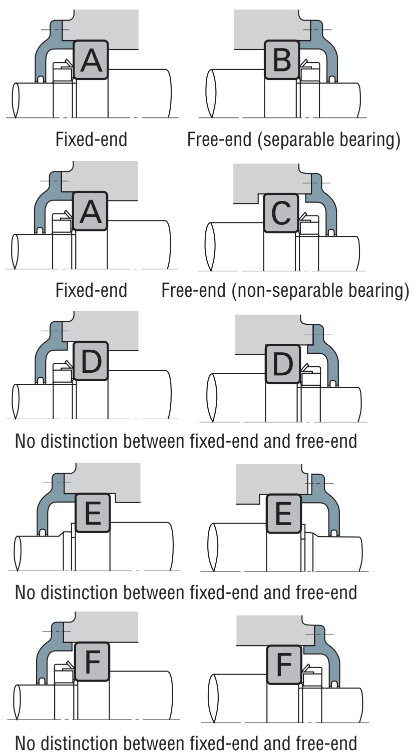
Other bearings must be "free-end" bearings that carry only radial loads to relieve the shaft's thermal elongation and contraction.

If measures to relieve a shaft's thermal elongation and contraction are insufficient, abnormal axial loads will be applied to the bearings, which can cause premature failure.

For free-end bearings, cylindrical roller bearings or needle roller bearings with separable inner and outer rings that are free to move axially (NU, N types, etc.) are recommended. Mounting and dismounting are also easier with these types.

When non-separable types are used as free-end bearings, usually the fit between the outer ring and housing is loose to allow axial movement of the running shaft together with the bearing. Sometimes, such elongation is relieved by a loose fit between the inner ring and shaft.

When the distance between the bearings is short and the influence of the shaft elongation and contraction is negligible, two opposed angular contact ball bearings or tapered roller bearings are used. The axial clearance (possible axial movement) after the mounting is adjusted using nuts or shims.



BEARING A
- Deep Groove Ball Bearing
- Matched Angular Contact Ball Bearing
- Double-Row Angular Contact Ball Bearing
- Self-Aligning Ball Bearing
- Cylindrical Roller Bearing with Ribs (NH, NUP types)
- Double-Row Tapered Roller Bearing
- Spherical Roller Bearing

BEARING B
- Cylindrical Roller Bearing (NU, N types)
- Needle Roller Bearing (NA type, etc.)

BEARING C(1)
- Deep Groove Ball Bearing
- Matched Angular Contact Ball Bearing (back-to-back)
- Double-Row Angular Contact Ball Bearing
- Self-Aligning Ball Bearing
- Double-Row Tapered Roller Bearing (KBE type)
- Spherical Roller Bearing

BEARING D,E(2)
- Angular Contact Ball Bearing
- Tapered Roller Bearing
- Magneto Bearing
- Cylindrical Roller Bearing (NJ, NF types)

BEARING F
- Deep Groove Ball Bearing
- Self-Aligning Ball Bearing
- Spherical Roller Bearing

Notes: (1) In the figure, shaft elongation and contraction are relieved at the outside surface of the outer ring, but sometimes this is done at the bore.
(2) For each type, two bearings are used in an opposed arrangement.

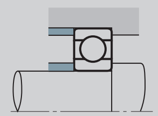
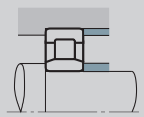
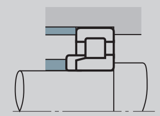
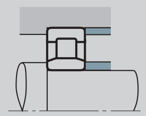
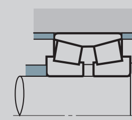
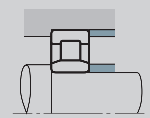
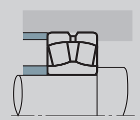
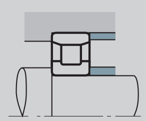
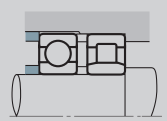
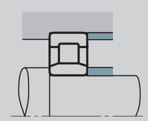
Fig. 3.1 Bearing Mounting Arrangements and Bearing Types

The distinction between free-end and fixed-end bearings and some possible bearing mounting arrangements for various bearing types are shown in Fig. 3.1.

3.2 Example Bearing Arrangements

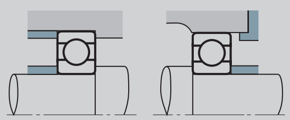
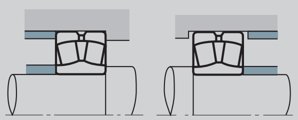
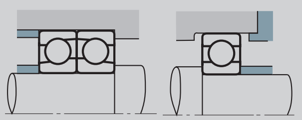
Some representative bearing mounting arrangements considering preload and rigidity of the entire assembly, shaft elongation and contraction, mounting error, etc. are shown in Table 3.1.

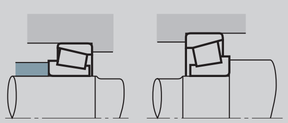
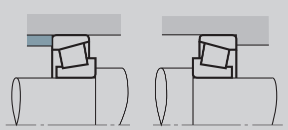
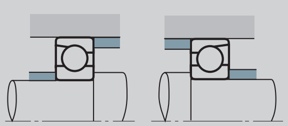
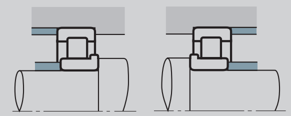
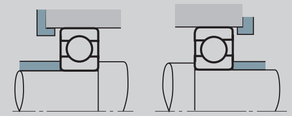
Table 3. 1 Representative Bearing Mounting Arrangements and Application Examples

Bearing Arrangements		Remarks	Application Examples
Fixed-end	Free-end		
		◯This is a common arrangement in which abnormal loads are not applied to bearings even if the shaft expands or contracts. ◯If mounting error is small, this is suitable for high speeds.	Medium-sized electric motors, blowers
		◯This arrangement can withstand heavy loads and shock loads and can take some axial load. ◯All cylindrical roller bearings are separable. This is helpful when interference is necessary for both the inner and outer rings.	Traction motors for rolling stock
		◯This arrangement is used when loads are relatively heavy. ◯A back-to-back type is used for maximum rigidity as a fixed-end bearing. ◯Both the shaft and housing must have high accuracy and the mounting error must be small.	Table rollers for steel mills, main spindles of lathes
		◯This arrangement is suitable when interference is necessary for both the inner and outer rings. Heavy axial loads cannot be applied.	Calender rolls of paper making machines, axles of diesel locomotives
		◯This arrangement is suitable for high speeds and heavy radial loads. Moderate axial loads can also be applied. ◯Some clearance is necessary between the outer ring of the deep groove ball bearing and the housing bore in order to avoid subjecting it to radial loads.	Reduction gears in diesel locomotives

Continued on next page

Table 3. 1 Representative Bearing Mounting Arrangements and Application Examples (cont'd)

Bearing Arrangements		Remarks	Application Examples
Fixed-end	Free-end		
		◯This is the most common arrangement. ◯It can sustain not only radial loads, but also moderate axial loads.	Double-suction volute pumps, automotive transmissions
		◯This is the most suitable arrangement when there is mounting error or shaft deflection. ◯It is often used for general and industrial applications in which heavy loads are applied.	Speed reducers, table rollers of steel mills, wheels for overhead travelling cranes
		◯This is suitable when there are rather heavy axial loads in both directions. ◯Double-row angular contact bearings may be used instead of an arrangement of two angular contact ball bearings.	Worm gear reducers

When there is no distinction between fixed-end and free-end	Remarks	Application Examples
 Back-to-back mounting  Face-to-face mounting	◯This arrangement is widely used since it can withstand heavy loads and shock loads. ◯The back-to-back arrangement is especially good when the distance between bearings is short and moment loads are applied. ◯Face-to-face mounting makes mounting easier when interference is necessary for the inner ring. In general, this arrangement is good when there is mounting error. ◯To use this arrangement with a preload, take extra care to ensure the correct amount of preload and clearance adjustment.	Pinion shafts of automotive differential gears, automotive front and rear axles, worm gear reducers
 Back-to-back mounting	◯This arrangement is used at high speeds when radial loads are not so heavy and axial loads are relatively heavy. ◯It provides good rigidity of the shaft by preloading. ◯For moment loads, back-to-back mounting is better than face-to-face mounting.	Grinding wheel shafts
 NJ + NJ mounting	◯This arrangement can withstand heavy loads and shock loads. ◯It can be used if interference is necessary for both the inner and outer rings. ◯Take care to ensure sufficient axial clearance during operation. ◯NF type + NF type mounting is also possible.	Final reduction gears of construction machines
	◯Sometimes a spring is used on the side of the outer ring of one bearing.	Small electric motors, small speed reducers, small pumps

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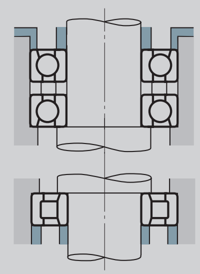
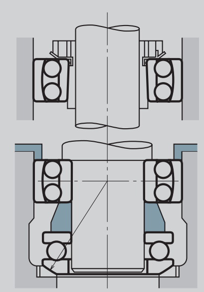
Vertical arrangements	Remarks	Application Examples
	◯Matched angular contact ball bearings are used on the fixed end. ◯A cylindrical roller bearing is used on the free end.	Vertical electric motors
	◯The spherical center of the self-aligning seat must coincide with that of the self-aligning ball bearing. ◯The upper bearing is on the free end.	Vertical openers (spinning and weaving machines)

4. SELECTION OF BEARING SIZE

4. SELECTION OF BEARING SIZE

4.1 Bearing Life

The functionality required of a rolling bearing varies per application and must be maintained for a specific period of time. Even if bearings are properly mounted and correctly operated, they will eventually fail to perform satisfactorily due to an increase in noise and vibration, loss of running accuracy, deterioration of grease, or fatigue flaking of the rolling surfaces.
Bearing life, in a broad sense of the term, is the period during which bearings continue to operate and satisfy their required functions. This bearing life may be defined as noise life, abrasion life, grease life, or rolling fatigue life, depending on which causes loss of bearing service.

Aside from the failure of bearings to function due to natural deterioration, bearings may fail when conditions such as heat-seizure, fracture, scoring of the rings, damage to the seals or cage, or other damage occurs.
Conditions such as these should not be interpreted as normal bearing failure since they often occur as a result of errors in bearing selection, improper design or manufacture of the bearing surroundings, incorrect mounting, or insufficient maintenance.

4.1.1 Rolling Fatigue Life and Basic Rating Life

When rolling bearings are operated under load, the raceways of their inner and outer rings and rolling elements are subjected to repeated cyclic stress. Because of metal fatigue of the rolling contact surfaces of the raceways and rolling elements, scaly particles may separate from the bearing material (Fig. 4.1). This phenomenon is called "spalling" or "flaking". Rolling fatigue life is represented by the total number of revolutions at which the bearing surface will start flaking due to stress. As shown in Fig. 4.2, even for seemingly identical bearings of the same type, size, and material that receive the same heat treatment and other processing, the rolling fatigue life varies greatly, even under identical operating conditions. This is because the flaking of materials due to fatigue is subject to many other variables. Consequently, "basic rating life", in which rolling fatigue life is treated as a statistical phenomenon, is used in preference to actual rolling fatigue life.

Suppose a number of bearings of the same type are operated individually under the same conditions. After a certain period of time, 10 % of them fail as a result of flaking caused by rolling fatigue. The total number of revolutions at this point is defined as the basic rating life or, if the speed is constant, the basic rating life is often expressed by the total number of operating hours completed when 10 % of the bearings become inoperable due to flaking.

In determining bearing life, basic rating life is often the only factor considered; however, other factors must also be taken into account. For example, the grease life of grease-lubricated bearings (refer to Section 11, Lubrication, Page A228) can be estimated. Since noise life and abrasion life are judged according to individual standards for different applications, specific values for noise or abrasion life must be determined empirically.

4.2 Basic Dynamic Load Rating and Fatigue Life

4.2.1 Basic Dynamic Load Rating

The basic dynamic load rating is defined as the constant load applied on bearings with stationary outer rings that the inner rings can endure for a rating life of one million revolutions (10^6 rev). The basic dynamic load rating of radial bearings is defined as a central radial load of constant direction and magnitude, while the basic load rating of thrust bearings is defined as an axial load of constant magnitude in the same direction as the central axis. Load ratings are listed under C_r for radial bearings and C_a for thrust bearings in the dimension tables.

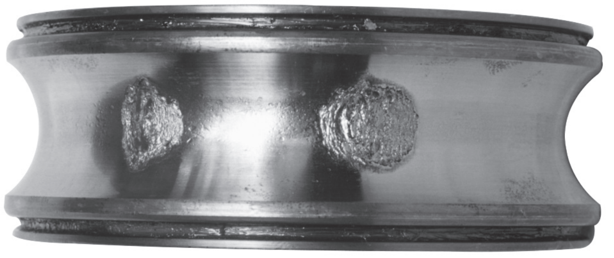


Fig. 4.1 Flaking Example

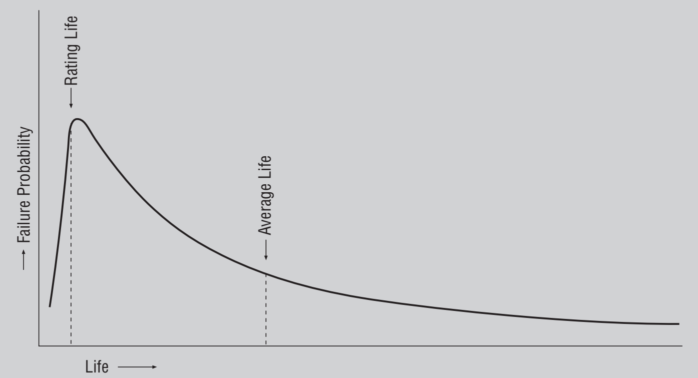


Fig. 4.2 Failure Probability and Bearing Life

4.2.2 Machinery in Which Bearings are Used and Projected Life

Selecting bearings with unnecessarily high load ratings is not advised, as such bearings may be too large and uneconomical. In addition, bearing life alone should not be the deciding factor in the selection of bearings. The strength, rigidity, and design of the shaft on which the bearings will be mounted should also be considered. Bearings are used in a wide range of applications and design life varies with specific applications and operating conditions. Table 4.1 gives empirical fatigue life factors derived from typical operating experience for various machines. Formulae for various life parameters can be found in Table 4.2.

Table 4.1 Fatigue Life Factor f_h for Various Bearing Applications

Operating Period	Fatigue Life Factor f_h				
	≤3	2–4	3–5	4–7	≥6
Infrequently used or only for short periods	·Small motors for home appliances, such as vacuum cleaners and washing machines ·Power tools	·Agricultural equipment			
Used only occasionally but reliability is important		·Motors for home heaters and air conditioners ·Construction equipment	·Conveyors ·Elevator cable sheaves		
Used intermittently for relatively long periods	·Rolling mill roll necks	·Small motors ·Deck cranes ·General cargo cranes ·Pinion stands ·Passenger cars	·Factory motors ·Machine tools ·Transmissions ·Vibrating screens ·Crushers	·Crane sheaves ·Compressors ·Specialized transmissions	
Used intermittently for more than eight hours daily		·Escalators	·Centrifugal separators ·Air conditioning equipment ·Blowers ·Woodworking machines ·Large motors ·Axle boxes on railway rolling stock	·Mine hoists ·Press flywheels ·Railway traction motors ·Locomotive axle boxes	·Paper making machines
Used continuously and high reliability is important					·Waterworks pumps ·Electric power stations ·Mine draining pumps

Table 4.2 Basic Rating Life, Fatigue Life Factor, and Speed Factor

Life Parameter	Ball Bearings	Roller Bearings
Basic Rating Life	$L_h=\frac{10^6}{60n}\left(\frac{C}{P}\right)^3=500f_h^3$	$L_h=\frac{10^6}{60n}\left(\frac{C}{P}\right)^{\frac{10}{3}}=500f_h^{\frac{10}{3}}$
Fatigue Life Factor	$f_h=f_n\frac{C}{P}$	$f_h=f_n\frac{C}{P}$
Speed Factor	$f_n=\left(\frac{10^6}{500\times60n}\right)^{\frac{1}{3}}=(0.03n)^{-\frac{1}{3}}$	$f_n=\left(\frac{10^6}{500\times60n}\right)^{\frac{3}{10}}=(0.03n)^{-\frac{3}{10}}$

n, f_n······Fig. 4.3 (See Page A036), Appendix Table 12 (See Page E018)
L_h, f_h····Fig. 4.4 (See Page A036), Appendix Table 13 (See Page E019)

4.2.3 Selection of Bearing Size Based on Basic Load Rating

The following relation exists between bearing load and basic rating life:

For ball bearings $L=\left(\frac{C}{P}\right)^3$ ··············· (4.1)

For roller bearings $L=\left(\frac{C}{P}\right)^{\frac{10}{3}}$ ·············· (4.2)

where L : Basic rating life (10^6 rev)
P : Bearing load (equivalent load) (N), {kgf} ....(Refer to Page A050)
C : Basic dynamic load rating (N), {kgf}
For radial bearings, C is written C_r
For thrust bearings, C is written C_a

It is convenient to express the fatigue life in terms of hours for bearings that run at a constant speed. In general, the fatigue life of bearings used in automobiles and other vehicles is given in terms of mileage.
By designating the basic rating life as L_h (h), bearing speed as n (min^{-1}), fatigue life factor as f_h, and speed factor as f_n, the relations shown in Table 4.2 are obtained.
If the bearing load P and speed n are known, it's possible to determine a fatigue life factor f_h appropriate for the desired life of the machine and then calculate the minimum basic load rating C with the following Equation:

$$C=\frac{f_h \cdot P}{f_n} \quad \cdots\cdots (4.3)$$

A bearing that satisfies this value of C should then be selected from the bearing tables.

4.2.4 Temperature Adjustment for Basic Load Rating

If rolling bearings are used at high temperatures, the hardness of the bearing steel decreases. Consequently, the basic load rating, which depends on the physical properties of the material, also decreases; therefore, the basic dynamic load rating should be adjusted for higher temperatures through the following equation:

$$C_t=f_t \cdot C \quad \cdots\cdots (4.4)$$

where C_t : Basic dynamic load rating after temperature correction (N), {kgf}
f_t : Temperature factor (See Table 4.3.)
C : Basic dynamic load rating before temperature adjustment (N), {kgf}

If bearings are used above 120 °C, they must be given a special dimensional stability heat treatment to prevent excessive dimensional changes. The basic dynamic load rating of bearings after such treatment may become lower than that listed in the bearing tables.

Table 4.3 Temperature Factor f_t

Bearing Temperature °C	125	150	175	200	250
Temperature Factor f_t	1.00	1.00	0.95	0.90	0.75

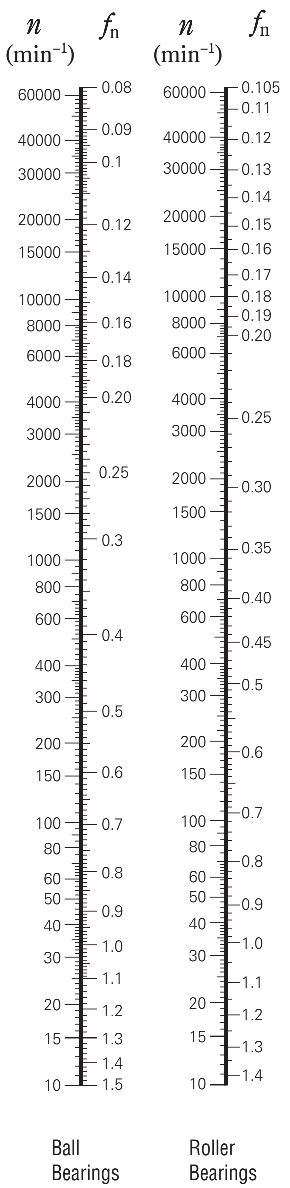


Fig. 4.3 Bearing Speed and Speed Factor

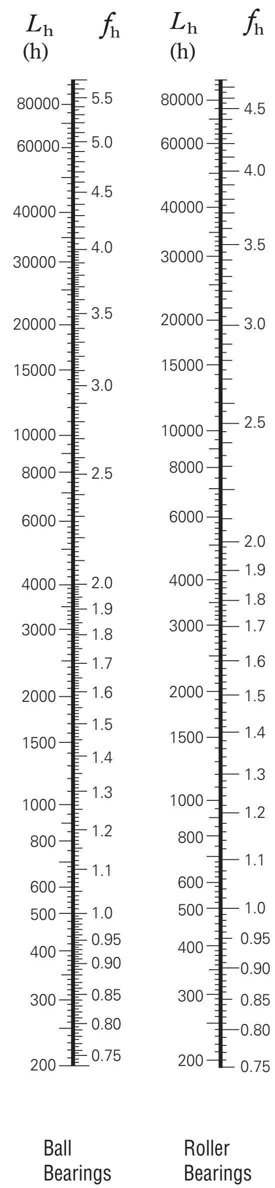


Fig. 4.4 Fatigue Life Factor and Fatigue Life

4.2.5 Correction of Basic Rating Life

As described previously, the basic equations for calculating basic rating life are as follows:

For ball bearings $L_{10} = \left(\frac{C}{P}\right)^3$ .... (**4.5**)

For roller bearings $L_{10} = \left(\frac{C}{P}\right)^{\frac{10}{3}}$ .... (**4.6**)

L_{10} life is defined as the basic rating life with a statistical reliability of 90%. Depending on the machines in which the bearings are used, sometimes a reliability higher than 90% may be required. However, recent improvements in bearing material have greatly extended fatigue life. In addition, the development of the Elasto-Hydrodynamic Theory of Lubrication proves that the thickness of the lubricating film in the contact zone between rings and rolling elements greatly influences bearing life. To reflect such improvements in the calculation of fatigue life, the basic rating life is adjusted using the following adjustment factors:

$L_{na} = a_1\, a_2\, a_3\, L_{10}$ .... (**4.7**)

where L_{na} : Adjusted rating life in which reliability, material improvements, lubricating conditions, etc. are considered

L_{10} : Basic rating life with a reliability of 90%

a_1 : Life adjustment factor for reliability

a_2 : Life adjustment factor for special bearing properties

a_3 : Life adjustment factor for operating conditions

The life adjustment factor for reliability a_1, is listed in Table 4.4 for reliabilities higher than 90%.
The life adjustment factor for special bearing properties a_2 is used to reflect improvements in bearing steel.
NSK now uses vacuum-degassed bearing steel, and test results show that life is greatly improved compared with earlier materials. The basic load ratings C_r and C_a listed in the bearing tables were calculated considering the extended life achieved by improvements in materials and manufacturing techniques. Consequently, when estimating life using Equation (4.7), you may assume that a_2 is greater than one.

Table 4.4 Reliability Factor a_1

Reliability (%)	90	95	96	97	98	99
a_1	1.00	0.64	0.55	0.47	0.37	0.25

The life adjustment factor for operating conditions a_3 is used to adjust for various factors, particularly lubrication. If there is no misalignment between the inner and outer rings and the thickness of the lubricating film in the contact zones of the bearing is sufficient, it is possible for a_3 to be greater than one; however, a_3 is less than one in the following cases:

- When the viscosity of the lubricant in the contact zones between the raceways and rolling elements is low.
- When the circumferential speed of the rolling elements is very slow.
- When bearing temperature is high.
- When lubricant is contaminated by water or foreign matter.
- When misalignment of the inner and outer rings is excessive.

It is difficult to determine the proper value of a_3 for specific operating conditions because there are still many unknowns. Since the special bearing property factor a_2 is also influenced by the operating conditions, there is a proposal to combine a_2 and a_3 into one quantity($a_2 \times a_3$), and not consider them independently. In this case, under normal lubricating and operating conditions, the product ($a_2 \times a_3$) should be assumed equal to one. However, if the viscosity of the lubricant is too low, the value drops to as low as 0.2.
If there is no misalignment and a lubricant with high viscosity is used so that a sufficient fluid-film thickness is secured, the product of ($a_2 \times a_3$) may be about two.

When selecting a bearing based on the basic load rating, it is best to choose an a_1 reliability factor appropriate for the projected use and an empirically determined C/P or f_h value derived from past results for lubrication, temperature, mounting conditions, etc. in similar machines.

The basic rating life Equations (4.1), (4.2), (4.5), and (4.6) give satisfactory results for a broad range of bearing loads. However, extra-heavy loads may cause detrimental plastic deformation at ball/raceway contact points. When P_r exceeds C_{0r} (basic static load rating) or 0.5 C_r, whichever is smaller, for radial bearings or P_a exceeds 0.5 C_a for thrust bearings, please consult NSK to establish the applicablity of the rating fatigue life equations.

4.2.6 Life Calculation of Multiple Bearings as a Group

When multiple rolling bearings are used in one machine, the fatigue life of individual bearings can be determined if the load acting on individual bearings is known. However, a machine generally becomes inoperative if a bearing in any part fails. It may therefore be necessary in certain cases to know the fatigue life of a group of bearings used in one machine. The fatigue life of the bearings varies greatly and our fatigue life calculation equation

$L=\left(\frac{C}{P}\right)^{p}$ applies to the 90% life (also called the rating fatigue life, which is either the gross number of revolutions or hours that 90% of multiple similar bearings operated under similar conditions can reach). In other words, the calculated fatigue life for one bearing has a probability of 90%. Since the endurance probability of a group of multiple bearings for a certain period is a product of the endurance probability of individual bearings for the same period, the rating fatigue life of a group of multiple bearings is not determined solely from the shortest rating fatigue life among the individual bearings. In fact, the group life is much shorter than the life of the bearing with the shortest fatigue life.

Assuming the rating fatigue life of individual bearings as L_1, L_2, L_3 ... and the rating fatigue life of the entire group of bearings as L, the equation below is obtained:

$$\frac{1}{L^{e}}=\frac{1}{L_1^{e}}+\frac{1}{L_2^{e}}+\frac{1}{L_3^{e}}+\cdots\cdots \quad \textbf{(4.8)}$$

where e=1.1 (both for ball and roller bearings)

L of Equation (4.8) can be determined easily by using Fig. 4.5.

To use this chart, find the L_1 value from Equation (4.8) on the L_1 scale and the value of L_2 on the L_2 scale, connect them with a straight line, and read where the line intersects the L scale. In this way, the value of L_A in

$$\frac{1}{L_A^{e}}=\frac{1}{L_1^{e}}+\frac{1}{L_2^{e}}$$

can be determined. Take this value L_A on the L_1 scale and the L_3 value on the L_2 scale, connect them with a straight line, and read where the line intersects the L scale. In this way, the value L in

$$\frac{1}{L^{e}}=\frac{1}{L_1^{e}}+\frac{1}{L_2^{e}}+\frac{1}{L_3^{e}}$$

can be determined.

Example

Assume that the calculated fatigue life values of automotive front wheel bearings are as follows:
280 000 km for inner bearing
320 000 km for outer bearing
Then, the fatigue life of the wheel bearings can be determined as 160 000 km from Fig. 4.5.
If the fatigue life of the bearing of the right-hand wheel bearing takes this value, the fatigue life of the left-hand wheel bearing will be the same. As a result, the fatigue life of the front wheels as a group is 85 000 km.

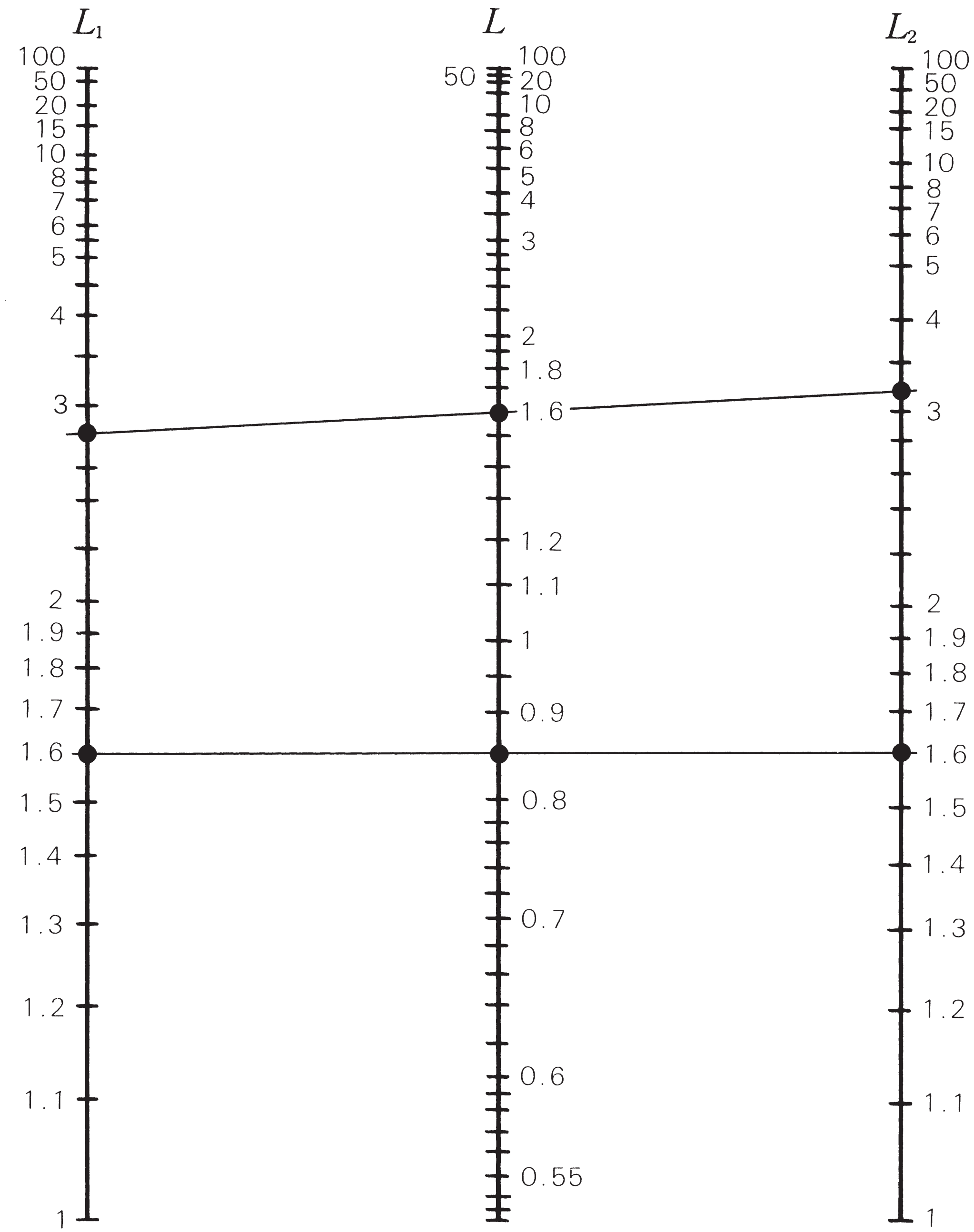


Fig. 4.5 Life Calculation Chart

4.2.7 New Life Theory

Bearing technology has advanced rapidly in recent years, particularly in the areas of dimensional accuracy and material cleanliness. As a result, bearings in a clean environment can now have a longer rolling fatigue life than the life obtained by the traditional ISO life calculation formula. This extended life is partly due to the important advancements in bearing-related technology, such as lubrication, cleanliness, and filtration.

The conventional life calculation formula (Equation 4.9) based on the theories of G. Lundberg and A. Palmgren (L-P theory, hereafter) addresses only subsurface-originated flaking. In this phenomenon, cracks initially occur due to dynamic shear stress immediately below the rolling surface and progressively reach the surface in the form of flaking.

$$ln\frac{1}{S} \propto \frac{\tau_o^{c} \cdot N^{e} \cdot V}{Z_o^{h}} \quad \cdots\cdots \quad (4.9)$$

NSK's new life calculation formula theorizes that rolling fatigue life is the sum total of the combined effects of both subsurface-originated flaking and surface-originated flaking occurring simultaneously.

NSK New Life Calculation Formula

(1) *Subsurface-originated flaking*
A precondition of subsurface-originated flaking in rolling bearings is contact of the rolling elements with the raceway via a sufficient and continuous oil film under clean lubrication conditions.
Fig. 4.6 plots the L_{10} life for each test condition with maximum surface contact pressure (P_{max}) and the number of repeated stresses cycles.
In the figure, line L_{10} theoretical refers to the theoretical line obtained using the conventional life calculation formula. As maximum surface contact pressure decreases, the line representing actual life separates from the conventionally calculated theoretical line towards longer life. This separation suggests the presence of a fatigue load limit P_u below which no rolling fatigue occurs. The difference between NSK's revised theory (Equation 4.10) and the conventional theory is better illustrated in Fig. 4.7.

$$ln\frac{1}{S} \propto N^{e}\int_{V} \frac{(\tau-\tau_u)^{c}}{Z_o^{h}}dV \quad \cdots\cdots \quad (4.10)$$

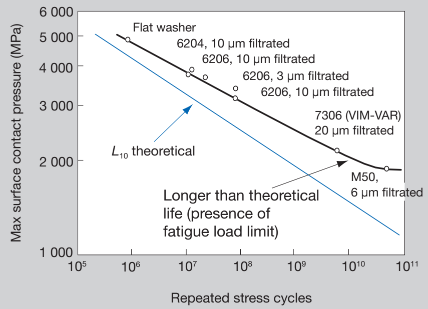


Fig. 4.6 Life Test Results Under Clean Lubrication

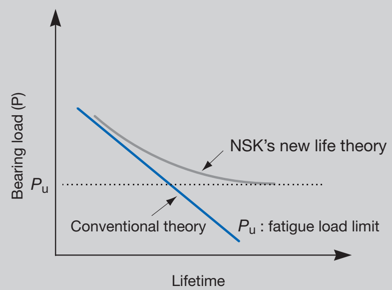


Fig. 4.7 NSK's New Life Theory That Considers Fatigue Limit

(2) *Surface-originated flaking*
Under actual bearing operation, the lubricant is often contaminated with foreign objects such as metal chips, burrs, cast sand, etc.
When foreign particles are mixed in the lubricant, they become pressed onto the raceways by the rolling elements and dents occur on the surfaces of the raceways and rolling elements. Stress concentrations occur at the edges of the dents, generating fine cracks that over time propagate into flaking of the raceways and rolling elements.
As shown in Fig. 4.8, actual life is shorter than conventional calculated life under contaminated lubrication conditions at low maximum surface pressure. This result shows that the actual life under contaminated lubrication is further shortened compared to the theoretical life because of the decrease in maximum surface contact pressure.

Table 4.5 Values of Contamination Coefficient a_c

	Very Clean	Clean	Normal	Contaminated	Heavily Contaminated
a_c Factor	1	0.8	0.5	0.4–0.1	0.05
Application Guide	10 μm filtration	10–30 μm filtration	30–100 μm filtration	Greater than 100 μm filtration or no filtration (oil bath, circulating lubrication, etc.)	No filtration, presence of many fine particles
Application Examples	Sealed grease-lubricated bearings for electrical appliances, information technology equipment, etc.	Sealed grease-lubricated bearings for electric motors, railway axle boxes, machine tools, etc.	Normal usage, automotive hub unit bearings, etc.	Bearings for automotive transmissions, industrial gearboxes, construction machines, etc.	—

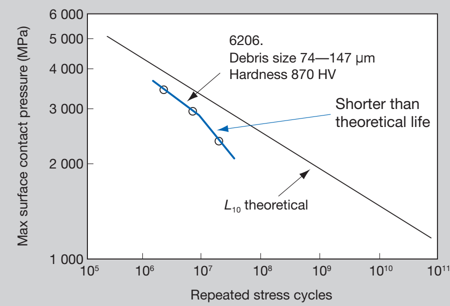


Fig. 4.8 Life Test Results Under Contaminated Lubrication Conditions

Therefore, the new NSK life calculation formula considers the trend of results in life tests under clean conditions and low load. Based on these results, the new life equation is a function of $(P\text{-}P_u)/C$ affected by specific lubrication conditions identified by the lubrication parameter. In addition, the effects of different types and shapes of foreign particles are assumed to be strongly influenced by bearing load and lubrication conditions. Such a relationship can be expressed as a function of the load parameter. This relationship of the new life calculation formula is defined as $(P\text{-}P_u)/C{\cdot}1/a_c$.
The Equation for calculating surface-originated flaking based on the above concept is as follows:

$$ln\frac{1}{S} \propto N^{e}\int_{V} \frac{(\tau-\tau_u)^{c}}{Z_o^{h}}dV \times \left\{\frac{1}{f(a_c,a_L)}-1\right\} \quad \cdots\cdots \quad (4.11)$$

(3) *Calculation of Contamination Coefficient a_c*
The contamination coefficient in Table 4.5 is determined in terms of lubrication cleanliness. Test results on ball and roller bearings with grease and filtered lubrication show life as a number of times longer than the conventional calculation would suggest. Yet, hardness becomes a factor when foreign particles harder than Hv350 enter the bearing, causing dents on the raceway. Fatigue damage from these dents can progress to flaking in a short time. Test results on ball and roller bearings under foreign particle contamination show from 1/3 to 1/10 the conventionally calculated life.
Based on these test results, the contamination coefficient a_c for NSK's new life theory is classified into five levels.

(4) *New life calculation formula* L_{able}

The following formula, which combines subsurface-originated flaking and surface-originated flaking, is proposed as the new life calculation formula:

$$ln\frac{1}{S} \propto N^e \int_V \frac{(\tau-\tau_u)^c}{Z_o^h} dV \times \left\{\frac{1}{f(a_c, a_L)}\right\} \cdots\cdots \textbf{(4.12)}$$

$$L_{able} = a_1 \cdot a_{NSK} \cdot L_{10} \cdots\cdots \textbf{(4.13)}$$

Life Correction Factor a_{NSK}

The life correction factor a_{NSK} is the function of lubrication parameter $(P\text{-}P_u)/C \cdot 1/a_c$ as shown below:

$$a_{NSK} \propto F\left\{\frac{P\text{-}P_u}{C} \cdot \frac{1}{a_c}, a_L\right\} \cdots\cdots \textbf{(4.14)}$$

NSK's new life theory considers the life-extending effect of improved material and heat treatment by correcting contamination factor a_c. The theory also utilizes viscosity ratio κ ($\kappa = \nu/\nu_1$ where ν is the operational viscosity and ν_1 the required viscosity) because the lubrication parameter a_L changes with the degree of oil film formation based on the lubricant and operating temperature. The theory indicates that the better the lubrication conditions (higher κ), the longer the life.

Figures 4.9 and 4.10 show the diagrams of correction factor a_{NSK} as a function of the new life calculation formula. In addition, point contact and line contact are considered separately for ball and roller bearings respectively in this new formula.

List of symbols used:
S : Probability that flaking does not occur after stress has been repeated N times
N : Number of repeated stresses
τ : Internal stress
τu : Internal stress at fatigue limit
V : Stress volume
Z0 : Depth at which maximum shear stress occurs
ac : Contamination coefficient
aL : Lubrication parameter
(a function of viscosity ratio κ)
P : Load applied to bearing
Pu : Fatigue load limit
C : Basic dynamic load rating
e, c, h : Constants

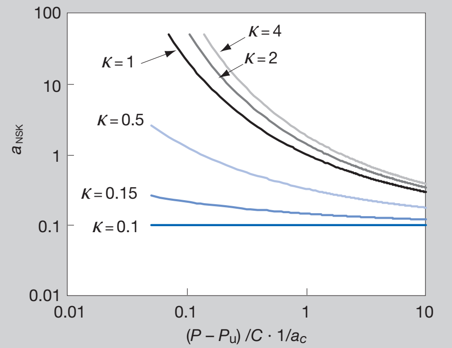


Fig. 4.9 New Life Calculation Diagram for Ball Bearings

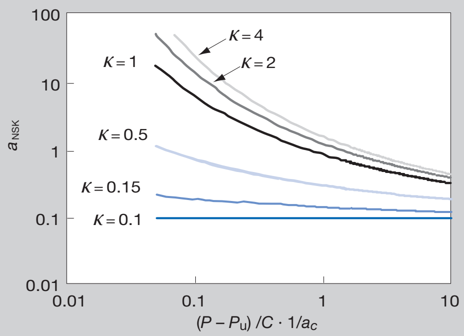


Fig. 4.10 New Life Calculation Diagram for Roller Bearings

Table 4.5 Reliability Factor a_1 Used in New Life Calculauton Formula L_{able}

Reliability (%)	90	95	96	97	98	99
a_1	1.00	0.62	0.53	0.44	0.33	0.21

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4.3 Calculation of Bearing Loads

The loads applied on bearings generally include the weight of the body to be supported by the bearings, the weight of the rotating elements themselves, the transmission power of gears and belting, the load produced by operation of the machine in which the bearings are used, and so on. These loads can be theoretically calculated, but some of them are difficult to estimate. Therefore, be sure to correct estimates using empirically derived data.

4.3.1 Load Factor

When a radial or axial load has been mathematically calculated, the actual load on the bearing may be greater than the calculated load because of vibration and shock present during machine operation. The actual load may be calculated using the following equation:

$$\left.\begin{aligned} F_r &= f_w \cdot F_{rc} \\ F_a &= f_w \cdot F_{ac} \end{aligned}\right\} \quad \cdots\cdots (4.15)$$

where F_r, F_a : Loads applied on bearing (N), {kgf}

F_{rc}, F_{ac} : Theoretically calculated load (N), {kgf}

f_w : Load factor

The values given in Table 4.6 are usually used for the load factor f_w.

Table 4.6 Values of Load Factor f_w

Operating Conditions	Typical Applications	f_w
Smooth operation free from shocks	Electric motors, Machine tools, Air conditioners	1 to 1.2
Normal operation	Air blowers, Compressors, Elevators, Cranes, Paper making machines	1.2 to 1.5
Operation accompanied by shock and vibration	Construction equipment, Crushers, Vibrating screens, Rolling mills	1.5 to 3

4.3.2 Bearing Loads in Belt or Chain Transmission Applications

The force acting on the pulley or sprocket wheel when power is transmitted with a belt or chain is calculated using the following equations:

$$\left.\begin{aligned} M &= 9\,550\,000H / n \;....(\mathrm{N \cdot mm}) \\ &= 974\,000H / n \;....\{\mathrm{kgf \cdot mm}\} \end{aligned}\right\} \quad \cdots\cdots (4.16)$$

$$P_k = M / r \quad \cdots\cdots (4.17)$$

where M : Torque acting on pulley or sprocket wheel (N·mm), {kgf·mm}

P_k : Effective force transmitted by belt or chain (N), {kgf}

H : Power transmitted(kW)

n : Speed (min^{-1})

r : Effective radius of pulley or sprocket wheel (mm)

When calculating the load on a pulley shaft, belt tension must be included. Thus, to calculate actual load K_b for belt transmissions, the effective transmitting power is multiplied by the belt factor f_b, which represents belt tension. The values of the belt factor f_b for different types of belts are shown in Table 4.7.

$$K_b = f_b \cdot P_k \quad \cdots\cdots (4.18)$$

For chain transmissions, values corresponding to f_b should be 1.25 to 1.5.

Table 4.7 Belt Factor f_b

Type of Belt	f_b
Toothed belts	1.3 to 2
V belts	2 to 2.5
Flat belts with tension pulley	2.5 to 3
Flat belts	4 to 5

4.3.3 Bearing Loads in Gear Transmission Applications

The loads imposed on gears in gear transmissions vary according to the type of gears used. In the simplest case of spur gears, the load is calculated as follows:

$$\left.\begin{aligned} M &= 9\,550\,000H / n \;....(\mathrm{N \cdot mm}) \\ &= 974\,000H / n \;....\{\mathrm{kgf \cdot mm}\} \end{aligned}\right\} \quad \cdots\cdots (4.19)$$

$$P_k = M / r \quad \cdots\cdots (4.20)$$

$$S_k = P_k \tan\theta \quad \cdots\cdots (4.21)$$

$$K_c = \sqrt{P_k^2 + S_k^2} = P_k \sec\theta \quad \cdots\cdots (4.22)$$

where M : Torque applied to gear (N · mm),{kgf · mm}

P_k : Tangential force on gear (N), {kgf}

S_k : Radial force on gear (N), {kgf}

K_c : Combined force imposed on gear (N), {kgf}

H : Power transmitted (kW)

n : Speed (min^{-1})

r : Pitch circle radius of drive gear (mm)

θ : Pressure angle

In addition to the theoretical load calculated above, vibration and shock, which depend on how accurately the gear is finished, should be included by multiplying the theoretically calculated load by gear factor f_g.
The values of f_g should generally fall within the ranges in Table 4.8. When vibration from other sources occurs, the actual load is obtained by multiplying the load factor by this gear factor.

Table 4.8 Values of Gear Factor f_g

Gear Finish Accuracy	f_g
Precision ground gears	1 ∼1.1
Ordinary machined gears	1.1∼1.3

4.3.4 Load Distribution on Bearings

In the simple examples shown in Figs. 4.11 and 4.12, the radial loads on bearings Ⅰ and Ⅱ can be calculated using the following equations:

$$F_{C\,\mathrm{I}} = \frac{b}{c}K \quad \cdots\cdots (4.23)$$

$$F_{C\,\mathrm{II}} = \frac{a}{c}K \quad \cdots\cdots (4.24)$$

where $F_{C\,\mathrm{I}}$: Radial load applied on bearing Ⅰ (N), {kgf}

$F_{C\,\mathrm{II}}$: Radial load applied on bearing Ⅱ (N), {kgf}

K : Shaft load (N), {kgf}

When these loads are applied simultaneously, first the radial load for each should be obtained, then the sum of the vectors may be calculated according to the load direction.

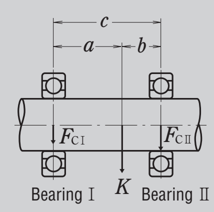


Fig. 4.11 Radial Load Distribution (1)

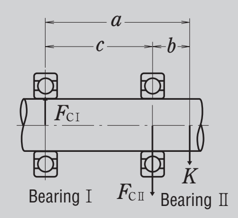


Fig. 4.12 Radial Load Distribution (2)

4.3.5 Average Fluctuating Load

When the load applied on bearings fluctuates, an average load that will yield the same bearing life as the fluctuating load should be calculated.

(1) When the relation between load and rotating speed is divided into the following steps (Fig. 4.13),

Load F_1 : Speed n_1 ; Operating time t_1
Load F_2 : Speed n_2 ; Operating time t_2
$\vdots$
Load F_n : Speed n_n ; Operating time t_n

the average load F_m may be calculated using the following equation:

$$F_m = \sqrt[p]{\frac{F_1^p n_1 t_1 + F_2^p n_2 t_2 + \cdots + F_n^p n_n t_n}{n_1 t_1 + n_2 t_2 + \cdots\cdots + n_n t_n}} \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.25})$$

where F_m : Average fluctuating load (N), {kgf}
$p = 3$ for ball bearings
$p = 10/3$ for roller bearings

The average speed n_m may be calculated as follows:

$$n_m = \frac{n_1 t_1 + n_2 t_2 + \cdots + n_n t_n}{t_1 + t_2 + \cdots\cdots + t_n} \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.26})$$

(2) When the load fluctuates almost linearly (Fig. 4.14), the average load may be calculated as follows:

$$F_m \fallingdotseq \frac{1}{3}(F_{min} + 2F_{max}) \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.27})$$

where F_{min} : Minimum fluctuating load value (N), {kgf}
F_{max} : Maximum fluctuating load value (N), {kgf}

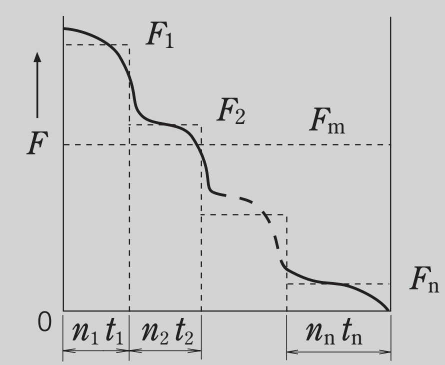


Fig. 4.13 Incremental Load Variation

(3) When the load fluctuation is similar to a sine wave (Fig. 4.15), an approximate value for the average load F_m may be calculated from the following equation:

In the case of Fig. 4.15 (a)

$$F_m \fallingdotseq 0.65\,F_{max} \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.28})$$

In the case of Fig. 4.15 (b)

$$F_m \fallingdotseq 0.75\,F_{max} \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.29})$$

(4) When both a rotating load and a stationary load are applied (Fig. 4.16),

F_R : Rotating load (N), {kgf}
F_S : Stationary load (N), {kgf}

an approximate value for the average load F_m may be calculated as follows:

a) Where $F_R \geqq F_S$

$$F_m \fallingdotseq F_R + 0.3F_S + 0.2\frac{F_S^2}{F_R} \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.30})$$

b) Where $F_R < F_S$

$$F_m \fallingdotseq F_S + 0.3F_R + 0.2\frac{F_R^2}{F_S} \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.31})$$

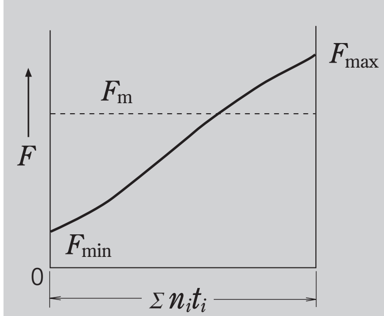


Fig. 4.14 Simple Load Fluctuation

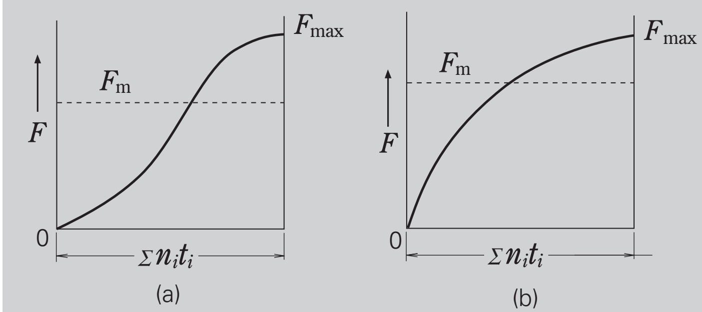


Fig. 4.15 Sinusoidal Load Variation

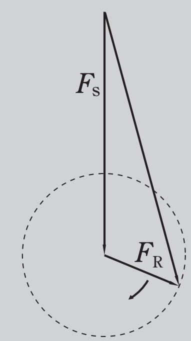


Fig. 4.16 Rotating Load and Stationary Load

4.3.6 Combination of Rotating and Stationary Loads

Generally, rotating, static, and indeterminate loads act on a rolling bearing. In certain cases, both the rotating load, which is caused by weight from unbalance or vibration, and the stationary load, which is caused by gravity or power transmission, may act simultaneously. The combined mean effective load can be calculated when indeterminate load is caused by rotating and static loads. There are two kinds of combined loads: rotating and stationary, which are classified depending on the magnitude of the load, as shown in Fig. 4.17.

Namely, the combined load becomes a running load with magnitude changing as shown in Fig. 4.17 (a) if the rotating load is larger than the static load. The combined load becomes an oscillating load with a magnitude changing as shown in Fig. 4.17 (b) if the rotating load is smaller than the stationary load.

In either case, the combined load F is expressed by the following equation:

$$F=\sqrt{F_R^2+F_S^2-2F_RF_S\cos\theta} \quad \cdots\cdots \quad \mathbf{(4.32)}$$

where F_R : Rotating load (N), {kgf}
F_S : Stationary load (N), {kgf}
θ : Angle defined by rotating and stationary loads

Combined load F can be approximated by Equations (4.33) and (4.34) which vary sinusoidally depending on the magnitude of F_R and F_S. F_R+F_S becomes the maximum load F_{max} and F_R-F_S becomes the minimum load F_{min} for $F_R \gg F_S$ or $F_R \ll F_S$.

$$F_R \gg F_S,\ F=F_R-F_S\cos\theta \quad \cdots\cdots \quad \mathbf{(4.33)}$$
$$F_R \ll F_S,\ F=F_S-F_R\cos\theta \quad \cdots\cdots \quad \mathbf{(4.34)}$$

Combined load F can also be approximated by Equations (4.35) and (4.36) when $F_R \fallingdotseq F_S$.

$F_R > F_S$,

$$F=F_R-F_S+2F_S\sin\frac{\theta}{2} \quad \cdots\cdots \quad \mathbf{(4.35)}$$

$F_R < F_S$,

$$F=F_S-F_R+2F_R\sin\frac{\theta}{2} \quad \cdots\cdots \quad \mathbf{(4.36)}$$

Curves of Equations (4.32), (4.33), (4.35), and (4.36) are shown in Fig. 4.18.

The mean load variation F_m as expressed by Equations (4.33) and (4.34) or (4.35) and (4.36) can be expressed respectively by Equations (4.37) and (4.38) or (4.39) and (4.40).

$$F_m=F_{min}+0.65\,(F_{max}-F_{min})$$
$$F_R \geqq F_S,\ F_m=F_R+0.3F_S \quad \cdots\cdots \quad \mathbf{(4.37)}$$
$$F_R \leqq F_S,\ F_m=F_S+0.3F_R \quad \cdots\cdots \quad \mathbf{(4.38)}$$

$$F_m=F_{min}+0.75\,(F_{max}-F_{min})$$
$$F_R \geqq F_S,\ F_m=F_R+0.5F_S \quad \cdots\cdots \quad \mathbf{(4.39)}$$
$$F_R \leqq F_S,\ F_m=F_S+0.5F_R \quad \cdots\cdots \quad \mathbf{(4.40)}$$

Generally, as the value F exists somewhere along Equations (4.37), (4.38), (4.39), and (4.40), the coefficient 0.3 or 0.5 of the second terms in these equations is assumed to change linearly along with F_S/F_R or F_R/F_S. Thus, these factors may be expressed as follows:

$$0.3+0.2\frac{F_S}{F_R},\ 0 \leqq \frac{F_S}{F_R} \leqq 1$$

or $$0.3+0.2\frac{F_R}{F_S},\ 0 \leqq \frac{F_R}{F_S} \leqq 1$$

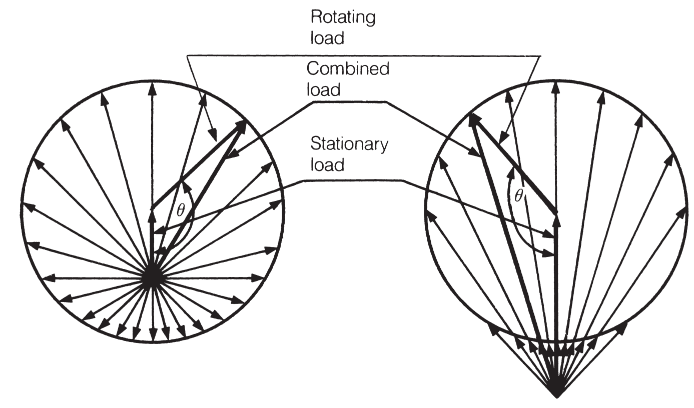
Accordingly, F_m can be expressed by the following equation:

$F_R \geqq F_S$,

$$F_m=F_R+\left(0.3+0.2\frac{F_S}{F_R}\right)F_S$$
$$=F_R+0.3F_S+0.2\frac{F_S^2}{F_R} \quad \cdots\cdots \quad \mathbf{(4.41)}$$

$F_R \leqq F_S$,

$$F_m=F_S+\left(0.3+0.2\frac{F_R}{F_S}\right)F_R$$
$$=F_S+0.3F_R+0.2\frac{F_R^2}{F_S} \quad \cdots\cdots \quad \mathbf{(4.42)}$$



(a) Rotating load > stationary load (b) Rotating load < stationary load

Fig. 4.17 Combined Rotating and Stationary Loads

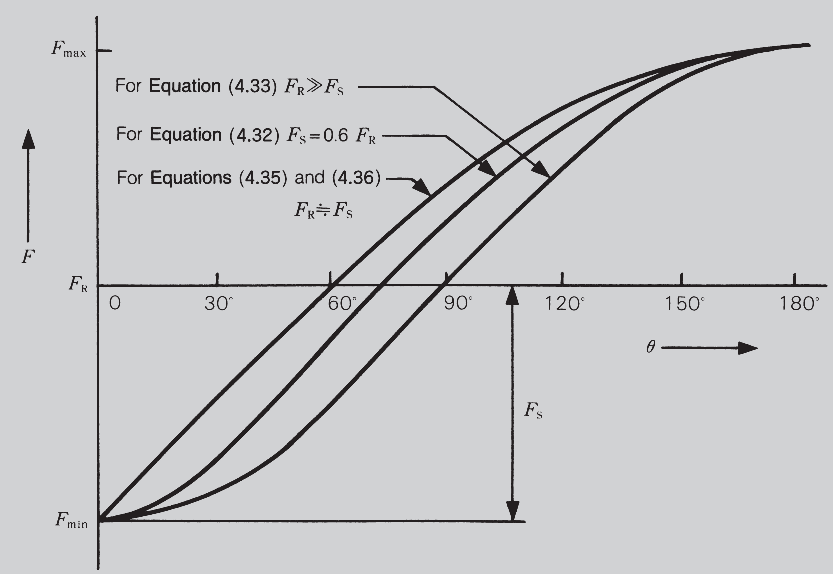


Fig. 4.18 Chart of Combined Loads

4.4 Equivalent Load

In some cases, the loads applied on bearings are purely radial or axial loads; however, in most cases, the loads are a combination of both. In addition, such loads usually fluctuate in both magnitude and direction. In such cases, the loads actually applied on bearings cannot be used for bearing life calculations; therefore, a hypothetical load should be estimated. This load should have a constant magnitude that passes through the center of the bearing and gives the same bearing life that the bearing would attain under actual load and rotation conditions. Such a hypothetical load is called the equivalent load.

4.4.1 Calculation of Equivalent Loads

The equivalent load on radial bearings may be calculated using the following equation:

$$P = XF_r + YF_a \quad \cdots\cdots (4.43)$$

where P : Equivalent Load (N), {kgf}
F_r : Radial load (N), {kgf}
F_a : Axial load (N), {kgf}
X : Radial load factor
Y : Axial load factor

The values of X and Y are listed in the bearing tables. The equivalent radial load for radial roller bearings with $\alpha = 0°$ is

$$P = F_r$$

In general, thrust ball bearings cannot take radial loads, but spherical thrust roller bearings can take some radial loads. In this case, the equivalent load may be calculated using the following equation:

$$P = F_a + 1.2F_r \quad \cdots\cdots (4.44)$$

where $\dfrac{F_r}{F_a} \leqq 0.55$

4.4.2 Axial Load Components in Angular Contact Ball Bearings and Tapered Roller Bearings

The effective load center of both angular contact ball bearings and tapered roller bearings is at the point of intersection between the shaft center line and a line representing the load applied on the rolling element by the outer ring, as shown in Fig. 4.19. This effective load center for each bearing is listed in the bearing tables. When radial loads are applied to these types of bearings, a component load is produced in the axial direction. In order to balance this component load, bearings of the same type are used in pairs, placed face-to-face or back-to-back. These axial loads can be calculated using the following equation:

$$F_{ai} = \frac{0.6}{Y} F_r \quad \cdots\cdots (4.45)$$

where F_{ai} : Component load in the axial direction (N), {kgf}
F_r : Radial load (N), {kgf}
Y : Axial load factor

Assume that radial loads F_{rI} and F_{rII} are applied on bearings I and II (Fig. 4.20) respectively and an external axial load F_{ae} is applied as shown. If the axial load factors are Y_I and Y_{II} and the radial load factor is X, then the equivalent loads P_I and P_{II} may be calculated as follows:

where $F_{ae} + \dfrac{0.6}{Y_{II}} F_{rII} \geqq \dfrac{0.6}{Y_I} F_{rI}$

$$\left.\begin{aligned} P_I &= XF_{rI} + Y_I\left(F_{ae} + \frac{0.6}{Y_{II}} F_{rII}\right) \\ P_{II} &= F_{rII} \end{aligned}\right\} \quad \cdots\cdots (4.46)$$

where $F_{ae} + \dfrac{0.6}{Y_{II}} F_{rII} < \dfrac{0.6}{Y_I} F_{rI}$

$$\left.\begin{aligned} P_I &= F_{rI} \\ P_{II} &= XF_{rII} + Y_{II}\left(\frac{0.6}{Y_I} F_{rI} - F_{ae}\right) \end{aligned}\right\} \quad \cdots\cdots (4.47)$$

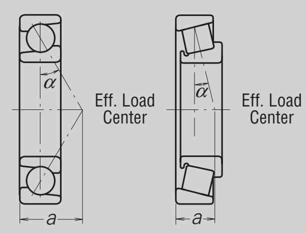


Fig. 4.19 Effective Load Centers

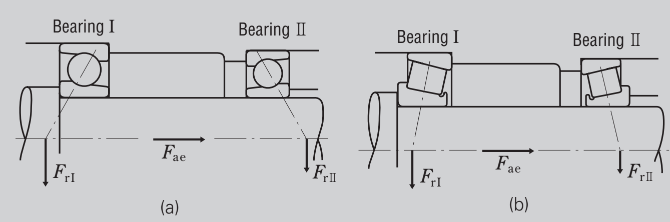


Fig. 4.20 Loads in Opposed Paired Mounting Arrangement

4.5 Static Load Ratings and Static Equivalent Loads

4.5.1 Static Load Ratings

When subjected to excessive or strong shock loads, rolling bearings may incur a local permanent deformation of the rolling elements and permanent deformation of the rolling elements and raceway surface if the elastic limit is exceeded. This nonelastic deformation increases in area and depth as load increases. When the load exceeds a certain limit, the smooth operation of the bearing is impeded.

The basic static load rating is defined as the static load that produces the following calculated contact stress between the raceway surface and center of the contact area of the rolling element subjected to the maximum stress:

For self-aligning ball bearings	4 600 MPa {469 kgf/mm²}
For other ball bearings	4 200 MPa {428 kgf/mm²}
For roller bearings	4 000 MPa {408 kgf/mm²}

In this most heavily stressed contact area, the sum of the permanent deformation of the rolling element and that of the raceway is nearly 0.0001 times the rolling element diameter. The basic static load rating C_o is defined as C_{or} for radial bearings and C_{oa} for thrust bearings in the bearing tables.

4.5.2 Static Equivalent Loads

The static equivalent load is a hypothetical load that produces a contact stress equal to the above maximum stress in the area of contact between the most heavily stressed rolling element and bearing raceway under actual conditions while the bearing is stationary (including very slow rotation or oscillation). The static radial load passing through the bearing center is taken as the static equivalent load for radial bearings, while the static axial load in the direction coinciding with the central axis is taken as the static equivalent load for thrust bearings.

(a) *Static equivalent load on radial bearings*

The greater of the two values calculated from the following equations should be adopted as the static equivalent load on radial bearings:

$$P_o = X_o F_r + Y_o F_a \quad \cdots\cdots \quad \textbf{(4.47)}$$

$$P_o = F_r \quad \cdots\cdots \quad \textbf{(4.48)}$$

where P_o : Static equivalent load (N), {kgf}
F_r : Radial load (N), {kgf}
F_a : Axial load (N), {kgf}
X_o : Static radial load factor
Y_o : Static axial load factor

(b) *Static equivalent load on thrust bearings*

$$P_o = X_o F_r + F_a \qquad \alpha \neq 90° \quad \cdots\cdots \quad \textbf{(4.49)}$$

where P_o : Static equivalent load (N), {kgf}
α : Contact angle

When $F_a < X_o F_r$, this equation becomes less accurate.
The values of X_o and Y_o for Equations (4.47) and (4.49) are listed in the bearing tables.
The static equivalent load for thrust roller bearings when

$\alpha = 90°$ is $P_o = F_a$

4.5.3 Permissible Static Load Factor

The permissible static equivalent load on bearings varies depending on the basic static load rating, application, and operating conditions.
The permissible static load factor f_s is a safety factor that is applied to the basic static load rating, and it is defined by the ratio in Equation (4.50). The generally recommended values of f_s are listed in Table 4.9.

$$f_s = \frac{C_o}{P_o} \quad \cdots\cdots \quad \textbf{(4.50)}$$

where C_o : Basic static load rating (N), {kgf}
P_o : Static equivalent load (N), {kgf}

For spherical thrust roller bearings, the value of f_s should be greater than 4.

Table 4.9 Values of Permissible Static Load Factor f_s

Operating Conditions	Lower Limit of f_s	
	Ball Bearings	Roller Bearings
Low-noise applications	2	3
Bearings subjected to vibration and shock loads	1.5	3
Standard operating conditions	1	1.5

4.6 Example Bearing Calculations

(Example 1)
Obtain the fatigue life factor f_h of single-row deep groove ball bearing **6208** when it is used under a radial load F_r=2 500 N and speed n =900 min^{-1}.

The basic load rating C_r of **6208** is 29 100 N (Bearing Table, Page C024). Since only a radial load is applied, the equivalent load P may be obtained as follows:

$$P = F_r = 2\,500 \text{ N}$$

Since the speed is n = 900 min^{-1}, the speed factor f_n can be obtained from the equation in Table 4.2 (Page A034) or Fig. 4.3 (Page A036).

$$f_n = 0.333$$

The fatigue life factor f_h, under these conditions, can be calculated as follows:

$$f_h = f_n \frac{C_r}{P} = 0.333 \times \frac{29\,100}{2\,500} = 3.88$$

This value is suitable for industrial applications, air conditioners in regular use, etc., and according to the equation in Table 4.2 or Fig. 4.4 (Page A036), this corresponds to approximately 29 000 hours of service life.

(Example 2)
Select a single-row deep groove ball bearing with a bore diameter of 50 mm and outside diameter under 100 mm that satisfies the following conditions:
Radial load F_r = 3 000 N
Speed n =1 900 min^{-1}
Basic rating life $L_h \geqq 10\,000$ h

The fatigue life factor f_h of ball bearings with a rating fatigue life longer than 10 000 hours is $f_h \geqq 2.72$.
Because f_n = 0.26, $P = F_r$ = 3 000 N,

$$f_h = f_n \frac{C_r}{P} = 0.26 \times \frac{C_r}{3\,000} \geqq 2.72$$

Therefore, $C_r \geqq 2.72 \times \frac{3\,000}{0.26} = 31\,380$ N

From the data listed in the bearing table on Page C026, **6210** should be selected, as it satisfies the above conditions.

(Example 3)
Obtain C_r / P or fatigue life factor f_h when an axial load F_a=1 000 N is added to the conditions of (Example 1)

When the radial load F_r and axial load F_a are applied on single-row deep groove ball bearing **6208**, the dynamic equivalent load P should be calculated in accordance with the following procedure:
Obtain the radial load factor X, axial load factor Y, and constant e, depending on the magnitude of $f_0 F_a / C_{0r}$, from the Dynamic Equivalent Load Table above the Bearing Table on Page C025.
The basic static load rating C_{0r} of ball bearing **6208** is 17 900 N (Page C024)

$$f_0 F_a / C_{0r} = 14.0 \times 1\,000/17\,900 = 0.782$$

$$e \fallingdotseq 0.26$$

and $F_a / F_r = 1\,000/2\,500 = 0.4 > e$

$$X = 0.56$$

Y = 1.67 (the value of Y is obtained by linear interpolation)

Therefore, the dynamic equivalent load P is

$$P = XF_r + YF_a$$
$$= 0.56 \times 2\,500 + 1.67 \times 1\,000$$
$$= 3\,070 \text{ N}$$

$$\frac{C_r}{P} = \frac{29\,100}{3\,070} = 9.48$$

$$f_h = f_n \frac{C_r}{P} = 0.333 \times \frac{29\,100}{3\,070} = 3.16$$

This value of f_h corresponds approximately to 15 800 hours of service life for ball bearings.

(Example 4)
Select a Series 231 spherical roller bearing that satisfies the following conditions:
Radial load F_r = 45 000 N
Axial load F_a = 8 000 N
Speed n = 500 min^{-1}
Basic rating life $L_h \geqq 30\,000$ h

The value of the fatigue life factor f_h which makes $L_h \geqq 30\,000$ h is greater than 3.45, according to Fig. 4.4 (Page A036).

The dynamic equivalent load P of spherical roller bearings is as follows:

when $F_a / F_r \leqq e$

$$P = XF_r + YX_a = F_r + Y_3 F_a$$

when $F_a / F_r > e$

$$P = XF_r + YF_a = 0.67\,F_r + Y_2 F_a$$

$$F_a / F_r = 8\,000/45\,000 = 0.18$$

We can see in the bearing table that the value of e is about 0.3 and that of Y_3 is about 2.2 for Series 231 bearings.

Therefore, $P = XF_r + YF_a = F_r + Y_3 F_a$
$= 45\,000 + 2.2 \times 8\,000$
$= 62\,600$ N

From the fatigue life factor f_h, the basic load rating can be obtained as follows:

$$f_h = f_n \frac{C_r}{P} = 0.444 \times \frac{C_r}{62\,600} \geqq 3.45$$

Consequently, $C_r \geqq 490\,000$ N
The smallest Series 231 spherical roller bearing satisfying this value of C_r is **23126CE4** (C_r = 505 000 N).
Once the bearing is determined, substitute the value of Y_3 in the equation and obtain the value of P.

$$P = F_r + Y_3 F_a = 45\,000 + 2.4 \times 8\,000$$
$$= 64\,200 \text{ N}$$

$$L_h = 500\left(f_n \frac{C_r}{P}\right)^{\frac{10}{3}}$$
$$= 500\left(0.444 \times \frac{505\,000}{64\,200}\right)^{\frac{10}{3}}$$
$$= 500 \times 3.49^{\frac{10}{3}} \fallingdotseq 32\,000 \text{ h}$$

(Example 5)
Assume that tapered roller bearings **HR30305DJ** and **HR30206J** are used in a back-to-back arrangement as shown in Fig. 4.21, and the distance between the outer ring back faces is 50 mm.
Calculate the basic rating life of each bearing when radial load F_r = 5 500 N and axial load F_{ae}=2 000 N are applied to **HR30305DJ** as shown in Fig. 4.21. The speed is 600 min^{-1}.

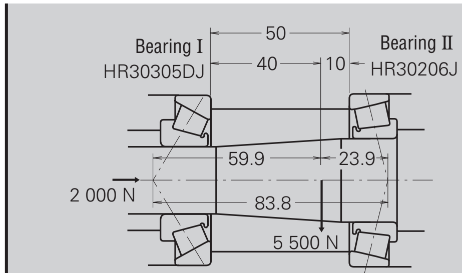


Fig. 4.21 Loads on Tapered Roller Bearings

To distribute the radial load F_r on bearings Ⅰ and Ⅱ, the effective load centers for tapered roller bearings must be located. Obtain the effective load center a for bearings Ⅰ and Ⅱ from the bearing table, then obtain the relative position of the radial load F_r and effective load centers. The result will be as shown in Fig. 4.21. Consequently, the radial load applied on bearings Ⅰ (**HR30305DJ**) and Ⅱ (**HR30206J**) can be obtained from the following equations:

$$F_{r\mathrm{I}} = 5\,500 \times \frac{23.9}{83.8} = 1\,569 \text{ N}$$

$$F_{r\mathrm{II}} = 5\,500 \times \frac{59.9}{83.8} = 3\,931 \text{ N}$$

The following values are obtained from data in the Bearing Table:

Bearings	Basic dynamic load rating C_r (N)	Axial load factor Y_1	Constant e
Bearing Ⅰ (**HR30305DJ**)	38 000	$Y_{\mathrm{I}} = 0.73$	0.83
Bearing Ⅱ (**HR30206J**)	43 000	$Y_{\mathrm{II}} = 1.6$	0.38

When radial loads are applied on tapered roller bearings, an axial load component is produced and must be considered to obtain the dynamic equivalent radial load (refer to Section 4.4.2, Page A051).

This is obtained by the following:

$$F_{ae}+\frac{0.6}{Y_{\mathrm{II}}}F_{r\mathrm{II}}=2\,000+\frac{0.6}{1.6}\times 3\,931$$

$$=3\,474\ \mathrm{N}$$

$$\frac{0.6}{Y_{\mathrm{I}}}F_{r\mathrm{I}}=\frac{0.6}{0.73}\times 1\,569=1\,290\ \mathrm{N}$$

Therefore, with this bearing arrangement, the axial load $F_{ae}+\frac{0.6}{Y_{\mathrm{II}}}F_{r\mathrm{II}}$ is applied on bearing Ⅰ but not on bearing Ⅱ.

For bearing Ⅰ,

$F_{r\mathrm{I}}=1\,569\ \mathrm{N}$

$F_{a\mathrm{I}}=3\,474\ \mathrm{N}$

since $F_{a\mathrm{I}}/F_{r\mathrm{I}}=2.2>e=0.83$

the dynamic equivalent load $P_{\mathrm{I}}=XF_{r\mathrm{I}}+Y_{\mathrm{I}}F_{a\mathrm{I}}$

$=0.4\times 1\,569+0.73\times 3\,474$

$=3\,164\ \mathrm{N}$

The fatigue life factor $f_h=f_n\frac{C_r}{P_{\mathrm{I}}}$

$$=\frac{0.42\times 38\,000}{3\,164}=5.04$$

and the rating fatigue life $L_h=500\times 5.04^{\frac{10}{3}}$

$=109\,750\ \mathrm{h}$

For bearing Ⅱ

since $F_{r\mathrm{II}}=3\,931\ \mathrm{N},\ F_{a\mathrm{II}}=0$

the dynamic equivalent load

$P_{\mathrm{II}}=F_{r\mathrm{II}}=3\,931\ \mathrm{N},$

fatigue life factor

$$f_h=f_n\frac{C_r}{P_{\mathrm{II}}}=\frac{0.42\times 43\,000}{3\,931}=4.59$$

and rating fatigue life $L_h=500\times 4.59^{\frac{10}{3}}=80\,400\ \mathrm{h}$ are obtained.

For face-to-face arrangements (DF type), please contact NSK.

(Example 6)
Select a bearing for a speed reducer under the following conditions:
Operating conditions

Radial load $F_r=245\,000\ \mathrm{N}$

Axial load $F_a=49\,000\ \mathrm{N}$

Speed $n=500\ \mathrm{min}^{-1}$

Size limitation

Shaft diameter: 300 mm

Housing bore: Less than 500 mm

In this application, heavy loads, shocks, and shaft deflection are expected; therefore, spherical roller bearings are appropriate.
The following spherical roller bearings satisfy the above size limits (refer to Page C284):

d	D	B	Bearing No.	Basic Dynamic Load Rating C_r (N)	Constant e	Factor Y_3
300	420	90	**23960 CAME4**	1 540 000	0.19	3.5
	460	118	**23060 CAME4**	2 400 000	0.24	2.8
	460	160	**24060 CAME4**	2 890 000	0.32	2.1
	500	160	**23160 CAME4**	3 350 000	0.31	2.2
	500	200	**24160 CAME4**	3 900 000	0.38	1.8

Since $F_a/F_r=0.20<e$, the dynamic equivalent load P is

$$P=F_r+Y_3F_a$$

Judging from the fatigue life factor f_h in Table 4.1 and example applications (refer to Page A034), a value of f_h, between 3 and 5 is appropriate.

$$f_h=f_n\frac{C_r}{P}=\frac{0.444\ C_r}{F_r+Y_3F_a}=3\text{ to }5$$

Assuming that $Y_3=2.1$, then the necessary basic load rating C_r can be obtained:

$$C_r=\frac{(F_r+Y_3F_a)\times(3\text{ to }5)}{0.444}$$

$$=\frac{(245\,000+2.1\times 49\,000)\times(3\text{ to }5)}{0.444}$$

$$=2\,350\,000\text{ to }3\,900\,000\ \mathrm{N}$$

Bearings that satisfy this range are **23060CAME4**, **24060CAME4**, **23160CAME4**, and **24160CAME4**.

4.7 Bearing Type and Allowable Axial Load

4.7.1 Change in Contact Angle of Radial Ball Bearings and Allowable Axial Load

(1) *Change in Contact Angle Due to Axial Load*

When an axial load acts on a radial ball bearing, the rolling element and raceway develop elastic deformation, resulting in an increase in contact angle and width. When heat generation or seizure occurs, the bearing should be disassembled and checked for a running trace to discover whether there has been a change in contact angle during operation. In this way, it is possible to see whether an abnormal axial load has been sustained.

The relation below reflects axial load on a bearing F_a, the load on rolling element Q, and the contact angle α when a load is applied. (see Equations (9.8), (9.9), and (9.10) in Section 9.6.2)

$$F_a = Z\,Q\sin\alpha$$

$$= K\,Z\,D_w^{\,2}\left\{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}-1\right\}^{3/2}\cdot\sin\alpha \quad \cdots\cdots \textbf{(4.51)}$$

$$\alpha = \sin^{-1}\frac{\sin\alpha_0+h}{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}} \quad \cdots\cdots \textbf{(4.52)}$$

$$h = \frac{\delta_a}{m_0} = \frac{\delta_a}{r_e+r_i-D_w}$$

Namely, δ_a refers to the change in Equation (4.52) to determine a contact angle α corresponding to the contact angle known from observation of the raceway. Thus, δ_a and α are introduced into Equation (4.51) to estimate the axial load F_a acting on the bearing. As specifications of the bearing are necessary for calculation, the contact angle α in this case was approximated from the axial load. The basic static load rating C_{0r} is expressed by Equation (4.53) for single-row radial ball bearings.

$$C_{0r} = f_0\,Z\,D_w^{\,2}\cos\alpha_0 \quad \cdots\cdots \textbf{(4.53)}$$

where f_0: Factor determined from the shape of bearing components and applicable stress level

Equation (4.54) is determined from Equations (4.51) and (4.53):

$$\frac{f_0}{C_{0r}}F_a = A\,F_a$$

$$= K\left\{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}-1\right\}^{3/2}\cdot\frac{\sin\alpha}{\cos\alpha_0} \quad \cdots\cdots \textbf{(4.54)}$$

where K: Constant determined from material and design of bearing

In other words, h is assumed and contact angle α is determined from Equation (4.52). Then h and α are introduced into Equation (4.54) to determine $A\,F_a$. This relation is used to show the value of A for each bore number of angular contact ball bearings in Table 4.14. The relationship between $A\,F_a$ and α is shown in Fig. 4.22.

Example 1

Calculate the change in the contact angle when pure axial load F_a = 35.0 kN (50% of basic static load rating) is applied to angular contact ball bearing 7215C.

A = 0.212 as determined from Table 4.10 and AF_a = 0.212 × 35.0 = 7.42 and $\alpha \fallingdotseq$ 26° according to Fig. 4.22.

The initial contact angle of 15° has changed to 26° under axial load.

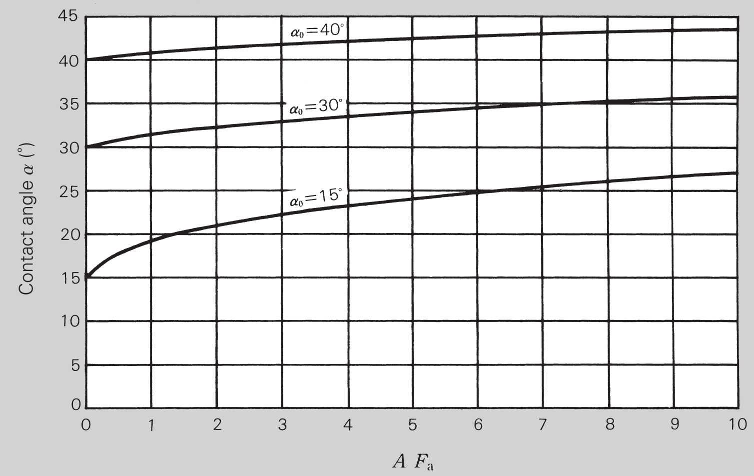


Fig. 4.22 Change in Contact Angle of Angular Contact Ball Bearings Under Axial Load

Table 4.10 Constant A Value of Angular Contact Ball Bearings

Units: kN^{-1}

Bearing Bore No.	Series 70 Bearings			Series 72 Bearings			Series 73 Bearings		
	15°	30°	40°	15°	30°	40°	15°	30°	40°
05	1.97	2.05	2.31	1.26	1.41	1.59	0.838	0.850	0.961
06	1.45	1.51	1.83	0.878	0.979	1.11	0.642	0.651	0.736
07	1.10	1.15	1.38	0.699	0.719	0.813	0.517	0.528	0.597
08	0.966	1.02	1.22	0.562	0.582	0.658	0.414	0.423	0.478
09	0.799	0.842	1.01	0.494	0.511	0.578	0.309	0.316	0.357
10	0.715	0.757	0.901	0.458	0.477	0.540	0.259	0.265	0.300
11	0.540	0.571	0.681	0.362	0.377	0.426	0.221	0.226	0.255
12	0.512	0.542	0.645	0.293	0.305	0.345	0.191	0.195	0.220
13	0.463	0.493	0.584	0.248	0.260	0.294	0.166	0.170	0.192
14	0.365	0.388	0.460	0.226	0.237	0.268	0.146	0.149	0.169
15	0.348	0.370	—	0.212	0.237	0.268	0.129	0.132	0.149
16	0.284	0.302	0.358	0.190	0.199	0.225	0.115	0.118	0.133
17	0.271	0.288	0.341	0.162	0.169	0.192	0.103	0.106	0.120
18	0.228	0.242	0.287	0.140	0.146	0.165	0.0934	0.0955	0.108
19	0.217	0.242	0.273	0.130	0.136	0.153	0.0847	0.0866	0.0979
20	0.207	0.231	0.261	0.115	0.119	0.134	0.0647	0.0722	0.0816

Values for deep groove ball bearings are similarly shown in Table 4.11 and Fig. 4.23.

Example 2

Calculate the change in the contact angle when a pure axial load F_a = 24.75 kN (50% of the basic static load rating) is applied to the deep groove ball bearing 6215. Note here that the radial internal clearance is calculated as the median (0.020 mm) of the normal clearance.

The initial contact angle of 10° is obtained from Fig. 3 on Page C015. A = 0.303 as determined from Table 4.11 and $A\ F_a$ = 0.303 × 24.75≒7.5, thus α≒24° based on Fig. 4.23.

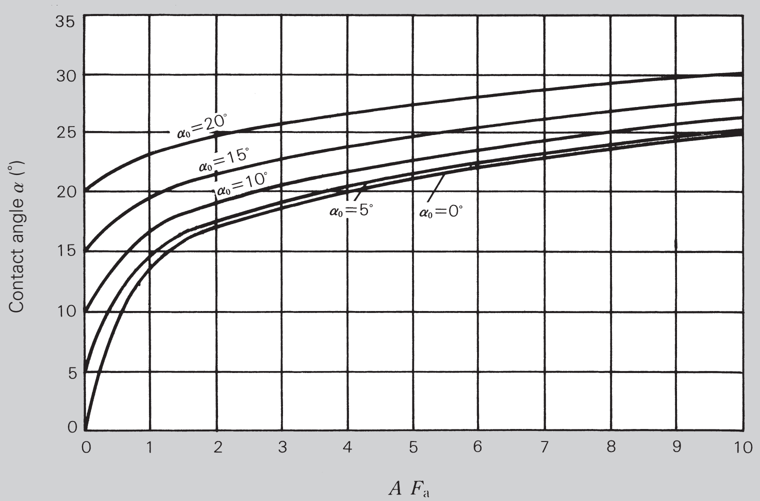


Fig. 4.23 Change in Contact Angle of Deep Groove Ball Bearings Under Axial Load

Table 4.11 Constant A Value of Deep Groove Ball Bearing

Units: kN^{-1}

Bearing Bore No.	Series 62 Bearings				
	0°	5°	10°	15°	20°
05	1.76	1.77	1.79	1.83	1.88
06	1.22	1.23	1.24	1.27	1.30
07	0.900	0.903	0.914	0.932	0.958
08	0.784	0.787	0.796	0.811	0.834
09	0.705	0.708	0.716	0.730	0.751
10	0.620	0.622	0.630	0.642	0.660
11	0.490	0.492	0.497	0.507	0.521
12	0.397	0.398	0.403	0.411	0.422
13	0.360	0.361	0.365	0.373	0.383
14	0.328	0.329	0.333	0.340	0.349
15	0.298	0.299	0.303	0.309	0.317
16	0.276	0.277	0.280	0.285	0.293
17	0.235	0.236	0.238	0.243	0.250
18	0.202	0.203	0.206	0.210	0.215
19	0.176	0.177	0.179	0.183	0.188
20	0.155	0.156	0.157	0.160	0.165

(2) ***Allowable Axial Load for a Deep Groove Ball Bearing***

The allowable axial load here refers to the limit load at which a contact ellipse is generated between the ball and raceway due to a change in the contact angle when a radial bearing, which is under an axial load, rides over the shoulder of the raceway groove. This is different from the limit value of a static equivalent load P_0 which is determined from the basic static load rating C_{0r} using the static axial load factor Y_0. Also note that the contact ellipse may ride over the shoulder even when the axial load on the bearing is below the limit value of P_0.

The allowable axial load $F_{a\,max}$ of a radial ball bearing can be determined through equations. The contact angle α for F_a is determined from the right terms of Equations (4.51) and Equation (4.52), while Q is calculated as follows:

$$Q=\frac{F_a}{Z\sin\alpha}$$

θ of Fig. 4.24 is determined as follows:

$$2a=A_2\,\mu\left(\frac{Q}{\Sigma\rho}\right)^{1/3}$$

$$\therefore\ \theta \fallingdotseq \frac{a}{r}$$

Accordingly, the allowable axial load may be determined as the maximum axial load at which the following relation is established.

$$\gamma \geqq \alpha+\theta$$

As the allowable axial load cannot be determined unless the internal specifications of a bearing are known, Fig. 4.25 shows the calculated results of allowable axial load for various deep groove radial ball bearings.

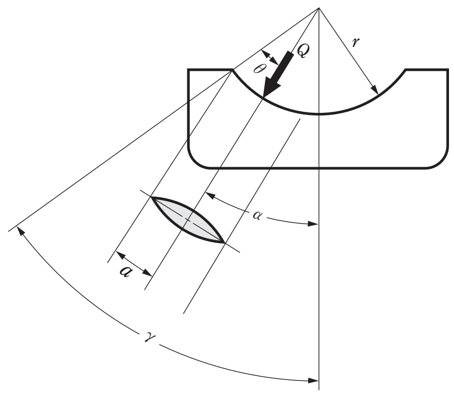


Fig. 4.24 Contact Ellipse

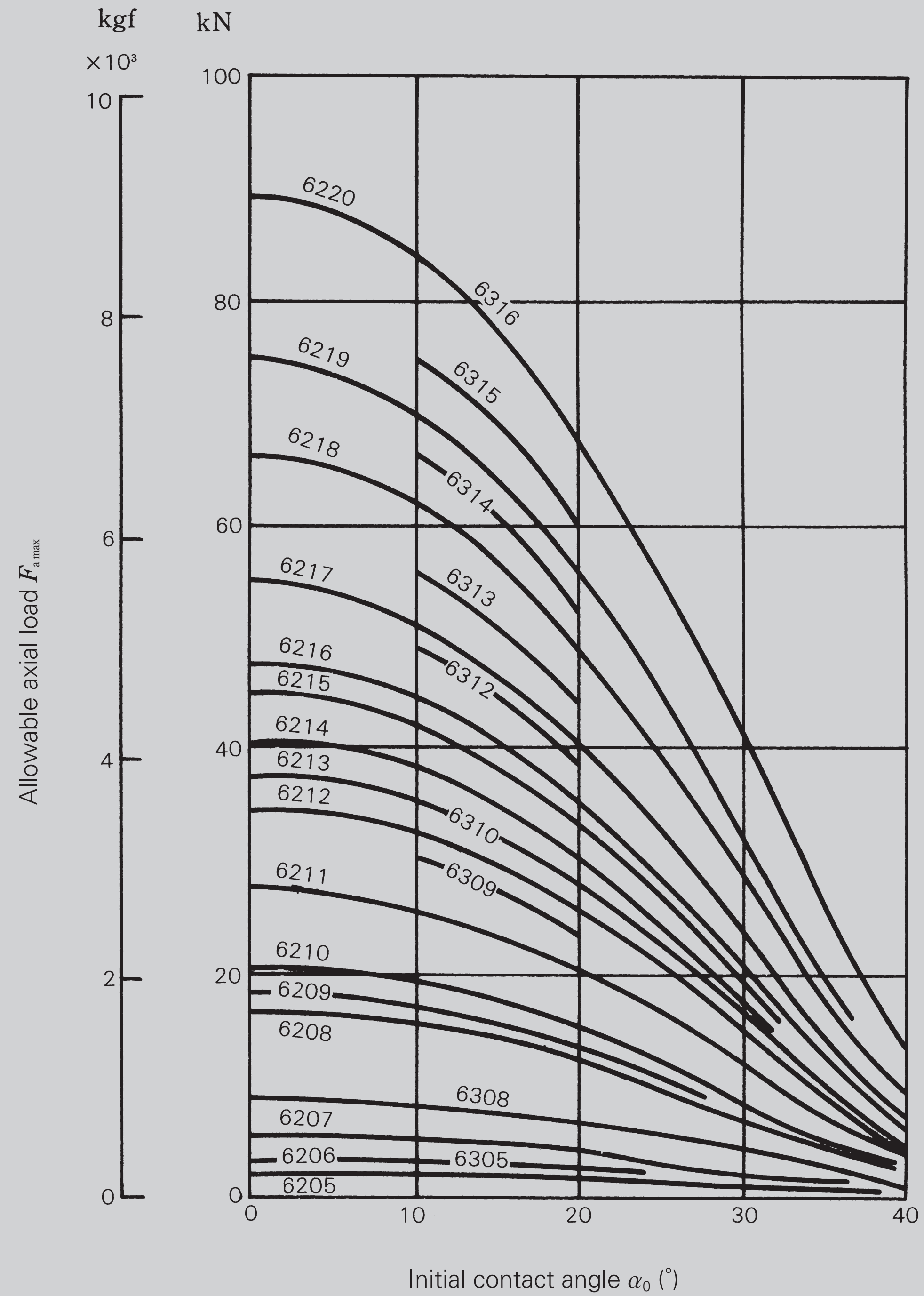


Fig. 4.25 Allowable Axial Load for Deep Groove Ball Bearings

4.7.2 Allowable Axial Load (Rib Breakdown Strength) for Cylindrical Roller Bearings

The inner and outer rings of cylindrical bearings with loose or fixed ribs can receive an axial load when operating under radial load. This allowable axial load is limited by the heat generated from sliding of the roller and rib faces, seizing, rib strength, and so on.
Fig 4.26 shows the allowable axial load (a load that considers heat generation between the end faces of the rollers and rib faces) when applied continuously to a Diameter Series 3 cylindrical roller bearing with grease or oil lubrication.

Grease lubrication (Empirical equation)

$$C_A = 9.8f\left\{\frac{900\ (k \cdot d)^2}{n+1\ 500} - 0.023 \times (k \cdot d)^{2.5}\right\} \text{(N)}$$

$$= f\left\{\frac{900\ (k \cdot d)^2}{n+1\ 500} - 0.023 \times (k \cdot d)^{2.5}\right\} \{\text{kgf}\} \quad \cdots\cdots \textbf{(4.55)}$$

Oil lubrication (Empirical equation)

$$C_A = 9.8f\left\{\frac{490\ (k \cdot d)^2}{n+1\ 000} - 0.000135 \times (k \cdot d)^{3.4}\right\} \text{(N)}$$

$$= f\left\{\frac{490\ (k \cdot d)^2}{n+1\ 000} - 0.000135 \times (k \cdot d)^{3.4}\right\} \{\text{kgf}\} \quad \cdots\cdots \textbf{(4.56)}$$

where C_A : Allowable axial load (N), {kgf}
d : Bearing bore diameter (mm)
n : Bearing speed (min^{-1})
f : Load factor
k : Dimensional factor

As Equations (4.55) and (4.56) do not consider rib strength, please consult with NSK regarding rib strength.
To enable a cylindrical roller bearing to maintain stable axial load capacity, note the following concerning the bearing and its surroundings:

- ○ Ensure that radial load is applied and that the magnitude of the radial load is at least 2.5 times that of the axial load.
- ○ Maintain sufficient lubricant between the roller end face and rib face.
- ○ Use a lubricant with an additive for extreme pressures.
- ○ Provide sufficient run-in time.
- ○ Confirm that bearing mounting accuracy is good.
- ○ Don' t use a bearing with an unnecessarily large internal clearance.

Moreover, take extra care and consideration for each bearing's lubrication, cooling method, and so on when:

- ○ Bearing speed is less than 200 min^{-1},
- ○ Bearing speed exceeds 50 % of the allowable speed in the bearing tables, or
- ○ Bearing bore diameter exceeds 200 mm.

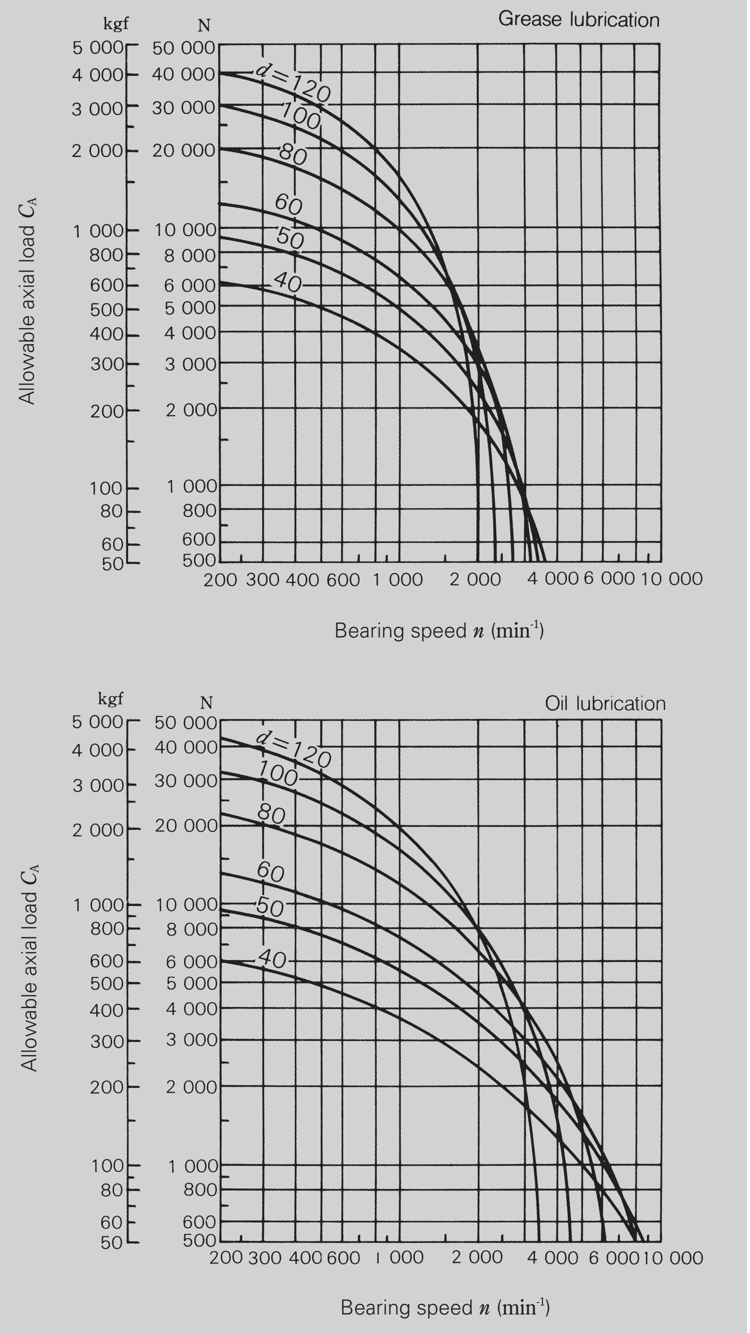
Please contact NSK for guidance in such cases.

f: Load Factor

	f value
Continuous Load	1
Intermittent Load	2
Short-Term Load	3

k: Dimensional Factor

	k value
Series 2 Bearing Diameter	0.75
Series 3 Bearing Diameter	1
Series 4 Bearing Diameter	1.2



For a Diameter Series 3 bearing (k = 1.0) under continuous load (f = 1)

Fig. 4.26 Allowable Axial Load for a Cylindrical Roller Bearing

4.8 Technical Data

4.8.1 Fatigue Life and Reliability

Where any part failure may result in damage to the entire machine and repair of damage is impossible, as in aircraft, satellite, or rocket applications, greatly increased reliability is demanded of each component. Recently, such high reliability requirements have been applied more generally to durable consumer goods and may also be utilized to achieve effective preventive maintenance of machines and equipment.

The rating fatigue life of a rolling bearing is the gross number of revolutions or the gross rotating period (when the rotating speed is constant) that 90 % of a group of similar bearings running individually under similar conditions can rotate without suffering material damage due to rolling fatigue. In other words, fatigue life is normally defined at 90 % reliability. There are other ways to describe the expected life. For example, an average value is frequently employed to describe the lifespan of human beings. However, if an average were used for bearings, then too many bearings would fail before the average life value was reached. On the other hand, if a low or minimum value was used as a criterion, then too many bearings would have a life much longer than the set value. With these considerations in mind, a 90 % value was chosen for common practice. A value of 95 % could have been taken as the statistical reliability, but the slightly looser reliability of 90 % was taken from a practical and economical viewpoint. A 90 % reliability however is not acceptable for aircraft parts, computers, or communication systems, and a 99 % or even 99.9 % reliability may be required in some of these cases.

The fatigue life distribution when a group of similar bearings are operated individually under similar conditions is shown in Fig. 4.27. The Weibull equation can be used to describe the fatigue life distribution within a damage ratio of 10 to 60 % (residual probability of 90 to 40 %). At a damage ratio of 10 % or less (residual probability of 90 % or more) however, the rolling fatigue life becomes longer than the theoretical curve of the Weibull distribution, as shown in Fig. 4.28. This is a conclusion drawn from the life test of numerous, widely-varying bearings and an analysis of collected data.

When bearing life with a failure ratio of 10 % or less (for example, the 95 % life or 98 % life) is to be considered on the basis of the above concept, the reliability factor a_1 shown in Table 4.12 is used to check the life. Assume here that the 98 % life L_2 is to be calculated for a bearing whose rating fatigue life L_{10} was calculated at 10 000 hours. The life can be calculated as $L_2 = 0.37 \times L_{10} =$ 3 700 hours. In this manner, the reliability of the bearing life can be matched to the equipment and difficulty of inspection and disassembly.

Table 4.12 Reliability Factor

Reliability, %	90	95	96	97	98	99
Life, L	L_{10} rating life	L_5	L_4	L_3	L_2	L_1
Reliability Factor, a_1	1	0.64	0.55	0.47	0.37	0.25

Apart from rolling fatigue, factors such as lubrication, wear, sound, and accuracy govern the durability of a bearing. These factors must be taken into account, but the endurance limit of these factors varies depending on application and conditions.

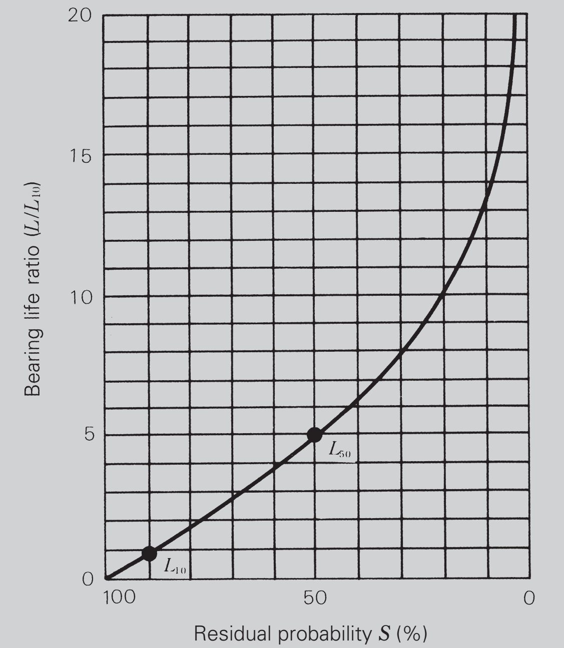


Fig. 4.27 Bearing Life and Residual Probability

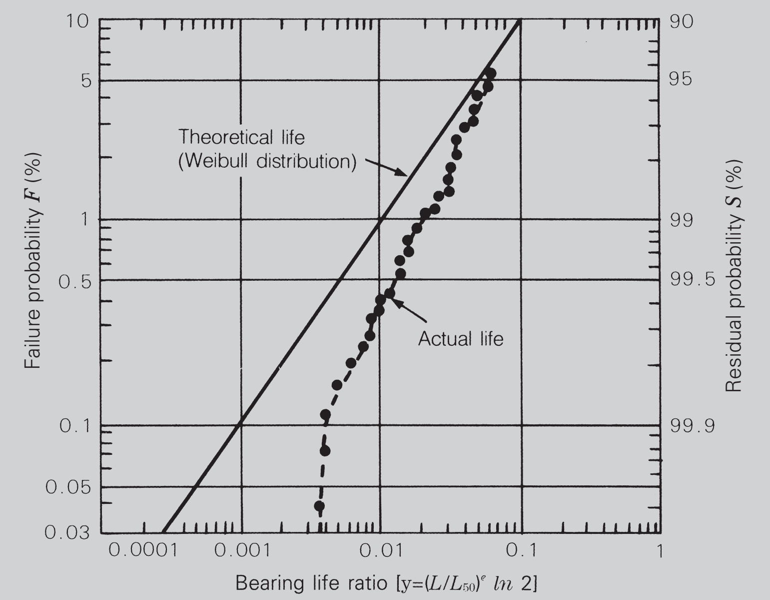


Fig. 4.28 Life Distribution in the Low-Failure Ratio Range

4.8.2 Radial Clearance and Fatigue Life

As shown in the catalog and elsewhere, the fatigue life calculation equation of rolling bearings is as follows:

$$L = \left(\frac{C}{P}\right)^{p} \quad\cdots\cdots\cdots\cdots \quad \mathbf{(4.57)}$$

where L : Rating fatigue life (10^6rev)
C : Basic dynamic load rating (N), {kgf}
P : Dynamic equivalent load (N), {kgf}
p : Index Ball bearing p=3,
Roller bearing p = $\frac{10}{3}$

The rating fatigue life L for a radial bearing in this case requires the load distribution in the bearing corresponds to the state with the load factor ε = 0.5 (Fig. 4.29).
The load distribution with ε=0.5 is obtained when the bearing radial internal clearance is zero. In this sense, the normal fatigue life calculation is intended to obtain a value when the clearance is zero. When the effect of the radial clearance is taken into account, the bearing fatigue life can be calculated. Equations (4.58) and (4.59) can be established between the bearing radial clearance Δ_r and a function $f(\varepsilon)$ of load factor ε:

For deep groove ball bearings:

$$\left.\begin{aligned} f(\varepsilon) &= \frac{\Delta_r \cdot D_w^{1/3}}{0.00044\left(\dfrac{F_r}{Z}\right)^{2/3}} \quad\cdots\cdots\cdots \quad (\mathrm{N}) \\ f(\varepsilon) &= \frac{\Delta_r \cdot D_w^{1/3}}{0.002\left(\dfrac{F_r}{Z}\right)^{2/3}} \quad\cdots\cdots\cdots \quad \{\mathrm{kgf}\} \end{aligned}\right\} \cdots\cdots \mathbf{(4.58)}$$

For cylindrical roller bearings:

$$\left.\begin{aligned} f(\varepsilon) &= \frac{\Delta_r \cdot L_{we}^{0.8}}{0.000077\left(\dfrac{F_r}{Z\cdot i}\right)^{0.9}} \quad\cdots\cdots\cdots \quad (\mathrm{N}) \\ f(\varepsilon) &= \frac{\Delta_r \cdot L_{we}^{0.8}}{0.0006\left(\dfrac{F_r}{Z\cdot i}\right)^{0.9}} \quad\cdots\cdots\cdots \quad \{\mathrm{kgf}\} \end{aligned}\right\} \cdots\cdots \mathbf{(4.59)}$$

where Δ_r : Radial clearance (mm)
F_r : Radial load (N), {kgf}
Z : Number of rolling elements
i : No. of rows of rolling elements
D_w : Ball diameter (mm)
L_{we} : Effective roller length (mm)
L_ε : Life with clearance of Δ_r
L : Life with zero clearance, obtained from Equation (4.57)

The relationship between load factor ε and $f(\varepsilon)$ and the life ratio L_ε/L, when radial internal clearance Δ_r exists is shown in Table 4.13.

Fig. 4.30 shows the relationship between the radial clearance and bearing fatigue life in example 6208 and NU208 bearings.

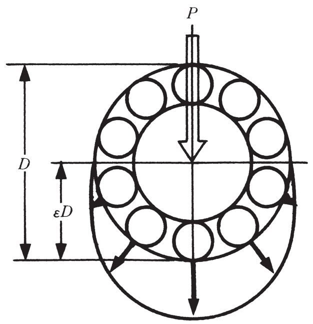


Fig. 4.29 Load Distribution With ε = 0.5

Table 4.13 ε and $f(\varepsilon)$, L_ε/L

ε	Deep Groove Ball Bearing		Cylindrical Roller Bearing	
	$f(\varepsilon)$	$\frac{L_\varepsilon}{L}$	$f(\varepsilon)$	$\frac{L_\varepsilon}{L}$
0.1	33.713	0.294	51.315	0.220
0.2	10.221	0.546	14.500	0.469
0.3	4.045	0.737	5.539	0.691
0.4	1.408	0.889	1.887	0.870
0.5	0	1.0	0	1.0
0.6	− 0.859	1.069	− 1.133	1.075
0.7	− 1.438	1.098	− 1.897	1.096
0.8	− 1.862	1.094	− 2.455	1.065
0.9	− 2.195	1.041	− 2.929	0.968
1.0	− 2.489	0.948	− 3.453	0.805
1.25	− 3.207	0.605	− 4.934	0.378
1.5	− 3.877	0.371	− 6.387	0.196
1.67	− 4.283	0.276	− 7.335	0.133
1.8	− 4.596	0.221	− 8.082	0.100
2.0	− 5.052	0.159	− 9.187	0.067
2.5	− 6.114	0.078	−11.904	0.029
3	− 7.092	0.043	−14.570	0.015
4	− 8.874	0.017	−19.721	0.005
5	−10.489	0.008	−24.903	0.002
10	−17.148	0.001	−48.395	0.0002

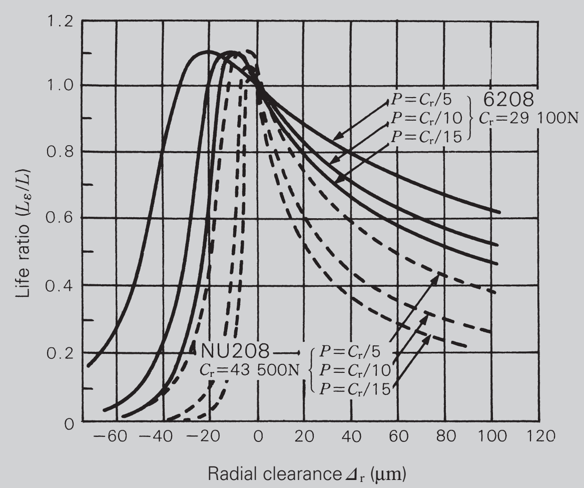


Fig. 4.30 Radial Clearance and Bearing Life Ratio

4.8.3 Misalignment of Inner/Outer Rings and Fatigue Life of Deep Groove Ball Bearings

A rolling bearing is manufactured with high accuracy, and careful attention to machining and assembly accuracies of the surrounding shafts and housing is critical for this accuracy is to be maintained. In practice however, the machining accuracy of parts around the bearing is limited, and bearings are subject to misalignment of inner/outer rings caused by the shaft deflection under external load.

The allowable misalignment is generally 0.0006 – 0.003 rad (2′ to 10′) but this varies depending on the size of the deep groove ball bearing, internal clearance during operation, and load.

This section introduces the relationship between the misalignment of inner/outer rings and fatigue life. Four Series 62 and 63 deep groove ball bearings of different sizes were selected as examples.

Assume the fatigue life without misalignment as $L_{\theta=0}$ and the fatigue life with misalignment as L_θ. The effect of the misalignment on the fatigue life may be found by calculating $L_\theta / L_{\theta=0}$. Results are shown in Figs. 4.31 to 4.34.

To represent ordinary running conditions in calculations, radial load F_r (N) {kgf} and axial load F_a (N) {kgf} were assumed to be approximately 10 % (normal load) and 1 % (light preload) of the dynamic load rating C_r (N) {kgf} respectively. Normal radial clearance was used and the shaft fit was set to around j5. Decrease of the internal clearance due to expansion of the inner ring was also taken into account.

Moreover, assuming that the temperature difference between the inner and outer rings was 5 °C during operation, inner/outer ring misalignment, $L_\theta / L_{\theta=0}$ was calculated for the maximum, minimum, and mean effective clearances.

As shown in Figs. 4.31 to 4.34, degradation of the fatigue life is limited to 5 to 10 % or less when the misalignment ranges from 0.0006 to 0.003 rad (2′ to 10′), thus not presenting much of a problem.

When misalignment exceeds a certain limit however, the fatigue life degrades rapidly as shown in the figures; therefore, pay careful attention to this matter.

When the clearance is small, not much effect is observed as long as misalignment is also small, as shown in the figures; however, life decreases substantially when misalignment increases. As previously mentioned, aim to minimize mounting error as much as possible.

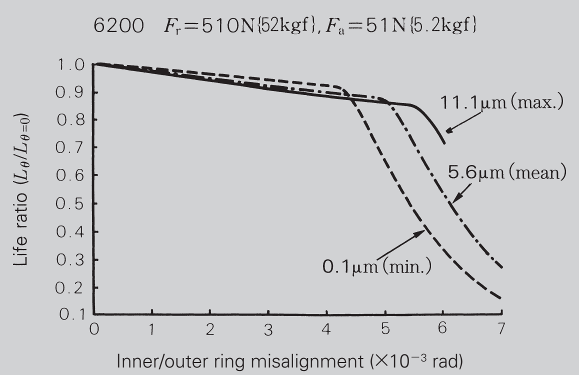


Fig. 4.31 Life Vs. Misalignment

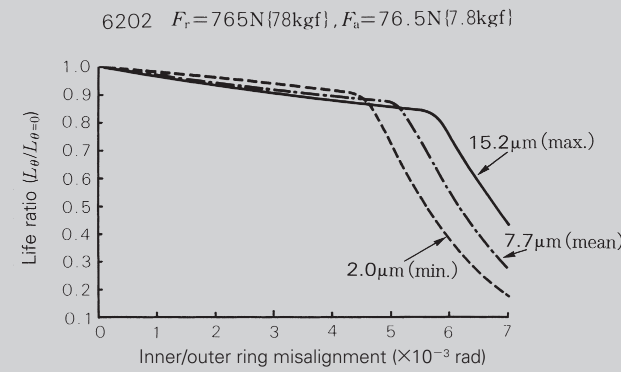


Fig. 4.32 Life Vs. Misalignment

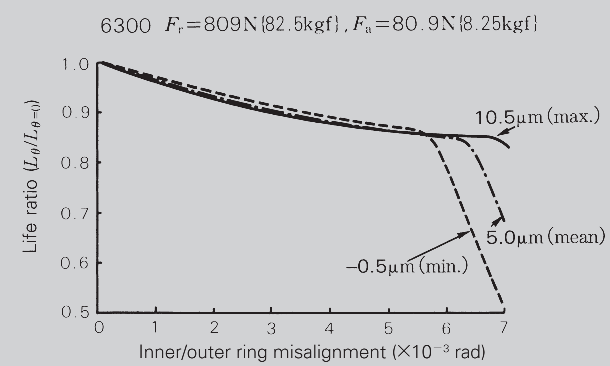


Fig. 4.33 Life Vs. Misalignment

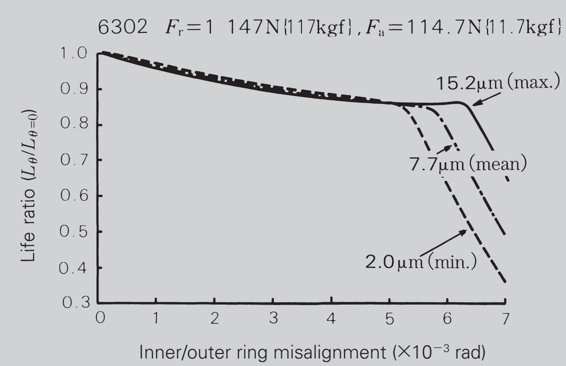


Fig. 4.34 Life Vs. Misalignment

4.8.4 Misalignment of Inner/Outer Rings and Fatigue Life of Cylindrical Roller Bearings

When a shaft supported by rolling bearings is deflected or there is some inaccuracy in a shoulder, misalignment arises between the inner and outer rings of the bearings, thereby lowering their fatigue life. The degree of degradation depends on bearing type, interior design, radial internal clearance, and magnitude of load during operation.

The relationship between the misalignment of inner/outer rings and fatigue life was determined with standard NU215 and NU315 cylindrical roller bearings, as shown in Figs. 4.35 to 4.38.

In these figures, the horizontal axis shows the misalignment of inner/outer rings (rad) while the vertical axis shows the fatigue life ratio $L_\theta / L_{\theta=0}$. The fatigue life without misalignment is $L_{\theta=0}$ and the fatigue life with misalignment is L_θ.

Figs. 4.35 and 4.36 show the life ratio for bearings under constant load (10 % of basic dynamic load rating C_r of a bearing) when the internal clearance is normal, C3, or C4. Figs. 4.37 and 4.38 show the life ratio for bearings with constant clearance (normal clearance) when the load is 5 %, 10 %, and 20 % of the basic dynamic load rating C_r.

Note that the median effective clearance in these examples was determined using m5/H7 fits and a temperature difference of 5°C between the inner and outer rings.

The fatigue life ratio for the clearance and load shows the same trend as in the case of other cylindrical roller bearings; however, the life ratio itself differs among bearing series and dimensions, with life degradation rapid in Series 22 and 23 bearings (wide type). Use of a specially designed bearing is advised when considerable misalignment is expected during application.

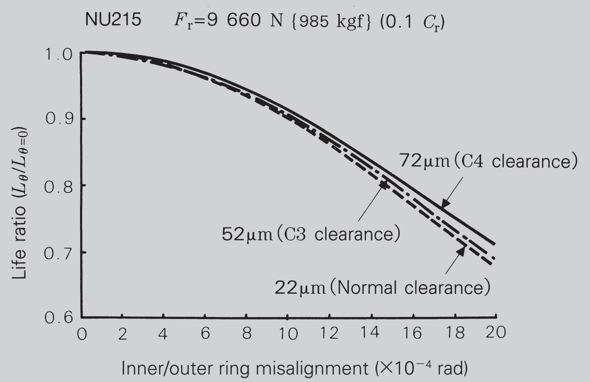


Fig. 4.35 Life Vs. Misalignment

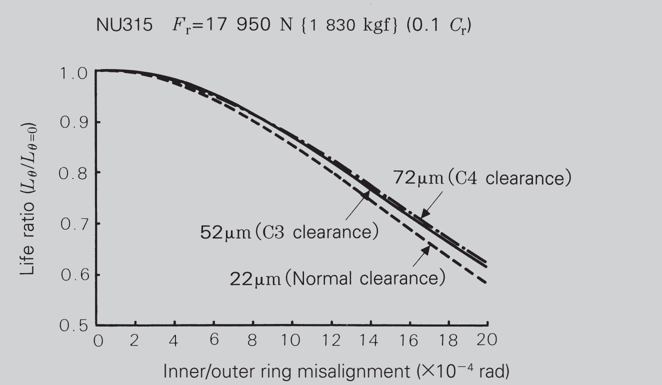


Fig. 4.36 Life Vs. Misalignment

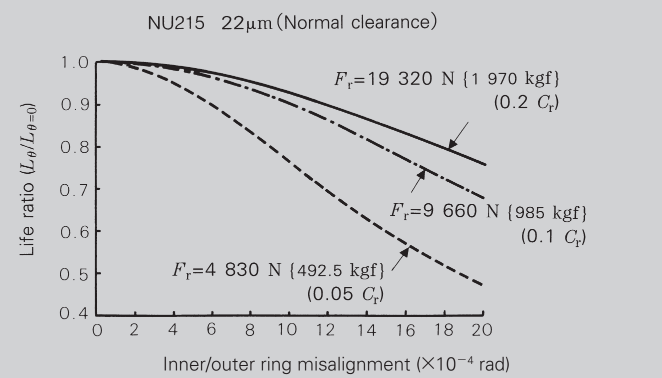


Fig. 4.37 Life Vs. Misalignment

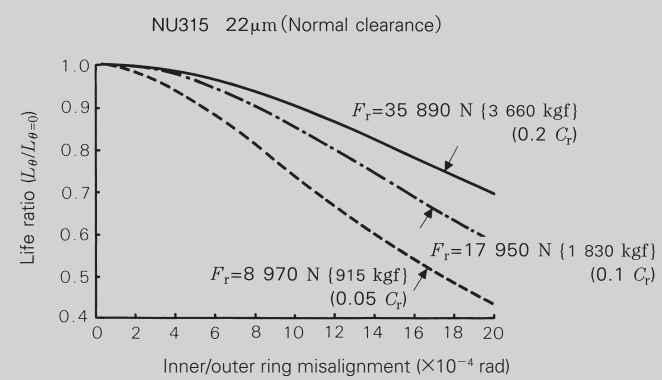


Fig. 4.38 Life Vs. Misalignment

4.8.5 Oil Film Parameters and Rolling Fatigue Life

As evidenced by numerous experiments and experiences, the rolling fatigue life of rolling bearings is closely related to lubrication.

The rolling fatigue life is expressed by the maximum number of rotations that a bearing can endure until the raceway or rolling surface of a bearing develops fatigue in the material resulting in flaking of the surface due to cyclic stress. Such flaking begins with either microscopic non-uniform portions (such as nonmetallic inclusions or cavities) in the material or with microscopic defects in the material surface (such as extremely small cracks, surface damage, or dents caused by contact between extremely small projections in the raceway or rolling surface). The former flaking is called subsurface-originating flaking while the latter is surface-originating flaking.

The oil film parameter Λ, which is the ratio between the resultant oil film thickness and surface roughness, expresses whether or not the lubrication at the rolling contact surface is satisfactory. The effect of the oil film grows with increasing Λ. Namely, when Λ is large (around 3 in general), surface-originating flaking due to contact between extremely small projections in the surface is less likely to occur. If the surface is free from defects (flaws, dents, etc.), the life is determined mainly by subsurface-originating flaking. On the other hand, a decrease in Λ tends to cause surface-originating flaking, resulting in degradation of the bearing's life. This state is shown in Fig. 4.39.

NSK has performed life experiments with about 370 bearings within the range of Λ = 0.3–3 and with different lubricants and bearing materials (● and ▲ in Fig. 4.40). Fig. 4.40 shows a summary of the principal experiments reported until now. As is evident in the figures, life decreases rapidly at around $\Lambda \fallingdotseq 1$ when compared with life values at around Λ = 3–4 where life changes at a slower rate. Life becomes about 1/10 or less at $\Lambda \leqq 0.5$ due to severe surface-originating flaking.

Accordingly, to extend the fatigue life of rolling bearings, increase the oil film parameter (ideally to a value above 3) by improving lubrication conditions.

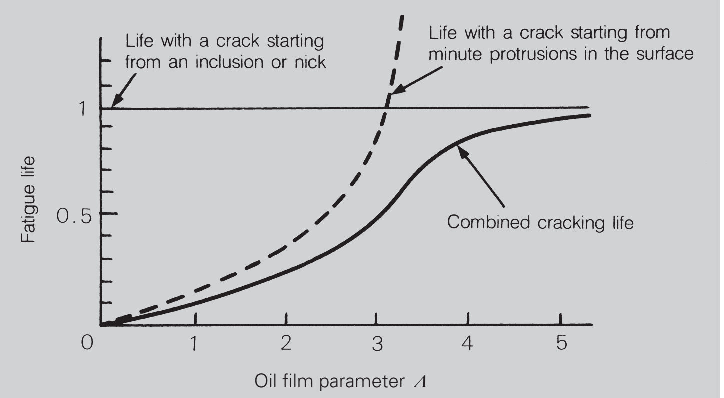


Fig. 4.39 Life According to Oil Film Parameter Λ (Tallian, et al.)

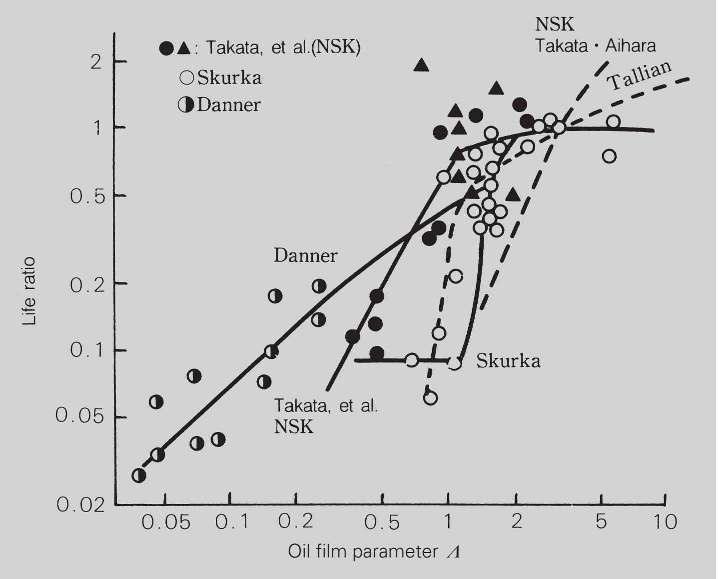


Fig. 4.40 Typical Experiment With Oil Film Parameter Λ and Rolling Fatigue Life
(Expressed with reference to the life at Λ =3)

4.8.6 EHL Oil Film Parameter Calculation Diagram

Lubrication of rolling bearings can be expressed by the theory of elastohydrodynamic lubrication (EHL). Introduced below is a method to determine the oil film parameter (oil film to surface roughness ratio), the most critical component in EHL.

(1) ***Oil Film Parameter***

The raceway surfaces and rolling surfaces of a bearing are extremely smooth, but have fine irregularities when viewed through a microscope. As the EHL oil film thickness is in the same order as the surface roughness, lubrication conditions cannot be discussed without considering this surface roughness. For example, given a particular mean oil film thickness, there are two conditions that may occur depending on the surface roughness. One consists of complete separation of the two surfaces by means of the oil film (Fig. 4.41 (a)). The other consists of metal contact between surface projections (Fig. 4.41 (b)). The degradation of lubrication and surface damage is attributed to case (b). The symbol lambda (Λ) represents the ratio between the oil film thickness and roughness. It is widely employed as the oil film parameter in the study and application of EHL.

$$\Lambda = h/\sigma \qquad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.60})$$

where h : EHL oil film thickness
σ : Combined roughness $\left(\sqrt{\sigma_1^2+\sigma_2^2}\right)$

σ_1, σ_2 : Root mean square (rms) roughness of each contact surface

The oil film parameter may be correlated to the formation of the oil film, and the degree of lubrication can be divided into three zones as shown in Fig. 4.42.

(2) ***Oil Film Parameter Calculation Diagram***

The ***Dowson-Higginson*** minimum oil film thickness equation shown below is used for figures:

$$H_{min} = 2.65\,\frac{G^{0.54}U^{0.7}}{W^{0.13}} \qquad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.61})$$

Use the oil film thickness that reflects the inner ring under maximum rolling element load (where thickness is minimum).

Equation (4.61) can be expressed by grouping variables as follows: (R) for speed, (A) for viscosity, (F) for load, and (J) for bearing technical specifications, with t as a constant:

$$\Lambda = t \cdot R \cdot A \cdot F \cdot J \qquad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.62})$$

R and A may be quantities not dependent on the bearing. When the load P is assumed to be between 98 N {10 kgf} and 98 kN {10 tf}, F changes by 2.54 times as $F \propto P^{-0.13}$. Since the actual load is usually determined from bearing size however, such change may be limited to 20 to 30 %. As a result, F is handled together with term J [$F = F(J)$]. Traditional Equation (4.62) can therefore be grouped as shown below:

$$\Lambda = T \cdot R \cdot A \cdot D \qquad \cdots\cdots\cdots\cdots\cdots\cdots \quad (\mathbf{4.63})$$

where T : Factor determined by the bearing **T**ype
R : Factor related to **R**otation speed
A : Factor related to viscosity (viscosity grade α: **A**lpha)
D : Factor related to bearing **D**imensions

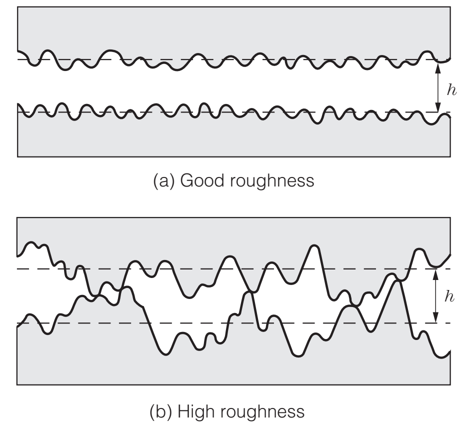


Fig. 4.41 Oil Film and Surface Roughness

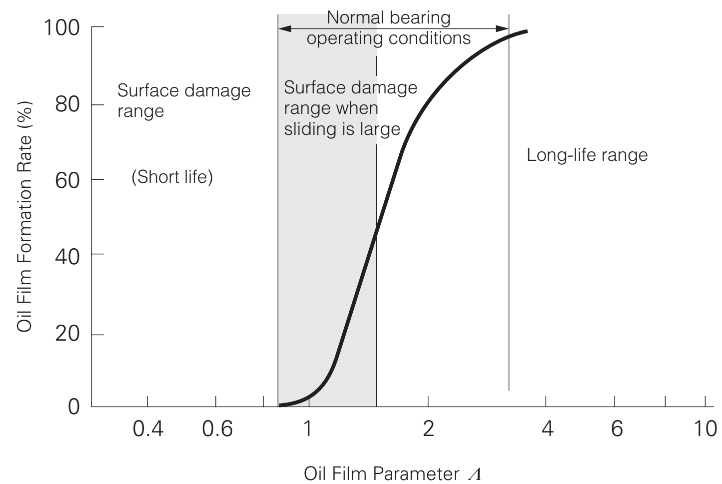


Fig. 4.42 Effect of Oil Film on Bearing Performance

The vitally important oil film parameter Λ is expressed by a simplified equation shown below. The fatigue life of rolling bearings becomes shorter when Λ is smaller. In the equation $\Lambda = T \cdot R \cdot A \cdot D$ variables include A for oil viscosity η_0 (mPa·s, {cp}), R for speed n (min^{-1}), and D for bearing bore diameter d (mm). The calculation procedure is described below.

(i) Determine the value of T from the bearing type (Table 4.14).

(ii) Determine the R value for n (min^{-1}) from Fig. 4.43.

(iii) Determine A from the absolute viscosity (mPa·s, {cp}) and kind of oil in Fig. 4.44.

Generally, the kinematic viscosity ν_0 (mm^2/s, {cSt}) is used and conversion is made as follows:

$$\eta_0 = \rho \cdot \nu_0 \quad \cdots\cdots \quad \textbf{(4.64)}$$

ρ refers to oil density (g/cm^3) and uses the approximate values shown below:

Mineral oil $\rho = 0.85$
Silicon oil $\rho = 1.0$
Diester oil $\rho = 0.9$

When the mineral oil could be naphthene or paraffin, use the paraffin curve shown in Fig. 4.44.

(iv) Determine the value of D from the Diameter Series and bore diameter d (mm) in Fig. 4.45.

(v) The product of the above values is used as the oil film parameter.

Table 4.14 T Value

Bearing Type	T Value
Ball bearing	1.5
Cylindrical roller bearing	1.0
Tapered roller bearing	1.1
Spherical roller bearing	0.8

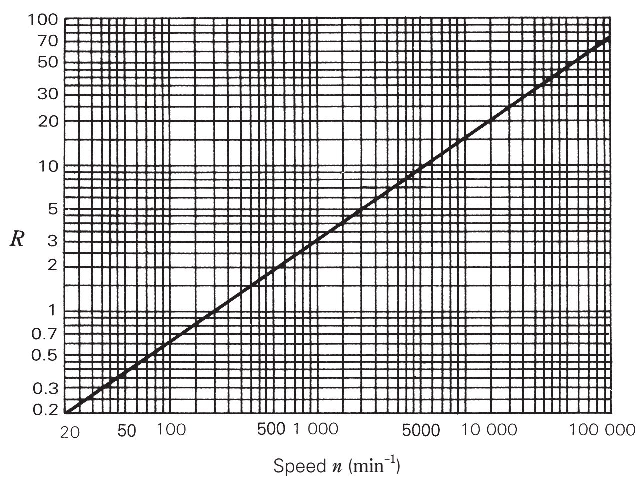


Fig. 4.43 Speed R

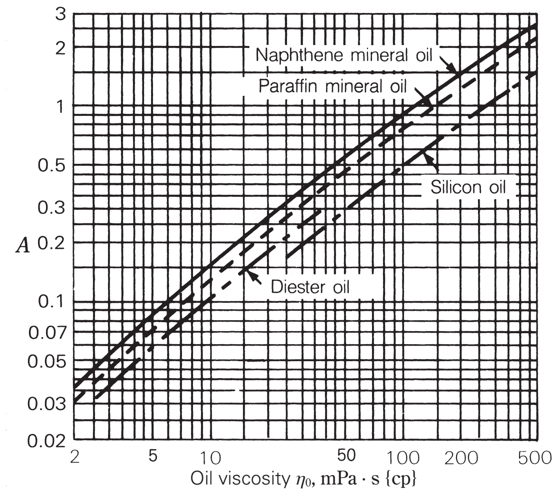


Fig. 4.44 Lubricant Viscosity A

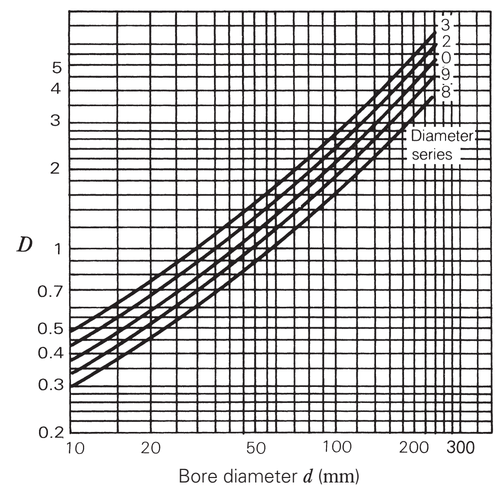


Fig. 4.45 Bearing Specifications D

Example EHL oil film parameter calculations are described below:

Example 1
Determine the oil film parameter when deep groove ball bearing 6312 is operated with paraffin mineral oil (η_0 = 30 mPa·s, {cp}) at a speed of n = 1 000 min^{-1}.

Solution
d = 60 mm and D = 130 mm from the bearing catalog.
T = 1.5 from Table 4.18
R = 3.0 from Fig. 4.43
A = 0.31 from Fig. 4.44
D = 1.76 from Fig. 4.45
Accordingly, Λ = 2.5

Example 2
Determine the oil film parameter when cylindrical roller bearing NU240 is operated with paraffin mineral oil (η_0 = 10 mPa·s, {cp}) at a speed of n = 2 500 min^{-1}.

Solution
d = 200 mm and D = 360 mm from the bearing catalog.
T = 1.0 from Table 4.18
R = 5.7 from Fig. 4.43
A = 0.13 from Fig. 4.44
D = 4.8 from Fig. 4.45
Accordingly, Λ = 3.6

(3) *Effects of Oil Shortage and Shearing Heat Generation*
The oil film parameter obtained above is applicable when the contact inlet is fully flooded with oil and the temperature at the inlet is constant (isothermal). However, these conditions may not be satisfied depending on lubrication and operating conditions.
One such condition is called starvation, and in this case, the actual oil film parameter may become smaller than that determined by Equation (4.64). Starvation may occur if lubrication becomes limited. In this condition, the guideline for adjusting the oil film parameter is 50 to 70 % of the value obtained from Equation (4.64).

Another effect is the localized rise of oil temperature in the contact inlet due to heavy shearing during high-speed operation, resulting in a decrease of oil viscosity. In this case, the oil film parameter becomes smaller than the theoretical isothermal value. The effect of shearing heat generation was analyzed by Murch and Wilson, who established a decrease factor for the oil film parameter. An approximation using the viscosity and speed (pitch diameter of rolling element set D_{pw} × rotating speed per minute n) is shown in Fig. 4.46. By multiplying the oil film parameter determined in the previous section by this decrease factor Hi, an oil film parameter considering the shearing heat generation is obtained:

$$\Lambda = Hi \cdot T \cdot R \cdot A \cdot D \quad \cdots\cdots\cdots\cdots \quad (\mathbf{4.65})$$

Note that the average of the bore and outside diameters of the bearings may be used as the pitch diameter D_{pw} (d_m) of the set of rolling elements.

Conditions for the calculation of Example 1 include $d_m n$ = 9.5 × 10^4 and η_0 = 30 mPa·s, {cp} and Hi nearly equivalent to 1, as evident from Fig. 4.46. Shearing heat generation therefore has almost no effect.
Conditions for Example 2 are $d_m n$ = 7×10^5 and η_0 = 10 mPa·s, {cp} and Hi = 0.76, meaning that the oil film parameter is smaller by about 25%. Accordingly, Λ is actually 2.7, not 3.6.

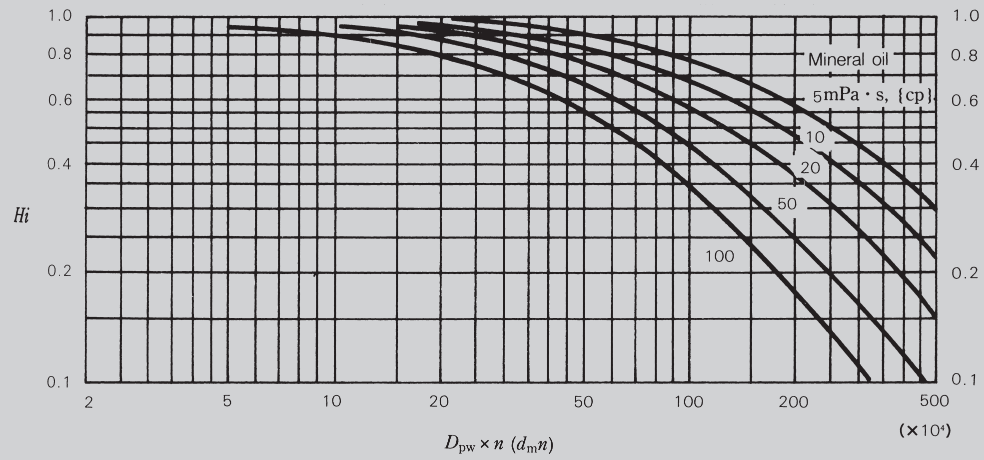


Fig. 4.46 Decrease of Oil Film Thickness Hi Due to Shearing Heat Generation

4.8.7 Load Calculation for Gears

(1) *Calculation of Loads on Spur, Helical, and Double-Helical Gears*

Since they are both mechanical elements, there is an extremely close relationship between gears and rolling bearings. Gear units, which are widely used in machines, are almost always used with bearings. Rating life calculation and selection of bearings to be used in gear units are based on the load at the gear meshing point.

The load at the gear meshing point is calculated as follows:

Spur Gear:

$$P_1 = P_2 = \frac{9\ 550\ 000H}{n_1\left(\frac{d_{p1}}{2}\right)} = \frac{9\ 550\ 000H}{n_2\left(\frac{d_{p2}}{2}\right)} \cdots\cdots (\mathrm{N})$$

$$= \frac{974\ 000H}{n_1\left(\frac{d_{p1}}{2}\right)} = \frac{974\ 000H}{n_2\left(\frac{d_{p2}}{2}\right)} \cdots\cdots \{\mathrm{kgf}\}$$

$$S_1 = S_2 = P_1 \tan\alpha$$

The magnitudes of forces P_2 and S_2 applied to the driven gear are the same as P_1 and S_1 respectively, but the direction is opposite.

Helical Gear:

$$P_1 = P_2 = \frac{9\ 550\ 000H}{n_1\left(\frac{d_{p1}}{2}\right)} = \frac{9\ 550\ 000H}{n_2\left(\frac{d_{p2}}{2}\right)} \cdots\cdots (\mathrm{N})$$

$$= \frac{974\ 000H}{n_1\left(\frac{d_{p1}}{2}\right)} = \frac{974\ 000H}{n_2\left(\frac{d_{p2}}{2}\right)} \cdots\cdots \{\mathrm{kgf}\}$$

$$S_1 = S_2 = \frac{P_1 \tan\alpha_n}{\cos\beta}$$

$$T_1 = T_2 = P_1 \tan\beta$$

The magnitudes of the forces P_2, S_2, and T_2 applied to the driven gear are the same as P_1, S_1, and T_1 respectively, but the direction is opposite.

Double-Helical Gear:

$$P_1 = P_2 = \frac{9\ 550\ 000H}{n_1\left(\frac{d_{p1}}{2}\right)} = \frac{9\ 550\ 000H}{n_2\left(\frac{d_{p2}}{2}\right)} \cdots\cdots (\mathrm{N})$$

$$= \frac{974\ 000H}{n_1\left(\frac{d_{p1}}{2}\right)} = \frac{974\ 000H}{n_2\left(\frac{d_{p2}}{2}\right)} \cdots\cdots \{\mathrm{kgf}\}$$

$$S_1 = S_2 = \frac{P_1 \tan\alpha_n}{\cos\beta}$$

where
- P : Tangential force (N), {kgf}
- S : Separating force (N), {kgf}
- T : Thrust (N), {kgf}
- H : Transmitted power (kW)
- n : Speed (min^{-1})
- d_p : Pitch diameter (mm)
- α : Gear pressure angle
- α_n : Gear normal pressure angle
- β : Twist angle

Subscript 1: Driving gear
Subscript 2: Driven gear

In the case of double-helical gears, thrust of the helical gears offsets each other and thus only tangential and separating forces act. For the directions of tangential, separating, and thrust forces, please refer to Figs. 4.47 and 4.48.

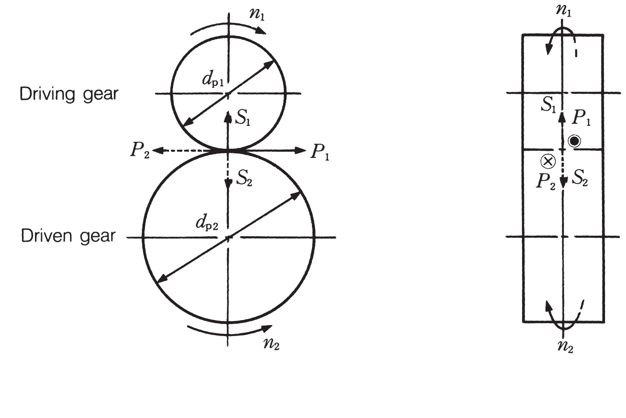


Fig. 4.47 Spur Gear

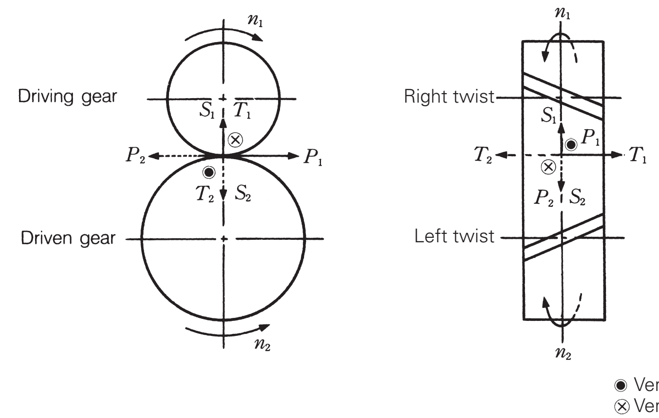


Fig. 4.48 Helical Gear

The thrust direction of the helical gear varies depending on the gear running direction, gear twist direction, and whether the gear is driving or driven.
The directions are as follows:

The force on the bearing is determined as follows:

Tangential force:

$$P_1 = P_2 = \frac{9\ 550\ 000H}{n_1\left(\frac{d_{p1}}{2}\right)} = \frac{9\ 550\ 000H}{n_2\left(\frac{d_{p2}}{2}\right)} \quad \cdots\cdots\cdots\cdots \text{(N)}$$

$$= \frac{974\ 000H}{n_1\left(\frac{d_{p1}}{2}\right)} = \frac{974\ 000H}{n_2\left(\frac{d_{p2}}{2}\right)} \quad \cdots\cdots \{\text{kgf}\}$$

Separating force: $S_1 = S_2 = P_1\dfrac{\tan\alpha_n}{\cos\beta}$

Thrust: $T_1 = T_2 = P_1 \cdot \tan\beta$

The same method can be applied to bearings C and D.

Table 4.15

Load Classification		Bearing A	Bearing B
Radial Load	From P_1	$P_A=\frac{b}{a+b}P_1$ ⊗	$P_B=\frac{a}{a+b}P_1$ ⊗
	From S_1	$S_A=\frac{b}{a+b}S_1$ ↑	$S_B=\frac{a}{a+b}S_1$ ↑
	From T_1	$U_A=\frac{d_{p1}/2}{a+b}T_1$ ↑	$U_B=\frac{d_{p1}/2}{a+b}T_1$ ↓
Combined Radial Load		$F_{rA}=\sqrt{P_A^2+(S_A+U_A)^2}$	$F_{rB}=\sqrt{P_B^2+(S_B-U_B)^2}$
Axial Load		$F_a=T_1$ ←	

Load directions shown reference the left side of Fig. 4.49.

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⊗ Vertically downward from the page

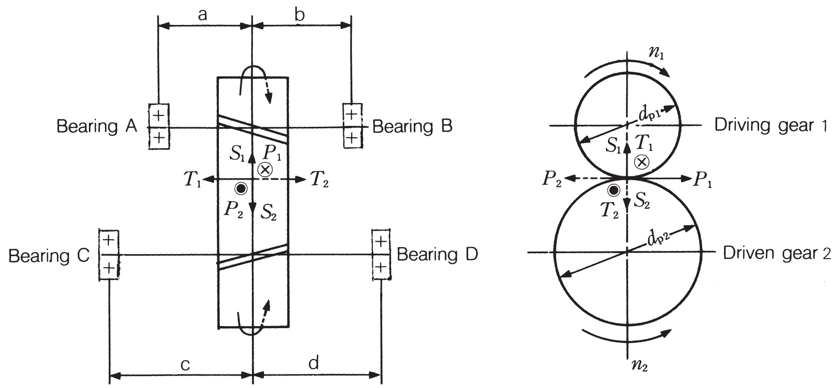


Fig. 4.49

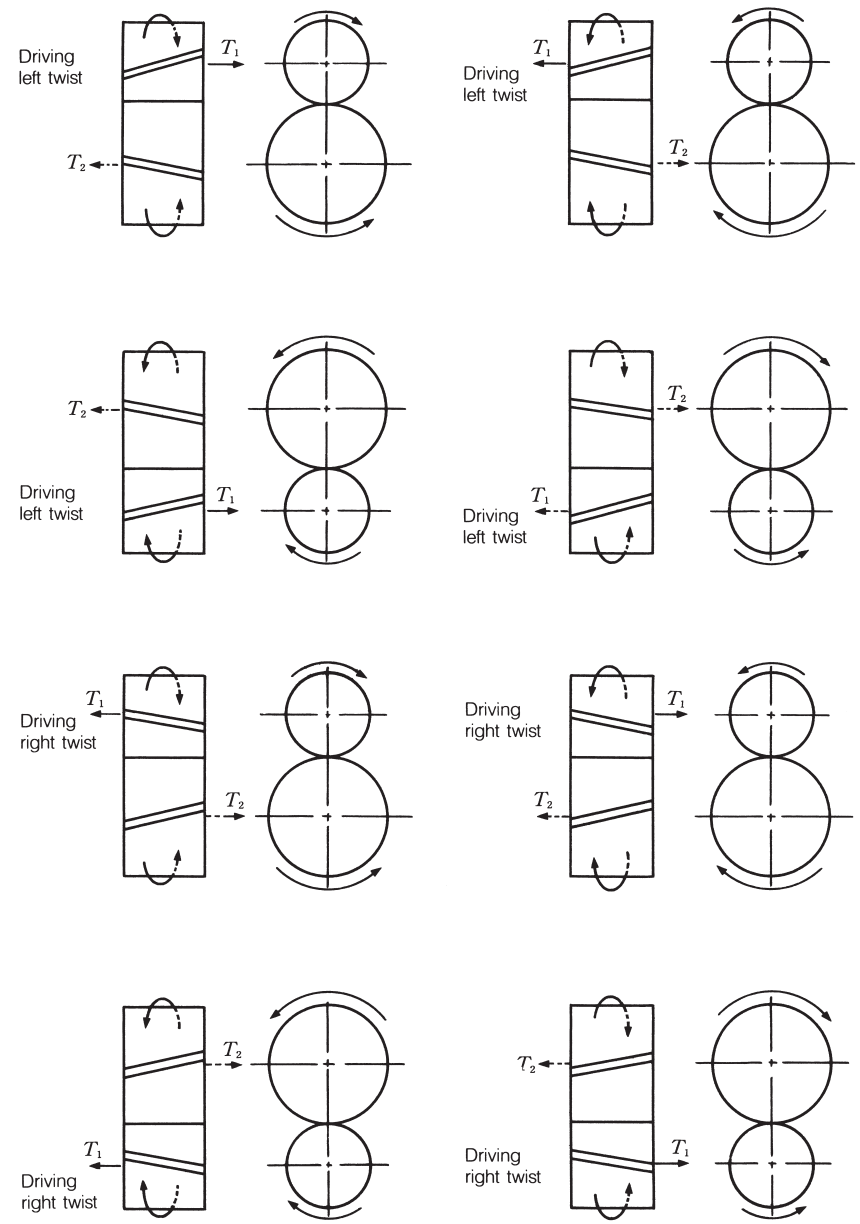


Fig. 4.50 Thrust Direction

(2) *Calculation of Load on Straight Bevel Gears*

The load at the meshing point of straight bevel gears is calculated as follows:

$$P_1 = P_2 = \frac{9\,550\,000H}{n_1\left(\frac{D_{m1}}{2}\right)} = \frac{9\,550\,000H}{n_2\left(\frac{D_{m2}}{2}\right)} \quad \cdots\cdots \text{(N)}$$

$$= \frac{974\,000H}{n_1\left(\frac{D_{m1}}{2}\right)} = \frac{974\,000H}{n_2\left(\frac{D_{m2}}{2}\right)} \quad \cdots\cdots \{\text{kgf}\}$$

$$D_{m1} = d_{p1} - w\sin\delta_1$$
$$D_{m2} = d_{p2} - w\sin\delta_2$$

$$S_1 = P_1\tan\alpha_n\cos\delta_1$$
$$S_2 = P_2\tan\alpha_n\cos\delta_2$$

$$T_1 = P_1\tan\alpha_n\cos\delta_1$$
$$T_2 = P_2\tan\alpha_n\cos\delta_2$$

where D_m : Average pitch diameter (mm)
d_p : Pitch diameter (mm)
w : Gear width (pitch line length) (mm)
α_n : Gear normal pressure angle
δ : Pitch cone angle

Generally, $\delta_1+\delta_2$ = 90°. In this case, S_1 and T_2 (or S_2 and T_1) are the same in magnitude but opposite in direction. S/P and T/P for δ are shown in Fig. 4.53. The load on the bearing can be calculated as shown below.

Table 4.16

◉ Vertically upward from the page
⊗ Vertically downward from the page

Load Classification		Bearing A	Bearing B	Bearing C	Bearing D
Radial Load	From P	$P_A=\frac{b}{a}P_1$ ◉	$P_B=\frac{a+b}{a}P_1$ ⊗	$P_C=\frac{d}{c+d}P_2$ ◉	$P_D=\frac{c}{c+d}P_2$ ◉
	From S	$S_A=\frac{b}{a}S_1$ ⬇	$S_B=\frac{a+b}{a}S_1$ ⬆	$S_C=\frac{d}{c+d}S_2$ ➡	$S_D=\frac{c}{c+d}S_2$ ➡
	From T	$U_A=\frac{D_{m1}}{2\cdot a}T_1$ ⬆	$U_B=\frac{D_{m1}}{2\cdot a}T_1$ ⬇	$U_C=\frac{D_{m2}}{2(c+d)}T_2$ ⬅	$U_D=\frac{D_{m2}}{2(c+d)}T_2$ ➡
Combined Radial Load		$F_{rA}=\sqrt{P_A^2+(S_A-U_A)^2}$	$F_{rB}=\sqrt{P_B^2+(S_B-U_B)^2}$	$F_{rC}=\sqrt{P_C^2+(S_C-U_C)^2}$	$F_{rD}=\sqrt{P_D^2+(S_D+U_D)^2}$
Axial Load		$F_a=T_1$ ⬅		$F_a=T_2$ ⬇	

Load directions shown reference Fig. 4.52.

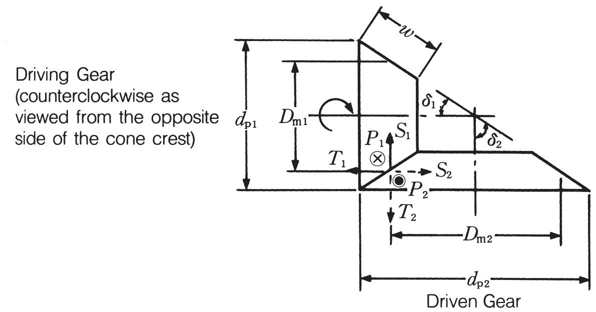


Fig. 4.51

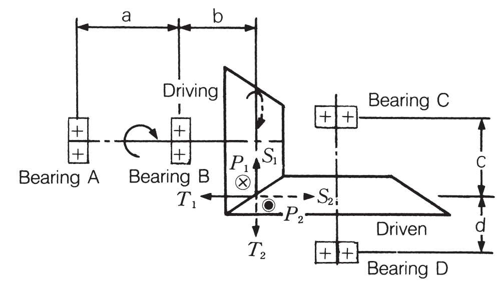


Fig. 4.52

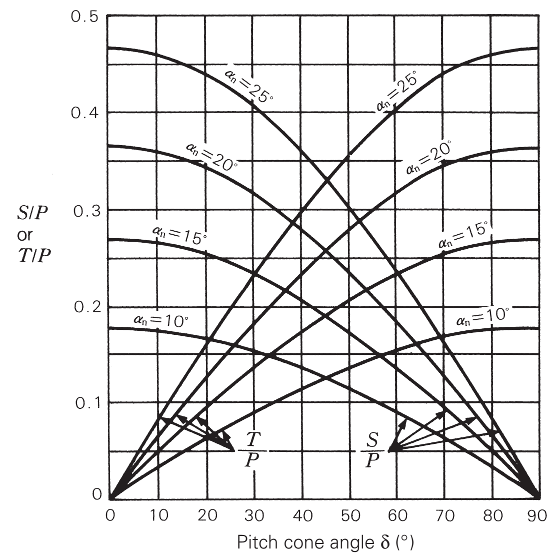


Fig. 4.53

(3) *Calculation of Load on Spiral Bevel Gears*

In the case of spiral bevel gears, the magnitude and direction of loads at the meshing point vary depending on the running direction and gear twist direction. The running direction is either clockwise or counterclockwise as viewed from the side opposite of the gears (Fig. 4.54). The gear twist direction is classified as shown in Fig. 4.55. The force at the meshing point is calculated as follows:

$$P_1 = P_2 = \frac{9\,550\,000H}{n_1\left(\dfrac{D_{m1}}{2}\right)} = \frac{9\,550\,000H}{n_2\left(\dfrac{D_{m2}}{2}\right)} \quad \cdots\cdots \text{(N)}$$

$$= \frac{974\,000H}{n_1\left(\dfrac{D_{m1}}{2}\right)} = \frac{974\,000H}{n_2\left(\dfrac{D_{m2}}{2}\right)} \quad \cdots\cdots \{\text{kgf}\}$$

where α_n : Gear normal pressure angle
β : Twisting angle
δ : Pitch cone angle
w : Gear width (mm)
D_m : Average pitch diameter (mm)
d_p : Pitch diameter (mm)

Note that the following applies:
$D_{m1} = d_{p1} - w\sin\delta_1$
$D_{m2} = d_{p2} - w\sin\delta_2$

The separating force S and thrust T depend on running direction and gear twist direction as follows:

(i) Clockwise with right twisting or counterclockwise with left twisting

Driving Gear
Separating Force

$$S_1 = \frac{P}{\cos\beta}(\tan\alpha_n \cos\delta_1 + \sin\beta \sin\delta_1)$$

Thrust

$$T_1 = \frac{P}{\cos\beta}(\tan\alpha_n \sin\delta_1 - \sin\beta \cos\delta_1)$$

Driven Gear
Separating Force

$$S_2 = \frac{P}{\cos\beta}(\tan\alpha_n \cos\delta_2 - \sin\beta \sin\delta_2)$$

Thrust

$$T_2 = \frac{P}{\cos\beta}(\tan\alpha_n \sin\delta_2 + \sin\beta \cos\delta_2)$$

(ii) Counterclockwise with right twist or clockwise with left twist

Driving Gear
Separating Force

$$S_1 = \frac{P}{\cos\beta}(\tan\alpha_n \cos\delta_1 - \sin\beta \sin\delta_1)$$

Thrust

$$T_1 = \frac{P}{\cos\beta}(\tan\alpha_n \sin\delta_1 + \sin\beta \cos\delta_1)$$

Driven Gear
Separating Force

$$S_2 = \frac{P}{\cos\beta}(\tan\alpha_n \cos\delta_2 + \sin\beta \sin\delta_2)$$

Thrust

$$T_2 = \frac{P}{\cos\beta}(\tan\alpha_n \sin\delta_2 - \sin\beta \cos\delta_2)$$

A positive (plus) calculation result indicates that the load is acting in a direction that separates the gears while a negative (minus) result indicates that the load is acting in a direction that brings the gears together. Generally, $\delta_1+\delta_2 = 90°$. In this case, T_1 and S_2 (S_1 and T_2) are the same in magnitude but opposite in direction. The load on the bearing can be calculated by the same method as described in Section 4.8.7 "(2) *Calculation of Load on Straight Bevel Gears*."

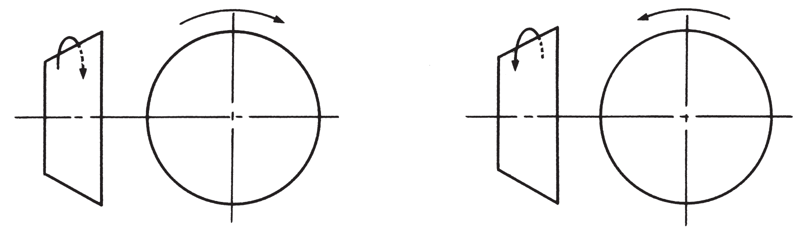


Fig. 4.54 Gear Running Direction

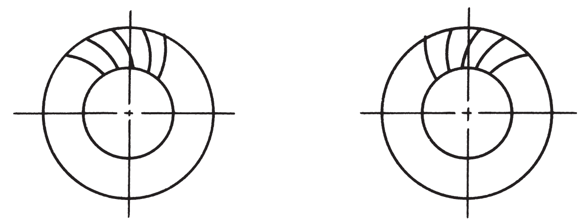


Fig. 4.55 Gear Twist Direction

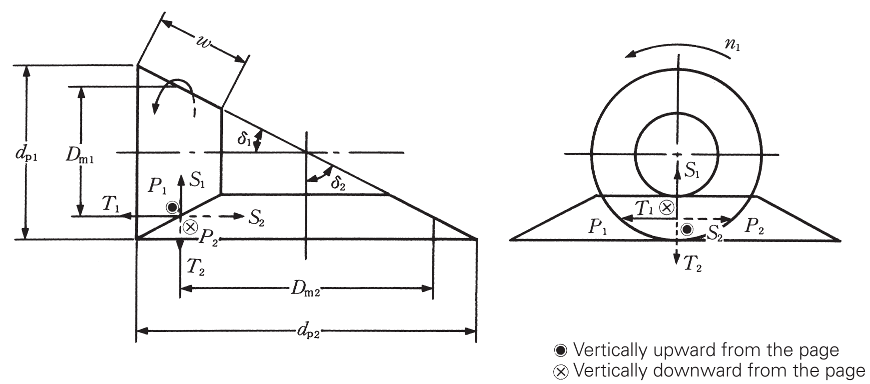


Fig. 4.56

(4) *Calculation of Load on Hypoid Gears*

The force acting at the meshing point of hypoid gears is calculated as follows:

$$P_1 = \frac{9\,550\,000H}{n_1\left(\frac{D_{m1}}{2}\right)} = \frac{\cos\beta_1}{\cos\beta_2}P_2 \cdots\cdots (N)$$

$$= \frac{974\,000H}{n_1\left(\frac{D_{m1}}{2}\right)} = \frac{\cos\beta_1}{\cos\beta_2}P_2 \cdots\cdots \{kgf\}$$

$$P_2 = \frac{9\,550\,000H}{n_2\left(\frac{D_{m2}}{2}\right)} \cdots\cdots (N)$$

$$= \frac{974\,000H}{n_2\left(\frac{D_{m2}}{2}\right)} \cdots\cdots \{kgf\}$$

$$D_{m1} = D_{m2}\frac{z_1}{z_2}\cdot\frac{\cos\beta_1}{\cos\beta_2}$$

$$D_{m2} = d_{p2} - w_2\sin\delta_2$$

where α_n : Gear normal pressure angle
β : Twisting angle
δ : Pitch cone angle
w : Gear width (mm)
D_m : Average pitch diameter (mm)
d_p : Pitch diameter (mm)
z : Number of teeth

The separating force S and thrust T depend on running direction and gear twist direction as follows:

(i) Clockwise with right twisting or counterclockwise with left twisting

Driving Gear
Separating Force

$$S_1 = \frac{P_1}{\cos\beta_1}(\tan\alpha_n \cos\delta_1 + \sin\beta_1 \sin\delta_1)$$

Thrust

$$T_1 = \frac{P_1}{\cos\beta_1}(\tan\alpha_n \sin\delta_1 - \sin\beta_1 \cos\delta_1)$$

Driven Gear
Separating Force

$$S_2 = \frac{P_2}{\cos\beta_2}(\tan\alpha_n \cos\delta_2 - \sin\beta_2 \sin\delta_2)$$

Thrust

$$T_2 = \frac{P_2}{\cos\beta_2}(\tan\alpha_n \sin\delta_2 + \sin\beta_2 \cos\delta_2)$$

(ii) Counterclockwise with right twist or clockwise with left twist

Driving Gear
Separating Force

$$S_1 = \frac{P_1}{\cos\beta_1}(\tan\alpha_n \cos\delta_1 - \sin\beta_1 \sin\delta_1)$$

Thrust

$$T_1 = \frac{P_1}{\cos\beta_1}(\tan\alpha_n \sin\delta_1 + \sin\beta_1 \cos\delta_1)$$

Driven Gear
Separating Force

$$S_2 = \frac{P_2}{\cos\beta_2}(\tan\alpha_n \cos\delta_2 + \sin\beta_2 \sin\delta_2)$$

Thrust

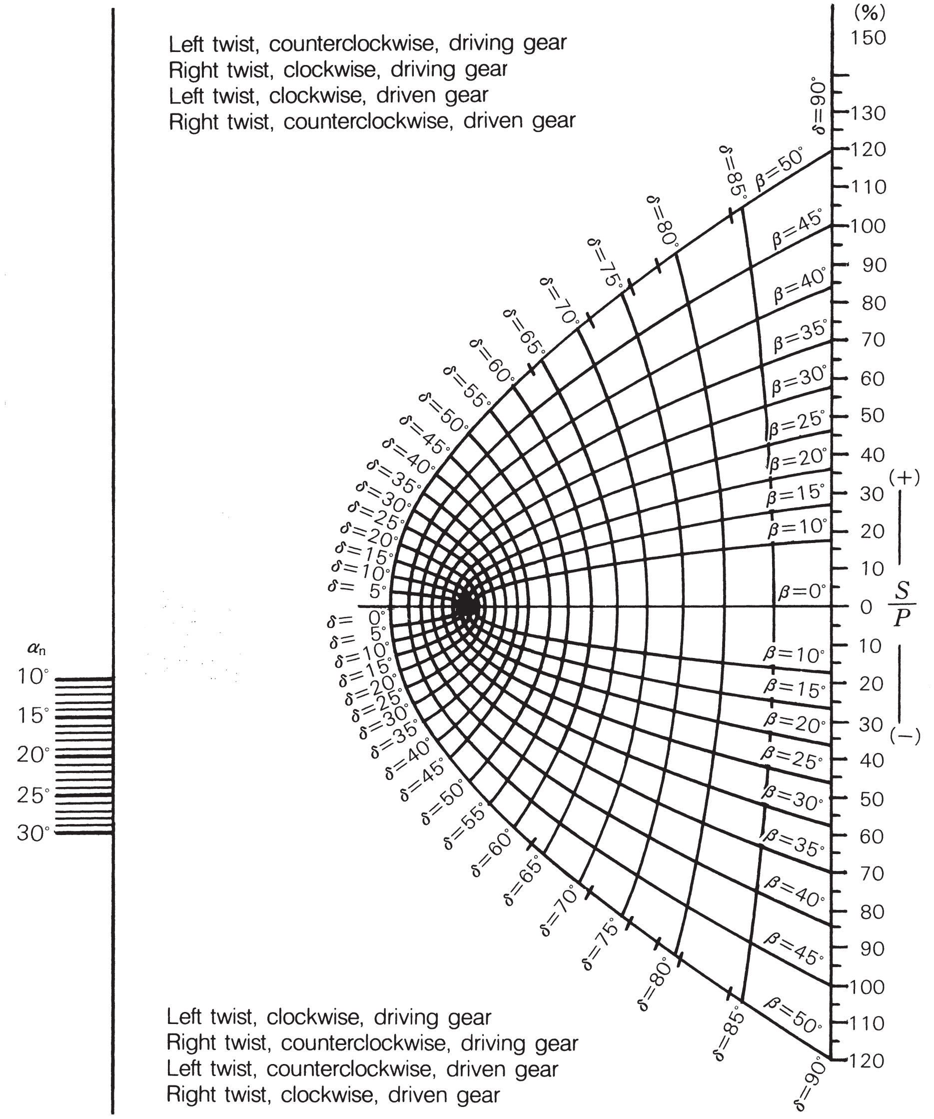
$$T_2 = \frac{P_2}{\cos\beta_2}(\tan\alpha_n \sin\delta_2 - \sin\beta_2 \cos\delta_2)$$

The positive (plus) calculation result indicates that the load is acting in a direction that separates the gears while a negative (minus) result indicates that the load is acting in a direction that brings the gears together.
For more information on running direction and gear twist direction, refer to Section 4.8.7 “(3) *Calculation of Load on Spiral Bevel Gears.*” The load on the bearing can be calculated by the same method as described in Section 4.8.7 “(2) *Calculation of Load Acting on Straight Bevel Gears.*”

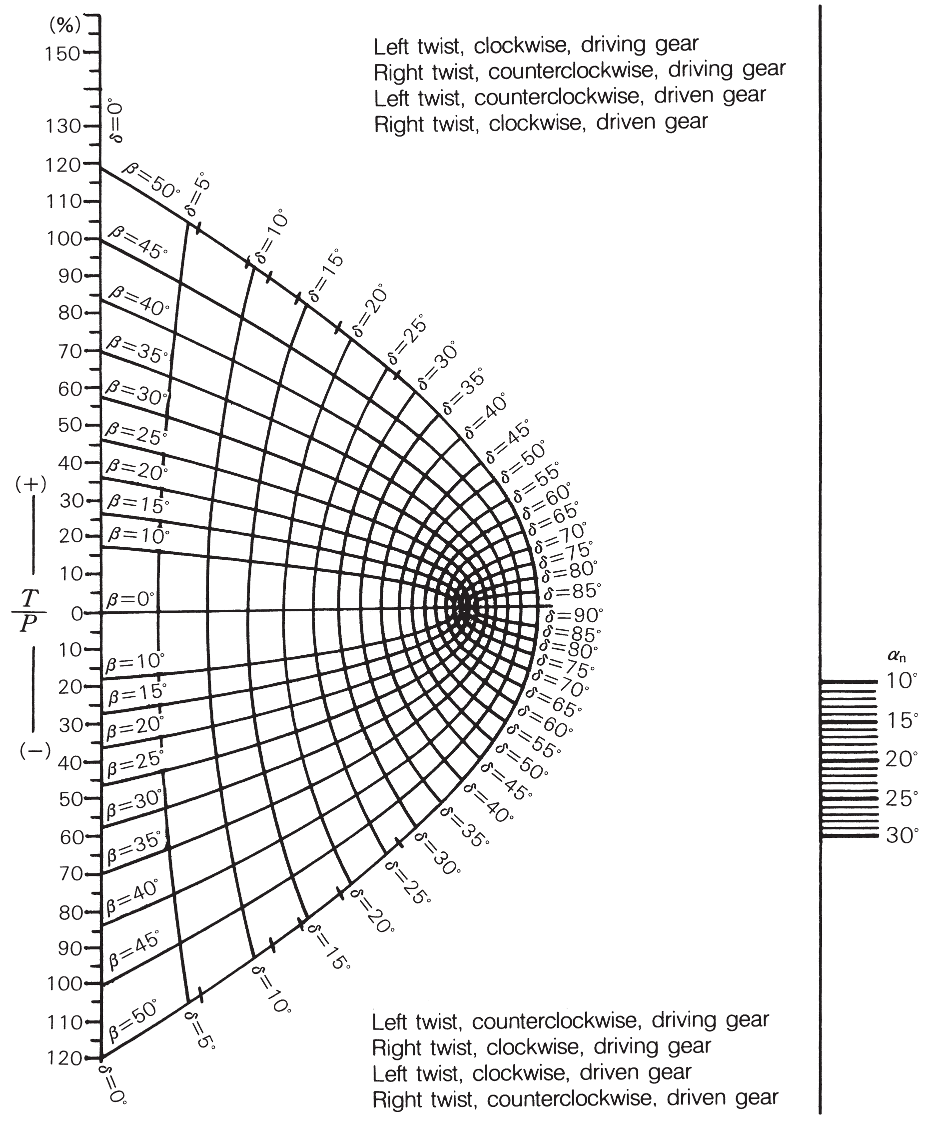
The following calculation diagrams are used to determine the approximate value and direction of separating force S and thrust T.

How To Use

1. Mark the gear normal pressure angle α_n on the vertical scale on the side of the appropriate diagram for S or T.
2. Determine the intersection between the pitch cone angle δ and twist angle β. Match your gear configuration to the text on either side of the β = 0 line, and use that side when determining the point of intersection.
3. Draw a line through the two points and the opposite vertical scale. The point where the line intersects the opposite vertical axis gives the ratio S/P or T/P (%) of the separating force S or thrust T to the tangential force P.



Calculation Diagram for Separating Force S



Calculation Diagram for Thrust T

(5) ***Calculation of Load on Worm Gears***

A worm gear is a kind of spigot gear, which can produce a high reduction ratio with small volume. The load at the meshing point of worm gears is calculated as shown in Table 4.17. Variables used in Table 4.17 are as follows:

i: Gear ratio $\left(i = \dfrac{Z_2}{Z_w}\right)$

η: Worm gear efficiency $\left[\eta = \dfrac{\tan\gamma}{\tan(\gamma+\psi)}\right]$

γ: Advance angle $\left(\gamma = \tan^{-1}\dfrac{d_{p2}}{i d_{p1}}\right)$

ψ: Frictional angle obtained from the following (as shown in Fig. 4.57):

$$V_R = \frac{\pi d_{p1} n_1}{\cos\gamma} \times \frac{10^{-3}}{60}$$

When V_R is 0.2 m/s or less, $\psi = 8°$.
When V_R exceeds 6 m/s, $\psi = 1°4'$.

α_n : Gear normal pressure angle
α_a : Shaft plane pressure angle
Z_w : No. of threads (No. of worm gear teeth)
Z_2 : No. of worm wheel teeth
Subscript 1: For driving worm gear
Subscript 2: For driven worm gear

In a worm gear, there are four combinations of interaction at the meshing point as shown below depending on the twist directions and rotating directions of the worm gear, as shown below.

The load on the bearing is obtained from the magnitude and direction of each component at the meshing point according to the method shown in Table 4.17.

Table 4.17 Gear Loads

Force	Worm	Worm Wheel
Tangential P	$\dfrac{9\,550\,000H}{n_1\left(\dfrac{d_{p1}}{2}\right)}$ …………(N) $\dfrac{974\,000H}{n_1\left(\dfrac{d_{p1}}{2}\right)}$ …………{kgf}	$\dfrac{9\,550\,000Hi\eta}{n_1\left(\dfrac{d_{p2}}{2}\right)} = \dfrac{P_1\,\eta}{\tan\gamma} = \dfrac{P_1}{\tan(\gamma+\psi)}$ …………(N) $\dfrac{974\,000Hi\eta}{n_1\left(\dfrac{d_{p2}}{2}\right)} = \dfrac{P_1\,\eta}{\tan\gamma} = \dfrac{P_1}{\tan(\gamma+\psi)}$ ………{kgf}
Thrust T	$\dfrac{9\,550\,000H\eta}{n_1\left(\dfrac{d_{p2}}{2}\right)} = \dfrac{P_1\,\eta}{\tan\gamma} = \dfrac{P_1}{\tan(\gamma+\psi)}$ …………(N) $\dfrac{974\,000H\eta}{n_1\left(\dfrac{d_{p2}}{2}\right)} = \dfrac{P_1\,\eta}{\tan\gamma} = \dfrac{P_1}{\tan(\gamma+\psi)}$ ………{kgf}	$\dfrac{9\,550\,000H}{n_1\left(\dfrac{d_{p1}}{2}\right)}$ …………(N) $\dfrac{974\,000H}{n_1\left(\dfrac{d_{p1}}{2}\right)}$ …………{kgf}
Separating S	$\dfrac{P_1\tan\alpha_n}{\sin(\gamma+\psi)} = \dfrac{P_1\tan\alpha_a}{\tan(\gamma+\psi)}$ …………(N), {kgf}	$\dfrac{P_1\tan\alpha_n}{\sin(\gamma+\psi)} = \dfrac{P_1\tan\alpha_a}{\tan(\gamma+\psi)}$ …………(N), {kgf}

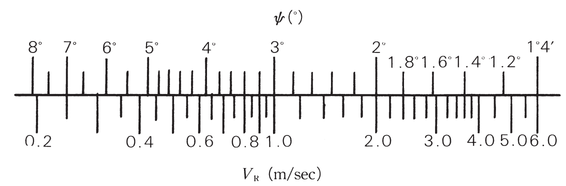


Fig. 4.57

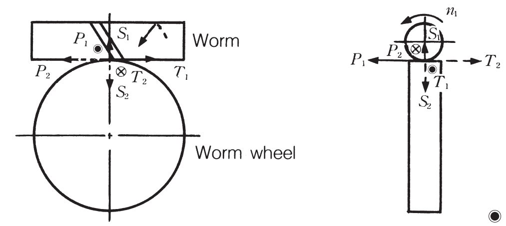


Fig. 4.58 Right Twist Worm Gear

◉ Vertically upward from the page
⊗ Vertically downward from the page

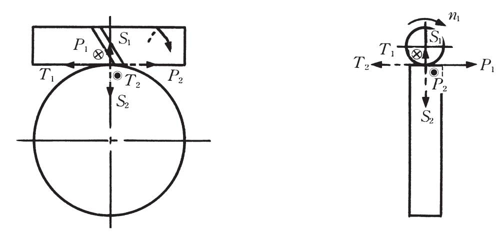


Fig. 4.59 Right Twist Worm Gear (Worm Rotation is Opposite Fig. 4.58)

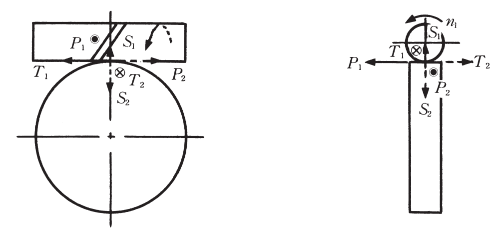


Fig. 4.60 Left Twist Worm Gear

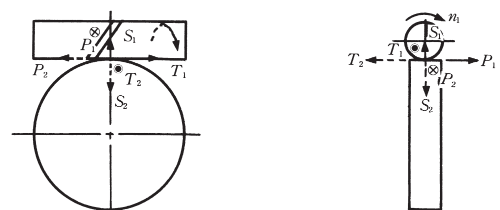
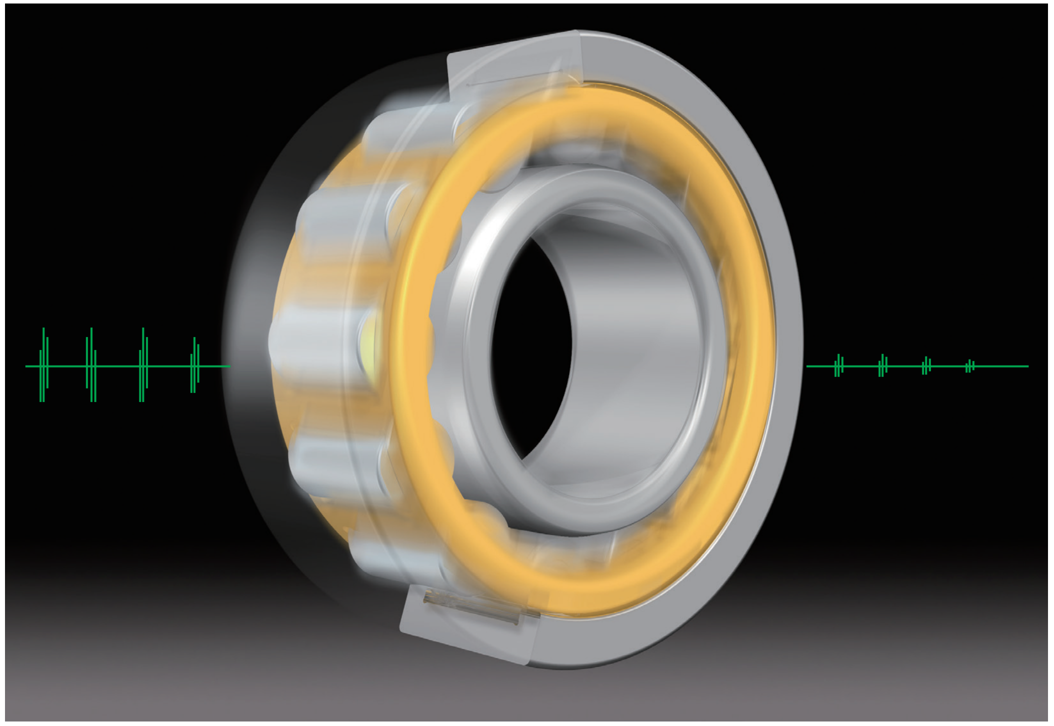


Fig. 4.61 Left Twist Worm Gear (Worm Rotation is Opposite Fig. 4.60)



5. SPEEDS

5. SPEEDS

NSK uses four definitions of speed, as shown in Table 5.1.

Table 5.1 Overview of Speeds

Speed	Overview	Applicable Lubrication Methods
Limiting Speed (Grease)	Empirically obtained and comprehensive bearing limiting speed when using grease lubrication.	Grease lubrication
Limiting Speed (Oil)	Empirically obtained and comprehensive bearing limiting speed when using oil-bath lubrication.	Oil-bath lubrication
Thermal Speed Rating([1])	Rotational speed at which equilibrium is reached between the heat generated by the bearing and the heat flow emitted through the shaft and housing under reference conditions defined by ISO 15312. This is one among various criteria that shows suitability for high-speed operation.	Oil-bath lubrication subject to conditions outlined in ISO 15312
Limiting Speed (Mechanical)([1])	Mechanical and kinematic limiting speed achievable under ideal conditions for lubrication, heat dissipation, and temperature.	Properly designed and controlled forced-circulation oil lubrication

Note ([1]) Thermal speed ratings and limiting speeds (mechanical) are only listed in the tables for single-row cylindrical roller bearings and spherical roller bearings.

5.1 Limiting Speed (Grease/Oil)

When bearings are in operation, the higher the speed, the higher the bearing temperature due to friction. The limiting speed is the empirically obtained value for the maximum speed at which bearings can be continuously operated without generating excessive heat or failing due to seizure. Consequently, the limiting speed of bearings varies depending on such factors as bearing type and size, cage shape and material, load, lubricating method, and heat dissipation of the bearing's surroundings.

The limiting speed (grease) and limiting speed (oil) in the bearing tables are applicable to bearings of standard design subjected to normal loads, i.e. $C/P \geqq 12$ and $F_a/F_r \leqq$ approximately 0.2. The limiting speed (oil) listed in the bearing tables is for conventional oil-bath lubrication.
Some types of lubricants are not suitable for high speed, even though they may be markedly superior in other respects. When speeds are more than 70 percent of the listed limiting speeds, be sure to select a grease or oil with good high speed characteristics.

Reference
Table 11.2 Grease Properties (Pages A236 and 237)
Table 11.5 Example Lubricating Oils for Bearing Operating Conditions (Page A239)
Table 11.6 Brands of Lubricating Grease (Pages A240 and A241)

5.1.1 Correction of Limiting Speed (Grease/Oil)

When bearing load P exceeds 8 % of the basic load rating C, or when the axial load F_a exceeds 20 % of the radial load F_r, the limiting speed (grease) and limiting speed (oil) must be corrected by multiplying the value found in the bearing tables by the correction factor shown in Figs. 5.1 and 5.2.
When the required speed exceeds the limiting speed (oil) of the desired bearing, then the accuracy grade, internal clearance, cage type and material, lubrication, etc. must be carefully studied in order to select a bearing capable of the required speed. In such a case, forced-circulation oil lubrication, jet lubrication, oil-mist lubrication, or oil-air lubrication must be used.
If all these conditions are considered, a corrected maximum permissible speed may be obtained by multiplying the limiting speed (oil) found in the bearing tables by the correction factor shown in table 5.2. Please consult with NSK regarding high-speed applications.

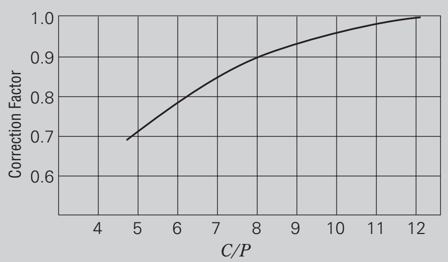


Fig. 5.1 Limiting Speed Correction Factor Variation With Load Ratio

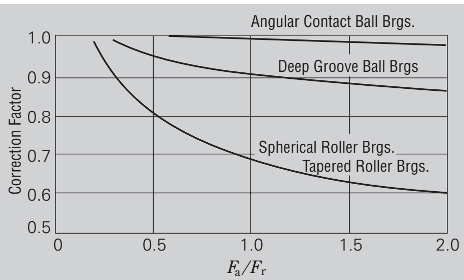


Fig. 5.2 Limiting Speed Correction Factor for Combined Radial and Axial Loads

Table 5.2 Limiting Speed Correction Factor for High-Speed Applications

Bearing Types	Correction Factor
Needle Roller Brgs.(except broad width)	2
Tapered Roller Brgs.	2
Deep Grooove Ball Brgs.	2.5
Angular Contact Ball Brgs.(except matched bearings)	1.5

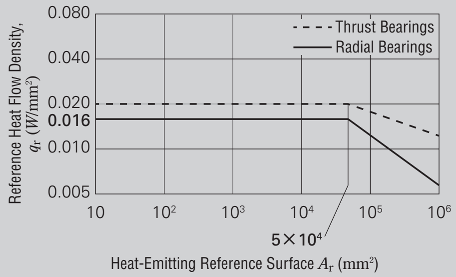


Fig. 5.3 Reference Heat Flow Density

5.1.2 Limiting Speed (Grease/Oil) for Rubber Contact Seals in Ball Bearings

The maximum permissible speed for rubber contact sealed bearings (DDU type) is determined mainly by the sliding surface speed of the inner circumference of the seal. Values for the limiting speed are listed in the bearing tables.

5.2 Thermal Speed Rating

The thermal speed rating is the rotational speed at which equilibrium is reached between the heat generated by the bearing and the heat flow emitted through the shaft and housing under reference conditions defined by ISO 15312. This is just one of various criteria used to determine suitability for operation at high speed.
The reference conditions below are defined by ISO 15312:
- -Rotation: Inner ring rotation with fixed outer ring
- -Ambient temperature: 20 °C
- -Outer ring temperature: 70 °C
- -Load (for radial bearings): 5% of static load rating ($0.05C_{0r}$)
- -Lubrication: Oil bath
- -Lubricant: ISO VG32 (for radial bearings)
- -Clearance: Normal

The amount of heat dissipated through the housing and shaft can be obtained from Fig. 5.3. In the figure, A_r (mm^2) refers to the heat-emitting reference surface area. ISO defines A_r as the total area of the bearing's inner ring bore surface and outer ring outside surface (for radial bearings) and q_r ($\mathrm{W/mm}^2$) as the heat flow density. Heat dissipation is calculated by multiplying this surface area (A_r) by the heat flow density (q_r).

5.3 Limiting Speed (Mechanical)

Limiting speed (mechanical) refers to the mechanical and kinematic limiting speed of bearings achievable under ideal lubrication, heat dissipation and temperature conditions, such as with properly designed and controlled forced-circulation oil lubrication for high-speed conditions.
The limiting speed (mechanical) considers the sliding speed and contact forces between the various bearing elements, centrifugal and gyratory forces, etc. The values in the tables are applicable to standard bearings subjected to normal loads (C/P = approximately 12).

In the bearing tables for single-row cylindrical roller bearings and spherical roller bearings, thermal speed ratings, limiting speeds (mechanical) and limiting speeds (grease) are listed. In the bearing tables for other bearing types, limiting speeds (grease) and limiting speeds (oil) are listed.

5.4 Technical Data

5.4.1 Rotation and Revolution Speed of Rolling Elements

When the rolling element rotates without slippage between the bearing rings, the distance the rolling element rolls on the inner ring raceway is equal to that on the outer ring raceway. This allows for a relationship between inner and outer ring speeds n_i and n_e and rolling element rotations n_a .
The revolution speed of the rolling element can be determined as the arithmetic mean of the circumferential speed on the inner ring raceway and that on the outer ring raceway (generally with either a stationary inner or outer ring). The rotations and revolutions of the rolling elements can be related as expressed by Equations (5.1) through (5.4).

No. of rotations

$$n_a = \left(\frac{D_{pw}}{D_w} - \frac{D_w\cos^2\alpha}{D_{pw}}\right)\frac{n_e - n_i}{2} \quad (\text{min}^{-1}) \quad \cdots\cdots (5.1)$$

Rotational circumferential speed

$$v_a = \frac{\pi D_w}{60\times10^3}\left(\frac{D_{pw}}{D_w} - \frac{D_w\cos^2\alpha}{D_{pw}}\right)\frac{n_e - n_i}{2} \quad (\text{m/s}) \quad \cdots\cdots (5.2)$$

No. of revolutions (No. of cage rotations)

$$n_c = \left(1 - \frac{D_w\cos\alpha}{D_{pw}}\right)\frac{n_i}{2} + \left(1 + \frac{D_w\cos\alpha}{D_{pw}}\right)\frac{n_e}{2} \quad (\text{min}^{-1}) \quad \cdots\cdots (5.3)$$

Revolutional circumferential speed
(cage speed at rolling element pitch diameter)

$$v_c = \frac{\pi D_{pw}}{60\times10^3}\left[\left(1 - \frac{D_w\cos\alpha}{D_{pw}}\right)\frac{n_i}{2} + \left(1 + \frac{D_w\cos\alpha}{D_{pw}}\right)\frac{n_e}{2}\right] \quad (\text{m/s}) \quad \cdots\cdots (5.4)$$

where D_{pw} : Pitch diameter of rolling elements (mm)
D_w : Diameter of rolling element (mm)
α : Contact angle (°)
n_e : Outer ring speed (min^{-1})
n_i : Inner ring speed (min^{-1})

Rotations and revolutions of the rolling elements are shown in Table 5.3 for inner ring rotating ($n_e = 0$) and outer ring rotating ($n_i = 0$) respectively at $0° \leqq \alpha < 90°$ and at $\alpha = 90°$.
Table 5.4 shows the rotation speed n_a and revolution speed n_c of the rolling elements during inner ring rotation of ball bearings 6210 and 6310.

Table 5.4 Rolling Element Rotation Speed n_a and Revolution Speed n_c for Ball Bearings 6210 and 6310

Ball Bearing	γ	n_a	n_c
6210	0.181	$-2.67n_i$	$0.41n_i$
6310	0.232	$-2.04n_i$	$0.38n_i$

Remarks $\gamma = \frac{D_w\cos\alpha}{D_{pw}}$

Table 5.3 Rolling Element Rotation Speed n_a, Rotational Circumferential Speed v_a, Revolution Speed n_c, and Revolutional Circumferential Speed v_c

Contact Angle	Rotation/Revolution Speed	Inner Ring Rotation ($n_e = 0$)	Outer Ring Rotation ($n_i = 0$)
$0° \leqq \alpha < 90°$	n_a (min^{-1})	$-\left(\frac{1}{\gamma} - \gamma\right)\frac{n_i}{2}\cdot\cos\alpha$	$\left(\frac{1}{\gamma} - \gamma\right)\frac{n_e}{2}\cdot\cos\alpha$
	v_a (m/s)	$\frac{\pi D_w}{60\times10^3}n_a$	
	n_c (min^{-1})	$(1-\gamma)\frac{n_i}{2}$	$(1+\gamma)\frac{n_e}{2}$
	v_c (m/s)	$\frac{\pi D_{pw}}{60\times10^3}n_c$	
$\alpha = 90°$	n_a (min^{-1})	$-\frac{1}{\gamma}\cdot\frac{n_i}{2}$	$\frac{1}{\gamma}\cdot\frac{n_e}{2}$
	v_a (m/s)	$\frac{\pi D_w}{60\times10^3}n_a$	
	n_c (min^{-1})	$\frac{n_i}{2}$	$\frac{n_e}{2}$
	v_c (m/s)	$\frac{\pi D_{pw}}{60\times10^3}n_c$	

Reference 1. ±: "+" indicates clockwise rotation while "−" indicates counterclockwise rotation.
2. $\gamma = \frac{D_w\cos\alpha}{D_{pw}}$ $(0° \leqq \alpha < 90°)$, $\gamma = \frac{D_w}{D_{pw}}$ $(\alpha = 90°)$

6. BOUNDARY DIMENSIONS AND BEARING DESIGNATIONS

6. BOUNDARY DIMENSIONS AND BEARING DESIGNATIONS

6.1 Boundary Dimensions and Dimensions of Snap Ring Grooves

6.1.1 Boundary Dimensions

The boundary dimensions of rolling bearings, which are shown in Figs.6.1 through 6.5, refer to the dimensions that define their external geometry. They include bore diameter d, outside diameter D, width B, assembled bearing width(or height) T, chamfer dimension r, etc. All of these dimensions are important when mounting a bearing on a shaft and in a housing. These boundary dimensions have been internationally standardized (ISO15) and adopted by JIS B 1512 (*Boundary Dimensions of Rolling Bearings*).

The boundary dimensions and Dimension Series of radial bearings, tapered roller bearings, and thrust bearings are listed in Tables 6.1 to 6.3 (Pages A106 to A115).

These tables list boundary dimensions for each Diameter and Dimension Series by bore number. A very large number of series are possible; however, not all are currently commercially available. Representative bearing types and series designations are shown across the top of the bearing tables (refer to Table 6.5, Bearing Series Designations on Page A121 for more information).

The relative cross-sectional dimensions of radial bearings (excluding tapered roller bearings) and thrust bearings for various series are shown in Figs. 6.6 and 6.7 respectively.

6.1.2 Dimensions of Snap Ring Grooves and Locating Snap Rings

The dimensions of snap ring grooves in the outer surfaces of bearings and the dimensions and accuracy of the locating snap rings themselves are specified by ISO 464. The dimensions of snap ring grooves and locating snap ring for bearings of Diameter Series 8, 9, 0, 2, 3, and 4 are shown in Table 6.4 (Pages A116 to A119).

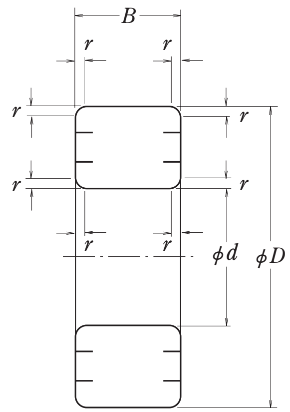


Fig. 6.1 Boundary Dimensions of Radial Ball and Roller Bearings

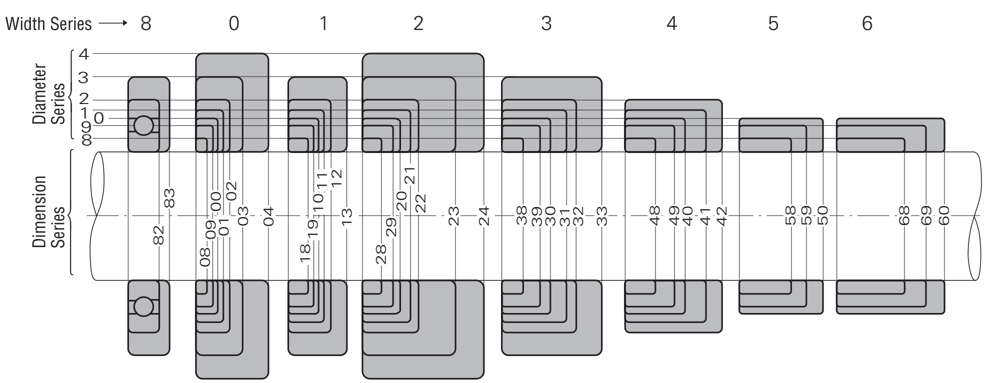


Fig. 6.6 Comparison of Cross Sections of Radial Bearings (Excluding Tapered Roller Bearings) for Various Dimensional Series

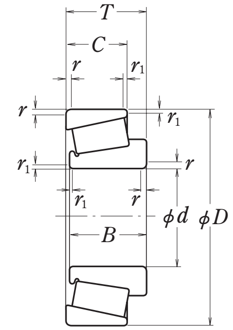


Fig. 6.2 Tapered Roller Bearings

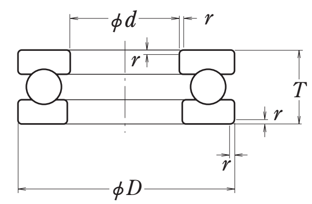


Fig. 6.3 Single-Direction Thrust Ball Bearings

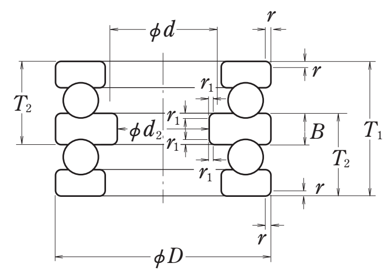


Fig. 6.4 Double-Direction Thrust Ball Bearings

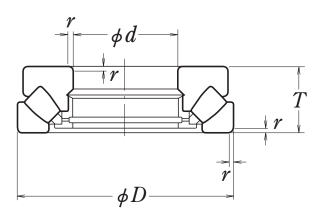


Fig. 6.5 Spherical Thrust Roller Bearings

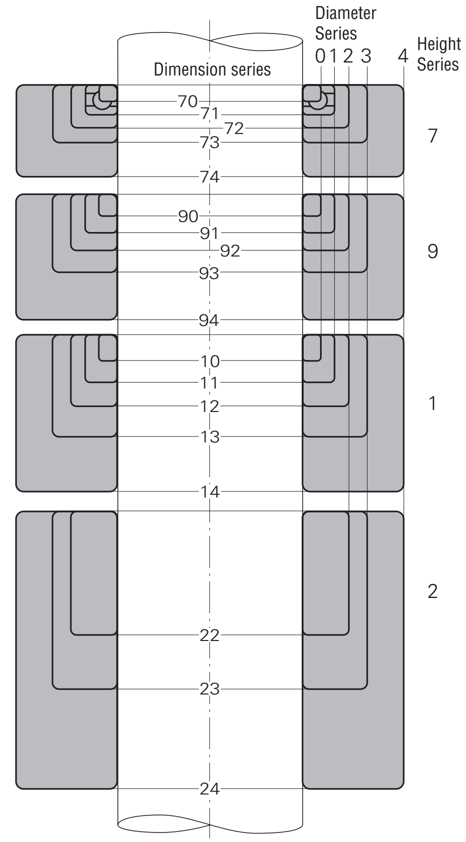


Fig. 6.7 Comparison of Cross Sections of Thrust Bearings (Excluding Diameter Series 5) for Various Dimension Series

Table 6. 1 Boundary Dimensions of Radial Bearings (Excluding Tapered Roller Bearings) — (1)

Units: mm

Single-Row Ball Bearings									68										69 79										160	60 70							
Double-Row Ball Bearings																																					
Cylindrical Roller Bearings										N28	NN38	NN48							N19	N29	NN39	NN49								N10	N20	NN30	NN40				
Needle Roller Bearings												NA48										NA49	NA59	NA69													
Spherical Roller Bearings																					239											230	240				
Bore Number	d	Diameter Series 7					Diameter Series 8										Diameter Series 9											Diameter Series 0									
		D	Dimension Series				D	Dimension Series							Dimension Series		D	Dimension Series							Dimension Series			D	Dimension Series							Dimension Series	
			17	27	37	17~37		08	18	28	38	48	58	68	08	18~68		09	19	29	39	49	59	69	09	19~39	49~69		00	10	20	30	40	50	60	00	10~60
			B			r (min.)		B							r (min.)			B							r (min.)				B							r (min.)	
—	0.6	2	0.8	—	—	0.05	2.5	—	1	—	1.4	—	—	—	—	0.05	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1	1	2.5	1	—	—	0.05	3	—	1	—	1.5	—	—	—	—	0.05	4	—	1.6	—	2.3	—	—	—	—	0.1	—	—	—	—	—	—	—	—	—	—	—
—	1.5	3	1	—	1.8	0.05	4	—	1.2	—	2	—	—	—	—	0.05	5	—	2	—	2.6	—	—	—	—	0.15	—	6	—	2.5	—	3	—	—	—	—	0.15
2	2	4	1.2	—	2	0.05	5	—	1.5	—	2.3	—	—	—	—	0.08	6	—	2.3	—	3	—	—	—	—	0.15	—	7	—	2.8	—	3.5	—	—	—	—	0.15
—	2.5	5	1.5	—	2.3	0.08	6	—	1.8	—	2.6	—	—	—	—	0.08	7	—	2.5	—	3.5	—	—	—	—	0.15	—	8	—	2.8	—	4	—	—	—	—	0.15
3	3	6	2	2.5	3	0.08	7	—	2	—	3	—	—	—	—	0.1	8	—	3	—	4	—	—	—	—	0.15	—	9	—	3	—	5	—	—	—	—	0.15
4	4	7	2	2.5	3	0.08	9	—	2.5	3.5	4	—	—	—	—	0.1	11	—	4	—	5	—	—	—	—	0.15	—	12	—	4	—	6	—	—	—	—	0.2
5	5	8	2	2.5	3	0.08	11	—	3	4	5	—	—	—	—	0.15	13	—	4	—	6	10	—	—	—	0.2	0.15	14	—	5	—	7	—	—	—	—	0.2
6	6	10	2.5	3	3.5	0.1	13	—	3.5	5	6	—	—	—	—	0.15	15	—	5	—	7	10	—	—	—	0.2	0.15	17	—	6	—	9	—	—	—	—	0.3
7	7	11	2.5	3	3.5	0.1	14	—	3.5	5	6	—	—	—	—	0.15	17	—	5	—	7	10	—	—	—	0.3	0.15	19	—	6	8	10	—	—	—	—	0.3
8	8	12	2.5	—	3.5	0.1	16	—	4	5	6	8	—	—	—	0.2	19	—	6	—	9	11	—	—	—	0.3	0.2	22	—	7	9	11	14	19	25	—	0.3
9	9	14	3	—	4.5	0.1	17	—	4	5	6	8	—	—	—	0.2	20	—	6	—	9	11	—	—	—	0.3	0.3	24	—	7	10	12	15	20	27	—	0.3
00	10	15	3	—	4.5	0.1	19	—	5	6	7	9	—	—	—	0.3	22	—	6	8	10	13	16	22	—	0.3	0.3	26	—	8	10	12	16	21	29	—	0.3
01	12	18	4	—	5	0.2	21	—	5	6	7	9	—	—	—	0.3	24	—	6	8	10	13	16	22	—	0.3	0.3	28	7	8	10	12	16	21	29	0.3	0.3
02	15	21	4	—	5	0.2	24	—	5	6	7	9	—	—	—	0.3	28	—	7	8.5	10	13	18	23	—	0.3	0.3	32	8	9	11	13	17	23	30	0.3	0.3
03	17	23	4	—	5	0.2	26	—	5	6	7	9	—	—	—	0.3	30	—	7	8.5	10	13	18	23	—	0.3	0.3	35	8	10	12	14	18	24	32	0.3	0.3
04	20	27	4	—	5	0.2	32	4	7	8	10	12	16	22	0.3	0.3	37	7	9	11	13	17	23	30	0.3	0.3	0.3	42	8	12	14	16	22	30	40	0.3	0.6
/22	22	—	—	—	—	—	34	4	7	—	10	—	16	22	0.3	0.3	39	7	9	11	13	17	23	30	0.3	0.3	0.3	44	8	12	14	16	22	30	40	0.3	0.6
05	25	32	4	—	5	0.2	37	4	7	8	10	12	16	22	0.3	0.3	42	7	9	11	13	17	23	30	0.3	0.3	0.3	47	8	12	14	16	22	30	40	0.3	0.6
/28	28	—	—	—	—	—	40	4	7	—	10	—	16	22	0.3	0.3	45	7	9	11	13	17	23	30	0.3	0.3	0.3	52	8	12	15	18	24	32	43	0.3	0.6
06	30	37	4	—	5	0.2	42	4	7	8	10	12	16	22	0.3	0.3	47	7	9	11	13	17	23	30	0.3	0.3	0.3	55	9	13	16	19	25	34	45	0.3	1
/32	32	—	—	—	—	—	44	4	7	—	10	—	16	22	0.3	0.3	52	7	10	13	15	20	27	36	0.3	0.6	0.6	58	9	13	16	20	26	35	47	0.3	1
07	35	—	—	—	—	—	47	4	7	8	10	12	16	22	0.3	0.3	55	7	10	13	15	20	27	36	0.3	0.6	0.6	62	9	14	17	20	27	36	48	0.3	1
08	40	—	—	—	—	—	52	4	7	8	10	12	16	22	0.3	0.3	62	8	12	14	16	22	30	40	0.3	0.6	0.6	68	9	15	18	21	28	38	50	0.3	1
09	45	—	—	—	—	—	58	4	7	8	10	13	18	23	0.3	0.3	68	8	12	14	16	22	30	40	0.3	0.6	0.6	75	10	16	19	23	30	40	54	0.6	1
10	50	—	—	—	—	—	65	5	7	10	12	15	20	27	0.3	0.3	72	8	12	14	16	22	30	40	0.3	0.6	0.6	80	10	16	19	23	30	40	54	0.6	1
11	55	—	—	—	—	—	72	7	9	11	13	17	23	30	0.3	0.3	80	9	13	16	19	25	34	45	0.3	1	1	90	11	18	22	26	35	46	63	0.6	1.1
12	60	—	—	—	—	—	78	7	10	12	14	18	24	32	0.3	0.3	85	9	13	16	19	25	34	45	0.3	1	1	95	11	18	22	26	35	46	63	0.6	1.1
13	65	—	—	—	—	—	85	7	10	13	15	20	27	36	0.3	0.6	90	9	13	16	19	25	34	45	0.3	1	1	100	11	18	22	26	35	46	63	0.6	1.1
14	70	—	—	—	—	—	90	8	10	13	15	20	27	36	0.3	0.6	100	10	16	19	23	30	40	54	0.6	1	1	110	13	20	24	30	40	54	71	0.6	1.1
15	75	—	—	—	—	—	95	8	10	13	15	20	27	36	0.3	0.6	105	10	16	19	23	30	40	54	0.6	1	1	115	13	20	24	30	40	54	71	0.6	1.1
16	80	—	—	—	—	—	100	8	10	13	15	20	27	36	0.3	0.6	110	10	16	19	23	30	40	54	0.6	1	1	125	14	22	27	34	45	60	80	0.6	1.1
17	85	—	—	—	—	—	110	9	13	16	19	25	34	45	0.3	1	120	11	18	22	26	35	46	63	0.6	1.1	1.1	130	14	22	27	34	45	60	80	0.6	1.1
18	90	—	—	—	—	—	115	9	13	16	19	25	34	45	0.3	1	125	11	18	22	26	35	46	63	0.6	1.1	1.1	140	16	24	30	37	50	67	90	1	1.5
19	95	—	—	—	—	—	120	9	13	16	19	25	34	45	0.3	1	130	11	18	22	26	35	46	63	0.6	1.1	1.1	145	16	24	30	37	50	67	90	1	1.5
20	100	—	—	—	—	—	125	9	13	16	19	25	34	45	0.3	1	140	13	20	24	30	40	54	71	0.6	1.1	1.1	150	16	24	30	37	50	67	90	1	1.5
21	105	—	—	—	—	—	130	9	13	16	19	25	34	45	0.3	1	145	13	20	24	30	40	54	71	0.6	1.1	1.1	160	18	26	33	41	56	75	100	1	2
22	110	—	—	—	—	—	140	10	16	19	23	30	40	54	0.6	1	150	13	20	24	30	40	54	71	0.6	1.1	1.1	170	19	28	36	45	60	80	109	1	2
24	120	—	—	—	—	—	150	10	16	19	23	30	40	54	0.6	1	165	14	22	27	34	45	60	80	0.6	1.1	1.1	180	19	28	36	46	60	80	109	1	2
26	130	—	—	—	—	—	165	11	18	22	26	35	46	63	0.6	1.1	180	16	24	30	37	50	67	90	1	1.5	1.5	200	22	33	42	52	69	95	125	1.1	2
28	140	—	—	—	—	—	175	11	18	22	26	35	46	63	0.6	1.1	190	16	24	30	37	50	67	90	1	1.5	1.5	210	22	33	42	53	69	95	125	1.1	2
30	150	—	—	—	—	—	190	13	20	24	30	40	54	71	0.6	1.1	210	19	28	36	45	60	80	109	1	2	2	225	24	35	45	56	75	100	136	1.1	2.1
32	160	—	—	—	—	—	200	13	20	24	30	40	54	71	0.6	1.1	220	19	28	36	45	60	80	109	1	2	2	240	25	38	48	60	80	109	145	1.5	2.1
34	170	—	—	—	—	—	215	14	22	27	34	45	60	80	0.6	1.1	230	19	28	36	45	60	80	109	1	2	2	260	28	42	54	67	90	122	160	1.5	2.1
36	180	—	—	—	—	—	225	14	22	27	34	45	60	80	0.6	1.1	250	22	33	42	52	69	95	125	1.1	2	2	280	31	46	60	74	100	136	180	2	2.1
38	190	—	—	—	—	—	240	16	24	30	37	50	67	90	1	1.5	260	22	33	42	52	69	95	125	1.1	2	2	290	31	46	60	75	100	136	180	2	2.1
40	200	—	—	—	—	—	250	16	24	30	37	50	67	90	1	1.5	280	25	38	48	60	80	109	145	1.5	2.1	2.1	310	34	51	66	82	109	150	200	2	2.1
44	220	—	—	—	—	—	270	16	24	30	37	50	67	90	1	1.5	300	25	38	48	60	80	109	145	1.5	2.1	2.1	340	37	56	72	90	118	160	218	2.1	3
48	240	—	—	—	—	—	300	19	28	36	45	60	80	109	1	2	320	25	38	48	60	80	109	145	1.5	2.1	2.1	360	37	56	72	92	118	160	218	2.1	3
52	260	—	—	—	—	—	320	19	28	36	45	60	80	109	1	2	360	31	46	60	75	100	136	180	2	2.1	2.1	400	44	65	82	104	140	190	250	3	4
56	280	—	—	—	—	—	350	22	33	42	52	69	95	125	1.1	2	380	31	46	60	75	100	136	180	2	2.1	2.1	420	44	65	82	106	140	190	250	3	4
60	300	—	—	—	—	—	380	25	38	48	60	80	109	145	1.5	2.1	420	37	56	72	90	118	160	218	2.1	3	3	460	50	74	95	118	160	218	290	4	4
64	320	—	—	—	—	—	400	25	38	48	60	80	109	145	1.5	2.1	440	37	56	72	90	118	160	218	2.1	3	3	480	50	74	95	121	160	218	290	4	4
68	340	—	—	—	—	—	420	25	38	48	60	80	109	145	1.5	2.1	460	37	56	72	90	118	160	218	2.1	3	3	520	57	82	106	133	180	243	325	4	5
72	360	—	—	—	—	—	440	25	38	48	60	80	109	145	1.5	2.1	480	37	56	72	90	118	160	218	2.1	3	3	540	57	82	106	134	180	243	325	4	5
76	380	—	—	—	—	—	480	31	46	60	75	100	136	180	2	2.1	520	44	65	82	106	140	190	250	3	4	4	560	57	82	106	135	180	243	325	4	5
80	400	—	—	—	—	—	500	31	46	60	75	100	136	180	2	2.1	540	44	65	82	106	140	190	250	3	4	4	600	63	90	118	148	200	272	355	5	5
84	420	—	—	—	—	—	520	31	46	60	75	100	136	180	2	2.1	560	44	65	82	106	140	190	250	3	4	4	620	63	90	118	150	200	272	355	5	5
88	440	—	—	—	—	—	540	31	46	60	75	100	136	180	2	2.1	600	50	74	95	118	160	218	290	4	4	4	650	67	94	122	157	212	280	375	5	6
92	460	—	—	—	—	—	580	37	56	72	90	118	160	218	2.1	3	620	50	74	95	118	160	218	290	4	4	4	680	71	100	128	163	218	300	400	5	6
96	480	—	—	—	—	—	600	37	56	72	90	118	160	218	2.1	3	650	54	78	100	128	170	230	308	4	5	5	700	71	100	128	165	218	300	400	5	6
/500	500	—	—	—	—	—	620	37	56	72	90	118	160	218	2.1	3	670	54	78	100	128	170	230	308	4	5	5	720	71	100	128	167	218	300	400	5	6
/530	530	—	—	—	—	—	650	37	56	72	90	118	160	218	2.1	3	710	57	82	106	136	180	243	325	4	5	5	780	80	112	145	185	250	335	450	6	6
/560	560	—	—	—	—	—	680	37	56	72	90	118	160	218	2.1	3	750	60	85	112	140	190	258	345	5	5	5	820	82	115	150	195	258	355	462	6	6
/600	600	—	—	—	—	—	730	42	60	78	98	128	175	236	3	3	800	63	90	118	150	200	272	355	5	5	5	870	85	118	155	200	272	365	488	6	6
/630	630	—	—	—	—	—	780	48	69	88	112	150	200	272	3	4	850	71	100	128	165	218	300	400	5	6	6	920	92	128	170	212	290	388	515	6	7.5
/670	670	—	—	—	—	—	820	48	69	88	112	150	200	272	3	4	900	73	103	136	170	230	308	412	5	6	6	980	100	136	180	230	308	425	560	6	7.5
/710	710	—	—	—	—	—	870	50	74	95	118	160	218	290	4	4	950	78	106	140	180	243	325	438	5	6	6	1030	103	140	185	236	315	438	580	6	7.5
/750	750	—	—	—	—	—	920	54	78	100	128	170	230	308	4	5	1000	80	112	145	185	250	335	450	6	6	6	1090	109	150	195	250	335	462	615	7.5	7.5
/800	800	—	—	—	—	—	980	57	82	106	136	180	243	325	4	5	1060	82	115	150	195	258	355	462	6	6	6	1150	112	155	200	258	345	475	630	7.5	7.5
/850	850	—	—	—	—	—	1030	57	82	106	136	180	243	325	4	5	1120	85	118	155	200	272	365	488	6	6	6	1220	118	165	212	272	365	500	670	7.5	7.5
/900	900	—	—	—	—	—	1090	60	85	112	140	190	258	345	5	5	1180	88	122	165	206	280	375	500	6	6	6	1280	122	170	218	280	375	515	690	7.5	7.5
/950	950	—	—	—	—	—	1150	63	90	118	150	200	272	355	5	5	1250	95	132	175	224	300	400	545	6	7.5	7.5	1360	132	180	236	300	412	560	730	7.5	7.5
/1000	1000	—	—	—	—	—	1220	71	100	128	165	218	300	400	5	6	1320	103	140	185	236	315	438	580	6	7.5	7.5	1420	136	185	243	308	412	560	750	7.5	7.5
/1060	1060	—	—	—	—	—	1280	71	100	128	165	218	300	400	5	6	1400	109	150	195	250	335	462	615	7.5	7.5	7.5	1500	140	195	250	325	438	600	800	9.5	9.5
/1120	1120	—	—	—	—	—	1360	78	106	140	180	243	325	438	5	6	1460	109	150	195	250	335	462	615	7.5	7.5	7.5	1580	145	200	265	345	462	615	825	9.5	9.5
/1180	1180	—	—	—	—	—	1420	78	106	140	180	243	325	438	5	6	1540	115	160	206	272	355	488	650	7.5	7.5	7.5	1660	155	212	272	355	475	650	875	9.5	9.5
/1250	1250	—	—	—	—	—	1500	80	112	145	185	250	335	450	6	6	1630	122	170	218	280	375	515	690	7.5	7.5	7.5	1750	—	218	290	375	500	—	—	—	9.5
/1320	1320	—	—	—	—	—	1600	88	122	165	206	280	375	500	6	6	1720	128	175	230	300	400	545	710	7.5	7.5	7.5	1850	—	230	300	400	530	—	—	—	12
/1400	1400	—	—	—	—	—	1700	95	132	175	224	300	400	545	6	7.5	1820	—	185	243	315	425	—	—	—	9.5	9.5	1950	—	243	315	412	545	—	—	—	12
/1500	1500	—	—	—	—	—	1820	—	140	185	243	315	—	—	—	7.5	1950	—	195	258	335	450	—	—	—	9.5	9.5	2120	—	272	355	462	615	—	—	—	12
/1600	1600	—	—	—	—	—	1950	—	155	200	265	345	—	—	—	7.5	2060	—	200	265	345	462	—	—	—	9.5	9.5	2240	—	280	365	475	630	—	—	—	12
/1700	1700	—	—	—	—	—	2060	—	160	206	272	355	—	—	—	7.5	2180	—	212	280	355	475	—	—	—	9.5	9.5	2360	—	290	375	500	650	—	—	—	15
/1800	1800	—	—	—	—	—	2180	—	165	218	290	375	—	—	—	9.5	2300	—	218	290	375	500	—	—	—	12	12	2500	—	308	400	530	690	—	—	—	15
/1900	1900	—	—	—	—	—	2300	—	175	230	300	400	—	—	—	9.5	2430	—	230	308	400	530	—	—	—	12	12	—	—	—	—	—	—	—	—	—	—
/2000	2000	—	—	—	—	—	2430	—	190	250	325	425	—	—	—	9.5	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

Remarks The chamfer dimensions listed in this table do not necessarily apply to the following:
(a) Chamfers of grooves in outer rings with snap ring grooves
(b) Chamfers on a side without ribs in thin-section cylindrical roller bearings
(c) Chamfers on the front-facing side in angular contact ball bearings
(d) Chamfers on inner rings in bearings with tapered bores

Table 6. 1 Boundary Dimensions of Radial Bearings (Excluding Tapered Roller Bearings) — (2)

Units: mm

Diameter Series 1

Bore Number	d	D	B 01	B 11	B 21	B 31	B 41	r (min.) 01	r (min.) 11~41
Single-Row Ball Bearings									
Double-Row Ball Bearings									
Cylindrical Roller Bearings						NN 31			
Needle Roller Bearings									
Spherical Roller Bearings						231	241		
—	0.6	—	—	—	—	—	—	—	—
1	1	—	—	—	—	—	—	—	—
—	1.5	—	—	—	—	—	—	—	—
2	2	—	—	—	—	—	—	—	—
—	2.5	—	—	—	—	—	—	—	—
3	3	—	—	—	—	—	—	—	—
4	4	—	—	—	—	—	—	—	—
5	5	—	—	—	—	7	—	—	—
6	6	—	—	—	8	10	—	—	—
7	7	—	—	—	9	11	14	—	—
8	8	—	—	—	10	12	15	—	—
9	9	—	—	—	10	12	16	—	—
00	10	—	—	—	12	14	18	—	—
01	12	—	—	—	12	14	18	—	—
02	15	—	—	—	12	14	18	—	—
03	17	—	—	—	13	15	20	—	—
04	20	—	—	—	15	18	24	—	—
/22	22	—	—	—	16	19	25	—	—
05	25	—	—	—	16	19	25	—	—
/28	28	—	—	—	17	20	27	—	—
06	30	—	—	—	18	21	28	—	—
/32	32	—	—	—	19	23	30	—	—
07	35	—	—	—	21	25	33	—	—
08	40	—	—	—	22	26	35	—	—
09	45	—	—	—	22	26	35	—	—
10	50	—	—	—	22	26	35	—	—
11	55	—	—	—	24	30	40	—	—
12	60	—	—	—	24	30	40	—	—
13	65	—	—	—	27	34	45	—	—
14	70	—	—	—	27	34	45	—	—
15	75	—	—	—	30	37	50	—	—
16	80	—	—	—	30	37	50	—	—
17	85	—	—	—	31	41	56	—	—
18	90	150	—	—	33	45	60	—	2
19	95	160	—	—	39	52	65	—	2
20	100	165	21	30	39	52	65	1.1	2
21	105	175	22	33	42	56	69	1.1	2
22	110	180	22	33	42	56	69	1.1	2
24	120	200	25	38	48	62	80	1.5	2
26	130	210	25	38	48	64	80	1.5	2
28	140	225	27	40	50	68	85	1.5	2.1
30	150	250	31	46	60	80	100	2	2.1
32	160	270	34	51	66	86	109	2	2.1
34	170	280	34	51	66	88	109	2	2.1
36	180	300	37	56	72	96	118	2.1	3
38	190	320	42	60	78	104	128	3	3
40	200	340	44	65	82	112	140	3	3
44	220	370	48	69	88	120	150	3	4
48	240	400	50	74	95	128	160	4	4
52	260	440	57	82	106	144	180	4	4
56	280	460	57	82	106	146	180	4	5
60	300	500	63	90	118	160	200	5	5
64	320	540	71	100	128	176	218	5	5
68	340	580	78	106	140	190	243	5	5
72	360	600	78	106	140	192	243	5	5
76	380	620	78	106	140	194	243	5	5
80	400	650	80	112	145	200	250	6	6
84	420	700	88	122	165	224	280	6	6
88	440	720	88	122	165	226	280	6	6
92	460	760	95	132	175	240	300	6	7.5
96	480	790	100	136	180	248	308	6	7.5
/500	500	830	106	145	190	264	325	7.5	7.5
/530	530	870	109	150	195	272	335	7.5	7.5
/560	560	920	115	160	206	280	355	7.5	7.5
/600	600	980	122	170	218	300	375	7.5	7.5
/630	630	1030	128	175	230	315	400	7.5	7.5
/670	670	1090	136	185	243	336	412	7.5	7.5
/710	710	1150	140	195	250	345	438	9.5	9.5
/750	750	1220	150	206	272	365	475	9.5	9.5
/800	800	1280	155	212	272	375	475	9.5	9.5
/850	850	1360	165	224	290	400	500	12	12
/900	900	1420	165	230	300	412	515	12	12
/950	950	1500	175	243	315	438	545	12	12
/1000	1000	1580	185	258	335	462	580	12	12
/1060	1060	1660	190	265	345	475	600	12	15
/1120	1120	1750	—	280	365	475	630	—	15
/1180	1180	1850	—	290	388	500	670	—	15
/1250	1250	1950	—	308	400	530	710	—	15
/1320	1320	2060	—	325	425	560	750	—	15
/1400	1400	2180	—	345	450	580	775	—	19
/1500	1500	2300	—	355	462	600	800	—	19

Diameter Series 2

Bore Number	d	D	B 82	B 02	B 12	B 22	B 32	B 42	r (min.) 82	r (min.) 02~42
Single-Row Ball Bearings				62 72		622	632			
Double-Row Ball Bearings				12		42 22	52 53			
Cylindrical Roller Bearings				N 2		N 22	N 32			
Needle Roller Bearings										
Spherical Roller Bearings						222	232			
—	0.6	—	—	—	—	—	—	—	—	—
1	1	—	—	—	—	—	—	—	—	—
—	1.5	—	—	—	—	—	—	—	—	—
2	2	—	—	—	—	—	—	—	—	—
—	2.5	—	—	—	—	—	—	—	—	—
3	3	10	2.5	4	—	—	5	—	0.1	0.15
4	4	13	3	5	—	—	7	—	0.15	0.2
5	5	16	3.5	5	—	—	8	—	0.15	0.3
6	6	19	4	6	—	—	10	—	0.2	0.3
7	7	22	5	7	—	—	11	—	0.3	0.3
8	8	24	5	8	—	—	12	—	0.3	0.3
9	9	26	6	8	—	—	13	—	0.3	0.3
00	10	30	7	9	—	14	14.3	—	0.3	0.6
01	12	32	7	10	—	14	15.9	—	0.3	0.6
02	15	35	8	11	—	14	15.9	20	0.3	0.6
03	17	40	8	12	—	16	17.5	22	0.3	0.6
04	20	47	9	14	—	18	20.6	27	0.3	1
/22	22	50	9	14	—	18	20.6	27	0.3	1
05	25	52	10	15	—	18	20.6	27	0.3	1
/28	28	58	10	16	—	19	23	30	0.6	1
06	30	62	10	16	—	20	23.8	32	0.6	1
/32	32	65	11	17	—	21	25	33	0.6	1
07	35	72	12	17	—	23	27	37	0.6	1.1
08	40	80	13	18	—	23	30.2	40	0.6	1.1
09	45	85	13	19	—	23	30.2	40	0.6	1.1
10	50	90	13	20	—	23	30.2	40	0.6	1.1
11	55	100	14	21	—	25	33.3	45	0.6	1.5
12	60	110	16	22	—	28	36.5	50	1	1.5
13	65	120	18	23	—	31	38.1	56	1	1.5
14	70	125	18	24	—	31	39.7	56	1	1.5
15	75	130	18	25	—	31	41.3	56	1	1.5
16	80	140	19	26	—	33	44.4	60	1	2
17	85	150	21	28	—	36	49.2	65	1.1	2
18	90	160	22	30	—	40	52.4	69	1.1	2
19	95	170	24	32	—	43	55.6	75	1.1	2.1
20	100	180	25	34	—	46	60.3	80	1.5	2.1
21	105	190	27	36	—	50	65.1	85	1.5	2.1
22	110	200	28	38	—	53	69.8	90	1.5	2.1
24	120	215	—	40	42	58	76	95	—	2.1
26	130	230	—	40	46	64	80	100	—	3
28	140	250	—	42	50	68	88	109	—	3
30	150	270	—	45	54	73	96	118	—	3
32	160	290	—	48	58	80	104	128	—	3
34	170	310	—	52	62	86	110	140	—	4
36	180	320	—	52	62	86	112	140	—	4
38	190	340	—	55	65	92	120	150	—	4
40	200	360	—	58	70	98	128	160	—	4
44	220	400	—	65	78	108	144	180	—	4
48	240	440	—	72	85	120	160	200	—	4
52	260	480	—	80	90	130	174	218	—	5
56	280	500	—	80	90	130	176	218	—	5
60	300	540	—	85	98	140	192	243	—	5
64	320	580	—	92	105	150	208	258	—	5
68	340	620	—	92	118	165	224	280	—	6
72	360	650	—	95	122	170	232	290	—	6
76	380	680	—	95	132	175	240	300	—	6
80	400	720	—	103	140	185	256	315	—	6
84	420	760	—	109	150	195	272	335	—	7.5
88	440	790	—	112	155	200	280	345	—	7.5
92	460	830	—	118	165	212	296	365	—	7.5
96	480	870	—	125	170	224	310	388	—	7.5
/500	500	920	—	136	185	243	336	412	—	7.5
/530	530	980	—	145	200	258	355	450	—	9.5
/560	560	1030	—	150	206	272	365	475	—	9.5
/600	600	1090	—	155	212	280	388	488	—	9.5
/630	630	1150	—	165	230	300	412	515	—	12
/670	670	1220	—	175	243	315	438	545	—	12
/710	710	1280	—	180	250	325	450	560	—	12
/750	750	1360	—	195	265	345	475	615	—	15
/800	800	1420	—	200	272	355	488	615	—	15
/850	850	1500	—	206	280	375	515	650	—	15
/900	900	1580	—	218	300	388	515	670	—	15
/950	950	1660	—	230	315	412	530	710	—	15
/1000	1000	1750	—	243	330	425	560	750	—	15
/1060	1060	—	—	—	—	—	—	—	—	—
/1120	1120	—	—	—	—	—	—	—	—	—
/1180	1180	—	—	—	—	—	—	—	—	—
/1250	1250	—	—	—	—	—	—	—	—	—
/1320	1320	—	—	—	—	—	—	—	—	—
/1400	1400	—	—	—	—	—	—	—	—	—
/1500	1500	—	—	—	—	—	—	—	—	—

Diameter Series 3 and Diameter Series 4

Bore Number	d	DS3 D	DS3 B 83	DS3 B 03	DS3 B 13	DS3 B 23	DS3 B 33	DS3 r (min.) 83	DS3 r (min.) 03~33	DS4 D	DS4 B 04	DS4 B 24	DS4 r (min.) 04~24
Single-Row Ball Bearings				63 73		623	633				64 74		
Double-Row Ball Bearings				13		43 23	53 33				104		
Cylindrical Roller Bearings				N 3		N 23	N 33				N 4		
Needle Roller Bearings													
Spherical Roller Bearings				213		223							
—	0.6	—	—	—	—	—	—	—	—	—	—	—	—
1	1	—	—	—	—	—	—	—	—	—	—	—	—
—	1.5	—	—	—	—	—	—	—	—	—	—	—	—
2	2	—	—	—	—	—	—	—	—	—	—	—	—
—	2.5	—	—	—	—	—	—	—	—	—	—	—	—
3	3	13	—	5	—	—	7	—	0.2	—	—	—	—
4	4	16	—	5	—	—	9	—	0.3	—	—	—	—
5	5	19	—	6	—	—	10	—	0.3	—	—	—	—
6	6	22	—	7	—	11	13	—	0.3	—	—	—	—
7	7	26	—	9	—	13	15	—	0.3	—	—	—	—
8	8	28	—	9	—	13	15	—	0.3	30	10	14	0.6
9	9	30	—	10	—	14	16	—	0.6	32	11	15	0.6
00	10	35	9	11	—	17	19	0.3	0.6	37	12	16	0.6
01	12	37	9	12	—	17	19	0.3	1	42	13	19	1
02	15	42	9	13	—	17	19	0.3	1	52	15	24	1.1
03	17	47	10	14	—	19	22.2	0.6	1	62	17	29	1.1
04	20	52	10	15	—	21	22.2	0.6	1.1	72	19	33	1.1
/22	22	56	11	16	—	21	25	0.6	1.1	—	—	—	—
05	25	62	12	17	—	24	25.4	0.6	1.1	80	21	36	1.5
/28	28	68	13	18	—	24	30	0.6	1.1	—	—	—	—
06	30	72	13	19	—	27	30.2	0.6	1.1	90	23	40	1.5
/32	32	75	14	20	—	28	32	0.6	1.1	—	—	—	—
07	35	80	14	21	—	31	34.9	0.6	1.5	100	25	43	1.5
08	40	90	16	23	—	33	36.5	1	1.5	110	27	46	2
09	45	100	17	25	—	36	39.7	1	1.5	120	29	50	2
10	50	110	19	27	—	40	44.4	1	2	130	31	53	2.1
11	55	120	21	29	—	43	49.2	1.1	2	140	33	57	2.1
12	60	130	22	31	—	46	54	1.1	2.1	150	35	60	2.1
13	65	140	24	33	—	48	58.7	1.1	2.1	160	37	64	2.1
14	70	150	25	35	—	51	63.5	1.5	2.1	180	42	74	3
15	75	160	27	37	—	55	68.3	1.5	2.1	190	45	77	3
16	80	170	28	39	—	58	68.3	1.5	2.1	200	48	80	3
17	85	180	30	41	—	60	73	2	3	210	52	86	4
18	90	190	30	43	—	64	73	2	3	225	54	90	4
19	95	200	33	45	—	67	77.8	2	3	240	55	95	4
20	100	215	36	47	51	73	82.6	2.1	3	250	58	98	4
21	105	225	37	49	53	77	87.3	2.1	3	260	60	100	4
22	110	240	42	50	57	80	92.1	3	3	280	65	108	4
24	120	260	44	55	62	86	106	3	3	310	72	118	5
26	130	280	48	58	66	93	112	3	4	340	78	128	5
28	140	300	50	62	70	102	118	4	4	360	82	132	5
30	150	320	—	65	75	108	128	—	4	380	85	138	5
32	160	340	—	68	79	114	136	—	4	400	88	142	5
34	170	360	—	72	84	120	140	—	4	420	92	145	5
36	180	380	—	75	88	126	150	—	4	440	95	150	6
38	190	400	—	78	92	132	155	—	5	460	98	155	6
40	200	420	—	80	97	138	165	—	5	480	102	160	6
44	220	460	—	88	106	145	180	—	5	540	115	180	6
48	240	500	—	95	114	155	195	—	5	580	122	190	6
52	260	540	—	102	123	165	206	—	6	620	132	206	7.5
56	280	580	—	108	132	175	224	—	6	670	140	224	7.5
60	300	620	—	109	140	185	236	—	7.5	710	150	236	7.5
64	320	670	—	112	155	200	258	—	7.5	750	155	250	9.5
68	340	710	—	118	165	212	272	—	7.5	800	165	265	9.5
72	360	750	—	125	170	224	290	—	7.5	850	180	280	9.5
76	380	780	—	128	175	230	300	—	7.5	900	190	300	9.5
80	400	820	—	136	185	243	308	—	7.5	950	200	315	12
84	420	850	—	136	190	250	315	—	9.5	980	206	325	12
88	440	900	—	145	200	265	345	—	9.5	1030	212	335	12
92	460	950	—	155	212	280	365	—	9.5	1060	218	345	12
96	480	980	—	160	218	290	375	—	9.5	1120	230	365	15
/500	500	1030	—	170	230	300	388	—	12	1150	236	375	15
/530	530	1090	—	180	243	325	412	—	12	1220	250	400	15
/560	560	1150	—	190	258	335	438	—	12	1280	258	412	15
/600	600	1220	—	200	272	355	462	—	15	1360	272	438	15
/630	630	1280	—	206	280	375	488	—	15	1420	280	450	15
/670	670	1360	—	218	300	400	515	—	15	1500	290	475	15
/710	710	1420	—	224	308	412	530	—	15	—	—	—	—
/750	750	1500	—	236	325	438	560	—	15	—	—	—	—
/800	800	1600	—	258	355	462	600	—	15	—	—	—	—
/850	850	1700	—	272	375	488	630	—	19	—	—	—	—
/900	900	1780	—	280	388	500	650	—	19	—	—	—	—
/950	950	1850	—	290	400	515	670	—	19	—	—	—	—
/1000	1000	1950	—	300	412	545	710	—	19	—	—	—	—
/1060	1060	—	—	—	—	—	—	—	—	—	—	—	—
/1120	1120	—	—	—	—	—	—	—	—	—	—	—	—
/1180	1180	—	—	—	—	—	—	—	—	—	—	—	—
/1250	1250	—	—	—	—	—	—	—	—	—	—	—	—
/1320	1320	—	—	—	—	—	—	—	—	—	—	—	—
/1400	1400	—	—	—	—	—	—	—	—	—	—	—	—
/1500	1500	—	—	—	—	—	—	—	—	—	—	—	—

Remarks The chamfer dimensions listed in this table do not necessarily apply to the following:
(a) Chamfers of grooves in outer rings with snap ring grooves
(b) Chamfers on a side without ribs in thin-section cylindrical roller bearings
(c) Chamfers on the front-facing side in angular contact ball bearings
(d) Chamfers on inner rings in bearings with tapered bores

BOUNDARY DIMENSIONS AND BEARING DESIGNATIONS

Table 6. 2 Boundary Dimensions of Tapered Roller Bearings

Units: mm

Tapered Roller Bearings		329									320 X				330					331					
		Diameter Series 9									Diameter Series 0									Diameter Series 1					
			Dimension Series 29						Chamfer Dimension			Dimension Series 20			Dimension Series 30			Chamfer Dimension			Dimension Series 31			Chamfer Dimension	
			I			II			I.R.	O.R.								I.R.	O.R.					I.R.	O.R.
Bore Number	d	D	B	C	T	B	C	T	r (min.)	r (min.)	D	B	C	T	B	C	T	r (min.)	r (min.)	D	B	C	T	r (min.)	r (min.)
00	10	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
01	12	—	—	—	—	—	—	—	—	—	28	11	—	11	13	—	13	0.3	0.3	—	—	—	—	—	—
02	15	—	—	—	—	—	—	—	—	—	32	12	—	12	14	—	14	0.3	0.3	—	—	—	—	—	—
03	17	—	—	—	—	—	—	—	—	—	35	13	—	13	15	—	15	0.3	0.3	—	—	—	—	—	—
04	20	37	11	—	11.6	12	9	12	0.3	0.3	42	15	12	15	17	—	17	0.6	0.6	—	—	—	—	—	—
/22	22	40	—	—	—	12	9	12	0.3	0.3	44	15	11.5	15	—	—	—	0.6	0.6	—	—	—	—	—	—
05	25	42	11	—	11.6	12	9	12	0.3	0.3	47	15	11.5	15	17	14	17	0.6	0.6	—	—	—	—	—	—
/28	28	45	—	—	—	12	9	12	0.3	0.3	52	16	12	16	—	—	—	1	1	—	—	—	—	—	—
06	30	47	11	—	11.6	12	9	12	0.3	0.3	55	17	13	17	20	16	20	1	1	—	—	—	—	—	—
/32	32	52	—	—	—	15	10	14	0.6	0.6	58	17	13	17	—	—	—	1	1	—	—	—	—	—	—
07	35	55	13	—	14	14	11.5	14	0.6	0.6	62	18	14	18	21	17	21	1	1	—	—	—	—	—	—
08	40	62	14	—	15	15	12	15	0.6	0.6	68	19	14.5	19	22	18	22	1	1	75	26	20.5	26	1.5	1.5
09	45	68	14	—	15	15	12	15	0.6	0.6	75	20	15.5	20	24	19	24	1	1	80	26	20.5	26	1.5	1.5
10	50	72	14	—	15	15	12	15	0.6	0.6	80	20	15.5	20	24	19	24	1	1	85	26	20	26	1.5	1.5
11	55	80	16	—	17	17	14	17	1	1	90	23	17.5	23	27	21	27	1.5	1.5	95	30	23	30	1.5	1.5
12	60	85	16	—	17	17	14	17	1	1	95	23	17.5	23	27	21	27	1.5	1.5	100	30	23	30	1.5	1.5
13	65	90	16	—	17	17	14	17	1	1	100	23	17.5	23	27	21	27	1.5	1.5	110	34	26.5	34	1.5	1.5
14	70	100	19	—	20	20	16	20	1	1	110	25	19	25	31	25.5	31	1.5	1.5	120	37	29	37	2	1.5
15	75	105	19	—	20	20	16	20	1	1	115	25	19	25	31	25.5	31	1.5	1.5	125	37	29	37	2	1.5
16	80	110	19	—	20	20	16	20	1	1	125	29	22	29	36	29.5	36	1.5	1.5	130	37	29	37	2	1.5
17	85	120	22	—	23	23	18	23	1.5	1.5	130	29	22	29	36	29.5	36	1.5	1.5	140	41	32	41	2.5	2
18	90	125	22	—	23	23	18	23	1.5	1.5	140	32	24	32	39	32.5	39	2	1.5	150	45	35	45	2.5	2
19	95	130	22	—	23	23	18	23	1.5	1.5	145	32	24	32	39	32.5	39	2	1.5	160	49	38	49	2.5	2
20	100	140	24	—	25	25	20	25	1.5	1.5	150	32	24	32	39	32.5	39	2	1.5	165	52	40	52	2.5	2
21	105	145	24	—	25	25	20	25	1.5	1.5	160	35	26	35	43	34	43	2.5	2	175	56	44	56	2.5	2
22	110	150	24	—	25	25	20	25	1.5	1.5	170	38	29	38	47	37	47	2.5	2	180	56	43	56	2.5	2
24	120	165	27	—	29	29	23	29	1.5	1.5	180	38	29	38	48	38	48	2.5	2	200	62	48	62	2.5	2
26	130	180	30	—	32	32	25	32	2	1.5	200	45	34	45	55	43	55	2.5	2	—	—	—	—	—	—
28	140	190	30	—	32	32	25	32	2	1.5	210	45	34	45	56	44	56	2.5	2	—	—	—	—	—	—
30	150	210	36	—	38	38	30	38	2.5	2	225	48	36	48	59	46	59	3	2.5	—	—	—	—	—	—
32	160	220	36	—	38	38	30	38	2.5	2	240	51	38	51	—	—	—	3	2.5	—	—	—	—	—	—
34	170	230	36	—	38	38	30	38	2.5	2	260	57	43	57	—	—	—	3	2.5	—	—	—	—	—	—
36	180	250	42	—	45	45	34	45	2.5	2	280	64	48	64	—	—	—	3	2.5	—	—	—	—	—	—
38	190	260	42	—	45	45	34	45	2.5	2	290	64	48	64	—	—	—	3	2.5	—	—	—	—	—	—
40	200	280	48	—	51	51	39	51	3	2.5	310	70	53	70	—	—	—	3	2.5	—	—	—	—	—	—
44	220	300	48	—	51	51	39	51	3	2.5	340	76	57	76	—	—	—	4	3	—	—	—	—	—	—
48	240	320	48	—	51	51	39	51	3	2.5	360	76	57	76	—	—	—	4	3	—	—	—	—	—	—
52	260	360	—	—	—	63.5	48	63.5	3	2.5	400	87	65	87	—	—	—	5	4	—	—	—	—	—	—
56	280	380	—	—	—	63.5	48	63.5	3	2.5	420	87	65	87	—	—	—	5	4	—	—	—	—	—	—
60	300	420	—	—	—	76	57	76	4	3	460	100	74	100	—	—	—	5	4	—	—	—	—	—	—
64	320	440	—	—	—	76	57	76	4	3	480	100	74	100	—	—	—	5	4	—	—	—	—	—	—
68	340	460	—	—	—	76	57	76	4	3	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
72	360	480	—	—	—	76	57	76	4	3	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

Remarks 1. Other Series not conforming to this table are also specified by ISO.
2. In Diameter Series 9, Classification I refers to specifications of the old standard, while Classification II refers to those specified by ISO. Dimension Series without classifications conform to dimensions (D, B, C, T) specified by ISO.
3. The chamfer dimensions listed are the minimum permissible dimensions specified by ISO. They do not apply to chamfers on the front face.

302				322			332					303 or 303D					313			323					Tapered Roller Bearings	
Diameter Series 2												Diameter Series 3														
	Dimension Series 02			Dimension Series 22			Dimension Series 32			Chamfer Dimension			Dimension Series 03				Dimension Series 13			Dimension Series 23			Chamfer Dimension			
										I.R.	O.R.												I.R.	O.R.		
D	B	C	T	B	C	T	B	C	T	r (min.)	r (min.)	D	B	C	C (1)	T	B	C	T	B	C	T	r (min.)	r (min.)	d	Bore Number
30	9	—	9.7	14	—	14.7	—	—	—	0.6	0.6	35	11	—	—	11.9	—	—	—	17	—	17.9	0.6	0.6	10	00
32	10	9	10.75	14	—	14.75	—	—	—	0.6	0.6	37	12	—	—	12.9	—	—	—	17	—	17.9	1	1	12	01
35	11	10	11.75	14	—	14.75	—	—	—	0.6	0.6	42	13	11	—	14.25	—	—	—	17	14	18.25	1	1	15	02
40	12	11	13.25	16	14	17.25	—	—	—	1	1	47	14	12	—	15.25	—	—	—	19	16	20.25	1	1	17	03
47	14	12	15.25	18	15	19.25	—	—	—	1	1	52	15	13	—	16.25	—	—	—	21	18	22.25	1.5	1.5	20	04
50	14	12	15.25	18	15	19.25	—	—	—	1	1	56	16	14	—	17.25	—	—	—	21	18	22.25	1.5	1.5	22	/22
52	15	13	16.25	18	15	19.25	22	18	22	1	1	62	17	15	13	18.25	—	—	—	24	20	25.25	1.5	1.5	25	05
58	16	14	17.25	19	16	20.25	24	19	24	1	1	68	18	15	14	19.75	—	—	—	24	20	25.75	1.5	1.5	28	/28
62	16	14	17.25	20	17	21.25	25	19.5	25	1	1	72	19	16	14	20.75	—	—	—	27	23	28.75	1.5	1.5	30	06
65	17	15	18.25	21	18	22.25	26	20.5	26	1	1	75	20	17	15	21.75	—	—	—	28	24	29.75	1.5	1.5	32	/32
72	17	15	18.25	23	19	24.25	28	22	28	1.5	1.5	80	21	18	15	22.75	—	—	—	31	25	32.75	2	1.5	35	07
80	18	16	19.75	23	19	24.75	32	25	32	1.5	1.5	90	23	20	17	25.25	—	—	—	33	27	35.25	2	1.5	40	08
85	19	16	20.75	23	19	24.75	32	25	32	1.5	1.5	100	25	22	18	27.25	—	—	—	36	30	38.25	2	1.5	45	09
90	20	17	21.75	23	19	24.75	32	24.5	32	1.5	1.5	110	27	23	19	29.25	—	—	—	40	33	42.25	2.5	2	50	10
100	21	18	22.75	25	21	26.75	35	27	35	2	1.5	120	29	25	21	31.5	—	—	—	43	35	45.5	2.5	2	55	11
110	22	19	23.75	28	24	29.75	38	29	38	2	1.5	130	31	26	22	33.5	—	—	—	46	37	48.5	3	2.5	60	12
120	23	20	24.75	31	27	32.75	41	32	41	2	1.5	140	33	28	23	36	—	—	—	48	39	51	3	2.5	65	13
125	24	21	26.25	31	27	33.25	41	32	41	2	1.5	150	35	30	25	38	—	—	—	51	42	54	3	2.5	70	14
130	25	22	27.25	31	27	33.25	41	31	41	2	1.5	160	37	31	26	40	—	—	—	55	45	58	3	2.5	75	15
140	26	22	28.25	33	28	35.25	46	35	46	2.5	2	170	39	33	27	42.5	—	—	—	58	48	61.5	3	2.5	80	16
150	28	24	30.5	36	30	38.5	49	37	49	2.5	2	180	41	34	28	44.5	—	—	—	60	49	63.5	4	3	85	17
160	30	26	32.5	40	34	42.5	55	42	55	2.5	2	190	43	36	30	46.5	—	—	—	64	53	67.5	4	3	90	18
170	32	27	34.5	43	37	45.5	58	44	58	3	2.5	200	45	38	32	49.5	—	—	—	67	55	71.5	4	3	95	19
180	34	29	37	46	39	49	63	48	63	3	2.5	215	47	39	—	51.5	51	35	56.5	73	60	77.5	4	3	100	20
190	36	30	39	50	43	53	68	52	68	3	2.5	225	49	41	—	53.5	53	36	58	77	63	81.5	4	3	105	21
200	38	32	41	53	46	56	—	—	—	3	2.5	240	50	42	—	54.5	57	38	63	80	65	84.5	4	3	110	22
215	40	34	43.5	58	50	61.5	—	—	—	3	2.5	260	55	46	—	59.5	62	42	68	86	69	90.5	4	3	120	24
230	40	34	43.75	64	54	67.75	—	—	—	4	3	280	58	49	—	63.75	66	44	72	93	78	98.75	5	4	130	26
250	42	36	45.75	68	58	71.75	—	—	—	4	3	300	62	53	—	67.75	70	47	77	102	85	107.75	5	4	140	28
270	45	38	49	73	60	77	—	—	—	4	3	320	65	55	—	72	75	50	82	108	90	114	5	4	150	30
290	48	40	52	80	67	84	—	—	—	4	3	340	68	58	—	75	79	—	87	114	95	121	5	4	160	32
310	52	43	57	86	71	91	—	—	—	5	4	360	72	62	—	80	84	—	92	120	100	127	5	4	170	34
320	52	43	57	86	71	91	—	—	—	5	4	380	75	64	—	83	88	—	97	126	106	134	5	4	180	36
340	55	46	60	92	75	97	—	—	—	5	4	400	78	65	—	86	92	—	101	132	109	140	6	5	190	38
360	58	48	64	98	82	104	—	—	—	5	4	420	80	67	—	89	97	—	107	138	115	146	6	5	200	40
400	65	54	72	108	90	114	—	—	—	5	4	460	88	73	—	97	106	—	117	145	122	154	6	5	220	44
440	72	60	79	120	100	127	—	—	—	5	4	500	95	80	—	105	114	—	125	155	132	165	6	5	240	48
480	80	67	89	130	106	137	—	—	—	6	5	540	102	85	—	113	123	—	135	165	136	176	6	6	260	52
500	80	67	89	130	106	137	—	—	—	6	5	580	108	90	—	119	132	—	145	175	145	187	6	6	280	56
540	85	71	96	140	115	149	—	—	—	6	5	—	—	—	—	—	—	—	—	—	—	—	—	—	300	60
580	92	75	104	150	125	159	—	—	—	6	5	—	—	—	—	—	—	—	—	—	—	—	—	—	320	64
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	340	68
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	360	72

Note (1) Steep-slope bearing 303D in DIN corresponds to 313 in JIS. Series 13 bearings with bore diameters larger than 100 mm are designated as 313.

Table 6. 3 Boundary Dimensions of Thrust Bearings (Flat Seats) — (1)

Units: mm

Thrust Ball Brgs.										511					512		522			
Spherical Thrust Roller Brgs.														292						
		Diameter Series 0					Diameter Series 1					Diameter Series 2								
			Dimension Series					Dimension Series					Dimension Series							
			70	90	10			71	91	11			72	92	12	22	22			
																	Central Washer			
Bore Number	d	D	T			r (min.)	D	T			r (min.)	D	T				d_2	B	r (min.)	r_1 (min.)
4	4	12	4	—	6	0.3	—	—	—	—	—	16	6	—	8	—	—	—	0.3	—
6	6	16	5	—	7	0.3	—	—	—	—	—	20	6	—	9	—	—	—	0.3	—
8	8	18	5	—	7	0.3	—	—	—	—	—	22	6	—	9	—	—	—	0.3	—
00	10	20	5	—	7	0.3	24	6	—	9	0.3	26	7	—	11	—	—	—	0.6	—
01	12	22	5	—	7	0.3	26	6	—	9	0.3	28	7	—	11	—	—	—	0.6	—
02	15	26	5	—	7	0.3	28	6	—	9	0.3	32	8	—	12	22	10	5	0.6	0.3
03	17	28	5	—	7	0.3	30	6	—	9	0.3	35	8	—	12	—	—	—	0.6	—
04	20	32	6	—	8	0.3	35	7	—	10	0.3	40	9	—	14	26	15	6	0.6	0.3
05	25	37	6	—	8	0.3	42	8	—	11	0.6	47	10	—	15	28	20	7	0.6	0.3
06	30	42	6	—	8	0.3	47	8	—	11	0.6	52	10	—	16	29	25	7	0.6	0.3
07	35	47	6	—	8	0.3	52	8	—	12	0.6	62	12	—	18	34	30	8	1	0.3
08	40	52	6	—	9	0.3	60	9	—	13	0.6	68	13	—	19	36	30	9	1	0.6
09	45	60	7	—	10	0.3	65	9	—	14	0.6	73	13	—	20	37	35	9	1	0.6
10	50	65	7	—	10	0.3	70	9	—	14	0.6	78	13	—	22	39	40	9	1	0.6
11	55	70	7	—	10	0.3	78	10	—	16	0.6	90	16	21	25	45	45	10	1	0.6
12	60	75	7	—	10	0.3	85	11	—	17	1	95	16	21	26	46	50	10	1	0.6
13	65	80	7	—	10	0.3	90	11	—	18	1	100	16	21	27	47	55	10	1	0.6
14	70	85	7	—	10	0.3	95	11	—	18	1	105	16	21	27	47	55	10	1	1
15	75	90	7	—	10	0.3	100	11	—	19	1	110	16	21	27	47	60	10	1	1
16	80	95	7	—	10	0.3	105	11	—	19	1	115	16	21	28	48	65	10	1	1
17	85	100	7	—	10	0.3	110	11	—	19	1	125	18	24	31	55	70	12	1	1
18	90	105	7	—	10	0.3	120	14	—	22	1	135	20	27	35	62	75	14	1.1	1
20	100	120	9	—	14	0.6	135	16	21	25	1	150	23	30	38	67	85	15	1.1	1
22	110	130	9	—	14	0.6	145	16	21	25	1	160	23	30	38	67	95	15	1.1	1
24	120	140	9	—	14	0.6	155	16	21	25	1	170	23	30	39	68	100	15	1.1	1.1
26	130	150	9	—	14	0.6	170	18	24	30	1	190	27	36	45	80	110	18	1.5	1.1
28	140	160	9	—	14	0.6	180	18	24	31	1	200	27	36	46	81	120	18	1.5	1.1
30	150	170	9	—	14	0.6	190	18	24	31	1	215	29	39	50	89	130	20	1.5	1.1
32	160	180	9	—	14	0.6	200	18	24	31	1	225	29	39	51	90	140	20	1.5	1.1
34	170	190	9	—	14	0.6	215	20	27	34	1.1	240	32	42	55	97	150	21	1.5	1.1
36	180	200	9	—	14	0.6	225	20	27	34	1.1	250	32	42	56	98	150	21	1.5	2
38	190	215	11	—	17	1	240	23	30	37	1.1	270	36	48	62	109	160	24	2	2
40	200	225	11	—	17	1	250	23	30	37	1.1	280	36	48	62	109	170	24	2	2
44	220	250	14	—	22	1	270	23	30	37	1.1	300	36	48	63	110	190	24	2	2
48	240	270	14	—	22	1	300	27	36	45	1.5	340	45	60	78	—	—	—	2.1	—
52	260	290	14	—	22	1	320	27	36	45	1.5	360	45	60	79	—	—	—	2.1	—
56	280	310	14	—	22	1	350	32	42	53	1.5	380	45	60	80	—	—	—	2.1	—
60	300	340	18	24	30	1	380	36	48	62	2	420	54	73	95	—	—	—	3	—
64	320	360	18	24	30	1	400	36	48	63	2	440	54	73	95	—	—	—	3	—

Remarks
1. Dimension Series 22, 23, and 24 are double-direction bearings.
2. The maximum permissible outside diameter of shaft and central washers and minimum permissible bore diameter of housing washers are omitted here (refer to the bearing tables for thrust bearings).

			513	523								514	524								Thrust Ball Brgs.	
		293									294										Spherical Thrust Roller Brgs.	
Diameter Series 3									Diameter Series 4									Diameter Series 5				
	Dimension Series									Dimension Series									Dimension Series			
	73	93	13	23	23					74	94	14	24	24					95			
					Central Washer									Central Washer								
D	T				d_2	B	r (min.)	r_1 (min.)	D	T				d_2	B	r (min.)	r_1 (min.)	D	T	r (min.)	d	Bore Number
20	7	—	11	—	—	—	0.6	—	—	—	—	—	—	—	—	—	—	—	—	—	4	4
24	8	—	12	—	—	—	0.6	—	—	—	—	—	—	—	—	—	—	—	—	—	6	6
26	8	—	12	—	—	—	0.6	—	—	—	—	—	—	—	—	—	—	—	—	—	8	8
30	9	—	14	—	—	—	0.6	—	—	—	—	—	—	—	—	—	—	—	—	—	10	00
32	9	—	14	—	—	—	0.6	—	—	—	—	—	—	—	—	—	—	—	—	—	12	01
37	10	—	15	—	—	—	0.6	—	—	—	—	—	—	—	—	—	—	—	—	—	15	02
40	10	—	16	—	—	—	0.6	—	—	—	—	—	—	—	—	—	—	52	21	1	17	03
47	12	—	18	—	—	—	1	—	—	—	—	—	—	—	—	—	—	60	24	1	20	04
52	12	—	18	34	20	8	1	0.3	60	16	21	24	45	15	11	1	0.6	73	29	1.1	25	05
60	14	—	21	38	25	9	1	0.3	70	18	24	28	52	20	12	1	0.6	85	34	1.1	30	06
68	15	—	24	44	30	10	1	0.3	80	20	27	32	59	25	14	1.1	0.6	100	39	1.1	35	07
78	17	22	26	49	30	12	1	0.6	90	23	30	36	65	30	15	1.1	0.6	110	42	1.5	40	08
85	18	24	28	52	35	12	1	0.6	100	25	34	39	72	35	17	1.1	0.6	120	45	2	45	09
95	20	27	31	58	40	14	1.1	0.6	110	27	36	43	78	40	18	1.5	0.6	135	51	2	50	10
105	23	30	35	64	45	15	1.1	0.6	120	29	39	48	87	45	20	1.5	0.6	150	58	2.1	55	11
110	23	30	35	64	50	15	1.1	0.6	130	32	42	51	93	50	21	1.5	0.6	160	60	2.1	60	12
115	23	30	36	65	55	15	1.1	0.6	140	34	45	56	101	50	23	2	1	170	63	2.1	65	13
125	25	34	40	72	55	16	1.1	1	150	36	48	60	107	55	24	2	1	180	67	3	70	14
135	27	36	44	79	60	18	1.5	1	160	38	51	65	115	60	26	2	1	190	69	3	75	15
140	27	36	44	79	65	18	1.5	1	170	41	54	68	120	65	27	2.1	1	200	73	3	80	16
150	29	39	49	87	70	19	1.5	1	180	42	58	72	128	65	29	2.1	1.1	215	78	4	85	17
155	29	39	50	88	75	19	1.5	1	190	45	60	77	135	70	30	2.1	1.1	225	82	4	90	18
170	32	42	55	97	85	21	1.5	1	210	50	67	85	150	80	33	3	1.1	250	90	4	100	20
190	36	48	63	110	95	24	2	1	230	54	73	95	166	90	37	3	1.1	270	95	5	110	22
210	41	54	70	123	100	27	2.1	1.1	250	58	78	102	177	95	40	4	1.5	300	109	5	120	24
225	42	58	75	130	110	30	2.1	1.1	270	63	85	110	192	100	42	4	2	320	115	5	130	26
240	45	60	80	140	120	31	2.1	1.1	280	63	85	112	196	110	44	4	2	340	122	5	140	28
250	45	60	80	140	130	31	2.1	1.1	300	67	90	120	209	120	46	4	2	360	125	6	150	30
270	50	67	87	153	140	33	3	1.1	320	73	95	130	226	130	50	5	2	380	132	6	160	32
280	50	67	87	153	150	33	3	1.1	340	78	103	135	236	135	50	5	2.1	400	140	6	170	34
300	54	73	95	165	150	37	3	2	360	82	109	140	245	140	52	5	3	420	145	6	180	36
320	58	78	105	183	160	40	4	2	380	85	115	150	—	—	—	5	—	440	150	6	190	38
340	63	85	110	192	170	42	4	2	400	90	122	155	—	—	—	5	—	460	155	7.5	200	40
360	63	85	112	—	—	—	4	—	420	90	122	160	—	—	—	6	—	500	170	7.5	220	44
380	63	85	112	—	—	—	4	—	440	90	122	160	—	—	—	6	—	540	180	7.5	240	48
420	73	95	130	—	—	—	5	—	480	100	132	175	—	—	—	6	—	580	190	9.5	260	52
440	73	95	130	—	—	—	5	—	520	109	145	190	—	—	—	6	—	620	206	9.5	280	56
480	82	109	140	—	—	—	5	—	540	109	145	190	—	—	—	6	—	670	224	9.5	300	60
500	82	109	140	—	—	—	5	—	580	118	155	205	—	—	—	7.5	—	710	236	9.5	320	64

Table 6. 3 Boundary Dimensions of Thrust Bearings (Flat Seats) — (2)

Units: mm

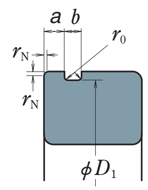
Bore Number	d	Diameter Series 0 D	Dimension Series 70 T	Dimension Series 90 T	Dimension Series 10 T	r (min.)	Diameter Series 1 D	Dimension Series 71 T	Dimension Series 91 T	Dimension Series 11 T	r (min.)	Diameter Series 2 D	Dimension Series 72 T	Dimension Series 92 T	Dimension Series 12 T	Dimension Series 22 T	Dimension Series 22 Central Washer d_2	Dimension Series 22 Central Washer B	r (min.)	r_1 (min.)
Thrust Ball Brgs.										511					512	522				
Spherical Thrust Roller Brgs.														292						
68	340	380	18	24	30	1	420	36	48	64	2	460	54	73	96	—	—	—	3	—
72	360	400	18	24	30	1	440	36	48	65	2	500	63	85	110	—	—	—	4	—
76	380	420	18	24	30	1	460	36	48	65	2	520	63	85	112	—	—	—	4	—
80	400	440	18	24	30	1	480	36	48	65	2	540	63	85	112	—	—	—	4	—
84	420	460	18	24	30	1	500	36	48	65	2	580	73	95	130	—	—	—	5	—
88	440	480	18	24	30	1	540	45	60	80	2.1	600	73	95	130	—	—	—	5	—
92	460	500	18	24	30	1	560	45	60	80	2.1	620	73	95	130	—	—	—	5	—
96	480	520	18	24	30	1	580	45	60	80	2.1	650	78	103	135	—	—	—	5	—
/500	500	540	18	24	30	1	600	45	60	80	2.1	670	78	103	135	—	—	—	5	—
/530	530	580	23	30	38	1.1	640	50	67	85	3	710	82	109	140	—	—	—	5	—
/560	560	610	23	30	38	1.1	670	50	67	85	3	750	85	115	150	—	—	—	5	—
/600	600	650	23	30	38	1.1	710	50	67	85	3	800	90	122	160	—	—	—	5	—
/630	630	680	23	30	38	1.1	750	54	73	95	3	850	100	132	175	—	—	—	6	—
/670	670	730	27	36	45	1.5	800	58	78	105	4	900	103	140	180	—	—	—	6	—
/710	710	780	32	42	53	1.5	850	63	85	112	4	950	109	145	190	—	—	—	6	—
/750	750	820	32	42	53	1.5	900	67	90	120	4	1000	112	150	195	—	—	—	6	—
/800	800	870	32	42	53	1.5	950	67	90	120	4	1060	118	155	205	—	—	—	7.5	—
/850	850	920	32	42	53	1.5	1000	67	90	120	4	1120	122	160	212	—	—	—	7.5	—
/900	900	980	36	48	63	2	1060	73	95	130	5	1180	125	170	220	—	—	—	7.5	—
/950	950	1030	36	48	63	2	1120	78	103	135	5	1250	136	180	236	—	—	—	7.5	—
/1000	1000	1090	41	54	70	2.1	1180	82	109	140	5	1320	145	190	250	—	—	—	9.5	—
/1060	1060	1150	41	54	70	2.1	1250	85	115	150	5	1400	155	206	265	—	—	—	9.5	—
/1120	1120	1220	45	60	80	2.1	1320	90	122	160	5	1460	—	206	—	—	—	—	9.5	—
/1180	1180	1280	45	60	80	2.1	1400	100	132	175	6	1520	—	206	—	—	—	—	9.5	—
/1250	1250	1360	50	67	85	3	1460	—	—	175	6	1610	—	216	—	—	—	—	9.5	—
/1320	1320	1440	—	—	95	3	1540	—	—	175	6	1700	—	228	—	—	—	—	9.5	—
/1400	1400	1520	—	—	95	3	1630	—	—	180	6	1790	—	234	—	—	—	—	12	—
/1500	1500	1630	—	—	105	4	1750	—	—	195	6	1920	—	252	—	—	—	—	12	—
/1600	1600	1730	—	—	105	4	1850	—	—	195	6	2040	—	264	—	—	—	—	15	—
/1700	1700	1840	—	—	112	4	1970	—	—	212	7.5	2160	—	276	—	—	—	—	15	—
/1800	1800	1950	—	—	120	4	2080	—	—	220	7.5	2280	—	288	—	—	—	—	15	—
/1900	1900	2060	—	—	130	5	2180	—	—	220	7.5	—	—	—	—	—	—	—	—	—
/2000	2000	2160	—	—	130	5	2300	—	—	236	7.5	—	—	—	—	—	—	—	—	—
/2120	2120	2300	—	—	140	5	2430	—	—	243	7.5	—	—	—	—	—	—	—	—	—
/2240	2240	2430	—	—	150	5	2570	—	—	258	9.5	—	—	—	—	—	—	—	—	—
/2360	2360	2550	—	—	150	5	2700	—	—	265	9.5	—	—	—	—	—	—	—	—	—
/2500	2500	2700	—	—	160	5	2850	—	—	272	9.5	—	—	—	—	—	—	—	—	—

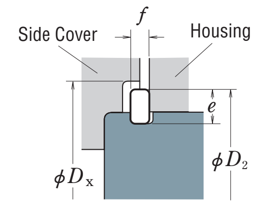
Diameter Series 3 D	Dimension Series 73 T	Dimension Series 93 T	Dimension Series 13 T	Dimension Series 23 T	Dimension Series 23 Central Washer d_2	Dimension Series 23 Central Washer B	r (min.)	r_1 (min.)	Diameter Series 4 D	Dimension Series 74 T	Dimension Series 94 T	Dimension Series 14 T	Dimension Series 24 T	Dimension Series 24 Central Washer d_2	Dimension Series 24 Central Washer B	r (min.)	r_1 (min.)	Diameter Series 5 D	Dimension Series 95 T	r (min.)	d	Bore Number
			513	523								514	524									Thrust Ball Brgs.
		293									294											Spherical Thrust Roller Brgs.
540	90	122	160	—	—	—	5	—	620	125	170	220	—	—	—	7.5	—	750	243	12	340	68
560	90	122	160	—	—	—	5	—	640	125	170	220	—	—	—	7.5	—	780	250	12	360	72
600	100	132	175	—	—	—	6	—	670	132	175	224	—	—	—	7.5	—	820	265	12	380	76
620	100	132	175	—	—	—	6	—	710	140	185	243	—	—	—	7.5	—	850	272	12	400	80
650	103	140	180	—	—	—	6	—	730	140	185	243	—	—	—	7.5	—	900	290	15	420	84
680	109	145	190	—	—	—	6	—	780	155	206	265	—	—	—	9.5	—	950	308	15	440	88
710	112	150	195	—	—	—	6	—	800	155	206	265	—	—	—	9.5	—	980	315	15	460	92
730	112	150	195	—	—	—	6	—	850	165	224	290	—	—	—	9.5	—	1000	315	15	480	96
750	112	150	195	—	—	—	6	—	870	165	224	290	—	—	—	9.5	—	1060	335	15	500	/500
800	122	160	212	—	—	—	7.5	—	920	175	236	308	—	—	—	9.5	—	1090	335	15	530	/530
850	132	175	224	—	—	—	7.5	—	980	190	250	335	—	—	—	12	—	1150	355	15	560	/560
900	136	180	236	—	—	—	7.5	—	1030	195	258	335	—	—	—	12	—	1220	375	15	600	/600
950	145	190	250	—	—	—	9.5	—	1090	206	280	365	—	—	—	12	—	1280	388	15	630	/630
1000	150	200	258	—	—	—	9.5	—	1150	218	290	375	—	—	—	15	—	1320	388	15	670	/670
1060	160	212	272	—	—	—	9.5	—	1220	230	308	400	—	—	—	15	—	1400	412	15	710	/710
1120	165	224	290	—	—	—	9.5	—	1280	236	315	412	—	—	—	15	—	—	—	—	750	/750
1180	170	230	300	—	—	—	9.5	—	1360	250	335	438	—	—	—	15	—	—	—	—	800	/800
1250	180	243	315	—	—	—	12	—	1440	—	354	—	—	—	—	15	—	—	—	—	850	/850
1320	190	250	335	—	—	—	12	—	1520	—	372	—	—	—	—	15	—	—	—	—	900	/900
1400	200	272	355	—	—	—	12	—	1600	—	390	—	—	—	—	15	—	—	—	—	950	/950
1460	—	276	—	—	—	—	12	—	1670	—	402	—	—	—	—	15	—	—	—	—	1000	/1000
1540	—	288	—	—	—	—	15	—	1770	—	426	—	—	—	—	15	—	—	—	—	1060	/1060
1630	—	306	—	—	—	—	15	—	1860	—	444	—	—	—	—	15	—	—	—	—	1120	/1120
1710	—	318	—	—	—	—	15	—	1950	—	462	—	—	—	—	19	—	—	—	—	1180	/1180
1800	—	330	—	—	—	—	19	—	2050	—	480	—	—	—	—	19	—	—	—	—	1250	/1250
1900	—	348	—	—	—	—	19	—	2160	—	505	—	—	—	—	19	—	—	—	—	1320	/1320
2000	—	360	—	—	—	—	19	—	2280	—	530	—	—	—	—	19	—	—	—	—	1400	/1400
2140	—	384	—	—	—	—	19	—	—	—	—	—	—	—	—	—	—	—	—	—	1500	/1500
2270	—	402	—	—	—	—	19	—	—	—	—	—	—	—	—	—	—	—	—	—	1600	/1600
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	1700	/1700
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	1800	/1800
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	1900	/1900
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	2000	/2000
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	2120	/2120
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	2240	/2240
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	2360	/2360
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	2500	/2500

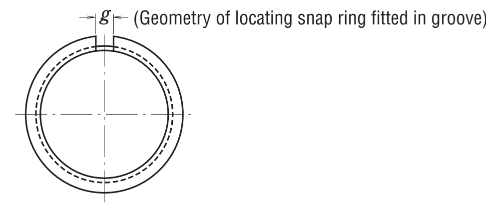
Remarks
1. Dimension Series 22, 23, and 24 are double-direction bearings.
2. The maximum permissible outside diameter of shaft and central washers and minimum permissible bore diameter of housing washers are omitted here (refer to the bearings tables for thrust bearings).

BOUNDARY DIMENSIONS AND BEARING DESIGNATIONS

Table 6. 4 Dimensions of Snap Ring Grooves and Locating Snap Rings — (1)
Bearings of Dimension Series 18 and 19







Units: mm

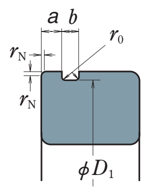
Applicable Bearings			Snap Ring Groove								
d		D	Snap Ring Groove Diameter D_1		Snap Ring Groove Position a				Snap Ring Groove Width b		Radius of Bottom Corners r_0
Dimension Series					Bearing Dimension Series 18		Bearing Dimension Series 19				
18	19		max.	min.	max.	min.	max.	min.	max.	min.	max.
—	10	**22**	20.8	20.5	—	—	1.05	0.9	1.05	0.8	0.2
—	12	**24**	22.8	22.5	—	—	1.05	0.9	1.05	0.8	0.2
—	15	**28**	26.7	26.4	—	—	1.3	1.15	1.2	0.95	0.25
—	17	**30**	28.7	28.4	—	—	1.3	1.15	1.2	0.95	0.25
20	—	**32**	30.7	30.4	1.3	1.15	—	—	1.2	0.95	0.25
22	—	**34**	32.7	32.4	1.3	1.15	—	—	1.2	0.95	0.25
25	20	**37**	35.7	35.4	1.3	1.15	1.7	1.55	1.2	0.95	0.25
—	22	**39**	37.7	37.4	—	—	1.7	1.55	1.2	0.95	0.25
28	—	**40**	38.7	38.4	1.3	1.15	—	—	1.2	0.95	0.25
30	25	**42**	40.7	40.4	1.3	1.15	1.7	1.55	1.2	0.95	0.25
32	—	**44**	42.7	42.4	1.3	1.15	—	—	1.2	0.95	0.25
—	28	**45**	43.7	43.4	—	—	1.7	1.55	1.2	0.95	0.25
35	30	**47**	45.7	45.4	1.3	1.15	1.7	1.55	1.2	0.95	0.25
40	32	**52**	50.7	50.4	1.3	1.15	1.7	1.55	1.2	0.95	0.25
—	35	**55**	53.7	53.4	—	—	1.7	1.55	1.2	0.95	0.25
45	—	**58**	56.7	56.4	1.3	1.15	—	—	1.2	0.95	0.25
—	40	**62**	60.7	60.3	—	—	1.7	1.55	1.2	0.95	0.25
50	—	**65**	63.7	63.3	1.3	1.15	—	—	1.2	0.95	0.25
—	45	**68**	66.7	66.3	—	—	1.7	1.55	1.2	0.95	0.25
55	50	**72**	70.7	70.3	1.7	1.55	1.7	1.55	1.2	0.95	0.25
60	—	**78**	76.2	75.8	1.7	1.55	—	—	1.6	1.3	0.4
—	55	**80**	77.9	77.5	—	—	2.1	1.9	1.6	1.3	0.4
65	60	**85**	82.9	82.5	1.7	1.55	2.1	1.9	1.6	1.3	0.4
70	65	**90**	87.9	87.5	1.7	1.55	2.1	1.9	1.6	1.3	0.4
75	—	**95**	92.9	92.5	1.7	1.55	—	—	1.6	1.3	0.4
80	70	**100**	97.9	97.5	1.7	1.55	2.5	2.3	1.6	1.3	0.4
—	75	**105**	102.6	102.1	—	—	2.5	2.3	1.6	1.3	0.4
85	80	**110**	107.6	107.1	2.1	1.9	2.5	2.3	1.6	1.3	0.4
90	—	**115**	112.6	112.1	2.1	1.9	—	—	1.6	1.3	0.4
95	85	**120**	117.6	117.1	2.1	1.9	3.3	3.1	1.6	1.3	0.4
100	90	**125**	122.6	122.1	2.1	1.9	3.3	3.1	1.6	1.3	0.4
105	95	**130**	127.6	127.1	2.1	1.9	3.3	3.1	1.6	1.3	0.4
110	100	**140**	137.6	137.1	2.5	2.3	3.3	3.1	2.2	1.9	0.6
—	105	**145**	142.6	142.1	—	—	3.3	3.1	2.2	1.9	0.6
120	110	**150**	147.6	147.1	2.5	2.3	3.3	3.1	2.2	1.9	0.6
130	120	**165**	161.8	161.3	3.3	3.1	3.7	3.5	2.2	1.9	0.6
140	—	**175**	171.8	171.3	3.3	3.1	—	—	2.2	1.9	0.6
—	130	**180**	176.8	176.3	—	—	3.7	3.5	2.2	1.9	0.6
150	140	**190**	186.8	186.3	3.3	3.1	3.7	3.5	2.2	1.9	0.6
160	—	**200**	196.8	196.3	3.3	3.1	—	—	2.2	1.9	0.6

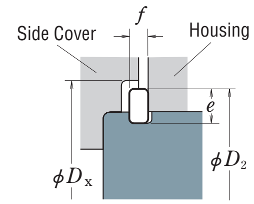
Remarks The minimum permissible chamfer dimensions r_N on the snap-ring-groove side of the outer rings are as follows:
Dimension series 18 : For outside diameters of 78 mm or less, use a 0.3 mm chamfer.
For all others exceeding 78 mm, use 0.5 mm chamfer.
Dimension series 19 : For outside diameters of 24 mm or less, use a 0.2 mm chamfer.
For 47mm or less, use a 0.3 mm chamfer.
For those exceeding 47 mm, use a 0.5 mm chamfer (However, for an outside diameter of 68 mm, use a 0.3 mm chamfer, though note this is not compliant with ISO 15).

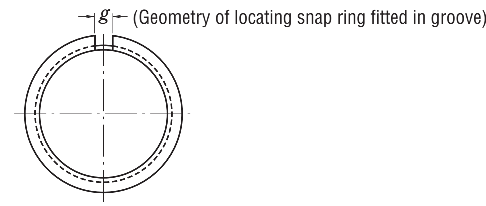
Locating Snap Ring							Side Cover
Locating Snap Ring Number	Cross Sectional Height e		Thickness f		Geometry of Snap Ring Fitted in Groove (Reference)		Stepped Bore Diameter (Reference) D_X
					Slit Width g	Snap Ring Outside Diameter D_2	
	max.	min.	max.	min.	approx.	max.	min.
NR 1022	2.0	1.85	0.7	0.6	2	24.8	25.5
NR 1024	2.0	1.85	0.7	0.6	2	26.8	27.5
NR 1028	2.05	1.9	0.85	0.75	3	30.8	31.5
NR 1030	2.05	1.9	0.85	0.75	3	32.8	33.5
NR 1032	2.05	1.9	0.85	0.75	3	34.8	35.5
NR 1034	2.05	1.9	0.85	0.75	3	36.8	37.5
NR 1037	2.05	1.9	0.85	0.75	3	39.8	40.5
NR 1039	2.05	1.9	0.85	0.75	3	41.8	42.5
NR 1040	2.05	1.9	0.85	0.75	3	42.8	43.5
NR 1042	2.05	1.9	0.85	0.75	3	44.8	45.5
NR 1044	2.05	1.9	0.85	0.75	4	46.8	47.5
NR 1045	2.05	1.9	0.85	0.75	4	47.8	48.5
NR 1047	2.05	1.9	0.85	0.75	4	49.8	50.5
NR 1052	2.05	1.9	0.85	0.75	4	54.8	55.5
NR 1055	2.05	1.9	0.85	0.75	4	57.8	58.5
NR 1058	2.05	1.9	0.85	0.75	4	60.8	61.5
NR 1062	2.05	1.9	0.85	0.75	4	64.8	65.5
NR 1065	2.05	1.9	0.85	0.75	4	67.8	68.5
NR 1068	2.05	1.9	0.85	0.75	5	70.8	72
NR 1072	2.05	1.9	0.85	0.75	5	74.8	76
NR 1078	3.25	3.1	1.12	1.02	5	82.7	84
NR 1080	3.25	3.1	1.12	1.02	5	84.4	86
NR 1085	3.25	3.1	1.12	1.02	5	89.4	91
NR 1090	3.25	3.1	1.12	1.02	5	94.4	96
NR 1095	3.25	3.1	1.12	1.02	5	99.4	101
NR 1100	3.25	3.1	1.12	1.02	5	104.4	106
NR 1105	4.04	3.89	1.12	1.02	5	110.7	112
NR 1110	4.04	3.89	1.12	1.02	5	115.7	117
NR 1115	4.04	3.89	1.12	1.02	5	120.7	122
NR 1120	4.04	3.89	1.12	1.02	7	125.7	127
NR 1125	4.04	3.89	1.12	1.02	7	130.7	132
NR 1130	4.04	3.89	1.12	1.02	7	135.7	137
NR 1140	4.04	3.89	1.7	1.6	7	145.7	147
NR 1145	4.04	3.89	1.7	1.6	7	150.7	152
NR 1150	4.04	3.89	1.7	1.6	7	155.7	157
NR 1165	4.85	4.7	1.7	1.6	7	171.5	173
NR 1175	4.85	4.7	1.7	1.6	10	181.5	183
NR 1180	4.85	4.7	1.7	1.6	10	186.5	188
NR 1190	4.85	4.7	1.7	1.6	10	196.5	198
NR 1200	4.85	4.7	1.7	1.6	10	206.5	208

BOUNDARY DIMENSIONS AND BEARING DESIGNATIONS

Table 6. 4 Dimensions of Snap Ring Grooves and Locating Snap Rings — (2)
Bearings of Diameter Series 0, 2, 3, and 4







Units: mm

Applicable Bearings					Snap Ring Groove								
d				D	Snap Ring Groove Diameter D_1		Snap Ring Groove Position a				Snap Ring Groove Width b		Radius of Bottom Corners r_0
Diameter Series							Bearing Diameter Series 0		Bearing Diameter Series 2, 3, 4				
0	2	3	4		max.	min.	max.	min.	max.	min.	max.	min.	max.
10	—	—	—	**26**	24.5	24.25	1.35	1.19	—	—	1.17	0.87	0.2
12	—	—	—	**28**	26.5	26.25	1.35	1.19	—	—	1.17	0.87	0.2
—	10	9	8	**30**	28.17	27.91	—	—	2.06	1.9	1.65	1.35	0.4
15	12	—	9	**32**	30.15	29.9	2.06	1.9	2.06	1.9	1.65	1.35	0.4
17	15	10	—	**35**	33.17	32.92	2.06	1.9	2.06	1.9	1.65	1.35	0.4
—	—	12	10	**37**	34.77	34.52	—	—	2.06	1.9	1.65	1.35	0.4
—	17	—	—	**40**	38.1	37.85	—	—	2.06	1.9	1.65	1.35	0.4
20	—	15	12	**42**	39.75	39.5	2.06	1.9	2.06	1.9	1.65	1.35	0.4
22	—	—	—	**44**	41.75	41.5	2.06	1.9	—	—	1.65	1.35	0.4
25	20	17	—	**47**	44.6	44.35	2.06	1.9	2.46	2.31	1.65	1.35	0.4
—	22	—	—	**50**	47.6	47.35	—	—	2.46	2.31	1.65	1.35	0.4
28	25	20	15	**52**	49.73	49.48	2.06	1.9	2.46	2.31	1.65	1.35	0.4
30	—	—	—	**55**	52.6	52.35	2.08	1.88	—	—	1.65	1.35	0.4
—	—	22	—	**56**	53.6	53.35	—	—	2.46	2.31	1.65	1.35	0.4
32	28	—	—	**58**	55.6	55.35	2.08	1.88	2.46	2.31	1.65	1.35	0.4
35	30	25	17	**62**	59.61	59.11	2.08	1.88	3.28	3.07	2.2	1.9	0.6
—	32	—	—	**65**	62.6	62.1	—	—	3.28	3.07	2.2	1.9	0.6
40	—	28	—	**68**	64.82	64.31	2.49	2.29	3.28	3.07	2.2	1.9	0.6
—	35	30	20	**72**	68.81	68.3	—	—	3.28	3.07	2.2	1.9	0.6
45	—	32	—	**75**	71.83	71.32	2.49	2.29	3.28	3.07	2.2	1.9	0.6
50	40	35	25	**80**	76.81	76.3	2.49	2.29	3.28	3.07	2.2	1.9	0.6
—	45	—	—	**85**	81.81	81.31	—	—	3.28	3.07	2.2	1.9	0.6
55	50	40	30	**90**	86.79	86.28	2.87	2.67	3.28	3.07	3	2.7	0.6
60	—	—	—	**95**	91.82	91.31	2.87	2.67	—	—	3	2.7	0.6
65	55	45	35	**100**	96.8	96.29	2.87	2.67	3.28	3.07	3	2.7	0.6
70	60	50	40	**110**	106.81	106.3	2.87	2.67	3.28	3.07	3	2.7	0.6
75	—	—	—	**115**	111.81	111.3	2.87	2.67	—	—	3	2.7	0.6
—	65	55	45	**120**	115.21	114.71	—	—	4.06	3.86	3.4	3.1	0.6
80	70	—	—	**125**	120.22	119.71	2.87	2.67	4.06	3.86	3.4	3.1	0.6
85	75	60	50	**130**	125.22	124.71	2.87	2.67	4.06	3.86	3.4	3.1	0.6
90	80	65	55	**140**	135.23	134.72	3.71	3.45	4.9	4.65	3.4	3.1	0.6
95	—	—	—	**145**	140.23	139.73	3.71	3.45	—	—	3.4	3.1	0.6
100	85	70	60	**150**	145.24	144.73	3.71	3.45	4.9	4.65	3.4	3.1	0.6
105	90	75	65	**160**	155.22	154.71	3.71	3.45	4.9	4.65	3.4	3.1	0.6
110	95	80	—	**170**	163.65	163.14	3.71	3.45	5.69	5.44	3.8	3.5	0.6
120	100	85	70	**180**	173.66	173.15	3.71	3.45	5.69	5.44	3.8	3.5	0.6
—	105	90	75	**190**	183.64	183.13	—	—	5.69	5.44	3.8	3.5	0.6
130	110	95	80	**200**	193.65	193.14	5.69	5.44	5.69	5.44	3.8	3.5	0.6

Locating Snap Ring							Side Cover
Locating Snap Ring Number	Cross Sectional Height e		Thickness f		Geometry of Snap Ring Fitted in Groove (Reference)		Stepped Bore Diameter (Reference) D_X
					Slit Width g	Snap Ring Outside Diameter D_2	
	max.	min.	max.	min.	approx.	max.	min.
NR 26 (1)	2.06	1.91	0.84	0.74	3	28.7	29.4
NR 28 (1)	2.06	1.91	0.84	0.74	3	30.7	31.4
NR 30	3.25	3.1	1.12	1.02	3	34.7	35.5
NR 32	3.25	3.1	1.12	1.02	3	36.7	37.5
NR 35	3.25	3.1	1.12	1.02	3	39.7	40.5
NR 37	3.25	3.1	1.12	1.02	3	41.3	42
NR 40	3.25	3.1	1.12	1.02	3	44.6	45.5
NR 42	3.25	3.1	1.12	1.02	3	46.3	47
NR 44	3.25	3.1	1.12	1.02	3	48.3	49
NR 47	4.04	3.89	1.12	1.02	4	52.7	53.5
NR 50	4.04	3.89	1.12	1.02	4	55.7	56.5
NR 52	4.04	3.89	1.12	1.02	4	57.9	58.5
NR 55	4.04	3.89	1.12	1.02	4	60.7	61.5
NR 56	4.04	3.89	1.12	1.02	4	61.7	62.5
NR 58	4.04	3.89	1.12	1.02	4	63.7	64.5
NR 62	4.04	3.89	1.7	1.6	4	67.7	68.5
NR 65	4.04	3.89	1.7	1.6	4	70.7	71.5
NR 68	4.85	4.7	1.7	1.6	5	74.6	76
NR 72	4.85	4.7	1.7	1.6	5	78.6	80
NR 75	4.85	4.7	1.7	1.6	5	81.6	83
NR 80	4.85	4.7	1.7	1.6	5	86.6	88
NR 85	4.85	4.7	1.7	1.6	5	91.6	93
NR 90	4.85	4.7	2.46	2.36	5	96.5	98
NR 95	4.85	4.7	2.46	2.36	5	101.6	103
NR 100	4.85	4.7	2.46	2.36	5	106.5	108
NR 110	4.85	4.7	2.46	2.36	5	116.6	118
NR 115	4.85	4.7	2.46	2.36	5	121.6	123
NR 120	7.21	7.06	2.82	2.72	7	129.7	131.5
NR 125	7.21	7.06	2.82	2.72	7	134.7	136.5
NR 130	7.21	7.06	2.82	2.72	7	139.7	141.5
NR 140	7.21	7.06	2.82	2.72	7	149.7	152
NR 145	7.21	7.06	2.82	2.72	7	154.7	157
NR 150	7.21	7.06	2.82	2.72	7	159.7	162
NR 160	7.21	7.06	2.82	2.72	7	169.7	172
NR 170	9.6	9.45	3.1	3	10	182.9	185
NR 180	9.6	9.45	3.1	3	10	192.9	195
NR 190	9.6	9.45	3.1	3	10	202.9	205
NR 200	9.6	9.45	3.1	3	10	212.9	215

Note (1) The locating snap rings and snap ring grooves of these bearings are not specified by ISO.

Remarks 1. The dimensions of these snap ring grooves are not applicable to bearings of Dimension Series 00, 82, and 83.
2. The minimum permissible chamfer dimension r_N on the snap-ring side of outer rings is 0.5 mm. However, for Diameter Series 0 bearings with outside diameters of 35 mm or below, it is 0.3 mm.

6.2 Formulation of Bearing Designations

Bearing designations (or "bearing numbers") are codes containing alphanumeric and non-alphanumeric characters that indicate bearing type, boundary dimensions, dimensional and running accuracies, internal clearance, and other related specifications. The boundary dimensions of commonly used bearings mostly conform to the organizational concept of ISO, and the bearing numbers of these standard bearings are specified by JIS B 1513 (*Rolling bearings-Designation*). Due to a need for more detailed classification, NSK uses auxiliary designations other than those specified by JIS.

Bearing designations consist of a basic designation and supplementary designation. The basic designation indicates the bearing series (type) and the width and diameter series as shown in Table 6.5. Basic designations, supplementary designations, and the meanings of common numbers and designations are listed in Table 6.6 (Pages A122 and A123). Contact angle and other supplementary designations are shown in successive columns from left to right in Table 6.6. For reference, some example designations are shown here:

(Example 1) 6 3 08 ZZ C3
- Radial Clearance **C3** (Internal Clearance Designation)
- Shields on Both Sides (Shield Designation)
- Nominal Bore Diameter **40 mm** (Bore Number)
- Diameter Series **3**
- Single-Row Deep Groove Ball Bearing

(Diameter Series 3 and Single-Row Deep Groove Ball Bearing: Bearing Series Designation)

(Example 2) 7 2 20 A DB C3
- Axial Clearance **C3**
- Back-to-Back Arrangement
- Contact Angle **30°**
- Nominal Bore Diameter **100 mm**
- Diameter Series **2**
- Single-Row Angular Contact Ball Bearing

(Example 3) 1 2 06 K +H206X
- Adapter with **25 mm** Bore
- Tapered Bore (Taper **1:12**)
- Nominal Bore Diameter **30 mm**
- Diameter Series **2**
- Self-Aligning Ball Bearing

(Example 4) NU 3 18 M CM
- Radial Clearance for Electric-Motor Bearings **CM**
- Machined Brass Cage
- Nominal Bore Diameter **90 mm**
- Diameter Series **3**
- **NU** Type Cylindrical Roller Bearing

(Example 5) NN 3 0 17 K CC1 P4
- Accuracy of ISO **Class 4**
- Radial Clearance in Non-Interchangeable Cylindrical Roller Bearings **CC1**
- Tapered Bore (Taper **1:12**)
- Nominal Bore Diameter **85 mm**
- Diameter Series **0**
- Width Series **3**
- **NN** Type Cylindrical Roller Bearing

(Example 6) HR 3 0 2 07 J
- Small Diameter of Outer Ring Raceway and Contact Angle Conform to ISO
- Nominal Bore Diameter **35 mm**
- Diameter Series **2**
- Width Series **0**
- Tapered Roller Bearing
- High Capacity Bearing

(Example 7) 2 4 0 /1000 CA M K30 E4 C3
- Radial Clearance **C3**
- Outer Ring with Oil Groove and Oil Holes
- Tapered Bore (Taper **1:30**)
- Machined Brass Cage
- Spherical Roller Bearing
- Nominal Bore Diameter **1 000 mm**
- Diameter Series **0**
- Width Series **4**
- Spherical Roller Bearing

(Example 8) 5 1 2 15
- Nominal Bore Diameter **75 mm**
- Diameter Series **2**
- Height Series **1**
- Thrust Ball Bearing

Table 6. 5 Bearing Series Designations

Bearing Type	Bearing Series	Type	Dimensions: Width	Dimensions: Diameter
Single-Row Deep Groove Ball Bearings	**68**	6	(1)	8
	69	6	(1)	9
	60	6	(1)	0
	62	6	(0)	2
	63	6	(0)	3
Single-Row Angular Contact Ball Bearings	**79**	7	(1)	9
	70	7	(1)	0
	72	7	(0)	2
	73	7	(0)	3
Self-Aligning Ball Bearings	**12**	1	(0)	2
	13	1	(0)	3
	22	(1)	2	2
	23	(1)	2	3
Single-Row Cylindrical Roller Bearings	**NU10**	NU	1	0
	NU2	NU	(0)	2
	NU22	NU	2	2
	NU3	NU	(0)	3
	NU23	NU	2	3
	NU4	NU	(0)	4
	NJ2	NJ	(0)	2
	NJ22	NJ	2	2
	NJ3	NJ	(0)	3
	NJ23	NJ	2	3
	NJ4	NJ	(0)	4
	NUP2	NUP	(0)	2
	NUP22	NUP	2	2
	NUP3	NUP	(0)	3
	NUP23	NUP	2	3
	NUP4	NUP	(0)	4
	N10	N	1	0
	N2	N	(0)	2
	N3	N	(0)	3
	N4	N	(0)	4
	NF2	NF	(0)	2
	NF3	NF	(0)	3
	NF4	NF	(0)	4

Bearing Type	Bearing Series	Type	Dimensions: Width or Height	Dimensions: Diameter
Double-Row Cylindrical Roller Bearings	**NNU49**	NNU	4	9
	NN30	NN	3	0
Needle Roller Bearings	**NA48**	NA	4	8
	NA49	NA	4	9
	NA59	NA	5	9
	NA69	NA	6	9
Tapered Roller Bearings	**329**	3	2	9
	320	3	2	0
	330	3	3	0
	331	3	3	1
	302	3	0	2
	322	3	2	2
	332	3	3	2
	303	3	0	3
	323	3	2	3
Spherical Roller Bearings	**230**	2	3	0
	231	2	3	1
	222	2	2	2
	232	2	3	2
	213 (1)	2	0	3
	223	2	2	3
Thrust Ball Bearings with Flat Seats	**511**	5	1	1
	512	5	1	2
	513	5	1	3
	514	5	1	4
	522	5	2	2
	523	5	2	3
	524	5	2	4
Spherical Thrust Roller Bearings	**292**	2	9	2
	293	2	9	3
	294	2	9	4

Note (1) Bearing Series 213 should logically be 203, but customarily it is numbered 213.
Remark Numbers in parentheses ( ) in the width column are usually omitted from the bearing designation.

Table 6. 6 Formulation of Bearing Designations

Basic Designation (Designations and Numbers Conform to JIS(5); Marked on Bearings)

Bearing Series (1)

Code	Meaning
68, 69, 60, :	Single-Row Deep Groove Ball Bearings
70, 72, 73, :	Single-Row Angular Contact Ball Bearings
12, 13, 22, :	Self-Aligning Ball Bearings
NU10, NJ 2, N 3, NN 30, :	Cylindrical Roller Bearings
NA48, NA49, NA69, :	Needle Roller Bearings
320, 322, 323, :	Tapered Roller Bearings (2)
230, 222, 223, :	Spherical Roller Bearings
511, 512, 513, :	Thrust Ball Bearing With Flat Seats
292, 293, 294, :	Thrust Spherical Roller Bearings
HR(4)	High Capacity Tapered Roller Bearings and Others

Bore Number

Code	Meaning
1	Bore Diam. 1 mm
2	2
3	3
⋮	⋮
9	9
00	10
01	12
02	15
03	17
/22	22
/28	28
/32	32
04(3)	20
05	25
06	30
⋮	⋮
88	440
92	460
96	480
/500	500
/530	530
/560	560
⋮	⋮
/2 360	2 360
/2 500	2 500

Contact Angle

Code	Meaning
	Angular Contact Ball Bearings
A	Standard Contact Angle of 30°
A5	Standard Contact Angle of 25°
B	Standard Contact Angle of 40°
C	Standard Contact Angle of 15°
	Tapered Roller Bearings
Omitted	Contact Angle Less Than 17°
C	Approx. 20° Contact Angle
D	Approx. 28° Contact Angle

Supplementary Designation

Internal Design (NSK Designations; Marked on Bearings)

Code	Meaning
A	Internal Design Differs From Standard
J	Smaller Diameter of Outer Ring Raceway, Contact Angle, and Outer Ring Width of Tapered Roller Bearings (Conforms to ISO 355)
	(For High Capacity Bearings)
C, CA, CD, EA	Spherical Roller Bearings
E	Cylindrical Roller Bearings
E	Spherical Thrust Roller Bearings
EA	Angular Contact Ball Bearings

Material (NSK Designations; Marked on Bearings)

Code	Meaning
g	Case-Hardened Steel Used in Rings, Rolling Elements
h	Stainless Steel Used in Rings, Rolling Elements

Cage (NSK Designations; Not Marked on Bearings)

Code	Meaning
M	Machined-Brass Cage
W	Pressed-Steel Cage
T	Synthetic Resin Cage
V	Without Cage

External Features — Seals, Shields (NSK Designations)

Code	Meaning
Z, ZS	Shield on One Side Only
ZZ, ZZS	Shields on Both Sides
DU	Rubber Contact Seal on One Side Only
DDU	Rubber Contact Seals on Both Sides
V	Rubber Non-Contact Seal on One Side Only
VV	Rubber Non-Contact Seals on Both Sides

External Features — Design of Rings (Partially Match JIS(5); In General, Marked on Bearings)

Code	Meaning
K	Tapered Bore of Inner Ring (Taper 1:12)
K30	Tapered Bore of Inner Ring (Taper 1:30)
E	Notch or Lubricating Groove in Ring
E4	Lubricating Groove in Outside Surface and Holes in Outer Ring
N	Snap Ring Groove in Outer Ring
NR	Snap Ring Groove With Snap Ring in Outer Ring

Arrangement (Match JIS(5); In General, Marked on Bearings)

Code	Meaning
DB	Back-to-Back Arrangement
DF	Face-to-Face Arrangement
DT	Tandem Arrangement

Internal Clearance / Preload (NSK Designations / Partially Match JIS(5)/ BAS(6); In General, Marked on Bearings)

Code	Meaning (radial clearance)
	For All Radial Brgs.
C1	Clearance Less Than C2
C2	Clearance Less Than CN
Omitted	CN Clearance
C3	Clearance Greater Than CN
C4	Clearance Greater Than C3
C5	Clearance Greater Than C4
	For Non-Interchangeable Cylindrical Roller Brgs.
CC1	Clearance Less Than CC2
CC2	Clearance Less Than CC
CC	Normal Clearance
CC3	Clearance Greater Than CC
CC4	Clearance Greater Than CC3
CC5	Clearance Greater Than CC4
	For Extra-Small and Miniature Ball Brgs.
MC1	Clearance Less Than MC2
MC2	Clearance Less Than MC3
MC3	Normal Clearance
MC4	Clearance Greater Than MC3
MC5	Clearance Greater Than MC4
MC6	Clearance Greater Than MC5
CM	Clearance in Deep Groove Ball Bearings for Electric Motors
CT, CM	Clearance in Cylindrical Roller Bearings for Electric Motors
	(Preload of Angular Contact Ball Bearing)
EL	Extra Light Preload
L	Light Preload
M	Medium Preload
H	Heavy Preload

Tolerance Class (Match JIS(5); In General, Marked on Bearings)

Code	Meaning
Omitted	ISO Normal
P6	ISO Class 6
P6X	ISO Class 6X
P5	ISO Class 5
P4	ISO Class 4
P2	ISO Class 2
	ABMA(7) Tapered Roller Bearing
Omitted	Class 4
PN2	Class 2
PN3	Class 3
PN0	Class 0
PN00	Class 00

Special Specification (NSK Designations, Partially Match JIS(5); In General, Marked on Bearings)

Code	Meaning
	Bearings Treated with Dimensional Stabilization
X26	Working Temperature Lower Than 150 °C
X28	Working Temperature Lower Than 200 °C
X29	Working Temperature Lower Than 250 °C
	Spherical Roller Bearings
S11	Dimensional Stabilizing Treatment Working Temperature Lower Than 200 °C

Spacer / Sleeve (NSK Designations, Partially Match JIS(5); Not Marked on Bearings)

Code	Meaning
+K	Bearings With Outer Ring Spacers
+L	Bearings With Inner Ring Spacers
+KL	Bearings With Both Inner and Outer Ring Spacers
H	Adapter
AH	Withdrawal Sleeve
HJ	Thrust Collar

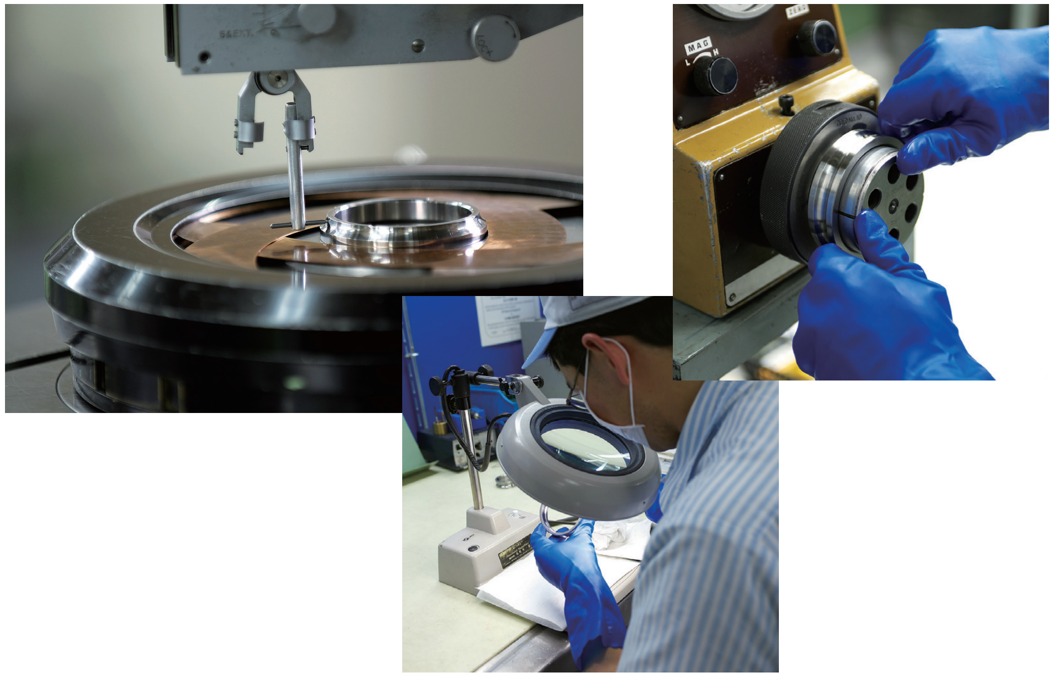
Grease (NSK Designations, Partially Match JIS(5); Not Marked on Bearings)

Code	Meaning
AS2	SHELL ALVANIA GREASE S2
ENS	ENS GREASE
NS7	NS HI-LUBE
PS2	MULTEMP PS No. 2

Notes
(1) Bearing Series designations conform to Table 6.5.
(2) For basic designations of tapered roller bearings in ISO's new series, refer to Page C182.
(3) The bore size (mm) is five times the bore number for Bore Numbers 04 through 96 (except double-direction thrust ball bearings).
(4) HR is an NSK Bearing Series designation.

Notes
(5) JIS: Japanese Industrial Standards.
(6) BAS: The Japan Bearing Industrial Association Standard.
(7) ABMA: The American Bearing Manufacturers Association.

7. BEARING TOLERANCES



7. BEARING TOLERANCES

7.1 Bearing Tolerance Standards

The tolerances for the boundary dimensions and running accuracy of rolling bearings are specified by ISO 492, 199, and 582. Tolerances are specified for the following items:

Class 2 is the highest bearing tolerance class in ISO but additional classes exist, including Class 6X (for tapered roller bearings), Class 6, Class 5, and Class 4. The applicable tolerance classes for each bearing type and the correspondence of these classes are shown in Table 7.1.

Tolerances of Rolling Bearings

- Dimensional Accuracy (Items necessary to mount bearings on shafts or in housings)
  - Tolerances for bore and outside diameters, ring width, and bearing width
  - Tolerances for inscribed and circumscribed circle diameters of rollers
  - Tolerances for chamfer dimensions
  - Tolerances for width variation
  - Tolerances for tapered bore diameters
- Running Accuracy (Items necessary to specify the runout of rotating machine parts)
  - Permissible radial runout of inner and outer rings
  - Permissible face runout with raceway inner and outer rings
  - Permissible perpendicularity of inner ring face with respect to the bore
  - Permissible perpendicularity of outer ring outside surface with respect to the face
  - Permissible raceway to back face thickness variation of thrust bearings

Table 7. 1 Bearing Types and Tolerance Classes

Deep Groove Ball Bearing			ISO 492	Normal	—	Class 6	Class 5	Class 4	Class 2
Angular Contact Ball Bearings				Normal	—	Class 6	Class 5	Class 4	Class 2
Self-Aligning Ball Bearings				Normal	—	Class 6 Equivalent	Class 5 Equivalent	—	—
Cylindrical Roller Bearings				Normal	—	Class 6	Class 5	Class 4	Class 2
Needle Roller Bearings				Normal	—	Class 6	Class 5	Class 4	—
Spherical Roller Bearings				Normal	—	Class 6	Class 5	—	—
Tapered Roller Bearings	Metric Design		ISO 492	Normal	Class 6X	Class 6	Class 5	Class 4	—
	Inch Design		ANSI/AFBMA Std.19.2	Class 4	—	Class 2	Class 3	Class 0	Class 00
	J Series		ANSI/AFBMA Std.19.1	Class K	Class N	—	Class C	Class B	—
Magneto Ball Bearings			BAS1061	Normal Equivalent	—	Class 6 Equivalent	Class 5 Equivalent	—	—
Thrust Ball Bearings			ISO 199	Normal	—	Class 6	Class 5	Class 4	—
Thrust Roller Bearings				Normal	—	—	—	—	—
Thrust Spherical Roller Bearings				Normal	—	—	—	—	—
Equivalent Standards (Reference)	JIS		JIS B 1514, 1536	Class 0	—	Class 6	Class 5	Class 4	Class 2
	JIS	Tapered Roller Bearings, Metric Design	JIS B 1514	Class 0	Class 6X	(Class 6)	Class 5	Class 4	—
	DIN		DIN620	P0	—	P6	P5	P4	P2
	ANSI/AFBMA	Ball Bearings	ANSI/AFBMA Std.20	ABEC1	—	ABEC3	ABEC5	ABEC7	ABEC9
		Roller Bearings		RBEC1	—	RBEC3	RBEC5	—	—
		Instrument Ball Bearings	ANSI/AFBMA Std.12.2	—	—	—	Class 5P	Class 7P	Class 9P
		Tapered Roller Bearings, Metric Design	ANSI/AFBMA Std.19.1	Class K	Class N	—	Class C	Class B	Class A
	BAS	Tapered Roller Bearings, Metric Design, Multi/Four-Row	BAS1002	Class 0	—	—	—	—	—

(Reference) Rough definitions of items related to running accuracy and their measuring methods are shown in Fig. 7.1. These are further described in detail in ISO 5593 (*Rolling Bearings-Vocabulary*), JIS B 1515, (*Rolling Bearings-Tolerances*) and elsewhere.

Supplementary Table

Running Accuracy	Inner Ring	Outer Ring	Dial Gauge
K_{ia}	Rotating	Stationary	A
K_{ea}	Stationary	Rotating	A
S_{ia}	Rotating	Stationary	B_1
S_{ea}	Stationary	Rotating	B_2
S_d	Rotating	Stationary	C
S_D	—	Rotating	D
S_i , S_e	Only the shaft, housing, or central washer is to be rotated.		E

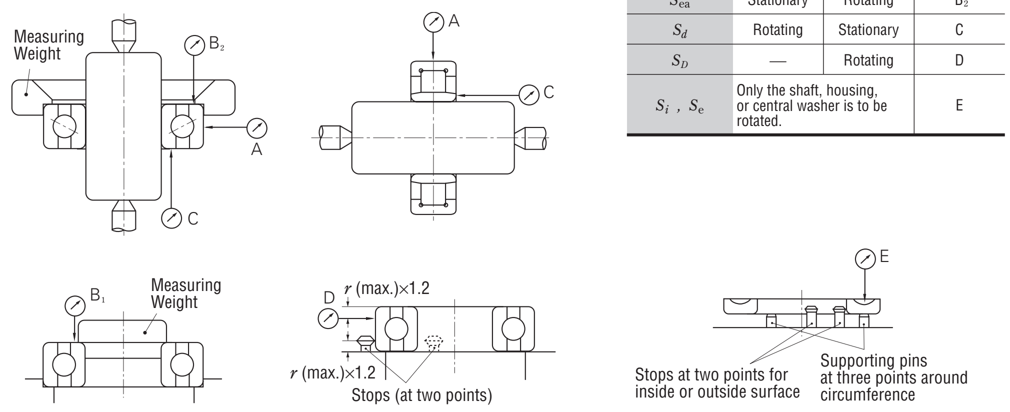


Fig. 7.1 Measuring Methods for Running Accuracy (Summarized)

Symbols for Boundary Dimensions and Running Accuracy

d	Nominal bore diameter
Δ_{ds}	Deviation of a single bore diameter
Δ_{dmp}	Single plane mean bore diameter deviation
V_{dp}	Bore diameter variation in a single radial plane
V_{dmp}	Mean bore diameter variation
V_{dsp}	Variation of bore diameter in a single plane
B	Inner ring width, nominal
Δ_{Bs}	Deviation of a single inner ring width
V_{Bs}	Inner ring width variation
K_{ia}	Radial runout of assembled brg. inner ring
S_d	Perpendicularity of inner ring face with respect to the bore
S_{ia}	Axial runout of inner ring of assembled bearing
S_i, S_e	Parallelism of inner ring raceway with respect to the face
T	Nominal (assembled) bearing width
Δ_{Ts}	Deviation of the actual brg. width
Δ_{T1s}	Deviation of the actual effective width of inner subunit
Δ_{T2s}	Deviation of the actual effective width of outer ring
D	Nominal outside diameter
Δ_{Ds}	Deviation of a single outside diameter
Δ_{Dmp}	Single plane mean outside diameter deviation
V_{Dp}	Outside diameter variation in a single radial plane
V_{Dmp}	Mean outside diameter variation
V_{Dsp}	Variation of outside diameter in a single plane
C	Nominal outer ring width
Δ_{Cs}	Deviation of a single outer ring width
V_{Cs}	Outer ring width variation
K_{ea}	Radial runout of assembled brg. outer ring
S_D	Perpendicularity of outer ring outside surface with respect to the face
S_{ea}	Axial runout of outer ring of assembled bearing

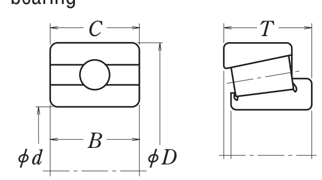


Table 7. 2 Tolerances for Radial Bearings (Excluding Tapered Roller Bearings)

Table 7. 2. 1 Tolerances for Inner Rings and Widths of Outer Rings

Nominal Bore Diameter d (mm)		Δ_{dmp} (2)										Δ_{ds} (2)			
		Normal Class		Class 6		Class 5		Class 4		Class 2		Class 4 Diameter Series 0, 1, 2, 3, 4		Class 2 Diameter Series 0, 1, 2, 3, 4	
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low
0.6(1)	**2.5**	0	− 8	0	− 7	0	− 5	0	− 4	0	−2.5	0	− 4	0	−2.5
2.5	**10**	0	− 8	0	− 7	0	− 5	0	− 4	0	−2.5	0	− 4	0	−2.5
10	**18**	0	− 8	0	− 7	0	− 5	0	− 4	0	−2.5	0	− 4	0	−2.5
18	**30**	0	− 10	0	− 8	0	− 6	0	− 5	0	−2.5	0	− 5	0	−2.5
30	**50**	0	− 12	0	−10	0	− 8	0	− 6	0	−2.5	0	− 6	0	−2.5
50	**80**	0	− 15	0	−12	0	− 9	0	− 7	0	−4	0	− 7	0	−4
80	**120**	0	− 20	0	−15	0	−10	0	− 8	0	−5	0	− 8	0	−5
120	**150**	0	− 25	0	−18	0	−13	0	−10	0	−7	0	−10	0	−7
150	**180**	0	− 25	0	−18	0	−13	0	−10	0	−7	0	−10	0	−7
180	**250**	0	− 30	0	−22	0	−15	0	−12	0	−8	0	−12	0	−8
250	**315**	0	− 35	0	−25	0	−18	—	—	—	—	—	—	—	—
315	**400**	0	− 40	0	−30	0	−23	—	—	—	—	—	—	—	—
400	**500**	0	− 45	0	−35	—	—	—	—	—	—	—	—	—	—
500	**630**	0	− 50	0	−40	—	—	—	—	—	—	—	—	—	—
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	—	—	—	—	—	—	—	—	—	—	—	—

Nominal Bore Diameter d (mm)		V_{dsp} (2)											V_{dmp} (2)				
		Normal Class Diameter Series			Class 6 Diameter Series			Class 5 Diameter Series		Class 4 Diameter Series		Class 2 Diameter Series	Normal Class	Class 6	Class 5	Class 4	Class 2
		9	0, 1	2, 3, 4	9	0, 1	2, 3, 4	9	1, 2, 3, 4	9	1, 2, 3, 4	0, 1, 2, 3, 4					
over	incl.	Max.			Max.			Max.		Max.		Max.	Max.	Max.	Max.	Max.	Max.
0.6	**2.5**	10	8	6	9	7	5	5	4	4	3	2.5	6	5	3	2	1.5
2.5	**10**	10	8	6	9	7	5	5	4	4	3	2.5	6	5	3	2	1.5
10	**18**	10	8	6	9	7	5	5	4	4	3	2.5	6	5	3	2	1.5
18	**30**	13	10	8	10	8	6	6	5	5	4	2.5	8	6	3	2.5	1.5
30	**50**	15	12	9	13	10	8	8	6	6	5	2.5	9	8	4	3	1.5
50	**80**	19	19	11	15	15	9	9	7	7	5	4	11	9	5	3.5	2
80	**120**	25	25	15	19	19	11	10	8	8	6	5	15	11	5	4	2.5
120	**150**	31	31	19	23	23	14	13	10	10	8	7	19	14	7	5	3.5
150	**180**	31	31	19	23	23	14	13	10	10	8	7	19	14	7	5	3.5
180	**250**	38	38	23	28	28	17	15	12	12	9	8	23	17	8	6	4
250	**315**	44	44	26	31	31	19	18	14	—	—	—	26	19	9	—	—
315	**400**	50	50	30	38	38	23	23	18	—	—	—	30	23	12	—	—
400	**500**	56	56	34	44	44	26	—	—	—	—	—	34	26	—	—	—
500	**630**	63	63	38	50	50	30	—	—	—	—	—	38	30	—	—	—
630	**800**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
800	**1 000**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 000	**1 250**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 250	**1 600**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 600	**2 000**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

Units : μm

Nominal Bore Diameter d (mm)		Δ_{Bs} (or Δ_{Cs})(3)																V_{Bs} (or V_{Cs})				
		Single Bearing								Combined Bearings								Inner Ring (or Outer Ring) (3)		Inner Ring		
		Normal Class		Class 6		Class 5		Class 4 Class 2		Class Normal (4) (5)		Class 6 (4) (5)		Class 5 (4) (5)		Class 4 (4) Class 2 (4)		Normal Class	Class 6	Class 5	Class 4	Class 2
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low	Max.	Max.	Max.	Max.	Max.
0.6	**2.5**	0	− 40	0	− 40	0	− 40	0	− 40	—	—	—	—	0	−250	0	−250	12	12	5	2.5	1.5
2.5	**10**	0	− 120	0	−120	0	− 40	0	− 40	0	−250	0	−250	0	−250	0	−250	15	15	5	2.5	1.5
10	**18**	0	− 120	0	−120	0	− 80	0	− 80	0	−250	0	−250	0	−250	0	−250	20	20	5	2.5	1.5
18	**30**	0	− 120	0	−120	0	−120	0	−120	0	−250	0	−250	0	−250	0	−250	20	20	5	2.5	1.5
30	**50**	0	− 120	0	−120	0	−120	0	−120	0	−250	0	−250	0	−250	0	−250	20	20	5	3	1.5
50	**80**	0	− 150	0	−150	0	−150	0	−150	0	−380	0	−380	0	−250	0	−250	25	25	6	4	1.5
80	**120**	0	− 200	0	−200	0	−200	0	−200	0	−380	0	−380	0	−380	0	−380	25	25	7	4	2.5
120	**150**	0	− 250	0	−250	0	−250	0	−250	0	−500	0	−500	0	−380	0	−380	30	30	8	5	2.5
150	**180**	0	− 250	0	−250	0	−250	0	−250	0	−500	0	−500	0	−380	0	−380	30	30	8	5	4
180	**250**	0	− 300	0	−300	0	−300	0	−300	0	−500	0	−500	0	−500	0	−500	30	30	10	6	5
250	**315**	0	− 350	0	−350	0	−350	—	—	0	−500	0	−500	0	−500	—	—	35	35	13	—	—
315	**400**	0	− 400	0	−400	0	−400	—	—	0	−630	0	−630	0	−630	—	—	40	40	15	—	—
400	**500**	0	− 450	0	−450	—	—	—	—	—	—	—	—	—	—	—	—	50	45	—	—	—
500	**630**	0	− 500	0	−500	—	—	—	—	—	—	—	—	—	—	—	—	60	50	—	—	—
630	**800**	0	− 750	—	—	—	—	—	—	—	—	—	—	—	—	—	—	70	—	—	—	—
800	**1 000**	0	−1 000	—	—	—	—	—	—	—	—	—	—	—	—	—	—	80	—	—	—	—
1 000	**1 250**	0	−1 250	—	—	—	—	—	—	—	—	—	—	—	—	—	—	100	—	—	—	—
1 250	**1 600**	0	−1 600	—	—	—	—	—	—	—	—	—	—	—	—	—	—	120	—	—	—	—
1 600	**2 000**	0	−2 000	—	—	—	—	—	—	—	—	—	—	—	—	—	—	140	—	—	—	—

Nominal Bore Diameter d (mm)		K_{ia}					S_d			S_{ia} (6)		
		Normal Class	Class 6	Class 5	Class 4	Class 2	Class 5	Class 4	Class 2	Class 5	Class 4	Class 2
over	incl.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.
0.6	**2.5**	10	5	4	2.5	1.5	7	3	1.5	7	3	1.5
2.5	**10**	10	6	4	2.5	1.5	7	3	1.5	7	3	1.5
10	**18**	10	7	4	2.5	1.5	7	3	1.5	7	3	1.5
18	**30**	13	8	4	3	2.5	8	4	1.5	8	4	2.5
30	**50**	15	10	5	4	2.5	8	4	1.5	8	4	2.5
50	**80**	20	10	5	4	2.5	8	5	1.5	8	5	2.5
80	**120**	25	13	6	5	2.5	9	5	2.5	9	5	2.5
120	**150**	30	18	8	6	2.5	10	6	2.5	10	7	2.5
150	**180**	30	18	8	6	5	10	6	4	10	7	5
180	**250**	40	20	10	8	5	11	7	5	13	8	5
250	**315**	50	25	13	—	—	13	—	—	15	—	—
315	**400**	60	30	15	—	—	15	—	—	20	—	—
400	**500**	65	35	—	—	—	—	—	—	—	—	—
500	**630**	70	40	—	—	—	—	—	—	—	—	—
630	**800**	80	—	—	—	—	—	—	—	—	—	—
800	**1 000**	90	—	—	—	—	—	—	—	—	—	—
1 000	**1 250**	100	—	—	—	—	—	—	—	—	—	—
1 250	**1 600**	120	—	—	—	—	—	—	—	—	—	—
1 600	**2 000**	140	—	—	—	—	—	—	—	—	—	—

Notes
(1) 0.6mm is included in this group.
(2) Applicable to bearings with cylindrical bores.
(3) Outer ring width tolerances or deviation depend on the values for the inner ring of the same bearing. Tolerances for the width variation of outer rings in Class 5, 4, and 2 are shown in Table 7.2.2.
(4) Applicable to individual rings manufactured for combined bearings.
(5) Also applicable to inner ring tapered bores with $d \geqq 50$ mm.
(6) Applicable to ball bearings such as deep groove ball bearings and angular contact ball bearings.

Remarks
1. The cylindrical bore diameter "no-go side" tolerance limit (high) specified in this table does not necessarily apply within a distance from the ring face to 1.2 times the chamfer dimension r (max.).
2. AISI/ABMA Std 20-1996: ABEC1·RBEC1, ABEC3·RBEC3, ABEC5·RBEC5, ABEC7·RBEC7, and ABEC9·RBEC9 are equivalent to Classes Normal, 6, 5, 4, and 2 respectively.

Table 7. 2 Tolerances for Radial Bearings (Excluding Tapered Roller Bearings)

Table 7. 2. 2 Tolerances for Outer Rings

Nominal Outside Diameter D (mm)		Δ_{Dmp}										Δ_{Ds}			
		Normal Class		Class 6		Class 5		Class 4		Class 2		Class 4		Class 2	
												Diameter Series 0, 1, 2, 3, 4			
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low
2.5(1)	**6**	0	− 8	0	− 7	0	− 5	0	− 4	0	− 2.5	0	− 4	0	− 2.5
6	**18**	0	− 8	0	− 7	0	− 5	0	− 4	0	− 2.5	0	− 4	0	− 2.5
18	**30**	0	− 9	0	− 8	0	− 6	0	− 5	0	− 4	0	− 5	0	− 4
30	**50**	0	− 11	0	− 9	0	− 7	0	− 6	0	− 4	0	− 6	0	− 4
50	**80**	0	− 13	0	−11	0	− 9	0	− 7	0	− 4	0	− 7	0	− 4
80	**120**	0	− 15	0	−13	0	−10	0	− 8	0	− 5	0	− 8	0	− 5
120	**150**	0	− 18	0	−15	0	−11	0	− 9	0	− 5	0	− 9	0	− 5
150	**180**	0	− 25	0	−18	0	−13	0	−10	0	− 7	0	−10	0	− 7
180	**250**	0	− 30	0	−20	0	−13	0	−11	0	− 8	0	−11	0	− 8
250	**315**	0	− 35	0	−25	0	−18	0	−13	0	− 8	0	−13	0	− 8
315	**400**	0	− 40	0	−28	0	−20	0	−15	0	−10	0	−15	0	−10
400	**500**	0	− 45	0	−33	0	−23	—	—	—	—	—	—	—	—
500	**630**	0	− 50	0	−38	0	−28	—	—	—	—	—	—	—	—
630	**800**	0	− 75	0	−45	0	−35	—	—	—	—	—	—	—	—
800	**1 000**	0	−100	0	−60	—	—	—	—	—	—	—	—	—	—
1 000	**1 250**	0	−125	—	—	—	—	—	—	—	—	—	—	—	—
1 250	**1 600**	0	−160	—	—	—	—	—	—	—	—	—	—	—	—
1 600	**2 000**	0	−200	—	—	—	—	—	—	—	—	—	—	—	—
2 000	**2 500**	0	−250	—	—	—	—	—	—	—	—	—	—	—	—

Nominal Outside Diameter D (mm)		V_{Dsp} (2)													V_{Dmp} (2)				
		Normal Class				Class 6				Class 5		Class 4		Class 2	Normal Class	Class 6	Class 5	Class 4	Class 2
		Open Bearings			Sealed/ Shielded Bearings	Open Bearings			Sealed/ Shielded Bearings	Open Bearings		Open Bearings		Open Bearings					
		Diameter Series				Diameter Series				Diameter Series		Diameter Series		Diameter Series					
		9	0, 1	2, 3, 4	2, 3, 4	9	0, 1	2, 3, 4	0, 1, 2, 3, 4	9	1, 2, 3, 4	9	1, 2, 3, 4	0, 1, 2, 3, 4					
over	incl.	Max.				Max.				Max.		Max.		Max.	Max.	Max.	Max.	Max.	Max.
2.5(1)	**6**	10	8	6	10	9	7	5	9	5	4	4	3	2.5	6	5	3	2	1.5
6	**18**	10	8	6	10	9	7	5	9	5	4	4	3	2.5	6	5	3	2	1.5
18	**30**	12	9	7	12	10	8	6	10	6	5	5	4	4	7	6	3	2.5	2
30	**50**	14	11	8	16	11	9	7	13	7	5	6	5	4	8	7	4	3	2
50	**80**	16	13	10	20	14	11	8	16	9	7	7	5	4	10	8	5	3.5	2
80	**120**	19	19	11	26	16	16	10	20	10	8	8	6	5	11	10	5	4	2.5
120	**150**	23	23	14	30	19	19	11	25	11	8	9	7	5	14	11	6	5	2.5
150	**180**	31	31	19	38	23	23	14	30	13	10	10	8	7	19	14	7	5	3.5
180	**250**	38	38	23	—	25	25	15	—	15	11	11	8	8	23	15	8	6	4
250	**315**	44	44	26	—	31	31	19	—	18	14	13	10	8	26	19	9	7	4
315	**400**	50	50	30	—	35	35	21	—	20	15	15	11	10	30	21	10	8	5
400	**500**	56	56	34	—	41	41	25	—	23	17	—	—	—	34	25	12	—	—
500	**630**	63	63	38	—	48	48	29	—	28	21	—	—	—	38	29	14	—	—
630	**800**	94	94	55	—	56	56	34	—	35	26	—	—	—	55	34	18	—	—
800	**1 000**	125	125	75	—	75	75	45	—	—	—	—	—	—	75	45	—	—	—
1 000	**1 250**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 250	**1 600**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 600	**2 000**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2 000	**2 500**	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

Units : μm

K_{ea}					S_D (4)			S_{ea} (3)(4)			V_{Cs} (5)			Nominal Outside Diameter D (mm)	
Normal Class	Class 6	Class 5	Class 4	Class 2	Class 5	Class 4	Class 2	Class 5	Class 4	Class 2	Class 5	Class 4	Class 2		
Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	over	incl.
15	8	5	3	1.5	8	4	1.5	8	5	1.5	5	2.5	1.5	**2.5**	**6**
15	8	5	3	1.5	8	4	1.5	8	5	1.5	5	2.5	1.5	**6**	**18**
15	9	6	4	2.5	8	4	1.5	8	5	2.5	5	2.5	1.5	**18**	**30**
20	10	7	5	2.5	8	4	1.5	8	5	2.5	5	2.5	1.5	**30**	**50**
25	13	8	5	4	8	4	1.5	10	5	4	6	3	1.5	**50**	**80**
35	18	10	6	5	9	5	2.5	11	6	5	8	4	2.5	**80**	**120**
40	20	11	7	5	10	5	2.5	13	7	5	8	5	2.5	**120**	**150**
45	23	13	8	5	10	5	2.5	14	8	5	8	5	2.5	**150**	**180**
50	25	15	10	7	11	7	4	15	10	7	10	7	4	**180**	**250**
60	30	18	11	7	13	8	5	18	10	7	11	7	5	**250**	**315**
70	35	20	13	8	13	10	7	20	13	8	13	8	7	**315**	**400**
80	40	23	—	—	15	—	—	23	—	—	15	—	—	**400**	**500**
100	50	25	—	—	18	—	—	25	—	—	18	—	—	**500**	**630**
120	60	30	—	—	20	—	—	30	—	—	20	—	—	**630**	**800**
140	75	—	—	—	—	—	—	—	—	—	—	—	—	**800**	**1 000**
160	—	—	—	—	—	—	—	—	—	—	—	—	—	**1 000**	**1 250**
190	—	—	—	—	—	—	—	—	—	—	—	—	—	**1 250**	**1 600**
220	—	—	—	—	—	—	—	—	—	—	—	—	—	**1 600**	**2 000**
250	—	—	—	—	—	—	—	—	—	—	—	—	—	**2 000**	**2 500**

Notes (1) 2.5 mm is included in this group.
(2) Applicable only when a locating snap ring is not used.
(3) Applicable to ball bearings, such as deep groove ball bearings and angular contact ball bearings.
(4) Not applicable to bearings with flanges.
(5) The tolerances for outer ring width variation in Normal Class and Class 6 bearings are shown in Table 7.2.1.

Remarks 1. The outside diameter "no-go side" tolerances (low) specified in this table do not necessarily apply within a distance from the ring face to 1.2 times the chamfer dimension r (max.).
2. AISI/ABMA Std 20-1996: ABEC1·RBEC1, ABEC3·RBEC3, ABEC5·RBEC5, ABEC7·RBEC7, and ABEC9·RBEC9 are equivalent to Classes Normal, 6, 5, 4, and 2 respectively.

Table 7. 3 Tolerances for Metric Series Tapered Roller Bearings

Table 7. 3. 1 Tolerances for Inner Ring Bore Diameter and Running Accuracy

Units : μm

Nominal Bore Diameter d (mm)		Δ_{dmp}								Δ_{ds}		V_{dsp}				V_{dmp}				K_{ia}				S_d		S_{ia}
		Normal Class Class 6X		(Class 6)		Class 5		Class 4		Class 4		Normal Class Class 6X	(Class 6)	Class 5	Class 4	Normal Class Class 6X	(Class 6)	Class 5	Class 4	Normal Class Class 6X	(Class 6)	Class 5	Class 4	Class 5	Class 4	Class 4
over	incl.	high	low	high	low	high	low	high	low	high	low	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.
10(1)	18	0	− 12	0	− 7	0	− 7	0	− 5	0	− 5	12	7	5	4	9	5	5	4	15	7	5	3	7	3	3
18	30	0	− 12	0	− 8	0	− 8	0	− 6	0	− 6	12	8	6	5	9	6	5	4	18	8	5	3	8	4	4
30	50	0	− 12	0	−10	0	−10	0	− 8	0	− 8	12	10	8	6	9	8	5	5	20	10	6	4	8	4	4
50	80	0	− 15	0	−12	0	−12	0	− 9	0	− 9	15	12	9	7	11	9	6	5	25	10	7	4	8	5	4
80	120	0	− 20	0	−15	0	−15	0	−10	0	−10	20	15	11	8	15	11	8	5	30	13	8	5	9	5	5
120	180	0	− 25	0	−18	0	−18	0	−13	0	−13	25	18	14	10	19	14	9	7	35	18	11	6	10	6	7
180	250	0	− 30	0	−22	0	−22	0	−15	0	−15	30	22	17	11	23	16	11	8	50	20	13	8	11	7	8
250	315	0	− 35	—	—	0	−25	0	−18	0	−18	35	—	19	12	26	—	13	9	60	—	13	9	13	8	9
315	400	0	− 40	—	—	0	−30	—	—	—	—	40	—	23	—	30	—	15	—	70	—	15	—	15	—	—
400	500	0	− 45	—	—	0	−35	—	—	—	—	45	—	28	—	34	—	17	—	80	—	20	—	17	—	—
500	630	0	− 60	—	—	0	−40	—	—	—	—	60	—	35	—	40	—	20	—	90	—	25	—	20	—	—
630	800	0	− 75	—	—	0	−50	—	—	—	—	75	—	45	—	45	—	25	—	100	—	30	—	25	—	—
800	1 000	0	−100	—	—	0	−60	—	—	—	—	100	—	60	—	55	—	30	—	115	—	37	—	30	—	—
1 000	1 250	0	−125	—	—	0	−75	—	—	—	—	125	—	75	—	65	—	37	—	130	—	45	—	40	—	—
1 250	1 600	0	−160	—	—	0	−90	—	—	—	—	160	—	90	—	80	—	45	—	150	—	55	—	50	—	—
1 600	2 000	0	−200	—	—	—	—	—	—	—	—	200	—	—	—	100	—	—	—	170	—	—	—	—	—	—

Notes (1) 10 mm is included in this group.

Remarks 1. The bore diameter "no-go side" tolerances (high) specified in this table do not necessarily apply within a distance from the ring face to 1.2 times the chamfer dimension r (max.).

2. Some of these tolerances conform to NSK standards.

Table 7. 3. 2 Tolerances for Outer Ring Outside Diameter and Running Accuracy

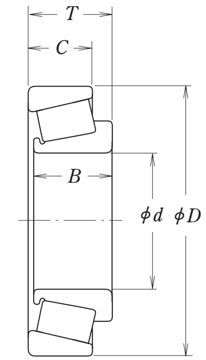
Units : μm

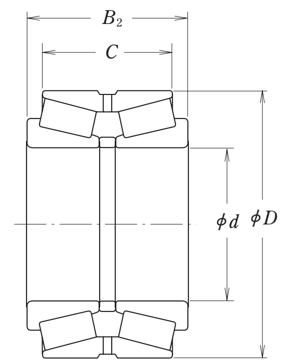
Nominal Outside Diameter D (mm)		Δ_{Dmp}								Δ_{Ds}		V_{Dsp}				V_{Dmp}				K_{ea}				S_D(2)		S_{ea}(2)
		Normal Class Class 6X		(Class 6)		Class 5		Class 4		Class 4		Normal Class Class 6X	(Class 6)	Class 5	Class 4	Normal Class Class 6X	(Class 6)	Class 5	Class 4	Normal Class Class 6X	(Class 6)	Class 5	Class 4	Class 5	Class 4	Class 4
over	incl.	high	low	high	low	high	low	high	low	high	low	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.
18(1)	30	0	− 12	0	− 8	0	− 8	0	− 6	0	− 6	12	8	6	5	9	6	5	4	18	9	6	4	8	4	5
30	50	0	− 14	0	− 9	0	− 9	0	− 7	0	− 7	14	9	7	5	11	7	5	5	20	10	7	5	8	4	5
50	80	0	− 16	0	−11	0	− 11	0	− 9	0	− 9	16	11	8	7	12	8	6	5	25	13	8	5	8	4	5
80	120	0	− 18	0	−13	0	− 13	0	−10	0	−10	18	13	10	8	14	10	7	5	35	18	10	6	9	5	6
120	150	0	− 20	0	−15	0	− 15	0	−11	0	−11	20	15	11	8	15	11	8	6	40	20	11	7	10	5	7
150	180	0	− 25	0	−18	0	− 18	0	−13	0	−13	25	18	14	10	19	14	9	7	45	23	13	8	10	5	8
180	250	0	− 30	0	−20	0	− 20	0	−15	0	−15	30	20	15	11	23	15	10	8	50	25	15	10	11	7	10
250	315	0	− 35	0	−25	0	− 25	0	−18	0	−18	35	25	19	14	26	19	13	9	60	30	18	11	13	8	10
315	400	0	− 40	0	−28	0	− 28	0	−20	0	−20	40	28	22	15	30	21	14	10	70	35	20	13	13	10	13
400	500	0	− 45	—	—	0	− 33	—	—	—	—	45	—	26	—	34	—	17	—	80	—	24	—	17	—	—
500	630	0	− 50	—	—	0	− 38	—	—	—	—	60	—	30	—	38	—	20	—	100	—	30	—	20	—	—
630	800	0	− 75	—	—	0	− 45	—	—	—	—	80	—	38	—	55	—	25	—	120	—	36	—	25	—	—
800	1000	0	−100	—	—	0	− 60	—	—	—	—	100	—	50	—	75	—	30	—	140	—	43	—	30	—	—
1 000	1 250	0	−125	—	—	0	− 80	—	—	—	—	130	—	65	—	90	—	38	—	160	—	52	—	38	—	—
1 250	1 600	0	−160	—	—	0	−100	—	—	—	—	170	—	90	—	100	—	50	—	180	—	62	—	50	—	—
1 600	2 000	0	−200	—	—	0	−125	—	—	—	—	210	—	120	—	110	—	65	—	200	—	73	—	65	—	—
2 000	2 500	0	−250	—	—	—	—	—	—	—	—	265	—	—	—	120	—	—	—	220	—	—	—	—	—	—

Notes (1) 18 mm is included in this group.
(2) Not applicable to bearings with flanges.

Remarks 1. The outside diameter "no-go side" tolerances (low) specified in this table do not necessarily apply within a distance from the ring face to 1.2 times the chamfer dimension r (max.).

2. Some of these tolerances conform to NSK standards.





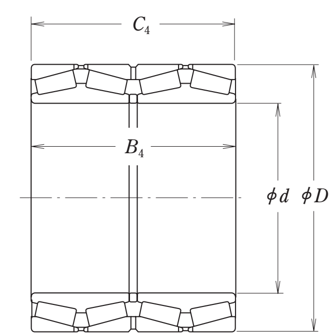


Table 7. 3 Tolerances for Metric Series Tapered Roller Bearings

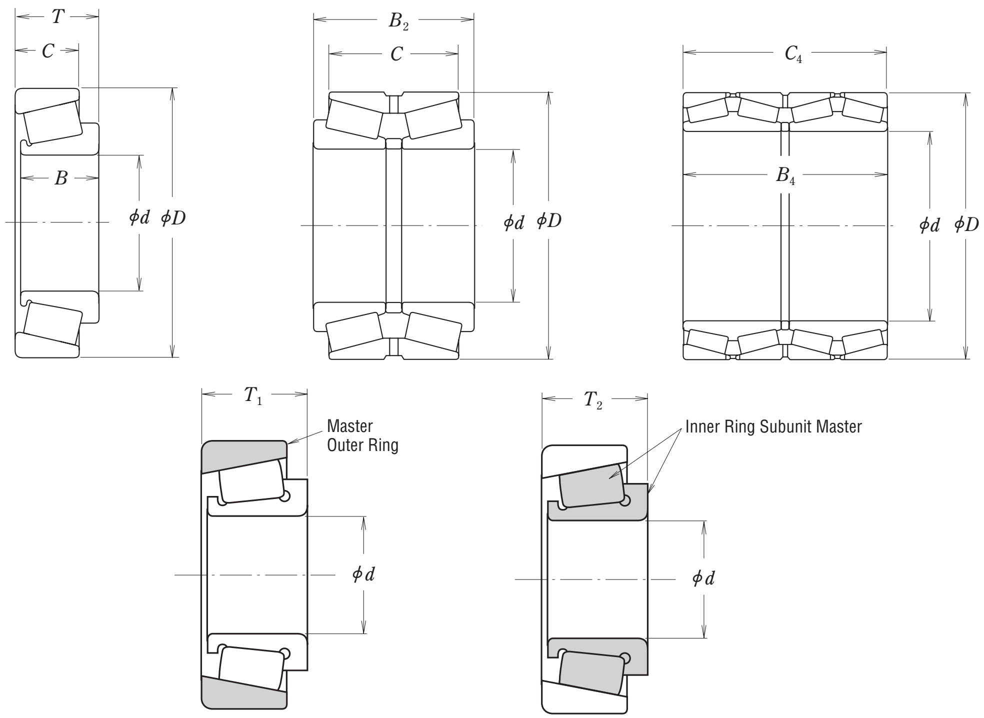
Table 7. 3. 3 Tolerances for Width, Overall Bearing Width, and Combined Bearing Width

Nominal Bore Diameter d (mm)		Δ_{Bs}										Δ_{Cs}									
		Normal Class		(Class 6)		Class 6X		Class 5		Class 4		Normal Class		(Class 6)		Class 6X		Class 5		Class 4	
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low
10[1]	18	0	−120	0	−120	0	−50	0	−200	0	−200	0	−120	0	−120	0	−100	0	−200	0	−200
18	30	0	−120	0	−120	0	−50	0	−200	0	−200	0	−120	0	−120	0	−100	0	−200	0	−200
30	50	0	−120	0	−120	0	−50	0	−240	0	−240	0	−120	0	−120	0	−100	0	−240	0	−240
50	80	0	−150	0	−150	0	−50	0	−300	0	−300	0	−150	0	−150	0	−100	0	−300	0	−300
80	120	0	−200	0	−200	0	−50	0	−400	0	−400	0	−200	0	−200	0	−100	0	−400	0	−400
120	180	0	−250	0	−250	0	−50	0	−500	0	−500	0	−250	0	−250	0	−100	0	−500	0	−500
180	250	0	−300	0	−300	0	−50	0	−600	0	−600	0	−300	0	−300	0	−100	0	−600	0	−600
250	315	0	−350	—	—	0	−50	0	−700	0	−700	0	−350	—	—	0	−100	0	−700	0	−700
315	400	0	−400	—	—	0	−50	0	−800	—	—	0	−400	—	—	0	−100	0	−800	—	—
400	500	0	−450	—	—	0	−50	0	−900	—	—	0	−450	—	—	0	−100	—	−900	—	—
500	630	0	−500	—	—	—	—	0	−1 100	—	—	0	−500	—	—	—	—	—	−1 100	—	—
630	800	0	−750	—	—	—	—	0	−1 600	—	—	0	−750	—	—	—	—	—	−1 600	—	—
800	1 000	0	−1 000	—	—	—	—	0	−2 000	—	—	0	−1 000	—	—	—	—	—	−2 000	—	—
1 000	1 250	0	−1 250	—	—	—	—	0	−2 000	—	—	0	−1 250	—	—	—	—	—	−2 000	—	—
1 250	1 600	0	−1 600	—	—	—	—	0	−2 000	—	—	0	−1 600	—	—	—	—	—	−2 000	—	—
1 600	2 000	0	−2 000	—	—	—	—	—	—	—	—	0	−2 000	—	—	—	—	—	—	—	—

Nominal Bore Diameter d (mm)		Δ_{Ts}										Δ_{T1s}							
		Normal Class		(Class 6)		Class 6X		Class 5		Class 4		Normal Class		Class 6X		Class 5		Class 4	
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low
10	18	+ 200	0	+200	0	+100	0	+200	−200	+200	−200	+100	0	+ 50	0	+100	−100	+100	−100
18	30	+ 200	0	+200	0	+100	0	+200	−200	+200	−200	+100	0	+ 50	0	+100	−100	+100	−100
30	50	+ 200	0	+200	0	+100	0	+200	−200	+200	−200	+100	0	+ 50	0	+100	−100	+100	−100
50	80	+ 200	0	+200	0	+100	0	+200	−200	+200	−200	+100	0	+ 50	0	+100	−100	+100	−100
80	120	+ 200	− 200	+200	−200	+100	0	+200	−200	+200	−200	+100	−100	+ 50	0	+100	−100	+100	−100
120	180	+ 350	− 250	+350	−250	+150	0	+350	−250	+350	−250	+150	−150	+ 50	0	+150	−150	+150	−150
180	250	+ 350	− 250	+350	−250	+150	0	+350	−250	+350	−250	+150	−150	+ 50	0	+150	−150	+150	−150
250	315	+ 350	− 250	+350	−250	+200	0	+350	−250	+350	−250	+150	−150	+100	0	+150	−150	+150	−150
315	400	+ 400	− 400	—	—	+200	0	+400	−400	—	—	+200	−200	+100	0	+200	−200	—	—
400	500	+ 400	− 400	—	—	+200	0	+400	−400	—	—	+225	−225	+100	0	+225	−225	—	—
500	630	+ 500	− 500	—	—	—	—	+500	−500	—	—	—	—	—	—	—	—	—	—
630	800	+ 600	− 600	—	—	—	—	+600	−600	—	—	—	—	—	—	—	—	—	—
800	1 000	+ 750	− 750	—	—	—	—	+750	−750	—	—	—	—	—	—	—	—	—	—
1 000	1 250	+ 900	− 900	—	—	—	—	+750	−750	—	—	—	—	—	—	—	—	—	—
1 250	1 600	+1 050	−1 050	—	—	—	—	+900	−900	—	—	—	—	—	—	—	—	—	—
1 600	2 000	+1 200	−1 200	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

Notes (1) 10 mm is included in this group.

Remarks The nominal effective width of the inner subunit T_1 is defined as the overall bearing width of the inner subunit combined with a master outer ring.
The nominal effective width of the outer ring T_2 is defined as the overall bearing width of the outer ring combined with a master inner subunit.



Units : μm

Δ_{T2s}								Overall Combined Bearing Width Deviation				Nominal Bore Diameter d (mm)	
Normal Class		Class 6X		Class 5		Class 4		Δ_{B2s} All Classes of Double-Row Bearings		Δ_{B4s}, Δ_{C4s} All Classes of Four-Row Bearings			
high	low	high	low	high	low	high	low	high	low	high	low	over	incl.
+100	0	+ 50	0	+100	−100	+100	−100	+ 200	− 200	—	—	10	18
+100	0	+ 50	0	+100	−100	+100	−100	+ 200	− 200	—	—	18	30
+100	0	+ 50	0	+100	−100	+100	−100	+ 200	− 200	—	—	30	50
+100	0	+ 50	0	+100	−100	+100	−100	+ 300	− 300	+ 300	− 300	50	80
+100	−100	+ 50	0	+100	−100	+100	−100	+ 300	− 300	+ 400	− 400	80	120
+200	−100	+100	0	+200	−100	+200	−100	+ 400	− 400	+ 500	− 500	120	180
+200	−100	+100	0	+200	−100	+200	−100	+ 450	− 450	+ 600	− 600	180	250
+200	−100	+100	0	+200	−100	+200	−100	+ 550	− 550	+ 700	− 700	250	315
+200	−200	+100	0	+200	−200	—	—	+ 600	− 600	+ 800	− 800	315	400
+225	−225	+100	0	+225	−225	—	—	+ 700	− 700	+ 900	− 900	400	500
—	—	—	—	—	—	—	—	+ 800	− 800	+ 1000	−1 000	500	630
—	—	—	—	—	—	—	—	+1 200	−1 200	+ 1500	−1 500	630	800
—	—	—	—	—	—	—	—	—	—	—	—	800	1 000
—	—	—	—	—	—	—	—	—	—	—	—	1 000	1 250
—	—	—	—	—	—	—	—	—	—	—	—	1 250	1 600
—	—	—	—	—	—	—	—	—	—	—	—	1 600	2 000

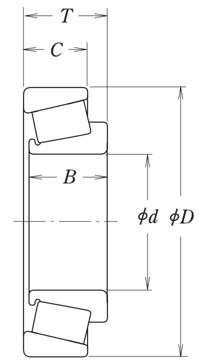
Table 7. 4 Tolerances for Inch Series Tapered Roller Bearings

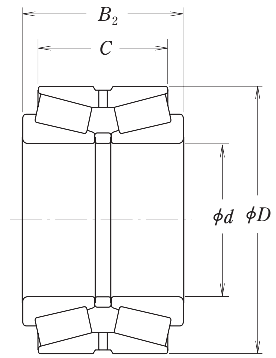
(Refer to Page A126 Table 7.1 for more information on "CLASS ⁎⁎" ANSI/ABMA tolerances.)

Table 7. 4. 1 Tolerances for Inner Ring Bore Diameter

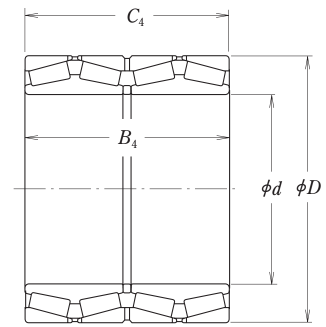
Units : μm

Nominal Bore Diameter d				Δ_{ds}					
over		incl.		CLASS 4, 2		CLASS 3, 0		CLASS 00	
(mm)	1/25.4	(mm)	1/25.4	high	low	high	low	high	low
–	–	**76.200**	3.0000	+13	0	+13	0	+8	0
76.200	3.0000	**266.700**	10.5000	+25	0	+13	0	+8	0
266.700	10.5000	**304.800**	12.0000	+25	0	+13	0	–	–
304.800	12.0000	**609.600**	24.0000	+51	0	+25	0	–	–
609.600	24.0000	**914.400**	36.0000	+76	0	+38	0	–	–
914.400	36.0000	**1 219.200**	48.0000	+102	0	+51	0	–	–
1 219.200	48.0000	–	–	+127	0	+76	0	–	–





KBE



KV

Table 7. 4. 2 Tolerances for Outer Ring Outside Diameter and Radial Runout of Inner and Outer Rings

Units : μm

Nominal Outside Diameter D				Δ_{Ds}						K_{ia}, K_{ea}				
over		incl.		CLASS 4, 2		CLASS 3, 0		CLASS 00		CLASS 4	CLASS 2	CLASS 3	CLASS 0	CLASS 00
(mm)	1/25.4	(mm)	1/25.4	high	low	high	low	high	low	max.	max.	max.	max.	max.
–	–	**266.700**	10.5000	+25	0	+13	0	+8	0	51	38	8	4	2
266.700	10.5000	**304.800**	12.0000	+25	0	+13	0	+8	0	51	38	8	4	2
304.800	12.0000	**609.600**	24.0000	+51	0	+25	0	–	–	51	38	18	–	–
609.600	24.0000	**914.400**	36.0000	+76	0	+38	0	–	–	76	51	51	–	–
914.400	36.0000	**1 219.200**	48.0000	+102	0	+51	0	–	–	76	–	76	–	–
1 219.200	48.0000	–	–	+127	0	+76	0	–	–	76	–	76	–	–

Table 7. 4. 3 Tolerances for Overall Width and Combined Width

Units : μm

Nominal Bore Diameter d				Δ_{Ts}									
over		incl.		CLASS 4		CLASS 2		CLASS 3 D≦508.000 (mm)		CLASS 3 D>508.000 (mm)		CLASS 0, 00	
(mm)	1/25.4	(mm)	1/25.4	high	low	high	low	high	low	high	low	high	low
–	–	**101.600**	4.0000	+203	0	+203	0	+203	−203	+203	−203	+203	−203
101.600	4.0000	**304.800**	12.0000	+356	−254	+203	0	+203	−203	+203	−203	+203	−203
304.800	12.0000	**609.600**	24.0000	+381	−381	+381	−381	+203	−203	+381	−381	–	–
609.600	24.0000	–	–	+381	−381	–	–	+381	−381	+381	−381	–	–

Nominal Bore Diameter d				Double-Row Bearings (KBE Type) Δ_{B2s}										Four-Row Bearings (KV Type) Δ_{B4s}, Δ_{C4s}	
over		incl.		CLASS 4		CLASS 2		CLASS 3 D≦508.000 (mm)		CLASS 3 D>508.000 (mm)		CLASS 0,00		CLASS 4, 3	
(mm)	1/25.4	(mm)	1/25.4	high	low	high	low	high	low	high	low	high	low	high	low
–	–	**101.600**	4.0000	+406	0	+406	0	+406	−406	+406	−406	+406	−406	+1 524	−1 524
101.600	4.0000	**304.800**	12.0000	+711	−508	+406	−203	+406	−406	+406	−406	+406	−406	+1 524	−1 524
304.800	12.0000	**609.600**	24.0000	+762	−762	+762	−762	+406	−406	+762	−762	–	–	+1 524	−1 524
609.600	24.0000	–	–	+762	−762	–	–	+762	−762	+762	−762	–	–	+1 524	−1 524

Table 7. 5 Tolerances for Magneto Bearings

Table 7. 5. 1 Tolerances for Inner Rings and Width of Outer Rings

Units : μm

Nominal Bore Diameter d (mm)		Δ_{dmp}						V_{dp}			V_{dmp}			Δ_{Bs} (or Δ_{Cs}) (1)				V_{Bs} (or V_{Cs}) (1)		Δ_{Ts}		K_{ia}			S_d	S_{ia}
		Normal Equivalent		Class 6 Equivalent		Class 5 Equivalent		Normal Equivalent	Class 6 Equivalent	Class 5 Equivalent	Normal Equivalent	Class 6 Equivalent	Class 5 Equivalent	Normal Class 6 Equivalent		Class 5 Equivalent		Normal Class 6 Equivalent	Class 5 Equivalent	Normal Class 6 Class 5 Equivalent		Normal Equivalent	Class 6 Equivalent	Class 5 Equivalent	Class 5 Equivalent	Class 5 Equivalent
over	incl.	high	low	high	low	high	low	max.	max.	max.	max.	max.	max.	high	low	high	low	max.	max.	high	low	max.	max.	max.	max.	max.
2.5	**10**	0	−8	0	−7	0	−5	6	5	4	6	5	3	0	−120	0	−40	15	5	+120	−120	10	6	4	7	7
10	**18**	0	−8	0	−7	0	−5	6	5	4	6	5	3	0	−120	0	−80	20	5	+120	−120	10	7	4	7	7
18	**30**	0	−10	0	−8	0	−6	8	6	5	8	6	3	0	−120	0	−120	20	5	+120	−120	13	8	4	8	8

Note (1) The actual width deviation and width variation of an outer ring is determined according to the inner ring of the same bearing.

Remark The bore diameter "no-go side" tolerances (high) specified in this table do not necessarily apply within a distance from the ring face to 1.2 times the chamfer dimension r (max.).

Table 7. 5. 2 Tolerances for Outer Rings

Units : μm

Nominal Outside Diameter D (mm)		Δ_{Dmp}												V_{Dp}			V_{Dmp}			K_{ea}			S_{ea}	S_D
		Bearing Series E						Bearing Series EN																
		Normal Equivalent		Class 6 Equivalent		Class 5 Equivalent		Normal Equivalent		Class 6 Equivalent		Class 5 Equivalent		Normal Equivalent	Class 6 Equivalent	Class 5 Equivalent	Normal Equivalent	Class 6 Equivalent	Class 5 Equivalent	Normal Equivalent	Class 6 Equivalent	Class 5 Equivalent	Class 5 Equivalent	Class 5 Equivalent
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.
6	**18**	+8	0	+7	0	+5	0	0	−8	0	−7	0	−5	6	5	4	6	5	3	15	8	5	8	8
18	**30**	+9	0	+8	0	+6	0	0	−9	0	−8	0	−6	7	6	5	7	6	3	15	9	6	8	8
30	**50**	+11	0	+9	0	+7	0	0	−11	0	−9	0	−7	8	7	5	8	7	4	20	10	7	8	8

Remark The outside diameter "no-go side" tolerances (low) do not necessarily apply within a distance from the ring face to 1.2 times the chamfer dimension r (max.).

Table 7. 6 Tolerances for Thrust Ball Bearings

Table 7. 6. 1 Tolerances for Shaft Washer Bore Diameter and Running Accuracy

Units : μm

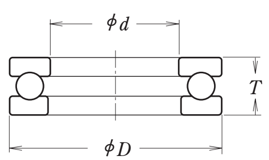
Nominal Bore Diameter d or d_2 (mm)		Δ_{dmp} or Δ_{d2mp}				V_{dsp} or V_{d2sp}		S_i or S_e (1)			
		Normal Class 6 Class 5		Class 4		Normal Class 6 Class 5	Class 4	Normal	Class 6	Class 5	Class 4
over	incl.	high	low	high	low	max.	max.	max.	max.	max.	max.
–	18	0	− 8	0	− 7	6	5	10	5	3	2
18	30	0	− 10	0	− 8	8	6	10	5	3	2
30	50	0	− 12	0	−10	9	8	10	6	3	2
50	80	0	− 15	0	−12	11	9	10	7	4	3
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	–	–	40	25	13	8
800	1 000	0	−100	–	–	–	–	45	30	15	–
1 000	1 250	0	−125	–	–	–	–	50	35	18	–

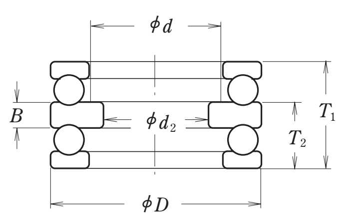
Note (1) For double-direction bearings, the thickness variation does not depend on the bore diameter d_2, but rather on d for single-direction bearings with the same D in the same Diameter Series.
The thickness variation of housing washers S_e applies only to flat-seat thrust bearings.

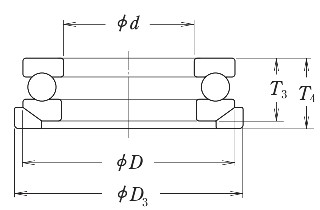
Table 7. 6. 2 Tolerances for Outside Diameter of Housing Washers and Aligning Seat Washers

Units : μm

Nominal Outside Diameter of Bearing or Aligning Seat Washer D or D_3 (mm)		Δ_{Dmp}						V_{Dsp}		Aligning Seat Washer Outside Diameter Deviation Δ_{D3s}	
		Flat Seat Type				Aligning Seat Washer Type					
		Normal Class 6 Class 5		Class 4		Normal Class 6		Normal Class 6 Class 5	Class 4	Normal Class 6	
over	incl.	high	low	high	low	high	low	max.	max.	high	low
10	18	0	− 11	0	− 7	0	− 17	8	5	0	− 25
18	30	0	− 13	0	− 8	0	− 20	10	6	0	− 30
30	50	0	− 16	0	− 9	0	− 24	12	7	0	− 35
50	80	0	− 19	0	−11	0	− 29	14	8	0	− 45
80	120	0	− 22	0	−13	0	− 33	17	10	0	− 60
120	180	0	− 25	0	−15	0	− 38	19	11	0	− 75
180	250	0	− 30	0	−20	0	− 45	23	15	0	− 90
250	315	0	− 35	0	−25	0	− 53	26	19	0	−105
315	400	0	− 40	0	−28	0	− 60	30	21	0	−120
400	500	0	− 45	0	−33	0	− 68	34	25	0	−135
500	630	0	− 50	0	−38	0	− 75	38	29	0	−180
630	800	0	− 75	0	−45	0	−113	55	34	0	−225
800	1 000	0	−100	–	–	–	–	75	–	–	–
1 000	1 250	0	−125	–	–	–	–	–	–	–	–
1 250	1 600	0	−160	–	–	–	–	–	–	–	–







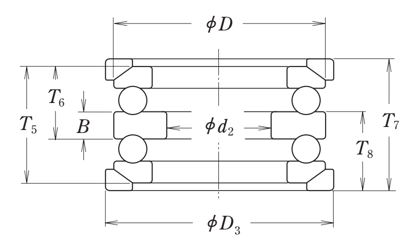


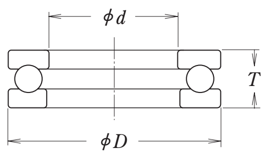
Table 7. 6. 3 Tolerances for Thrust Ball Bearing Height and Central Washer Height

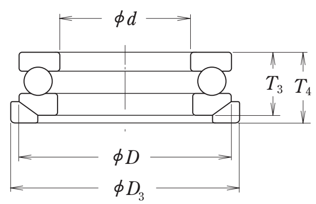
Units : μm

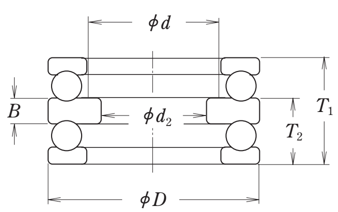
Nominal Bore Diameter d (1) (mm)		Flat Seat Type				Aligning Seat Washer Type				With Aligning Seat Washer				Height Deviation of Central Washer	
		Δ_{Ts} or Δ_{T2s}		Δ_{T1s}		Δ_{T3s} or Δ_{T6s}		Δ_{T5s}		Δ_{T4s} or Δ_{T8s}		Δ_{T7s}		Δ_{Bs}	
		Normal, Class 6 Class 5, Class 4		Normal, Class 6 Class 5, Class 4		Normal Class 6		Normal Class 6		Normal Class 6		Normal Class 6		Normal, Class 6 Class 5, Class 4	
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low
—	**30**	0	− 75	+ 50	−150	0	− 75	+ 50	−150	+ 50	− 75	+150	−150	0	− 50
30	**50**	0	−100	+ 75	−200	0	−100	+ 75	−200	+ 50	−100	+175	−200	0	− 75
50	**80**	0	−125	+100	−250	0	−125	+100	−250	+ 75	−125	+250	−250	0	−100
80	**120**	0	−150	+125	−300	0	−150	+125	−300	+ 75	−150	+275	−300	0	−125
120	**180**	0	−175	+150	−350	0	−175	+150	−350	+100	−175	+350	−350	0	−150
180	**250**	0	−200	+175	−400	0	−200	+175	−400	+100	−200	+375	−400	0	−175
250	**315**	0	−225	+200	−450	0	−225	+200	−450	+125	−225	+450	−450	0	−200
315	**400**	0	−300	+250	−600	0	−300	+250	−600	+150	−275	+550	−550	0	−250

Note (1) For double-direction bearings, classification depends on d for single-direction bearings with the same D in the same Diameter Series.

Remark Δ_{Ts} in the table is the deviation in the respective heights T in the figures below.







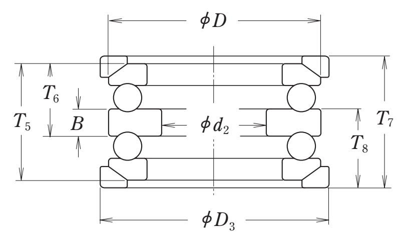


Table 7. 7 Tolerances for Tapered Roller Thrust Bearings

Table 7. 7. 1 Tolerances for Bore Diameters of Shaft Washers and Height (Metric, Normal Class)

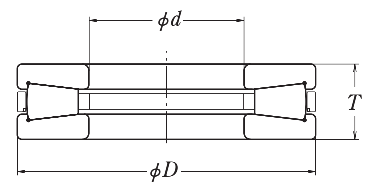
Units : μm

Nominal Bore Diameter d (mm)		Δ_{dmp}		Δ_{Ts}	
over	incl	high	low	high	low
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	−450
630	**800**	0	−75	0	−550
800	**1 000**	0	−100	0	−700
1 000	**1 250**	0	−125	0	−900
1 250	**1 600**	0	−160	0	−1 200

Table 7. 7. 2 Tolerances for Housing Washer Outside Diameters (Metric, Normal Class)

Units : μm

Nominal Outside Diameter D (mm)		Δ_{Dmp}	
over	incl	high	low
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
1 600	**2 000**	0	−200



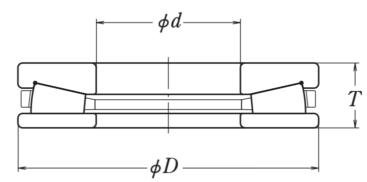


Table 7. 7 Tolerances for Tapered Roller Thrust Bearings

Table 7. 7. 3 Tolerances for Bore Diameters of Shaft Washers and Height (Inch, Class 4)

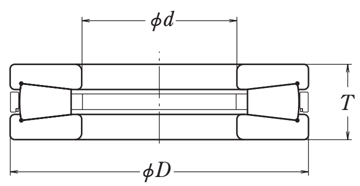
Units : μm

Nominal Bore Diameter d				$\Delta_{d\text{mp}}$		Δ_{Ts}	
over		incl					
(mm)	(inch)	(mm)	(inch)	high	low	high	low
—	—	**304.800**	12.0000	+25	0	+381	−381
304.800	12.0000	**609.600**	24.0000	+51	0	+381	−381
609.600	24.0000	**914.400**	36.0000	+76	0	+381	−381
914.400	36.0000	**1 219.200**	48.0000	+102	0	+381	−381

Table 7. 7. 4 Tolerances for Housing Washer Outside Diameters (Inch, Class 4)

Units : μm

Nominal Outside Diameter D				$\Delta_{D\text{mp}}$	
over		incl			
(mm)	(inch)	(mm)	(inch)	high	low
—	—	**304.800**	12.0000	+25	0
304.800	12.0000	**609.600**	24.0000	+51	0
609.600	24.0000	**914.400**	36.0000	+76	0
914.400	36.0000	**1 219.200**	48.0000	+102	0
1 219.200	48.0000	—	—	+127	0



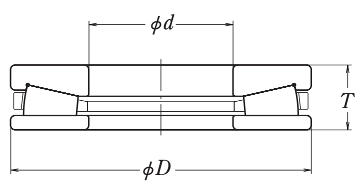


Table 7. 8 Tolerances for Thrust Spherical Roller Bearings

Table 7. 8. 1 Tolerances for Bore Diameters of Shaft Rings and Height (Normal Class)

Units : μm

Nominal Bore Diameter d (mm)		$\Delta_{d\text{mp}}$		$V_{d\text{sp}}$	Reference		
					S_d	Δ_{Ts}	
over	incl.	high	low	max.	max.	high	low
50	**80**	0	−15	11	25	+150	−150
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

Remark The bore diameter "no-go side" tolerances (high) specified in this table do not necessarily apply within a distance from the ring face to 1.2 times the chamfer dimension r (max.).

Table 7. 8. 2 Tolerances for Housing Ring Diameter (Normal Class)

Units : μm

Nominal Outside Diameter D (mm)		$\Delta_{D\text{mp}}$	
over	incl.	high	low
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

Remark The outside diameter "no-go side" tolerances (low) specified in this table do not necessarily apply within a distance from the ring face to 1.2 times the chamfer dimension r (max.).

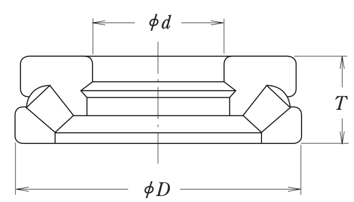


Table 7. 9 Tolerances of Instrument Ball Bearings (Inch Series)

CLASS 5P, CLASS 7P, and CLASS 9P (ANSI/ABMA Equivalent)

(1) Tolerances for Inner Rings and Width of Outer Rings

Units : μm

Nominal Bore Diameter d (mm)		Δ_{dmp}				Δ_{ds}				V_{dp}		V_{dmp}		Δ_{Bs}	
		CLASS 5P CLASS 7P		CLASS 9P		CLASS 5P CLASS 7P		CLASS 9P		CLASS 5P CLASS 7P	CLASS 9P	CLASS 5P CLASS 7P	CLASS 9P	Single Brgs CLASS 5P CLASS 7P CLASS 9P	
over	incl.	high	low	high	low	high	low	high	low	max.	max.	max.	max.	high	low
—	10	0	−5.1	0	−2.5	0	−5.1	0	−2.5	2.5	1.3	2.5	1.3	0	−25.4
10	18	0	−5.1	0	−2.5	0	−5.1	0	−2.5	2.5	1.3	2.5	1.3	0	−25.4
18	30	0	−5.1	0	−2.5	0	−5.1	0	−2.5	2.5	1.3	2.5	1.3	0	−25.4

Nominal Bore Diameter d (mm)		(or Δ_{Cs})		V_{Bs}			K_{ia}			S_{ia}			S_d		
		Combined Brgs (1) CLASS 5P CLASS 7P CLASS 9P		CLASS 5P	CLASS 7P	CLASS 9P	CLASS 5P	CLASS 7P	CLASS 9P	CLASS 5P	CLASS 7P	CLASS 9P	CLASS 5P	CLASS 7P	CLASS 9P
over	incl.	high	low	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.
—	10	0	−400	5.1	2.5	1.3	3.8	2.5	1.3	7.6	2.5	1.3	7.6	2.5	1.3
10	18	0	−400	5.1	2.5	1.3	3.8	2.5	1.3	7.6	2.5	1.3	7.6	2.5	1.3
18	30	0	−400	5.1	2.5	1.3	3.8	3.8	2.5	7.6	3.8	1.3	7.6	3.8	1.3

Note (1) Applicable to bearings for which the axial clearance (preload) is to be adjusted by combining two selected bearings.
Remark Please consult with NSK regarding CLASS 3P and the tolerances of Metric Series instrument ball bearings.

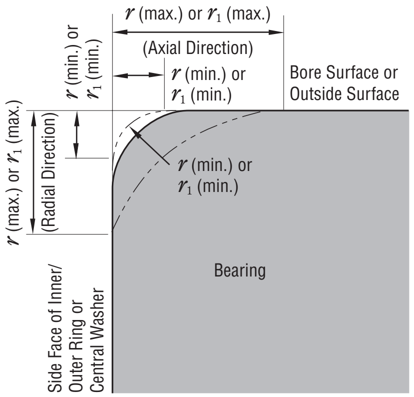
(2) Tolerances for Outer Rings

Units : μm

Nominal Outside Diameter D (mm)		Δ_{Dmp}				Δ_{Ds}						V_{Dp}			V_{Dmp}		
		CLASS 5P CLASS 7P		CLASS 9P		CLASS 5P CLASS 7P				CLASS 9P		CLASS 5P CLASS 7P		CLASS 9P	CLASS 5P CLASS 7P		CLASS 9P
						Open		Shielded Sealed		Open		Open	Shielded Sealed	Open	Open	Shielded Sealed	Open
over	incl.	high	low	high	low	high	low	high	low	high	low	max.	max.	max.	max.	max.	max.
—	18	0	−5.1	0	−2.5	0	−5.1	+1	−6.1	0	−2.5	2.5	5.1	1.3	2.5	5.1	1.3
18	30	0	−5.1	0	−3.8	0	−5.1	+1	−6.1	0	−3.8	2.5	5.1	2	2.5	5.1	2
30	50	0	−5.1	0	−3.8	0	−5.1	+1	−6.1	0	−3.8	2.5	5.1	2	2.5	5.1	2

Nominal Outside Diameter D (mm)		V_{Cs} (1)			S_D			K_{ea}			S_{ea}			Deviation of Flange Outside Diameter Δ_{D1s}		Deviation of Flange Width Δ_{C1s}		Flange Backface Runout with Raceway (2) S_{ea1}
		CLASS 5P	CLASS 7P	CLASS 9P	CLASS 5P	CLASS 7P	CLASS 9P	CLASS 5P	CLASS 7P	CLASS 9P	CLASS 5P	CLASS 7P	CLASS 9P	CLASS 5P CLASS 7P		CLASS 5P CLASS 7P		CLASS 5P CLASS 7P
over	incl.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	high	low	high	low	max.
—	18	5.1	2.5	1.3	7.6	3.8	1.3	5.1	3.8	1.3	7.6	5.1	1.3	0	−25.4	0	−50.8	7.6
18	30	5.1	2.5	1.3	7.6	3.8	1.3	5.1	3.8	2.5	7.6	5.1	2.5	0	−25.4	0	−50.8	7.6
30	50	5.1	2.5	1.3	7.6	3.8	1.3	5.1	5.1	2.5	7.6	5.1	2.5	0	−25.4	0	−50.8	7.6

Notes (1) Applicable to flange width variation for flanged bearings.
(2) Applicable to the flange back face.



r : Chamfer dimension of inner/outer ring
r_1 : Chamfer dimension of inner/outer ring (front side) or of central washer of thrust ball bearings

Remark The precise shape of chamfer surfaces has not been specified but its profile in the axial plane must not intersect an arc of radius r (min.) or r_1 (min.) that touches the side face of an inner ring or central washer and bore surface or the side face of an outer ring and outside surface.

Table 7. 10 Chamfer Dimension Limits (for Metric Series Bearings)

Table 7. 10. 1 Chamfer Dimension Limits for Radial Bearings (Excluding Tapered Roller Bearings)

Units : mm

Permissible Chamfer Dimension for Inner/ Outer Rings r (min.) or r_1 (min.)	Nominal Bore Diameter d		Permissible Chamfer Dimension for Inner/Outer Rings r (max.) or r_1 (max.)		Reference Corner Radius of Shaft or Housing r_a
	over	incl.	Radial Direction	Axial Direction (1)	max.
0.05	–	–	0.1	0.2	0.05
0.08	–	–	0.16	0.3	0.08
0.1	–	–	0.2	0.4	0.1
0.15	–	–	0.3	0.6	0.15
0.2	–	–	0.5	0.8	0.2
0.3	–	40	0.6	1	0.3
	40	–	0.8	1	
0.6	–	40	1	2	0.6
	40	–	1.3	2	
1	–	50	1.5	3	1
	50	–	1.9	3	
1.1	–	120	2	3.5	1
	120	–	2.5	4	
1.5	–	120	2.3	4	1.5
	120	–	3	5	
2	–	80	3	4.5	2
	80	220	3.5	5	
	220	–	3.8	6	
2.1	–	280	4	6.5	2
	280	–	4.5	7	
2.5	–	100	3.8	6	2
	100	280	4.5	6	
	280	–	5	7	
3	–	280	5	8	2.5
	280	–	5.5	8	
4	–	–	6.5	9	3
5	–	–	8	10	4
6	–	–	10	13	5
7.5	–	–	12.5	17	6
9.5	–	–	15	19	8
12	–	–	18	24	10
15	–	–	21	30	12
19	–	–	25	38	15

Note (1) For bearings with nominal widths less than 2 mm, the value of r (max.) in the axial direction is the same as that in the radial direction.

Table 7. 10. 2 Chamfer Dimension Limits for Tapered Roller Bearings

Units : mm

Permissible Chamfer Dimension for Inner/ Outer Rings r (min.)	Nominal Bore or Nominal Outside Diameter (1) d or D		Permissible Chamfer Dimension for Inner/ Outer Rings r (max.)		Reference Corner Radius of Shaft or Housing r_a
	over	incl.	Radial Direction	Axial Direction	max.
0.15	–	–	0.3	0.6	0.15
0.3	–	40	0.7	1.4	0.3
	40	–	0.9	1.6	
0.6	–	40	1.1	1.7	0.6
	40	–	1.3	2	
1	–	50	1.6	2.5	1
	50	–	1.9	3	
1.5	–	120	2.3	3	1.5
	120	250	2.8	3.5	
	250	–	3.5	4	
2	–	120	2.8	4	2
	120	250	3.5	4.5	
	250	–	4	5	
2.5	–	120	3.5	5	2
	120	250	4	5.5	
	250	–	4.5	6	
3	–	120	4	5.5	2.5
	120	250	4.5	6.5	
	250	400	5	7	
	400	–	5.5	7.5	
4	–	120	5	7	3
	120	250	5.5	7.5	
	250	400	6	8	
	400	–	6.5	8.5	
5	–	180	6.5	8	4
	180	–	7.5	9	
6	–	180	7.5	10	5
	180	–	9	11	

Note (1) Inner rings are classified by d and outer rings by D.

Table 7. 10. 3 Chamfer Dimension Limits for Thrust Bearings

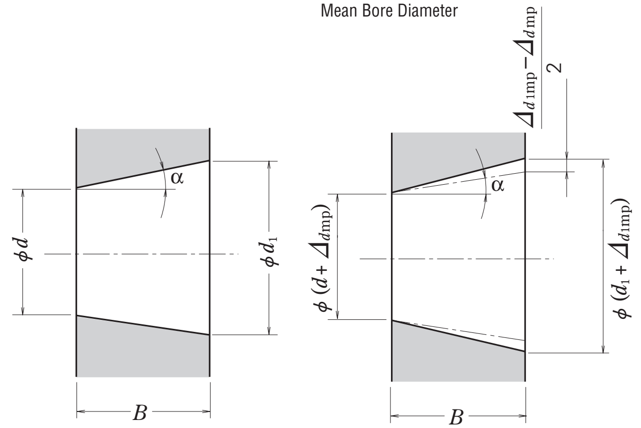
Units : mm

Permissible Chamfer Dimension for Shaft (or Central)/Housing Washers r (min.) or r_1 (min.)	Permissible Chamfer Dimension for Shaft (or Central)/Housing Washers r (max.) or r_1 (max.)	Reference Corner Radius of Shaft or Housing r_a
	Radial and Axial Direction	max.
0.05	0.1	0.05
0.08	0.16	0.08
0.1	0.2	0.1
0.15	0.3	0.15
0.2	0.5	0.2
0.3	0.8	0.3
0.6	1.5	0.6
1	2.2	1
1.1	2.7	1
1.5	3.5	1.5
2	4	2
2.1	4.5	2
3	5.5	2.5
4	6.5	3
5	8	4
6	10	5
7.5	12.5	6
9.5	15	8
12	18	10
15	21	12
19	25	15

Table 7.11 Tolerances for Tapered Bores (Normal Class)

Nominal Tapered Bore

Tapered Bore With Deviation in Single Plane Mean Bore Diameter



d : Nominal bore diameter

d_1 : Theoretical diameter of larger end of tapered bore

Taper 1:12 $d_1 = d + 1/12\,B$ Taper 1:30 $d_1 = d + /30\,B$

Δ_{dmp} : Single plane mean bore diameter deviation in theoretical diameter of smaller end of bore

Δ_{d1mp} : Single plane mean bore diameter deviation in theoretical diameter of larger end of bore

V_{dp} : Bore diameter variation in a single radial plane

B : Nominal inner ring width

α : Half of taper angle of tapered bore

Taper 1:12	Taper 1:30
α =2°23´9.4	α =57´17.4
=2.38594°	=0.95484°
=0.041643 rad	=0.016665 rad

Taper 1 : 12

Units : μm

Nominal Bore Diameter d (mm)		Δ_{dmp}		$\Delta_{d1mp}-\Delta_{dmp}$		V_{dp} (1) (2)
over	incl.	high	low	high	low	max.
18	**30**	+33	0	+21	0	13
30	**50**	+39	0	+25	0	16
50	**80**	+46	0	+30	0	19
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	–

Notes (1) Applicable to all radial planes of tapered bores.
(2) Not applicable to Diameter Series 7 and 8.

Taper 1 : 30

Units : μm

Nominal Bore Diameter d (mm)		Δ_{dmp}		$\Delta_{d1mp}-\Delta_{dmp}$		V_{dp} (1) (2)
over	incl.	high	low	high	low	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

Notes (1) Applicable to all radial planes of tapered bores.
(2) Not applicable to diameter series 7 and 8.

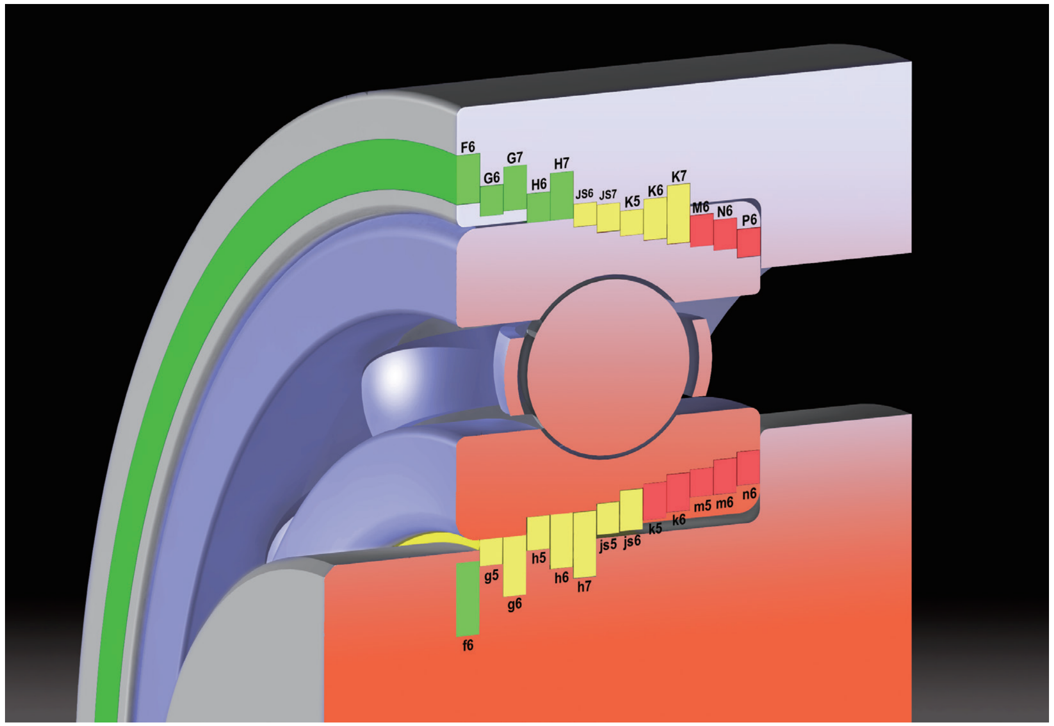
Remark For a value exceeding 630 mm, please contact NSK.

7.2 Selection of Tolerance Classes

For general applications, Normal Class tolerances are adequate in nearly all cases for satisfactory performance, However, bearings with Class 5, 4, or higher tolerances are more suitable for the following applications in Table 7.12.
Example reference applications and appropriate tolerance classes are listed for various bearing requirements and operating conditions.

Table 7.12 Typical Tolerance Classes for Specific Applications (Reference)

Bearing Requirements, Operating Conditions	Example Applications	Tolerance Classes
High Running Accuracy	VTR Drum Spindles	P5
	Magnetic Disk Spindles for Computers	P5, P4, P2
	Machine-Tool Main Spindles	P5, P4, P2
	Rotary Printing Presses	P5
	Rotary Tables of Vertical Presses, etc.	P5, P4
	Roll Necks of Cold Rolling Mill Backup Rolls	Higher than P4
	Slewing Bearings for Parabolic Antennas	Higher than P4
Extra High Speed	Dental Drills	CLASS 7P, CLASS 5P
	Gyroscopes	CLASS 7P, P4
	High Frequency Spindles	CLASS 7P, P4
	Superchargers	P5, P4
	Centrifugal Separators	P5, P4
	Main Shafts of Jet Engines	Higher than P4
Low Torque and Low Torque Variation	Gyroscope Gimbals	CLASS 7P, P4
	Servomechanisms	CLASS 7P, CLASS 5P
	Potentiometric Controllers	CLASS 7P



8. FITS AND INTERNAL CLEARANCES

8. FITS AND INTERNAL CLEARANCES

8.1 Fits

8.1.1 Importance of Proper Fits

In cases where a rolling bearing has its inner ring fitted to the shaft with only slight interference, harmful circumferential slippage may occur between the inner ring and shaft. This slipping of the inner ring, which is called "creep", results in a circumferential displacement of the ring relative to the shaft if the interference fit is not sufficiently tight. When creep occurs, the fitted surfaces become abraded, causing wear and considerable damage to the shaft. Abnormal heating and vibration may also occur due to abrasive metallic particles entering the interior of the bearing.

It is important to prevent creep by having sufficient interference to firmly secure the ring that rotates to either the shaft or housing. Creep cannot always be eliminated using only axial tightening through the bearing ring faces. Generally, it is not necessary to provide interference for rings subjected only to stationary loads. Fits are sometimes made without any interference for either the inner or outer ring to accommodate certain operating conditions or to facilitate mounting and dismounting. In these cases, lubrication or other applicable methods should be considered to prevent damage to the fitting surfaces due to creep.

8.1.2 Selection of Fit

(1) *Load Conditions and Fit*

The proper fit can be selected from Table 8.1 based on the load and operating conditions.

(2) *Magnitude of Load and Interference*

The interference of the inner ring is slightly reduced by bearing load; therefore, the loss of interference should be estimated using the following equations:

$$\left.\begin{aligned}\Delta d_F &= 0.08\sqrt{\frac{d}{B}F_r}\times 10^{-3} \;\ldots\ldots\; \text{(N)}\\ \Delta d_F &= 0.25\sqrt{\frac{d}{B}F_r}\times 10^{-3} \;\ldots\; \{\text{kgf}\}\end{aligned}\right\}\;\ldots\;\textbf{(8.1)}$$

where Δd_F : Interference decrease of inner ring (mm)
d : Bearing bore diameter (mm)
B : Nominal inner ring width (mm)
F_r : Radial load applied on bearing (N), {kgf}

Therefore, the effective interference Δd should be larger than the interference given by Equation (8.1).

However, interference often becomes insufficient with heavy loads where the radial load exceeds 20% of the basic static load rating C_{0r} under normal operating conditions. In these cases, interference should be estimated using Equation (8.2):

$$\left.\begin{aligned}\Delta d &\geqq 0.02\frac{F_r}{B}\times 10^{-3} \;\ldots\ldots\; \text{(N)}\\ \Delta d &\geqq 0.2\frac{F_r}{B}\times 10^{-3} \;\ldots\ldots\; \{\text{kgf}\}\end{aligned}\right\}\;\ldots\ldots\;\textbf{(8.2)}$$

where Δd : Effective interference (mm)
F_r : Radial load applied on bearing (N), {kgf}
B : Nominal inner ring width (mm)

Creep experiments conducted by NSK with NU219 bearings showed a linear relation between radial load (load at creep occurrence limit) and required effective interference. It was confirmed that this line agrees well with the straight line of Equation (8.2).

When subjected to loads heavier than 0.25 C_{0r}, the interference given by Equation (8.1) for NU219 bearings becomes insufficient and creep occurs.

Generally speaking, the necessary interference for loads heavier than 0.25 C_{0r} should be calculated using Equation (8.2). When doing this, verify that the fit does not cause excessive circumferential stress.

Calculation example

For NU219, B = 32 (mm) and assume
F_r = 98 100 N
C_{0r} = 183 000 N

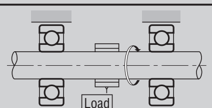
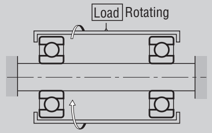
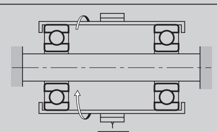
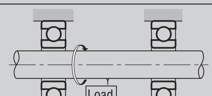
$$\frac{F_r}{C_{0r}} = \frac{98\ 100}{183\ 000} = 0.536 > 0.2$$

Therefore, the required effective interference is calculated using Equation (8.2).

$$\Delta d = 0.02\times\frac{98\ 100}{32}\times 10^{-3} = 0.061\ \text{(mm)}$$

This result agrees well with Fig. 8.1.

Table 8.1 Loading Conditions and Fits

Load Application	Bearing Operation: Inner Ring	Bearing Operation: Outer Ring	Load Conditions	Fitting: Inner Ring	Fitting: Outer Ring
 Load Stationary	Rotating	Stationary	Rotating Inner Ring Load / Stationary Outer Ring Load	Tight Fit	Loose Fit
 Load Rotating	Stationary	Rotating	Rotating Inner Ring Load / Stationary Outer Ring Load	Tight Fit	Loose Fit
 Load Stationary	Stationary	Rotating	Rotating Outer Ring Load / Stationary Inner Ring Load	Loose Fit	Tight Fit
 Load Rotating	Rotating	Stationary	Rotating Outer Ring Load / Stationary Inner Ring Load	Loose Fit	Tight Fit
Indeterminate load direction due to variation of direction or unbalanced load	Rotating or Stationary	Rotating or Stationary	Direction of Load Indeterminate	Tight Fit	Tight Fit

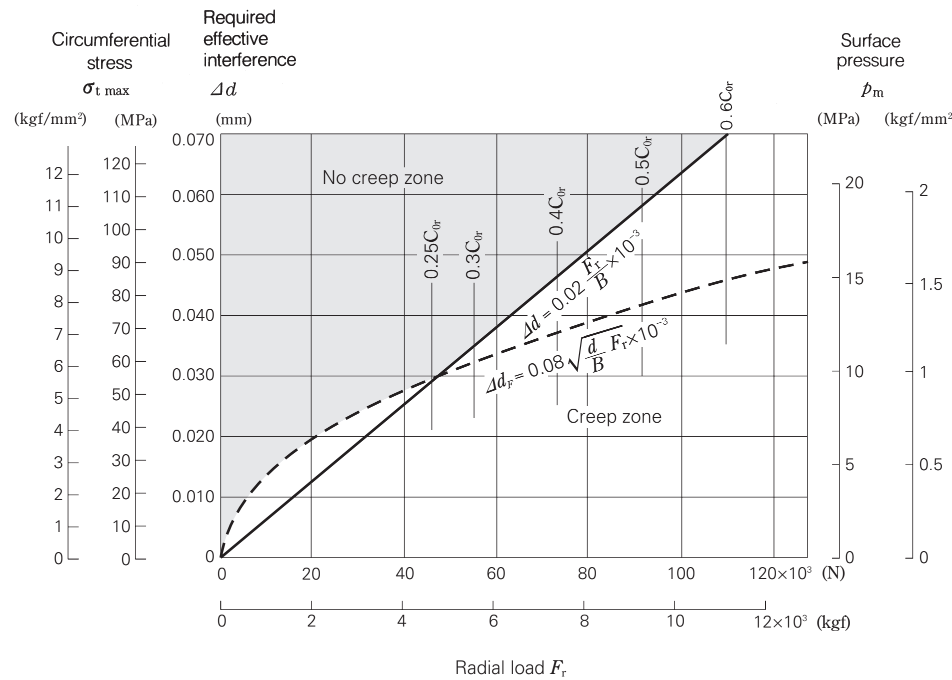


Fig. 8.1 Load and Required Effective Interference for Fit

(3) *Interference Variation Caused by Temperature Differences Between Bearing and Shaft or Housing*

The effective interference decreases due to the increasing bearing temperature during operation. If the temperature difference between the bearing and housing is ΔT (°C), then the temperature difference between the fitted surfaces of the shaft and inner ring is estimated to be about (0.1–0.15) ΔT when the shaft is cooled. The decrease in the interference of the inner ring due to this temperature difference Δd_T may be calculated using Equation (8.3):

$$\Delta d_T = (0.10 \text{ to } 0.15) \times \Delta T \cdot \alpha \cdot d$$
$$\fallingdotseq 0.0015 \Delta T \cdot d \times 10^{-3} \quad \cdots\cdots \textbf{(8.3)}$$

where Δd_T : Decrease in interference of inner ring due to temperature difference (mm)
ΔT : Temperature difference between bearing interior and surrounding parts (°C)
α : Coefficient of linear expansion of bearing steel $\fallingdotseq 12.5 \times 10^{-6}$ (1/°C)
d : Bearing nominal bore diameter (mm)

In addition, depending on the temperature difference between the outer ring and housing, or difference in their coefficients of linear expansion, interference may increase.

(4) *Effective Interference and Finish of Shaft and Housing*

Since the roughness of fitted surfaces is reduced during fitting, the effective interference becomes less than the apparent interference. The amount of this interference decrease varies depending on the roughness of the surfaces and may be estimated using the following equations:

For ground shafts $$\Delta d = \frac{d}{d+2} \Delta d_a \quad \cdots\cdots \textbf{(8.4)}$$

For machined shafts $$\Delta d = \frac{d}{d+3} \Delta d_a \quad \cdots\cdots \textbf{(8.5)}$$

where Δd : Effective interference (mm)
Δd_a : Apparent interference (mm)
d : Bearing nominal bore diameter (mm)

According to Equations (8.4) and (8.5), the effective interference of bearings with a bore diameter of 30 to 150 mm is about 95% of the apparent interference.

(5) *Fitting Stress and Ring Expansion and Contraction*

When bearings are mounted with interference on a shaft or in a housing, the rings either expand or contract and stress is produced. Excessive interference may damage the bearings; therefore, as a general rule, the maximum interference should be kept under approximately 7/10 000 the shaft diameter.
The pressure between fitted surfaces, expansion or contraction of the rings, and circumferential stress may be calculated using the equations in Table 8.2.

Table 8.2 Fit Conditions

	Inner Ring and Shaft	Outer Ring and Housing
Surface Pressure p_m (MPa) {kgf/mm²}	Hollow shaft $p_m = \frac{\Delta d}{d} \frac{1}{\left[\frac{m_s-1}{m_s E_s} - \frac{m_i-1}{m_i E_i}\right] + 2\left[\frac{k_0^2}{E_s(1-k_0^2)} + \frac{1}{E_i(1-k^2)}\right]}$ Solid shaft $p_m = \frac{\Delta d}{d} \frac{1}{\left[\frac{m_s-1}{m_s E_s} - \frac{m_i-1}{m_i E_i}\right] + \frac{2}{E_i(1-k^2)}}$	Housing outside diameter $p_m = \frac{\Delta D}{D} \frac{1}{\left[\frac{m_e-1}{m_e E_e} - \frac{m_h-1}{m_h E_h}\right] + 2\left[\frac{h^2}{E_e(1-h^2)} + \frac{1}{E_h(1-h_0^2)}\right]}$
Expansion of Inner Ring Raceway ΔD_i (mm) Contraction of Outer Ring Raceway ΔD_e (mm)	$\Delta D_i = 2d \frac{p_m}{E_i} \frac{k}{1-k^2}$ $= \Delta d \cdot k \frac{1-k_0^2}{1-k^2 k_0^2}$ (hollow shaft) $= \Delta d \cdot k$ (solid shaft)	$\Delta D_e = 2D \frac{p_m}{E_e} \frac{h}{1-h^2}$ $= \Delta D \cdot h \frac{1-h_0^2}{1-h^2 h_0^2}$
Maximum Stress $\sigma_{t\,max}$ (MPa) {kgf/mm²}	Maximum circumferential stress at inner ring bore fitting surface. $\sigma_{t\,max} = p_m \frac{1+k^2}{1-k^2}$	Maximum circumferential stress at outer ring outer surface. $\sigma_{t\,max} = p_m \frac{2}{1-h^2}$
Symbols	d : Shaft diameter, inner ring bore d_0 : Hollow shaft bore D_i : Inner ring raceway diameter $k = d/D_i$, $k_0 = d_0/d$ E_i : Inner ring Young's modulus, 208 000 MPa {21 200 kgf/mm²} E_s : Shaft Young's modulus m_i : Inner ring Poisson's number, 3.33 m_s : Shaft Poisson's number	D : Housing bore diameter, outer ring outside diameter D_0 : Housing outside diameter D_e : Outer ring raceway diameter $h = D_e/D$, $h_0 = D/D_0$ E_e : Outer ring Young's modulus, 208 000 MPa {21 200 kgf/mm²} E_h : Housing Young's modulus m_e : Outer ring Poisson's number, 3.33 m_h : Housing Poisson's number

(6) *Surface Pressure and Maximum Stress on Fitting Surfaces*

In order for rolling bearings to achieve their full life expectancy, their fitting must be appropriate. Usually an interference fit is chosen for a rotating inner ring, and a loose fit is used for a fixed outer ring. To select the fit, the magnitude of the load, the temperature differences among the bearing and shaft and housing, material characteristics of the shaft and housing, level of finish, material thickness, and bearing mounting/dismounting method must all be considered.

If the interference is insufficient for the operating conditions, ring loosening, creep, fretting, heat generation, or other problems may occur. If the interference is excessive, the ring may crack due to circumferential stress. The magnitude of the interference is usually satisfactory if it follows recommendations for the size of the shaft or housing listed in the bearing catalog. To determine surface pressure and stress on the fitting surfaces, calculations can be made assuming a thick-walled cylinder with uniform internal and external pressures; necessary equations for this are summarized in Table 8.2. For convenience in fitting bearing inner rings on solid steel shafts, which are the most common type of shaft, the surface pressure and maximum stress are shown in Figs. 8.3 and 8.4.

Fig. 8.3 shows the surface pressure p_m and maximum stress $\sigma_{t\ max}$ for given combinations of bearing bores for mean interference at various tolerance grades. Fig. 8.4 shows the maximum surface pressure p_m and maximum stress $\sigma_{t\ max}$ when maximum interference occurs.

Fig. 8.4 is convenient for checking if $\sigma_{t\ max}$ exceeds acceptable limits. The tensile strength of hardened bearing steel is about 1 570 to 1 960 MPa {160 to 200 kgf/mm^2}. However, for safety, plan for a maximum fitting stress of 127 MPa {13 kgf/mm^2}. For reference, the distributions of circumferential stress σ_t and radial stress σ_r in an inner ring are shown in Fig. 8.2.

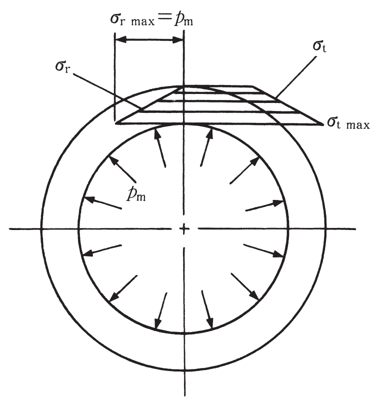


Fig. 8.2 Distribution of Circumferential Stress σ_t and Radial Stress σ_r

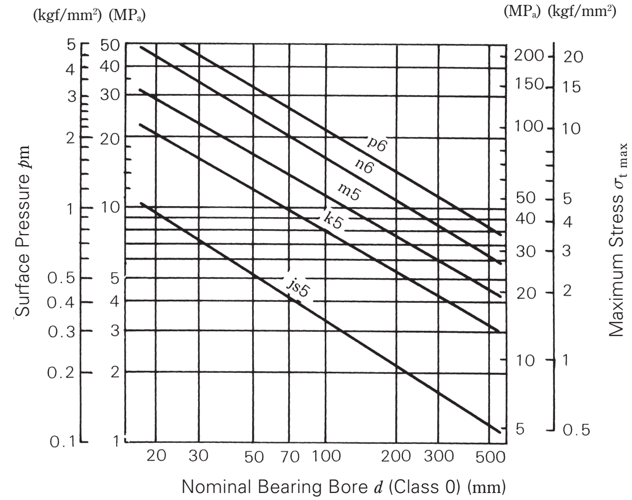


Fig. 8.3 Surface Pressure p_m and Maximum Stress $\sigma_{t\ max}$ for Mean Interference in Various Tolerance Grades

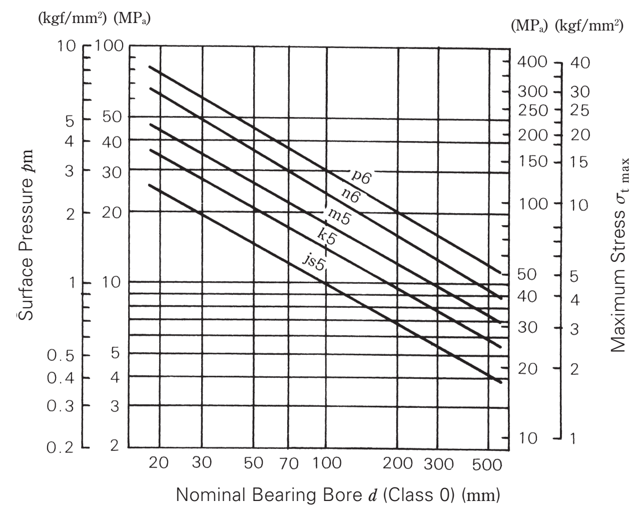


Fig. 8.4 Surface Pressure p_m and Maximum Stress $\sigma_{t\ max}$ for Maximum Interference in Various Tolerance Grades

(7) *Press-Fit Force and Withdrawal Force*

The force needed to mount bearings on shafts or in a housing hole with interference can be obtained using the thick-walled cylinder theory.

The press-fit force (or withdrawal force) depends upon the contact area, surface pressure, and coefficient of friction between the fitting surfaces.

The press-fit force (or withdrawal force) K needed to mount inner rings on shafts is given by Equation (8.6).

$$K=\mu\, p_m\, \pi\, d\, B \text{ (N), \{kgf\}} \quad \cdots\cdots \quad \textbf{(8.6)}$$

where μ: Coefficient of friction between fitting surfaces
μ=0.12 (for press-fitting)
μ=0.18 (for withdrawal)

p_m: Surface pressure (MPa), {kgf/mm²}
For example, inner ring surface pressure can be obtained using Table 8.2.

$$p_m=\frac{E}{2}\,\frac{\Delta d}{d}\,\frac{(1-k^2)\,(1-k_0^{\,2})}{1-k^2\,k_0^{\,2}}$$

d: Shaft diameter (mm)
B: Bearing width (mm)
Δd: Effective interference (mm)
E: Young's modulus of steel (MPa), {kgf/mm²}
E=208 000 MPa {21 200 kgf/mm²}
k: Inner ring thickness ratio
$k=d/D_i$
D_i: Inner ring raceway diameter (mm)
k_0: Hollow shaft thickness ratio
$k_0=d_0/d$
d_0: Bore diameter of hollow shaft (mm)

For solid shafts, d_0=0, consequently k_0=0. The value of k varies depending on bearing type and size, but it usually ranges between k=0.7 and 0.9. Assuming that k=0.8 and the shaft is solid, Equation (8.6) becomes the following:

$$\left.\begin{aligned} K &= 118\,000\mu\ \Delta d\ B \text{ (N)} \\ &= 12\,000\mu\ \Delta d\ B \text{ \{kgf\}} \end{aligned}\right\} \quad \cdots\cdots \quad \textbf{(8.7)}$$

Equation (8.7) is shown graphically in Fig. 8.5. The press-fit and withdrawal forces for outer rings and housings have also been calculated and the results are shown in Fig. 8.6.

The actual press-fit and withdrawal forces can become much higher than the calculated values if the bearing ring and shaft (or housing) are slightly misaligned or load is applied unevenly to the circumference of the bearing ring. Consequently, the values obtained from Figs. 8.5 and 8.6 should be considered only as guides when designing withdrawal tools. Tool strength should be five to six times higher than that indicated by the figures.

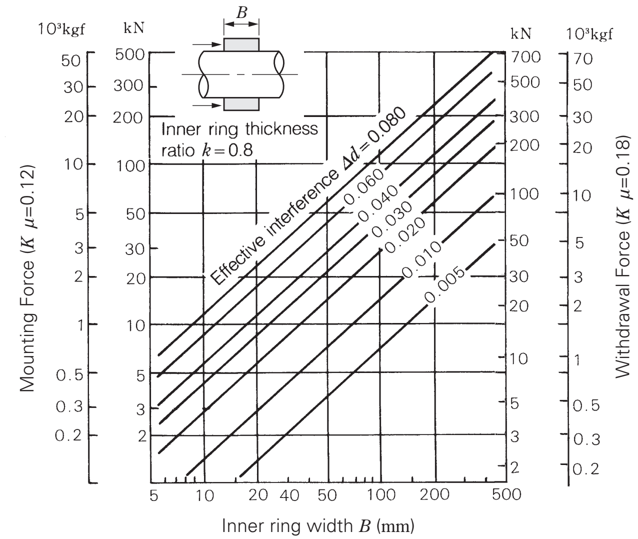


Fig. 8.5 Press-Fit and Withdrawal Forces for Inner Rings

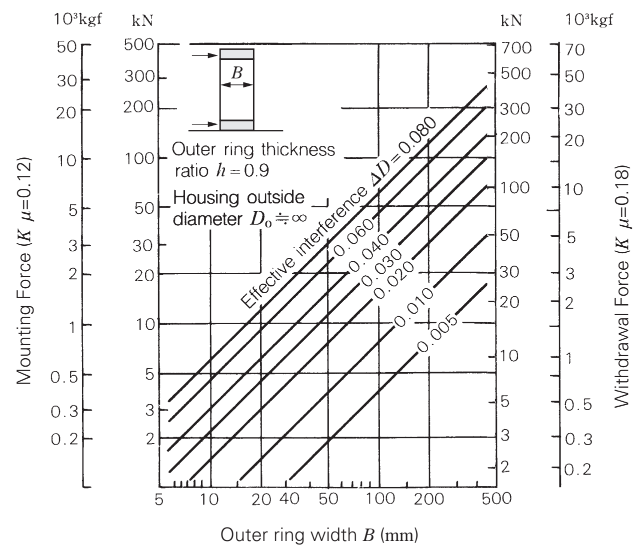


Fig. 8.6 Press-Fit and Withdrawal Forces for Outer Rings

8.1.3 Recommended Fits

As described previously, many factors, such as the characteristics and magnitude of bearing load, temperature differences, and means of bearing mounting and dismounting, must be considered when selecting the proper fit.

If the housing is thin or the bearing is mounted on a hollow shaft, a tighter than usual fit is necessary. A split housing often deforms the bearing into an oval shape; therefore, a split housing should be avoided when a tight fit with the outer ring is required.

The fits of both the inner and outer rings should be tight in applications where the shaft is subjected to considerable vibration.

The recommended fits for some common applications are shown in Tables 8.3 to 8.8. Please consult NSK regarding unusual operating conditions. For the tolerances and surface finish of shafts and housings, please refer to Section 13.1 (Page A270).

Table 8.3 Fits of Radial Bearings (Normal Class, Class 6X, and Class 6) With Shafts

Load Conditions		Examples	Shaft Diameter (mm) Ball Brgs	Shaft Diameter (mm) Cylindrical Roller Brgs, Tapered Roller Brgs	Shaft Diameter (mm) Spherical Roller Brgs	Tolerance of Shaft	Remarks
Radial Bearings With Cylindrical Bores							
Rotating Outer Ring Load	Easy axial displacement of inner ring on shaft is necessary.	Wheels on Stationary Axles	All Shaft Diameters			g6	Use g5 and h5 where accuracy is required. f6 can be used in large bearings to allow easy axial movement.
	Easy axial displacement of inner ring on shaft is unnecessary.	Tension Pulleys Rope Sheaves	All Shaft Diameters			h6	
Rotating Inner Ring Load or Indeterminate Direction of Load	Light Loads or Variable Loads ($<0.06C_r$(1))	Electrical Home Appliances Pumps, Blowers, Transport Vehicles, Precision Machinery, Machine Tools	<18	—	—	js5	Use Class 5 and high-precision bearings where accuracy is required. Use h5 for high-precision ball bearings with bore diameters of 18 mm or less.
			18 to 100	<40	—	js6(j6)	
			100 to 200	40 to 140	—	k6	
			—	140 to 200	—	m6	
	Normal Loads (0.06 to $0.13C_r$(1))	General Bearing Applications, Medium and Large Motors(3), Turbines, Pumps, Engine Main Bearings, Gears, Woodworking Machines	<18	—	—	js5 or js6 (j5 or j6)	k6 and m6 can be used for single-row tapered roller bearings and single-row angular contact ball bearings instead of k5 and m5.
			18 to 100	<40	<40	k5 or k6	
			100 to 140	40 to 100	40 to 65	m5 or m6	
			140 to 200	100 to 140	65 to 100	m6	
			200 to 280	140 to 200	100 to 140	n6	
			—	200 to 400	140 to 280	p6	
			—	—	280 to 500	r6	
			—	—	over 500	r7	
	Heavy Loads or Shock Loads ($>0.13C_r$(1))	Railway Axleboxes, Industrial Vehicles, Traction Motors, Construction Equipment, Crushers	—	50 to 140	50 to 100	n6	A bearing internal clearance greater than CN is necessary.
			—	140 to 200	100 to 140	p6	
			—	over 200	140 to 200	r6	
			—	—	200 to 500	r7	
Axial Loads Only		All Types of Bearing Applications	All Shaft Diameters			js6 (j6)	—
Radial Bearings With Tapered Bores and Sleeves							
All Types of Loading		General Bearing Applications, Railway Axleboxes	All Shaft Diameters			h9/IT5(2)	The deviation of the shaft from its true geometric form, e. g. roundness and cylindricity should be within the tolerances of IT5 and IT7.
		Transmission Shafts, Woodworking Spindles	All Shaft Diameters			h10/IT7(2)	

Notes (1) C_r represents the basic load rating of the bearing.
(2) Refer to Appendix Table 11 on Page E016 for the values of IT standard tolerance grades.
(3) Refer to Tables 8.14.1 and 8.14.2 for the recommended fits of shafts used in electric motors for deep groove ball bearings with bore diameters ranging from 10 mm to 160 mm and for cylindrical roller bearings with bore diameters ranging from 24 mm to 200 mm.

Remark This table applies only to solid steel shafts.

Table 8.4 Fits of Thrust Bearings With Shafts

Load Conditions		Examples	Shaft Diameter (mm)	Tolerance of Shaft	Remarks
Central Axial Load Only		Main Shafts of Lathes	All Shaft Diameters	h6 or js6 (j6)	—
Combined Radial and Axial Loads (Spherical Thrust Roller Bearings)	Stationary Inner Ring Load	Cone Crushers	All Shaft Diameters	js6 (j6)	
	Rotating Inner Ring Load or Indeterminate Direction of Load	Paper Pulp Refiners, Plastic Extruders	<200	k6	
			200 to 400	m6	
			over 400	n6	

Table 8.5 Fits of Radial Bearings (Normal Class, Class 6X, and Class 6) With Housings

Load Conditions			Examples	Tolerances for Housing Bores	Axial Displacement of Outer Ring	Remarks
Solid Housings	Rotating Outer Ring Load	Heavy Loads on Bearing in Thin-Walled Housing or Heavy Shock Loads	Automotive Wheel Hubs (Roller Bearings) Crane Travelling Wheels	P7	Impossible	—
		Normal or Heavy Loads	Automotive Wheel Hubs (Ball Bearings) Vibrating Screens	N7		
		Light or Variable Loads	Conveyor Rollers Rope Sheaves Tension Pulleys	M7		
	Indeterminate Direction of Load	Heavy Shock Loads	Traction Motors			
		Normal or Heavy Loads	Pumps Crankshaft Main Bearings Medium and Large Motors(1)	K7	Generally Impossible	If axial displacement of the outer ring is not required.
Solid or Split Housings		Normal or Light Loads		JS7 (J7)	Possible	Axial displacement of outer ring is necessary.
	Rotating Inner Ring Load	Any Kind of Load	General Bearing Applications, Railway Axleboxes	H7	Easily possible	—
		Normal or Light Loads	Plummer Blocks	H8		
		High Temperature Rise of Inner Ring Through Shaft	Paper Dryers	G7		
Solid Housing		Accurate Running Required Under Normal or Light Loads	Grinding Spindle Rear Ball Bearings High Speed Centrifugal Compessor Free Bearings	JS6 (J6)	Possible	—
	Indeterminate Direction of Load		Grinding Spindle Front Ball Bearings High Speed Centrifugal Compressor Fixed Bearings	K6	Generally Impossible	For heavy loads, an interference fit tighter than K may be used. When high accuracy is required, very strict tolerances should be used for fitting.
	Rotating Inner Ring Load	Accurate Running and High Rigidity Required Under Variable Loads	Cylindrical Roller Bearings for Machine Tool Main Spindle	M6 or N6	Impossible	
		Minimal noise is required.	Electrical Home Appliances	H6	Easily Possible	—

Note (1) Refer to Tables 8.14.1 and 8.14.2 for the recommended fits of housing bores of deep groove ball bearings and cylindrical roller bearings for electric motors.

Remarks 1. This table is applicable to cast-iron and steel housings. For housings made of light alloys, the interference should be tighter than listed in this table.
2. Refer to the introductory section of the bearing tables for special fits, such as those for drawn cup needle roller bearings.

Table 8.6 Fits of Thrust Bearings With Housings

Load Conditions		Bearing Types	Tolerances for Housing Bores	Remarks
Axial Loads Only		Thrust Ball Bearings	Clearance over 0.25mm	For General Applications
			H8	When precision is required
		Spherical Thrust Roller Bearings Steep Angle Tapered Roller Bearings	Outer ring has radial clearance.	When radial loads are sustained by other bearings
Combined Radial and Axial Loads	Stationary Outer Ring Loads	Spherical Thrust Roller Bearings	H7 or JS7 (J7)	—
	Rotating Outer Ring Loads or Indeterminate Direction of Load		K7	Normal Loads
			M7	Relatively Heavy Radial Loads

Table 8.7 Fits of Inch Series Tapered Roller Bearings With Shafts

(1) Bearings of Precision Classes 4 and 2

Units : μm

Operating Conditions		Nominal Bore Diameters d over (mm)	over 1/25.4	incl. (mm)	incl. 1/25.4	Bore Diameter Tolerances Δ_{ds} high	low	Shaft Diameter Tolerances high	low	Remarks
Rotating Inner Ring Loads	Normal Loads	—		76.200	3.0000	+13	0	+ 38	+ 25	For bearings with $d \leqq 152.4$ mm, clearance is usually larger than CN.
		76.200	3.0000	304.800	12.0000	+25	0	+ 64	+ 38	
		304.800	12.0000	609.600	24.0000	+51	0	+127	+ 76	
		609.600	24.0000	914.400	36.0000	+76	0	+190	+114	
	Heavy Loads Shock Loads High Speeds	—		76.200	3.0000	+13	0	+ 64	+ 38	In general, bearings with a clearance larger than CN are used. ※ indicates that the average interference is about 0.0005 d.
		76.200	3.0000	304.800	12.0000	+25	0	※		
		304.800	12.0000	609.600	24.0000	+51	0	※		
		609.600	24.0000	914.400	36.0000	+76	0	+381	+305	
Rotating Outer Ring Loads	Normal Loads Without Shocks	—		76.200	3.0000	+13	0	+ 13	0	The inner ring cannot be displaced axially. When heavy or shock loads exist, the figures above (rotating inner ring loads, heavy or shock loads) apply.
		76.200	3.0000	304.800	12.0000	+25	0	+ 25	0	
		304.800	12.0000	609.600	24.0000	+51	0	+ 51	0	
		609.600	24.0000	914.400	36.0000	+76	0	+ 76	0	
		—		76.200	3.0000	+13	0	0	− 13	The inner ring can be displaced axially.
		76.200	3.0000	304.800	12.0000	+25	0	0	− 25	
		304.800	12.0000	609.600	24.0000	+51	0	0	− 51	
		609.600	24.0000	914.400	36.0000	+76	0	0	− 76	

(2) Bearings of Precision Classes 3 and 0 ([1])

Units : μm

Operating Conditions		Nominal Bore Diameters d over (mm)	over 1/25.4	incl. (mm)	incl. 1/25.4	Bore Diameter Tolerances Δ_{ds} high	low	Shaft Diameter Tolerances high	low	Remarks
Rotating Inner Ring Loads	Precision Machine-Tool Main Spindles	—		76.200	3.0000	+13	0	+ 30	+18	—
		76.200	3.0000	304.800	12.0000	+13	0	+ 30	+18	
		304.800	12.0000	609.600	24.0000	+25	0	+ 64	+38	
		609.600	24.0000	914.400	36.0000	+38	0	+102	+64	
	Heavy Loads Shock Loads High Speeds	—		76.200	3.0000	+13	0	—		A minimum interference of about 0.00025 d is used.
		76.200	3.0000	304.800	12.0000	+13	0	—		
		304.800	12.0000	609.600	24.0000	+25	0	—		
		609.600	24.0000	914.400	36.0000	+38	0	—		
Rotating Outer Ring Loads	Precision Machine-Tool Main Spindles	—		76.200	3.0000	+13	0	+ 30	+18	—
		76.200	3.0000	304.800	12.0000	+13	0	+ 30	+18	
		304.800	12.0000	609.600	24.0000	+25	0	+ 64	+38	
		609.600	24.0000	914.400	36.0000	+38	0	+102	+64	

Note ([1]) For bearings with d greater than 304.8 mm, Class 0 does not exist.

Table 8.8 Fits of Inch Series Tapered Roller Bearings With Housings

(1) Bearings of Precision Classes 4 and 2

Units : μm

Operating Conditions		Nominal Outside Diameters D over (mm)	over 1/25.4	incl. (mm)	incl. 1/25.4	Outside Diameter Tolerances Δ_{Ds} high	low	Housing Bore Diameter Tolerances high	low	Remarks
Rotating Inner Ring Loads	Used either on free-end or fixed-end	—		76.200	3.0000	+25	0	+ 76	+ 51	The outer ring can be easily displaced axially.
		76.200	3.0000	127.000	5.0000	+25	0	+ 76	+ 51	
		127.000	5.0000	304.800	12.0000	+25	0	+ 76	+ 51	
		304.800	12.0000	609.600	24.0000	+51	0	+152	+102	
		609.600	24.0000	914.400	36.0000	+76	0	+229	+152	
	The outer ring position can be adjusted axially.	—		76.200	3.0000	+25	0	+ 25	0	The outer ring can be displaced axially.
		76.200	3.0000	127.000	5.0000	+25	0	+ 25	0	
		127.000	5.0000	304.800	12.0000	+25	0	+ 51	0	
		304.800	12.0000	609.600	24.0000	+51	0	+ 76	+ 25	
		609.600	24.0000	914.400	36.0000	+76	0	+127	+ 51	
	The outer ring position cannot be adjusted axially.	—		76.200	3.0000	+25	0	− 13	− 38	Generally, the outer ring is fixed axially.
		76.200	3.0000	127.000	5.0000	+25	0	− 25	− 51	
		127.000	5.0000	304.800	12.0000	+25	0	− 25	− 51	
		304.800	12.0000	609.600	24.0000	+51	0	− 25	− 76	
		609.600	24.0000	914.400	36.0000	+76	0	− 25	−102	
Rotating Outer Ring Loads	Normal Loads The outer ring position cannot be adjusted axially.	—		76.200	3.0000	+25	0	− 13	− 38	The outer ring is fixed axially.
		76.200	3.0000	127.000	5.0000	+25	0	− 25	− 51	
		127.000	5.0000	304.800	12.0000	+25	0	− 25	− 51	
		304.800	12.0000	609.600	24.0000	+51	0	− 25	− 76	
		609.600	24.0000	914.400	36.0000	+76	0	− 25	−102	

(2) Bearings of Precision Classes 3 and 0 ([1])

Units : μm

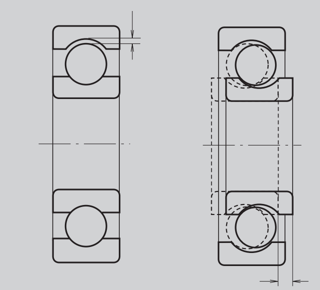
Operating Conditions		Nominal Outside Diameters D over (mm)	over 1/25.4	incl. (mm)	incl. 1/25.4	Outside Diameter Tolerances Δ_{Ds} high	low	Housing Bore Diameter Tolerances high	low	Remarks
Rotating Inner Ring Loads	Used on free-end	—		152.400	6.0000	+13	0	+38	+25	The outer ring can be easily displaced axially.
		152.400	6.0000	304.800	12.0000	+13	0	+38	+25	
		304.800	12.0000	609.600	24.0000	+25	0	+64	+38	
		609.600	24.0000	914.400	36.0000	+38	0	+89	+51	
	Used on fixed-end	—		152.400	6.0000	+13	0	+25	+13	The outer ring can be displaced axially.
		152.400	6.0000	304.800	12.0000	+13	0	+25	+13	
		304.800	12.0000	609.600	24.0000	+25	0	+51	+25	
		609.600	24.0000	914.400	36.0000	+38	0	+76	+38	
	The outer ring position can be adjusted axially.	—		152.400	6.0000	+13	0	+13	0	Generally, the outer ring is fixed axially.
		152.400	6.0000	304.800	12.0000	+13	0	+25	0	
		304.800	12.0000	609.600	24.0000	+25	0	+25	0	
		609.600	24.0000	914.400	36.0000	+38	0	+38	0	
	The outer ring position cannot be adjusted axially.	—		152.400	6.0000	+13	0	0	−13	The outer ring is fixed axially.
		152.400	6.0000	304.800	12.0000	+13	0	0	−25	
		304.800	12.0000	609.600	24.0000	+25	0	0	−25	
		609.600	24.0000	914.400	36.0000	+38	0	0	−38	
Rotating Outer Ring Loads	Normal Loads The outer ring position cannot be adjusted axially.	—		76.200	3.0000	+13	0	−13	−25	The outer ring is fixed axially.
		76.200	3.0000	152.400	6.0000	+13	0	−13	−25	
		152.400	6.0000	304.800	12.0000	+13	0	−13	−38	
		304.800	12.0000	609.600	24.0000	+25	0	−13	−38	
		609.600	24.0000	914.400	36.0000	+38	0	−13	−51	

Note ([1]) For bearings with D greater than 304.8 mm, Class 0 does not exist.

8.2 Bearing Internal Clearances

8.2.1 Internal Clearances and Their Standards

The internal clearance of rolling bearings in operation greatly influences bearing performance including fatigue life, vibration, noise, heat generation, etc. Consequently, the selection of proper internal clearance is one of the most important tasks when choosing a bearing after the type and size have been determined.
This bearing internal clearance refers to the combined clearances between the inner/outer rings and rolling elements. The radial and axial internal clearances are defined as the total amount that one ring can be displaced relative to the other in the radial and axial directions respectively (Fig. 8.7).



Radial internal Clearance   Axial internal Clearance

Fig. 8.7 Bearing Internal Clearance

To obtain accurate measurements, the clearance is generally measured by applying a specified measuring load on the bearing. This "measured clearance" is always slightly larger than the theoretical internal clearance ("geometrical clearance" for radial bearings) by the amount of elastic deformation caused by the measuring load.
Therefore, the theoretical internal clearance may be obtained by correcting the measured clearance by the amount of elastic deformation. However, in the case of roller bearings, this elastic deformation is negligibly small.
Usually the clearance before mounting is specified by the theoretical internal clearance.
In Table 8.9, reference table and page numbers are listed by bearing types.

Table 8.9 Index for Radial Internal Clearances by Bearing Type

Bearing Type		Table Number	Page Number
Deep Groove Ball Bearings		8.10	A169
Extra Small and Miniature Ball Bearings		8.11	A169
Magneto Bearings		8.12	A169
Self-Aligning Ball Bearings		8.13	A170
Deep Groove Ball Bearings	For Motors	8.14.1	A170
Cylindrical Roller Bearings	For Motors	8.14.2	A170
Cylindrical Roller Bearings	With Cylindrical Bores With Cylindrical Bores (Matched) With Tapered Bores (Matched)	8.15	A171
Spherical Roller Bearings	With Cylindrical Bores With Tapered Bores	8.16	A172
Double-Row and Combined Tapered Roller Bearings		8.17	A173
Combined Angular Contact Ball Bearings (1)		8.18	A174
Four-Point-Contact Ball Bearings (1)		8.19	A174

Note (1) Values given are axial internal clearances.

Table 8.10 Radial Internal Clearances in Deep Groove Ball Bearings

Units : μm

Nominal Bore Diameter d (mm)		Clearance									
		C2		CN		C3		C4		C5	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
10 only		0	7	2	13	8	23	14	29	20	37
10	**18**	0	9	3	18	11	25	18	33	25	45
18	**24**	0	10	5	20	13	28	20	36	28	48
24	**30**	1	11	5	20	13	28	23	41	30	53
30	**40**	1	11	6	20	15	33	28	46	40	64
40	**50**	1	11	6	23	18	36	30	51	45	73
50	**65**	1	15	8	28	23	43	38	61	55	90
65	**80**	1	15	10	30	25	51	46	71	65	105
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	510
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	690
630	**710**	20	130	110	260	240	400	380	570	540	760
710	**800**	20	140	120	290	270	450	430	630	600	840

Remarks To obtain the measured values, use the clearance correction values in the table below.
For the C2 clearance class, the smaller value should be used for bearings with minimum clearance and the larger value for bearings near the maximum clearance range.

Units : μm

Nominal Bore Dia. d (mm)		Measuring Load		Radial Clearance Correction Amount				
over	incl.	(N)	{kgf}	C2	CN	C3	C4	C5
10 (incl)	**18**	24.5	{2.5}	3 to 4	4	4	4	4
18	**50**	49	{5}	4 to 5	5	6	6	6
50	**280**	147	{15}	6 to 8	8	9	9	9

Remark For values exceeding 280 mm, please contact NSK.

Table 8.11 Radial Internal Clearances in Extra Small and Miniature Ball Bearings

Units : μm

Clearance Symbol	MC1		MC2		MC3		MC4		MC5		MC6	
Clearance	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
	0	5	3	8	5	10	8	13	13	20	20	28

Remarks 1. The standard clearance is MC3.
2. To obtain the measured value, add correction amount from the table below.

Units : μm

Clearance Symbol	MC1	MC2	MC3	MC4	MC5	MC6
Clearance Correction Value	1	1	1	1	2	2

The measuring loads are as follows:
For miniature ball bearings∗
2.5N {0.25kgf}
For extra small ball bearings∗
4.4N {0.45kgf}
∗For classification details, refer to Table 1 on Page C054.

Table 8.12 Radial Internal Clearances in Magneto Bearings

Units : μm

Nominal Bore Diameter d (mm)		Bearing Series	Clearance	
over	incl.		min.	max.
2.5	**30**	EN	10	50
		E	30	60

Table 8.13 Radial Internal Clearances in Self-Aligning Ball Bearings

Units : μm

Nominal Bore Dia. d (mm)		Clearance in Bearings With Cylindrical Bores										Clearance in Bearings With Tapered Bores									
		C2		CN		C3		C4		C5		C2		CN		C3		C4		C5	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
2.5	**6**	1	8	5	15	10	20	15	25	21	33	—	—	—	—	—	—	—	—	—	—
6	**10**	2	9	6	17	12	25	19	33	27	42	—	—	—	—	—	—	—	—	—	—
10	**14**	2	10	6	19	13	26	21	35	30	48	—	—	—	—	—	—	—	—	—	—
14	**18**	3	12	8	21	15	28	23	37	32	50	—	—	—	—	—	—	—	—	—	—
18	**24**	4	14	10	23	17	30	25	39	34	52	7	17	13	26	20	33	28	42	37	55
24	**30**	5	16	11	24	19	35	29	46	40	58	9	20	15	28	23	39	33	50	44	62
30	**40**	6	18	13	29	23	40	34	53	46	66	12	24	19	35	29	46	40	59	52	72
40	**50**	6	19	14	31	25	44	37	57	50	71	14	27	22	39	33	52	45	65	58	79
50	**65**	7	21	16	36	30	50	45	69	62	88	18	32	27	47	41	61	56	80	73	99
65	**80**	8	24	18	40	35	60	54	83	76	108	23	39	35	57	50	75	69	98	91	123
80	**100**	9	27	22	48	42	70	64	96	89	124	29	47	42	68	62	90	84	116	109	144
100	**120**	10	31	25	56	50	83	75	114	105	145	35	56	50	81	75	108	100	139	130	170
120	**140**	10	38	30	68	60	100	90	135	125	175	40	68	60	98	90	130	120	165	155	205
140	**160**	15	44	35	80	70	120	110	161	150	210	45	74	65	110	100	150	140	191	180	240

Table 8.14 Radial Internal Clearances in Bearings for Electric Motors

Table 8.14. 1 Deep Groove Ball Bearings for Electric Motors

Units : μm

Nominal Bore Dia. d (mm)		Clearance		Remarks	
		CM		Recommended fit	
over	incl.	min.	max.	Shaft	Housing Bore
10 (incl)	**18**	4	11	js5 (j5)	H6, H7(¹) or JS6, JS7 (J6, J7)(²)
18	**30**	5	12	k5	
30	**50**	9	17		
50	**80**	12	22		
80	**100**	18	30		
100	**120**	18	30	m5	
120	**160**	24	38		

Notes (¹) Applicable to outer rings that require movement in the axial direction.
(²) Applicable to outer rings that do not require movement in the axial direction.

Remark The radial internal clearance increase caused by the measuring load is equal to the correction amount for CN clearance listed in the remarks under Table 8.10.

Table 8.14.2 Cylindrical Roller Bearings for Electric Motors

Units : μm

Nominal Bore Dia. d (mm)		Clearance				Remarks	
		Interchangeable CT		Non-Interchangeable CM		Recommended Fit	
over	incl.	min.	max.	min.	max.	Shaft	Housing Bore
24	**40**	15	35	15	30	k5	JS6, JS7 (J6, J7)(¹) or K6, K7(²)
40	**50**	20	40	20	35	m5	
50	**65**	25	45	25	40		
65	**80**	30	50	30	45		
80	**100**	35	60	35	55		
100	**120**	35	65	35	60		
120	**140**	40	70	40	65		
140	**160**	50	85	50	80		
160	**180**	60	95	60	90	n6	
180	**200**	65	105	65	100		

Notes (¹) Applicable to outer rings that require movement in the axial direction.
(²) Applicable to outer rings that do not require movement in the axial direction.

Table 8.15 Radial Internal Clearances in Cylindrical Roller Bearings and Solid Needle Roller Bearings

Units : μm

Nominal Bore Dia. d (mm)		Clearances in Bearings With Cylindrical Bores										Clearances in Noninterchangeable Bearings With Cylindrical Bores											
		C2		CN		C3		C4		C5		CC1		CC2		CC (¹)		CC3		CC4		CC5	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
—	**10**	0	25	20	45	35	60	50	75	—	—	—	—	—	—	—	—	—	—	—	—	—	—
10	**24**	0	25	20	45	35	60	50	75	65	90	5	15	10	20	20	30	35	45	45	55	65	75
24	**30**	0	25	20	45	35	60	50	75	70	95	5	15	10	25	25	35	40	50	50	60	70	80
30	**40**	5	30	25	50	45	70	60	85	80	105	5	15	12	25	25	40	45	55	55	70	80	95
40	**50**	5	35	30	60	50	80	70	100	95	125	5	18	15	30	30	45	50	65	65	80	95	110
50	**65**	10	40	40	70	60	90	80	110	110	140	5	20	15	35	35	50	55	75	75	90	110	130
65	**80**	10	45	40	75	65	100	90	125	130	165	10	25	20	40	40	60	70	90	90	110	130	150
80	**100**	15	50	50	85	75	110	105	140	155	190	10	30	25	45	45	70	80	105	105	125	155	180
100	**120**	15	55	50	90	85	125	125	165	180	220	10	30	25	50	50	80	95	120	120	145	180	205
120	**140**	15	60	60	105	100	145	145	190	200	245	10	35	30	60	60	90	105	135	135	160	200	230
140	**160**	20	70	70	120	115	165	165	215	225	275	10	35	35	65	65	100	115	150	150	180	225	260
160	**180**	25	75	75	125	120	170	170	220	250	300	10	40	35	75	75	110	125	165	165	200	250	285
180	**200**	35	90	90	145	140	195	195	250	275	330	15	45	40	80	80	120	140	180	180	220	275	315
200	**225**	45	105	105	165	160	220	220	280	305	365	15	50	45	90	90	135	155	200	200	240	305	350
225	**250**	45	110	110	175	170	235	235	300	330	395	15	50	50	100	100	150	170	215	215	265	330	380
250	**280**	55	125	125	195	190	260	260	330	370	440	20	55	55	110	110	165	185	240	240	295	370	420
280	**315**	55	130	130	205	200	275	275	350	410	485	20	60	60	120	120	180	205	265	265	325	410	470
315	**355**	65	145	145	225	225	305	305	385	455	535	20	65	65	135	135	200	225	295	295	360	455	520
355	**400**	100	190	190	280	280	370	370	460	510	600	25	75	75	150	150	225	255	330	330	405	510	585
400	**450**	110	210	210	310	310	410	410	510	565	665	25	85	85	170	170	255	285	370	370	455	565	650
450	**500**	110	220	220	330	330	440	440	550	625	735	25	95	95	190	190	285	315	410	410	505	625	720

Note (¹) CC denotes normal clearance for noninterchangeable cylindrical roller bearings and solid needle roller bearings.

Units : μm

Nominal Bore Dia. d (mm)		Clearances in Noninterchangeable Bearings with Tapered Bores															
		CC9 (¹)		CC0		CC1		CC2		CC (²)		CC3		CC4		CC5	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
10	**24**	5	10	—	—	10	20	20	30	35	45	45	55	55	65	75	85
24	**30**	5	10	8	15	10	25	25	35	40	50	50	60	60	70	80	95
30	**40**	5	12	8	15	12	25	25	40	45	55	55	70	70	80	95	110
40	**50**	5	15	10	20	15	30	30	45	50	65	65	80	80	95	110	125
50	**65**	5	15	10	20	15	35	35	50	55	75	75	90	90	110	130	150
65	**80**	10	20	15	30	20	40	40	60	70	90	90	110	110	130	150	170
80	**100**	10	25	20	35	25	45	45	70	80	105	105	125	125	150	180	205
100	**120**	10	25	20	35	25	50	50	80	95	120	120	145	145	170	205	230
120	**140**	15	30	25	40	30	60	60	90	105	135	135	160	160	190	230	260
140	**160**	15	35	30	50	35	65	65	100	115	150	150	180	180	215	260	295
160	**180**	15	35	30	50	35	75	75	110	125	165	165	200	200	240	285	320
180	**200**	20	40	30	50	40	80	80	120	140	180	180	220	220	260	315	355
200	**225**	20	45	35	60	45	90	90	135	155	200	200	240	240	285	350	395
225	**250**	25	50	40	65	50	100	100	150	170	215	215	265	265	315	380	430
250	**280**	25	55	40	70	55	110	110	165	185	240	240	295	295	350	420	475
280	**315**	30	60	—	—	60	120	120	180	205	265	265	325	325	385	470	530
315	**355**	30	65	—	—	65	135	135	200	225	295	295	360	360	430	520	585
355	**400**	35	75	—	—	75	150	150	225	255	330	330	405	405	480	585	660
400	**450**	40	85	—	—	85	170	170	255	285	370	370	455	455	540	650	735
450	**500**	45	95	—	—	95	190	190	285	315	410	410	505	505	600	720	815

Notes (¹) Clearance CC9 is applicable to cylindrical roller bearings with tapered bores in ISO Tolerance Classes 5 and 4.
(²) CC denotes normal clearance for noninterchangeable cylindrical roller bearings and solid needle roller bearings.

Table 8.16 Radial Internal Clearances in Spherical Roller Bearings

Units : μm

Nominal Bore Dia. d (mm)		Clearance in Bearings With Cylindrical Bores										Clearance in Bearings With Tapered Bores									
		C2		CN		C3		C4		C5		C2		CN		C3		C4		C5	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
24	**30**	15	25	25	40	40	55	55	75	75	95	20	30	30	40	40	55	55	75	75	95
30	**40**	15	30	30	45	45	60	60	80	80	100	25	35	35	50	50	65	65	85	85	105
40	**50**	20	35	35	55	55	75	75	100	100	125	30	45	45	60	60	80	80	100	100	130
50	**65**	20	40	40	65	65	90	90	120	120	150	40	55	55	75	75	95	95	120	120	160
65	**80**	30	50	50	80	80	110	110	145	145	180	50	70	70	95	95	120	120	150	150	200
80	**100**	35	60	60	100	100	135	135	180	180	225	55	80	80	110	110	140	140	180	180	230
100	**120**	40	75	75	120	120	160	160	210	210	260	65	100	100	135	135	170	170	220	220	280
120	**140**	50	95	95	145	145	190	190	240	240	300	80	120	120	160	160	200	200	260	260	330
140	**160**	60	110	110	170	170	220	220	280	280	350	90	130	130	180	180	230	230	300	300	380
160	**180**	65	120	120	180	180	240	240	310	310	390	100	140	140	200	200	260	260	340	340	430
180	**200**	70	130	130	200	200	260	260	340	340	430	110	160	160	220	220	290	290	370	370	470
200	**225**	80	140	140	220	220	290	290	380	380	470	120	180	180	250	250	320	320	410	410	520
225	**250**	90	150	150	240	240	320	320	420	420	520	140	200	200	270	270	350	350	450	450	570
250	**280**	100	170	170	260	260	350	350	460	460	570	150	220	220	300	300	390	390	490	490	620
280	**315**	110	190	190	280	280	370	370	500	500	630	170	240	240	330	330	430	430	540	540	680
315	**355**	120	200	200	310	310	410	410	550	550	690	190	270	270	360	360	470	470	590	590	740
355	**400**	130	220	220	340	340	450	450	600	600	750	210	300	300	400	400	520	520	650	650	820
400	**450**	140	240	240	370	370	500	500	660	660	820	230	330	330	440	440	570	570	720	720	910
450	**500**	140	260	260	410	410	550	550	720	720	900	260	370	370	490	490	630	630	790	790	1 000
500	**560**	150	280	280	440	440	600	600	780	780	1 000	290	410	410	540	540	680	680	870	870	1 100
560	**630**	170	310	310	480	480	650	650	850	850	1 100	320	460	460	600	600	760	760	980	980	1 230
630	**710**	190	350	350	530	530	700	700	920	920	1 190	350	510	510	670	670	850	850	1 090	1 090	1 360
710	**800**	210	390	390	580	580	770	770	1 010	1 010	1 300	390	570	570	750	750	960	960	1 220	1 220	1 500
800	**900**	230	430	430	650	650	860	860	1 120	1 120	1 440	440	640	640	840	840	1 070	1 070	1 370	1 370	1 690
900	**1 000**	260	480	480	710	710	930	930	1 220	1 220	1 570	490	710	710	930	930	1 190	1 190	1 520	1 520	1 860
1 000	**1 120**	290	530	530	780	780	1 020	1 020	1 330	—	—	530	770	770	1 030	1 030	1 300	1 300	1 670	—	—
1 120	**1 250**	320	580	580	860	860	1 120	1 120	1 460	—	—	570	830	830	1 120	1 120	1 420	1 420	1 830	—	—
1 250	**1 400**	350	640	640	950	950	1 240	1 240	1 620	—	—	620	910	910	1 230	1 230	1 560	1 560	2 000	—	—

Table 8.17 Radial Internal Clearances in Double-Row and Combined Tapered Roller Bearings

Units : μm

Cylindrical Bore / Tapered Bore / Nominal Bore Dia. d (mm)		Clearance											
Cylindrical Bore		C1		C2		CN		C3		C4		C5	
Tapered Bore		—		C1		C2		CN		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
—	**18**	0	10	10	20	20	30	35	45	50	60	65	75
18	**24**	0	10	10	20	20	30	35	45	50	60	65	75
24	**30**	0	10	10	20	20	30	40	50	50	60	70	80
30	**40**	0	12	12	25	25	40	45	60	60	75	80	95
40	**50**	0	15	15	30	30	45	50	65	65	80	95	110
50	**65**	0	15	15	35	35	55	60	80	80	100	110	130
65	**80**	0	20	20	40	40	60	70	90	90	110	130	150
80	**100**	0	25	25	50	50	75	80	105	105	130	155	180
100	**120**	5	30	30	55	55	80	90	115	120	145	180	210
120	**140**	5	35	35	65	65	95	100	130	135	165	200	230
140	**160**	10	40	40	70	70	100	110	140	150	180	220	260
160	**180**	10	45	45	80	80	115	125	160	165	200	250	290
180	**200**	10	50	50	90	90	130	140	180	180	220	280	320
200	**225**	20	60	60	100	100	140	150	190	200	240	300	340
225	**250**	20	65	65	110	110	155	165	210	220	270	330	380
250	**280**	20	70	70	120	120	170	180	230	240	290	370	420
280	**315**	30	80	80	130	130	180	190	240	260	310	410	460
315	**355**	30	80	80	130	140	190	210	260	290	350	450	510
355	**400**	40	90	90	140	150	200	220	280	330	390	510	570
400	**450**	45	95	95	145	170	220	250	310	370	430	560	620
450	**500**	50	100	100	150	190	240	280	340	410	470	620	680
500	**560**	60	110	110	160	210	260	310	380	450	520	700	770
560	**630**	70	120	120	170	230	290	350	420	500	570	780	850
630	**710**	80	130	130	180	260	310	390	470	560	640	870	950
710	**800**	90	140	150	200	290	340	430	510	630	710	980	1 060
800	**900**	100	150	160	210	320	370	480	570	700	790	1 100	1 200
900	**1 000**	120	170	180	230	360	410	540	630	780	870	1 200	1 300
1 000	**1 120**	130	190	200	260	400	460	600	700	—	—	—	—
1 120	**1 250**	150	210	220	280	450	510	670	770	—	—	—	—
1 250	**1 400**	170	240	250	320	500	570	750	870	—	—	—	—

Remark Axial internal clearance $\Delta_a = \Delta_r \cot\alpha \fallingdotseq \frac{1.5}{e}\Delta_r$

where Δ_r : Radial internal clearance
α : Contact angle
e : Constant (listed in bearing tables)

Table 8.18 Axial Internal Clearances in Combined Angular Contact Ball Bearings (Measured Clearance)

Units : μm

Nominal Bore Diameter. d (mm)		Axial Internal Clearance											
		Contact Angle 30°						Contact Angle 40°					
		CN		C3		C4		CN		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
—	10	9	29	29	49	49	69	6	26	26	46	46	66
10	18	10	30	30	50	50	70	7	27	27	47	47	67
18	24	19	39	39	59	59	79	13	33	33	53	53	73
24	30	20	40	40	60	60	80	14	34	34	54	54	74
30	40	26	46	46	66	66	86	19	39	39	59	59	79
40	50	29	49	49	69	69	89	21	41	41	61	61	81
50	65	35	60	60	85	85	110	25	50	50	75	75	100
65	80	38	63	63	88	88	115	27	52	52	77	77	100
80	100	49	74	74	99	99	125	35	60	60	85	85	110
100	120	72	97	97	120	120	145	52	77	77	100	100	125
120	140	85	115	115	145	145	175	63	93	93	125	125	155
140	160	90	120	120	150	150	180	66	96	96	125	125	155
160	180	95	125	125	155	155	185	68	98	98	130	130	160
180	200	110	140	140	170	170	200	80	110	110	140	140	170

Remark This table is applicable to bearings with Normal and Class 6 tolerances. Please consult NSK regarding the internal axial clearances of bearings with Class 5 tolerance or better and contact angles of 15° and 25°.

Table 8.19 Axial Internal Clearance in Four-Point-Contact Ball Bearings (Measured Clearances)

Units : μm

Nominal Bore Dia. d (mm)		Axial Internal Clearance					
		CN		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.
10	18	45	85	75	125	115	165
18	40	56	106	96	146	136	186
40	60	76	126	116	166	156	206
60	80	86	136	126	176	166	226
80	100	96	156	136	196	186	246
100	140	116	176	156	216	206	266
140	180	136	196	176	246	226	296
180	220	156	226	206	276	256	326
220	260	175	245	225	305	285	365
260	300	195	275	255	335	315	395
300	350	215	305	275	365	345	425
350	400	245	335	315	405	385	475
400	500	285	385	355	455	435	525

8.2.2 Selection of Bearing Internal Clearances

CN Clearance is adequate for standard operating conditions. Clearance becomes progressively smaller from C2 to C1 and larger from C3 to C5.

Standard operating conditions are defined as those where the inner ring speed is less than approximately 50 % of the limiting speed listed in the bearing tables, the load is less than normal ($P \fallingdotseq 0.1C_r$), and the bearing has a tight fit on the shaft.

To reduce bearing noise, the radial internal clearance range is narrower than the normal class and the values are somewhat smaller for deep groove ball bearings and cylindrical roller bearings for electric motors (refer to Table 8.14.1 and 8.14.2).

Internal clearance varies with the fit and temperature differences in operation. The changes in radial internal clearance in a roller bearing are shown in Fig. 8.8.

(1) Decrease in Radial Clearance Caused by Fitting and Residual Clearance

When the inner ring or the outer ring has a tight fit on a shaft or in a housing, a decrease in the radial internal clearance is caused by the expansion or contraction of the bearing rings. The decrease varies according to the bearing type, bearing size, and design of the shaft and housing. The amount of this decrease is approximately 70 to 90% of the interference (refer to Section 8.1.2, Selection of Fit, *(5) Fitting Stress and Ring Expansion and Contraction*, Pages A156 and A157). The internal clearance after subtracting this decrease from the theoretical internal clearance Δ_0 is called the residual clearance Δ_f.

(2) Decrease in Radial Internal Clearance Caused by Temperature Differences Between Inner and Outer Rings and Effective Clearance

The frictional heat generated during operation is conducted away from the bearing through the shaft and housing. Since housings generally conduct heat better than shafts, the temperature of the inner ring and the rolling elements is usually higher than that of the outer ring by 5 to 10 °C. If the shaft is heated or the housing is cooled, the difference in temperature between the inner and outer rings increases. The radial clearance decreases due to the thermal expansion caused by the temperature difference between the inner and outer rings. The amount of this decrease can be calculated using the following equations:

$$\delta_t \fallingdotseq \alpha \Delta_t D_e \quad \cdots\cdots (8.8)$$

where δ_t : Decrease in radial clearance due to temperature difference between inner and outer rings (mm)

α : Coefficient of linear expansion of bearing steel $\fallingdotseq 12.5\times10^{-6}$ (1/°C)

Δ_t : Temperature difference between inner and outer rings (°C)

D_e : Outer ring raceway diameter (mm)

For ball bearings

$$D_e \fallingdotseq \frac{1}{5}(4D+d) \quad \cdots\cdots (8.9)$$

For roller bearings

$$D_e \fallingdotseq \frac{1}{4}(3D+d) \quad \cdots\cdots (8.10)$$

The clearance after subtracting δ_t from the residual clearance Δ_f is called the effective clearance Δ. Theoretically, the longest life of a bearing can be achieved when the effective clearance is slightly negative. However, it is difficult to achieve such an ideal condition, and an excessive negative clearance will greatly shorten the bearing life. Therefore, a clearance of zero or a slightly positive amount, instead of a negative one, should be selected. When single-row angular contact ball bearings or tapered roller bearings are used facing each other, there should be a small effective clearance, unless a preload is required. When two cylindrical roller bearings with a rib on one side are used facing each other, adequate axial clearance is necessary to allow for shaft elongation during operation.

The radial clearances used in some specific applications are given in Table 8.20. Please consult NSK regarding special operating conditions.

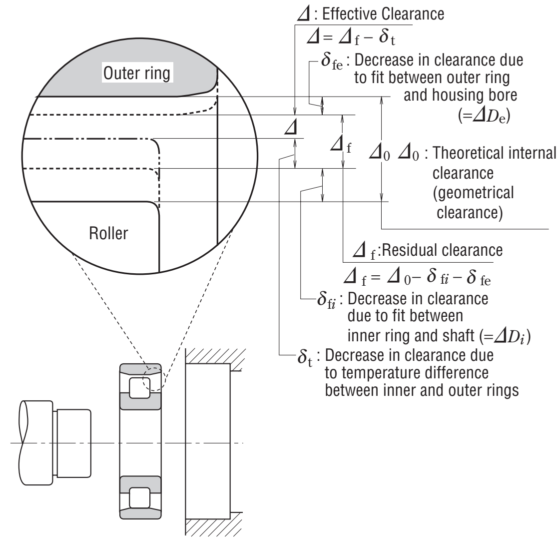


Fig. 8.8 Changes in Radial Internal Clearance of Bearings

Table 8. 20 Example Clearances for Specific Applications

Operating Conditions	Examples	Internal Clearance
When shaft deflection is large	Semi-floating rear wheels of automobiles	C5 or equivalent
When steam passes through hollow shafts or roller shafts are heated	Dryers in paper making machines	C3, C4
	Table rollers for rolling mills	C3
When impact loads and vibration are severe or when both the inner and outer rings are tight-fitted	Traction motors for railways	C4
	Vibrating screens	C3, C4
	Fluid couplings	C4
	Final reduction gears for tractors	C4
When both the inner and outer rings are loose-fitted	Rolling mill roll necks	C2 or equivalent
When noise and vibration restrictions are severe	Small motors with special specifications	C1, C2, CM
When clearance is adjusted after mounting to prevent shaft deflection, etc.	Main shafts of lathes	CC9, CC1

8.3 Technical Data

8.3.1 Temperature Rise and Dimensional Change

Rolling bearings are extremely precise mechanical elements; any change in dimensional accuracy due to temperature cannot be ignored. Accordingly, as a rule, measurement of a bearing must be performed at 20 °C and the dimensions set forth in the standards must be expressed by values at 20 °C.

Dimensional changes due to temperature change not only affect dimensional accuracy, but also cause changes in the internal clearance of a bearing during operation. Dimensional change may cause interference between the inner ring and shaft or between the outer ring and housing bore. It is possible to achieve shrink fitting with large interference by utilizing dimensional changes induced by temperature differences. The dimensional change Δl due to temperature rise can be expressed as Equation (8.11) below:

$$\Delta l = \Delta T \, \alpha \, l \ (\text{mm}) \quad \cdots\cdots (\mathbf{8.11})$$

where, Δl: Dimensional change (mm)
ΔT: Temperature rise (°C)
α: Coefficient of linear expansion for bearing steel
α=12.5×10^{-6} (1/°C)
l: Original dimension (mm)

Equation (8.11) may be illustrated as shown in Fig. 8.9. In the following cases, Fig. 8.9 can be utilized to easily obtain an approximate numerical values for dimensional change when there is, need to:

(1) Correct dimensional measurements according to the ambient air temperature
(2) Find the change in bearing internal clearance due to a temperature difference between the inner and outer rings during operation
(3) Find the relationship between interference and heating temperature during shrink fitting
(4) Find the change in interference when a temperature difference exists on the fit surface

Example

If an inner ring with a 110 mm bore is to be shrink-fitted to an n6 tolerance shaft, how much should it be heated?
The maximum interference between an inner ring with a 110 mm bore and an n6 shaft is 0.065 mm. To enable insertion of the inner ring with ease on the shaft, there must be a clearance of 0.03 to 0.04 mm. Accordingly, the inner ring must expand by 0.095 to 0.105 mm.
By using Fig 8.9, the intersection of the vertical axis when Δl = 0.105 and the horizontal axis when l = 110 can be determined. ΔT is located in the temperature range between 70 °C and 80 °C (ΔT≒77°C). Therefore, it's sufficient to set the inner ring heating temperature to the room temperature +80 °C.

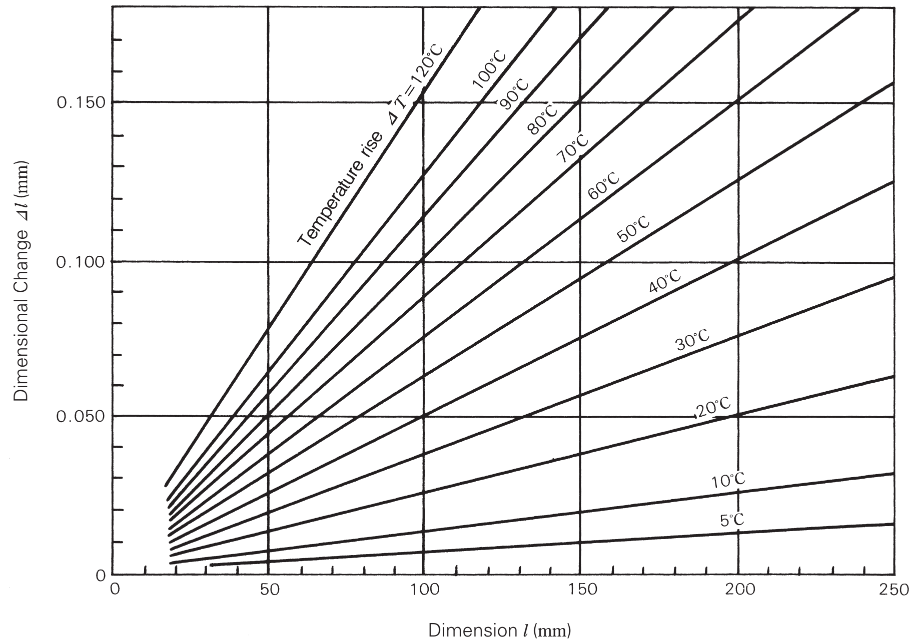


Fig. 8.9 Temperature Rise and Dimensional Change of Bearing Steel

8.3.2 Interference Deviation Due to Temperature Rise (Aluminum Housing, Plastic Housing)

Bearing housing materials such as aluminum, light alloys, or plastics (polyacetal resin, etc.) are often used to reduce weight and cost or improve the performance of equipment.

When non-ferrous materials are used in housings, any temperature rise occurring during operation affects the interference or clearance of the outer ring due to the difference in the coefficients of linear expansion. This change is significant for plastics which have high coefficients of linear expansion.

The deviation of clearance or interference of a fitting surface ΔD_T of a bearing's outer ring due to temperature rise is expressed by the following equation:

$$\Delta D_T=(\alpha_1\cdot\Delta T_1-\alpha_2\cdot\Delta T_2)D \text{ (mm)} \quad \cdots\cdots\cdots\cdots\cdots (\mathbf{8.12})$$

where ΔD_T: Change of clearance or interference at fitting surface due to temperature rise
α_1: Coefficient of linear expansion of housing (1/°C)
ΔT_1: Housing temperature rise near fitting surface (°C)
α_2: Coefficient of linear expansion of bearing outer ring
Bearing steel .... α_2=12.5×10^{-6} (1/°C)
ΔT_2: Outer ring temperature rise near fitting surface (°C)
D: Nominal outside diameter (mm)

In general, housing temperature rise and outer ring temperature rise are somewhat different, but if we assume they are approximately equal near the fitting surfaces ($\Delta T_1 \fallingdotseq \Delta T_2=\Delta T$), Equation (8.12) becomes Equation (8.13):

$$\Delta D_T=(\alpha_1-\alpha_2)\ \Delta T\cdot D \text{ (mm)} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots (\mathbf{8.13})$$

where ΔT: Temperature rise of outer ring and housing near fitting surfaces (°C)

Equation (8.13) for aluminum housings (α_1 = 23.7 × 10^{-6}), can be shown graphically as Fig. 8.10.

Polyacetal resin is the plastic most-often used for bearing housings. The coefficients of linear expansion of plastics may vary or show directional characteristics. For molded products made with polyacetal resin, the coefficient of linear expansion is approximately 9×10^{-5}. Equation (8.13) for polyacetal-resin housings can be expressed as Fig. 8.11.

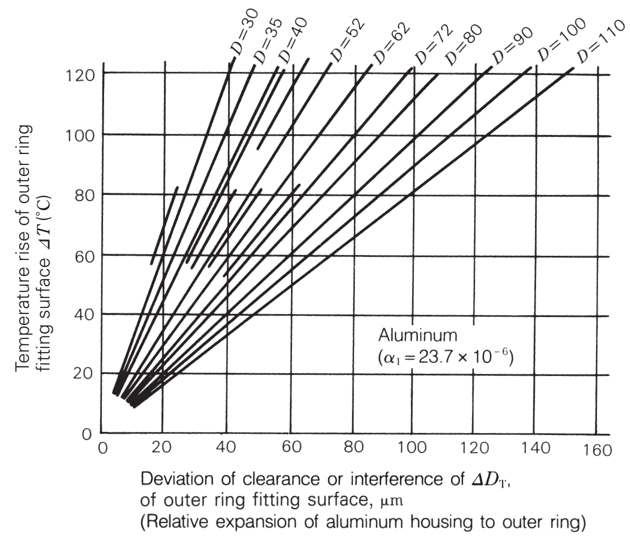


Fig. 8.10 Aluminum Housing

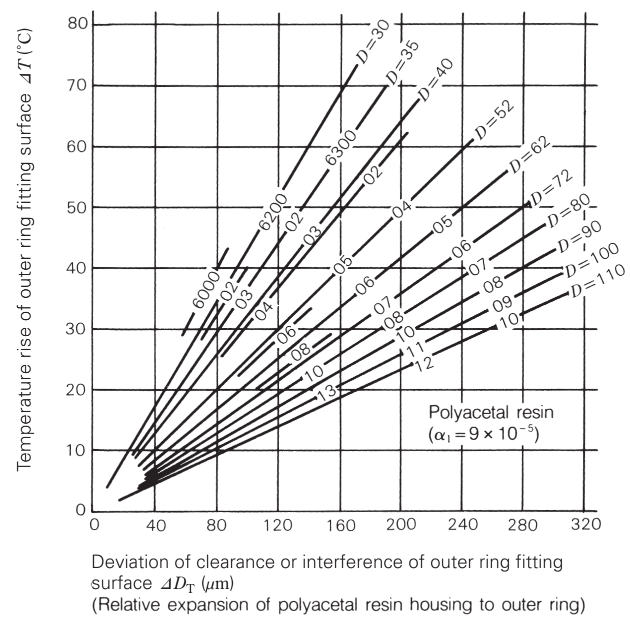


Fig. 8.11 Polyacetal-Resin Housing

8.3.3 Calculating Residual Internal Clearance After Mounting

The various types of internal bearing clearance were discussed in Section 8.2.2. This section will explain the step by step procedures for calculating residual internal clearance.

When the inner ring of a bearing is press-fit onto a shaft or when the outer ring is press fit into a housing, it stands to reason that radial internal clearance will decrease due to the resulting expansion or contraction of the bearing raceways. Generally, most bearing applications have a rotating shaft which requires a tight fit between the inner ring and shaft and a loose fit between the outer ring and housing; therefore, generally only the effect of the interference on the inner ring needs to be taken into account.

Below we have selected a 6310 single row deep groove ball bearing for our representative calculations. The shaft is set as k5 and the housing set as H7. An interference fit is applied only to the inner ring.

Shaft diameter, bore size, and radial clearance are standard bearing measurements. Assuming that 99.7% of the parts are within tolerance, the mean value of residual clearance $m_{\Delta f}$ and standard deviation of the internal clearance after mounting (residual clearance) $\sigma_{\Delta f}$ can be calculated. Measurements are given in millimeters (mm).

$$\sigma_s = \frac{R_s/2}{3} = 0.0018$$

$$\sigma_i = \frac{R_i/2}{3} = 0.0020$$

$$\sigma_{\Delta 0} = \frac{R_{\Delta 0}/2}{3} = 0.0028$$

$$\sigma_f^2 = \sigma_s^2 + \sigma_i^2$$

$$m_{\Delta f} = m_{\Delta 0} - \lambda_i\ (m_s - m_i) = 0.0035$$

$$\sigma_{\Delta f} = \sqrt{\sigma_{\Delta 0}^2 + \lambda_i^2\ \sigma_f^2} = 0.0035$$

where,
- σ_s: Standard deviation of shaft diameter
- σ_i: Standard deviation of bore diameter
- σ_f: Standard deviation of interference
- $\sigma_{\Delta 0}$: Standard deviation of radial clearance (before mounting)
- $\sigma_{\Delta f}$: Standard deviation of residual clearance (after mounting)
- m_s: Mean value of shaft diameter (ϕ50+0.008)
- m_i: Mean value of bore diameter (ϕ50–0.006)
- $m_{\Delta 0}$: Mean value of radial clearance (before mounting) (0.014)
- $m_{\Delta f}$: Mean value of residual clearance (after mounting)
- R_s: Shaft tolerance (0.011)
- R_i: Bearing bore tolerance (0.012)
- $R_{\Delta 0}$: Range in radial clearance (before mounting) (0.017)
- λ_i: Rate of raceway expansion from apparent interference (0.75 from Fig. 8.12)

The average amount of raceway expansion and contraction from apparent interference is calculated from $\lambda_i\ (m_m - m_i)$.

To determine, within a 99.7% probability, the variation in internal clearance after mounting $R_{\Delta f}$, use the following equation:

$$R_{\Delta f} = m_{\Delta f} \pm 3\sigma_{\Delta f} = +0.014 \text{ to } -0.007$$

In other words, the mean value of residual clearance ($m_{\Delta f}$) is +0.0035, and the range is from −0.007 to +0.014 for bearing 6310.

Units : mm

Shaft diameter	ϕ50	$^{+0.013}_{+0.002}$
Bearing bore diameter, (d)	ϕ50	$^{0}_{-0.012}$
Radial internal clearance (Δ_0)	0.006 to 0.023([1])	

Note ([1]) Standard internal clearance, unmounted

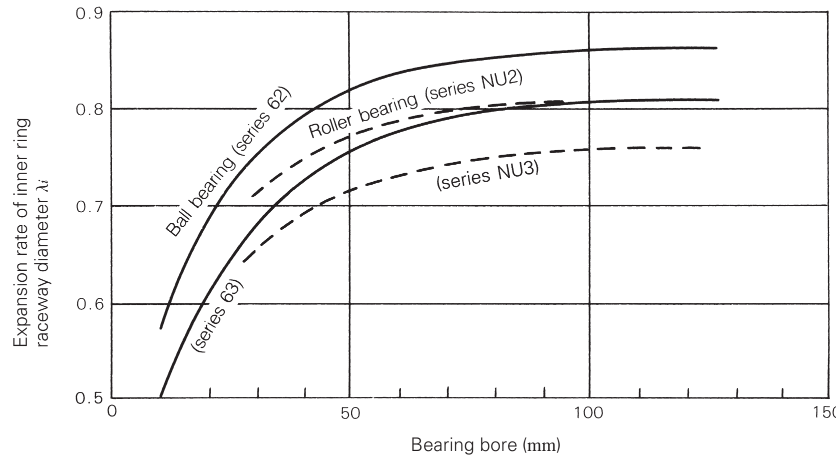


Fig. 8.12 λ_i : Rate of Inner Ring Raceway Expansion From Apparent Interference

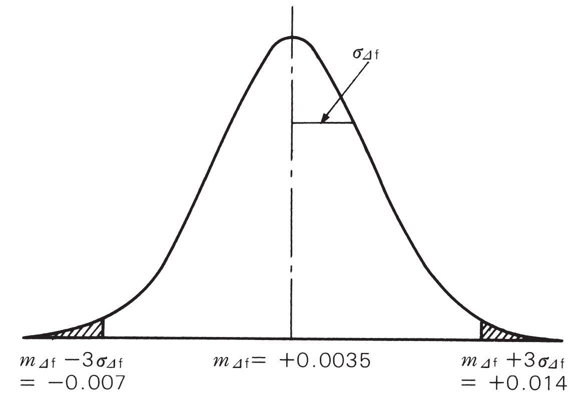


Fig. 8.13 Distribution of Residual Internal Clearance

8.3.4 Effect of Interference Fits on Bearing Raceways (Fit of Inner Ring)

One important factor related to radial clearance is the reduction in radial clearance resulting from the mounting fit. When an inner ring is mounted on a shaft with an interference fit and the outer ring is secured in a housing with an interference fit, the inner ring will expand and the outer ring will contract.

The means of calculating the amount of ring expansion or contraction were previously noted in Section 8.1.2 (5); however, the equation for establishing the amount of inner raceway expansion ΔD_i is given in Equation (8.14) below:

$$\Delta D_i = \Delta d\ k\ \frac{1-k_0^{\ 2}}{1-k^2\ k_0^{\ 2}} \quad \cdots\cdots\cdots\cdots\cdots (\mathbf{8.14})$$

where, Δd: Effective interference (mm)
k: Ratio of bore to inner raceway diameter; $k=d/D_i$
k_0: Ratio of inside to outside diameter of hollow shaft; $k_0=d_0/D_i$
d: Bore or shaft diameter (mm)
D_i: Inner raceway diameter (mm)
d_0: Inside diameter of hollow shaft (mm)

Equation (8.14) is represented in a clearer graphical form in Fig. 8.14.

The vertical axis of Fig. 8.14 represents inner raceway diameter expansion in relation to the amount of interference. The horizontal axis represents k_0: the ratio of the inside diameter of a hollow shaft to the outside diameter.

Generally, the decrease in radial clearance is calculated to be approximately 80% of the interference for solid shaft mountings only. For hollow shaft mountings the decrease in radial clearance varies with the ratio of inside shaft diameter to outside shaft diameter. Since the general 80% rule is based on average bearing bore size to inner raceway diameter ratios, the change will vary with different bearing types, sizes, and series. Typical plots for single-row deep groove ball bearings and for cylindrical roller bearings are shown in Figs. 8.15 and 8.16. Values in Fig. 8.14 apply only for steel shafts.

As an example, let us determine the decrease in the radial clearance of a 6220 ball bearing mounted on a hollow shaft (diameter d=100 mm, inside diameter d_0=65 mm) with a fit class of m5.

The ratio between bore diameter and raceway diameter k is 0.87 as shown in Fig. 8.15. The ratio of the shaft inside diameter to the shaft outside diameter k_0, is $k_0=d_0/d$=0.65. Thus, reading from Fig. 8.14, the rate of raceway expansion is 73%.

Given that an interference of m5 has a mean value of 30 μm, the amount of raceway expansion, or, the amount of decrease in the radial clearance from the fit is 0.73 × 30 = 22 μm.

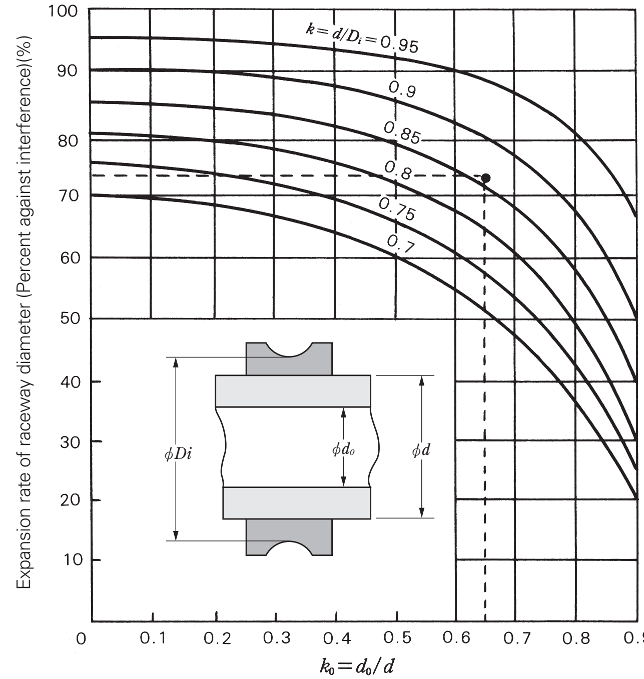


Fig. 8.14 Raceway Expansion in Relation to Bearing Fit (Inner Ring Fit on Steel Shaft)

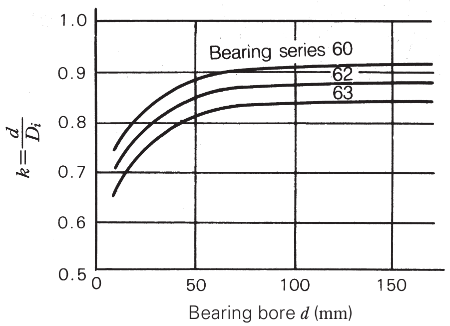


Fig. 8.15 Ratio of Bore Size to Raceway Diameter for Single-Row Deep Groove Ball Bearings

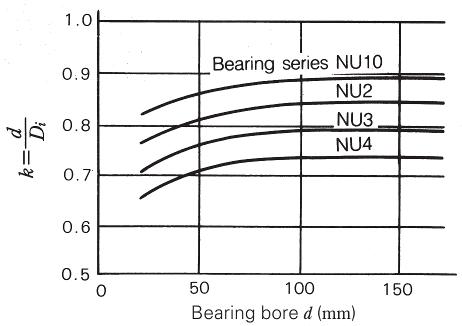


Fig. 8.16 Ratio of Bore Size to Raceway Diameter for Cylindrical Roller Bearings

8.3.5 Effect of Interference Fits on Bearing Raceways (Fit of Outer Ring)

When a bearing load is applied on a rotating inner ring (outer ring carrying a static load), an interference fit is adopted for the inner ring and the outer ring is mounted either with a transition fit or a clearance fit. However, when load is applied on a rotating outer ring (inner ring carrying a static load) or when there is an indeterminate load and the outer ring must be mounted with an interference fit, a decrease in radial internal clearance caused by the fit begins to contribute in the same way as when the inner ring is mounted with an interference fit.

Because the amount of interference that can be applied to the outer ring is limited by stress, the constraints of most bearing applications make it difficult to apply a large amount of interference to the outer ring. In addition, instances where there is an indeterminate load are quite rare compared to those where a rotating inner ring carries the load; therefore, there are few occasions where it is necessary to be cautious about the decrease in radial clearance caused by outer ring interference.

The decrease in outer raceway diameter ΔD_e is calculated using Equation (8.15) below:

$$\Delta D_e = \Delta D \cdot h \frac{1-h_0^{\,2}}{1-h^2 h_0^{\,2}} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots (\mathbf{8.15})$$

where
- ΔD: Effective interference (mm)
- h: Ratio between raceway dia. and outside dia. of outer ring, $h=D_e/D$
- h_0: Housing thickness ratio, $h_0=D/D_0$
- D: Bearing outside diameter (housing bore diameter) (mm)
- D_e: Raceway diameter of outer ring (mm)
- D_0: Outside diameter of housing (mm)

Fig. 8.17 represents the above equation graphically. The vertical axis shows the outer ring raceway contraction as a percentage of interference, and the horizontal axis shows the housing thickness ratio h_0. The data are plotted for constant values of outer ring thickness ratios from 0.7 through 1.0 in increments of 0.05. The value of thickness ratio h will differ with bearing type, size, and diameter series. Representative values for single-row deep groove ball bearings and cylindrical roller bearings are given in Figs. 8.18 and 8.19 respectively.

Loads applied on rotating outer rings occur in such applications as automotive front axles, tension pulleys, conveyor systems, and other pulley systems. As an example, let us estimate the decrease in radial clearance for a 6207 ball bearing mounted in a steel housing with an N7 fit. The outside diameter of the housing is assumed as D_0 = 95, and the bearing outside diameter D = 72. From Fig. 8.18, the outer ring thickness ratio, h, is 0.9. Because $h_0 = D/D_0$ = 0.76, the raceway contraction is 71 %, as indicated in Fig 8.17. Taking the mean value for N7 interference as 18 μm, the contraction of the outer raceway, or decrease in radial clearance is 0.71 × 18 ≒13 μm.

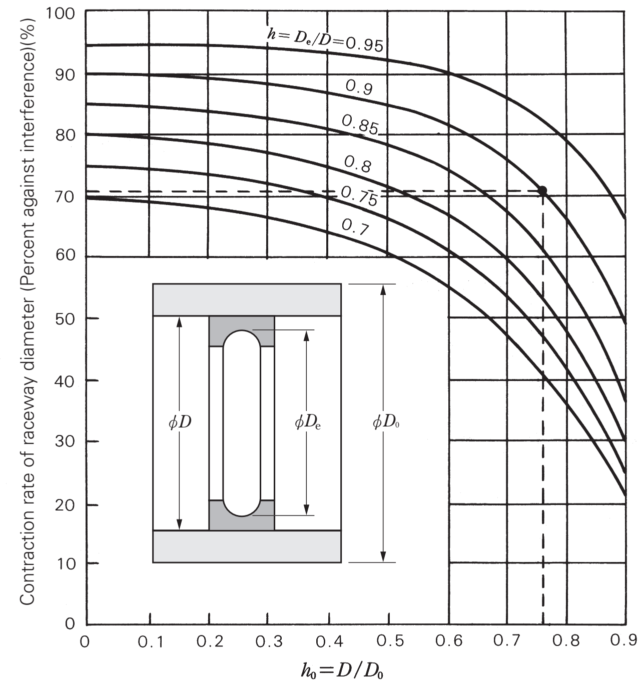


Fig. 8.17 Raceway Contraction in Relation to Bearing Fit (Outer Ring Fit in Steel Housing)

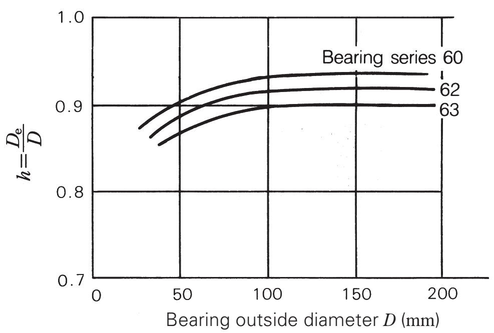


Fig. 8.18 Ratio of Outside Diameter to Raceway Diameter for Single-Row Deep Groove Ball Bearings

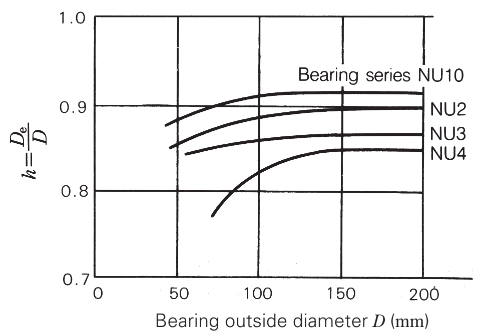


Fig. 8.19 Ratio of Outside Diameter to Raceway Diameter for Cylindrical Roller Bearings

8.3.6 Measuring Internal Clearance of Combined Tapered Roller Bearings (Offset Measuring Method)

Combined tapered roller bearings are available in two types: a back-to-back arrangement (DB type) and a face-to-face arrangement (DF type) (see Figs. 8.20 and Fig. 8.21). These arrangements have certain advantages and can be assembled as a set or combined with other bearings as a fixed- or free-side bearing.

The cages of DB tapered roller bearing arrangements protrude from the back side of the outer ring; therefore, the outer ring spacer (K spacer in Fig. 8.20) is mounted to prevent mutual contact of cages. An inner ring spacer (L spacer in Fig. 8.20), with the appropriate width is provided for the inner ring to secure the clearance. For the DF type, a K spacer is used, as shown in Fig. 8.21.

In general, to use such a bearing arrangement requires an appropriate clearance that accounts for heat generated during operation or an applied preload that increases the rigidity of the bearings. The spacer width should be adjusted so as to provide an appropriate clearance or preload (minus clearance) after mounting.

The clearance measurement method for a DB arrangement is as follows:

(1) As shown in Fig. 8.22, put bearing A on the surface plate and after stabilizing the rollers by rotating the outer ring (over 10 turns), measure the offset $f_A = T_A - B_A$ (consult Figs. 8.22 through 8.24 for these symbols).

(2) Next, as shown in Fig. 8.23, use the same procedure to measure bearing B for its offset $f_B = T_B - B_B$.

(3) Last, measure the width of the K and L spacers as shown in Fig. 8.24.

From the results of the above measurements, the axial clearance Δ_a of the arrangement can be obtained by Equation (8.16):

$$\Delta_a = (L - K) - (f_A + f_B) \quad \cdots\cdots (\mathbf{8.16})$$

As an example, let's confirm the clearance of tapered roller bearing arrangement HR32232JDB + KLR10AC3 to specifications. First, refer to Table 8.17 and note that the C3 clearance range is Δ_r=110 to 140 μm.

To compare this specification with the offset measurement results, convert it into axial clearance Δ_a by using Equation (8.17):

$$\Delta_a = \Delta_r \cot\alpha \doteqdot \Delta_r \frac{1.5}{e} \quad \cdots\cdots (\mathbf{8.17})$$

where e: Constant determined for each bearing No. (listed in the bearing tables of this catalog)

By referring to Page C205, we find e=0.44 and the following:

$$\Delta_a = (110 \text{ to } 140) \times \frac{1.5}{e}$$

$$\doteqdot 380 \text{ to } 480\ \mu\text{m}$$

We can confirm that the bearing clearance is C3 by verifying that the axial clearance Δ_a of Equation (8.16) (obtained by the bearing offset measurement) is within the above-mentioned range.

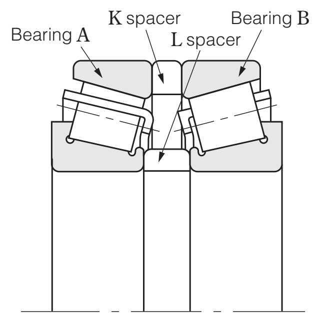


Fig. 8.20 DB Arrangement

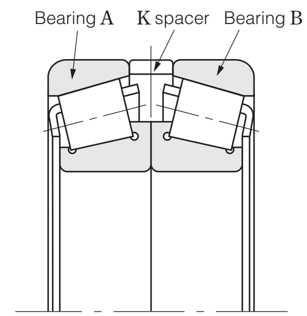


Fig. 8.21 DF Arrangement

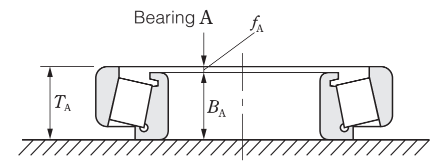


Fig. 8.22

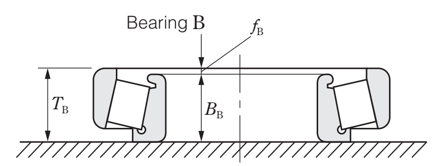


Fig. 8.23

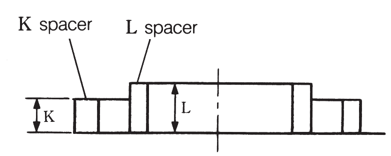


Fig. 8.24

8.3.7 Internal Clearance Adjustment When Mounting a Tapered Roller Bearing

Two single-row tapered roller bearings are usually arranged in a configuration opposite each other with clearance adjusted in the axial direction. There are two types of opposite placement methods: back-to-back (DB arrangement) and face-to-face (DF arrangement).
Clearance adjustment for a back-to-back arrangement is performed by tightening the inner ring by a shaft nut or a shaft end bolt. In Fig. 8.25, an example using a shaft-end bolt is shown. In this case, the fit on the tightened side of the inner ring with the shaft must be loose to allow displacement of the inner ring in the axial direction.
For a face-to-face arrangement (Fig. 8.26), a shim is inserted between the cover, which retains the outer ring in the axial direction, and the housing in order to allow adjustment to the specified axial clearance. In this case, use a loose fit between the tightened side of the outer ring and the housing in order to allow appropriate displacement of the outer ring in the axial direction. This is not necessary when the surrounding structure is designed to install the outer ring into a retaining cover (Fig. 8.27), allowing for both easy mounting and dismounting.
Theoretically, fatigue life is longest when bearing clearance is slightly negative during operation; however if the negative clearance is excessive, fatigue life becomes very short and heat generation quickly increases. Thus, we generally recommend that the clearance be slightly positive (slightly above zero) during operation. The bearing clearance after mounting should be decided while considering the reduction in clearance caused by the temperature difference between inner and outer rings during operation and thermal expansion of the shaft and housing in the axial direction.
In practice, clearances C1 or C2 are frequently adopted, as detailed in Table 8.17 on Page A173. In addition, the relationship between radial clearance Δ_r and axial clearance Δ_a is as follows:

$$\Delta_a = \Delta_r \cot\alpha \doteqdot \Delta_r \frac{1.5}{e}$$

where α: Contact angle
e: Constant determined for each bearing No. (listed in the bearing tables of this catalog)

Tapered roller bearings, which are used for head spindles of machine tools, automotive final reduction gears, etc., are set to a negative clearance for the purpose of obtaining bearing rigidity. This method is called a preload. There are two different types of preloading: constant-pressure preload and the more commonly used position preload.
There are two methods of position preload: one uses an already adjusted arrangement of bearings, while the other achieves the specified preload by tightening an adjustment nut or using an adjustment shim.
Constant pressure preload applies the appropriate preload to the bearing by means of a spring, hydraulic pressure, etc.
Fig. 8.28 shows an automotive final reduction gear.
For pinion gears, preload is adjusted by use of an inner ring spacer and shim. For large gears on the other hand, preload is controlled by the tightening torque of the outer ring retaining screw.
Fig. 8.29 shows the rear wheel of a truck. In this application, preload is applied by tightening the inner ring in the axial direction with a shaft nut. The preload is controlled by measuring the starting friction moment of the bearing.
Fig. 8.30 shows an example lathe head spindle where preload is controlled by tightening the shaft nut.
Fig. 8.31 shows an example of constant-pressure preload by spring displacement. In this case, the relationship between the spring's preload and displacement is used to establish the constant-pressure preload.

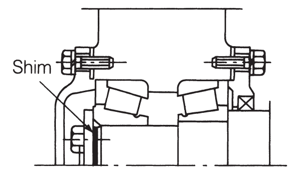


Fig. 8.25 DB Arrangement With Clearance Adjusted by Inner Rings.

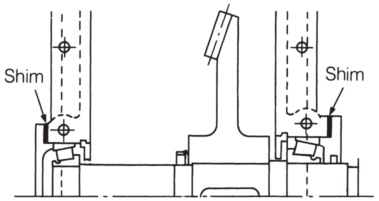


Fig. 8.26 DF Arrangement With Clearance Adjusted by Outer Rings.

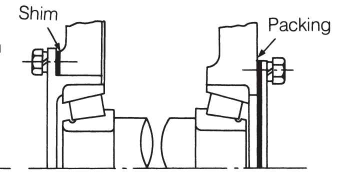


Fig. 8.27 Clearance Adjusted by Shim Thickness of Outer Ring Cover

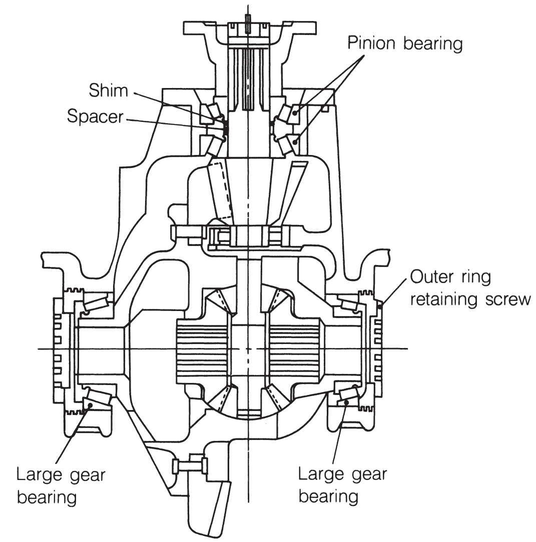


Fig. 8.28 Automotive Final Reduction Gear

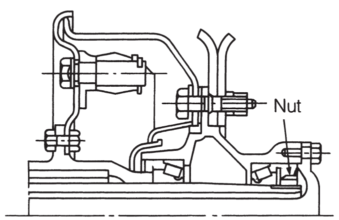


Fig. 8.29 Rear Wheel of Truck

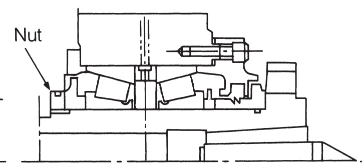


Fig. 8.30 Head Spindle of Lathe

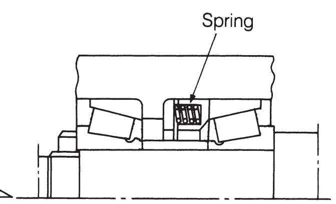
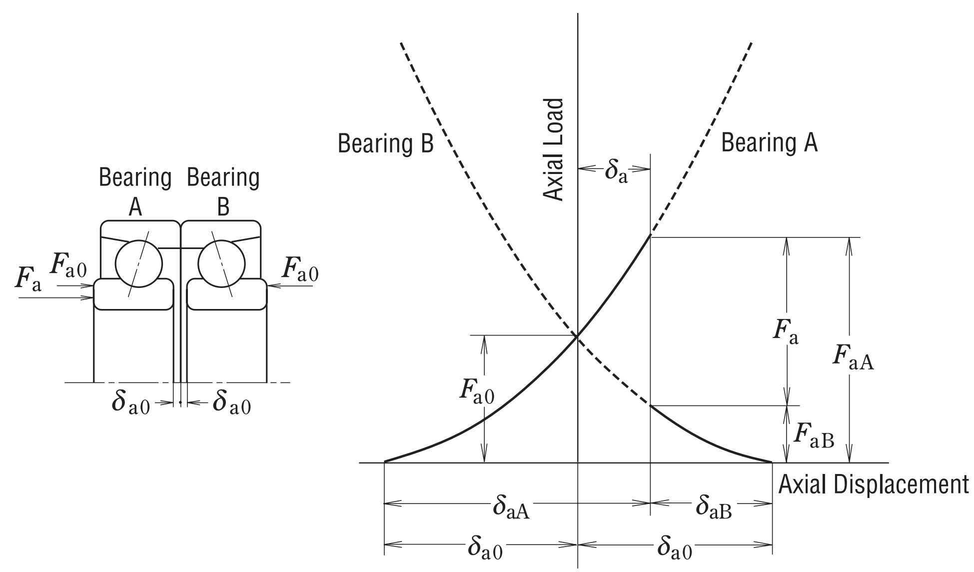


Fig. 8.31 Constant-Pressure Preload Applied by Spring


Bearing B
Axial Load
Bearing A
Bearing Bearing
A B
F_a F_{a0}
F_{a0}
δ_a
F_a
F_{aA}
F_{a0}
F_{aB}
δ_{a0} δ_{a0}
Axial Displacement
δ_{aA} δ_{aB}
δ_{a0} δ_{a0}

9. PRELOAD

9. PRELOAD

Rolling bearings usually retain some internal clearance while in operation. In some cases however, a negative clearance is desirable to keep them internally stressed. This is called "preloading". A preload is usually applied to bearings whose clearance can be adjusted during mounting, such as angular contact ball bearings or tapered roller bearings. Usually, two bearings are mounted face-to-face or back-to-back to form a paired mounting with a preload.

9.1 Purpose of Preload

The main purposes and some typical applications of preloaded bearings are as follows:

(1) To maintain the bearings in an exact position both radially and axially and to maintain the running accuracy of the shaft.
Applications: Main shafts of machine tools, precision instruments, etc.
(2) To increase bearing rigidity
Applications: Main shafts of machine tools, pinion shafts of final drive gears of automobiles, etc.
(3) To minimize noise due to axial vibration and resonance
Applications: Small electric motors, etc.
(4) To prevent sliding between the rolling elements and raceways due to unwanted gyratory sliding and spin sliding
Applications: High speed or high acceleration applications of angular contact ball bearings and thrust ball bearings
(5) To maintain the rolling elements in their proper position with the bearing rings
Applications: Thrust ball bearings and spherical thrust roller bearings mounted on a horizontal shaft

9.2 Preloading Methods

9.2.1 Position Preload

A position preload is achieved by fixing two axially opposed bearings in such a way that a preload is imposed on them. Their position, once fixed, remains unchanged while in operation.
In practice, the following three methods are generally used to obtain a position preload:

(1) Installing a paired mounting with previously adjusted stand-out dimensions (see Page A007, Fig. 1.1) and axial clearance.
(2) Using spacers or shims that have been specifically sized to obtain the required spacing and preload. (refer to Fig. 9.1)
(3) Utilizing bolts or nuts to allow adjustment of the axial preload. In this case, the starting torque should be measured to verify the proper preload.

9.2.2 Constant-Pressure Preload

A constant-pressure preload is achieved by using a coil or disc spring to impose a constant preload. Even if the relative position of the bearings changes during operation, the spring ensures that the magnitude of the preload remains relatively constant (refer to Fig. 9.2).

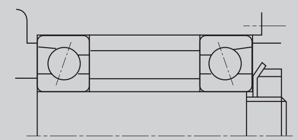


Fig. 9.1 Position Preload

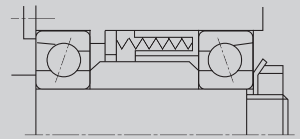


Fig. 9.2 Constant-Pressure Preload

9.3 Preload and Rigidity

9.3.1 Position Preload and Rigidity

The inner rings of Bearing A and B in the paired mounting shown in Fig. 9.3 are each displaced by δ_{a0}. When they are fixed axially, this clearance between inner rings $2\delta_{a0}$ is eliminated. Under this condition, preload F_{a0} is imposed on each bearing. A preload diagram showing bearing rigidity, or the relation between load and displacement with given axial load F_a imposed on a paired mounting, is shown in Fig. 9.4.

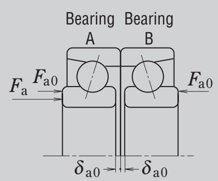


Fig. 9.3 Preloaded Back-to-Back Arrangement

9.3.2 Constant-Pressure Preload and Rigidity

A preload diagram for paired mountings under constant-pressure preload is shown in Fig. 9.5. The deflection curve of the spring is nearly parallel to the horizontal axis because the rigidity of the spring is lower than that of the bearing. As a result, rigidity under a constant-pressure preload is approximately equal to that of a single bearing with a preload F_{a0} applied to it. Fig. 9.6 presents a comparison of rigidity between a bearing with a position preload and one with a constant-pressure preload.

9.4 Selection of Preloading Method and Amount of Preload

9.4.1 Comparison of Preloading Methods

A comparison of the rigidity of both preloading methods is shown in Fig. 9.6. Position preload and constant-pressure preload may be compared as follows:

(1) When both preloads are equal, position preload provides greater bearing rigidity. In other words, bearings with position preload experience less deflection due to external loads.
(2) Under position preload, the amount of preload varies depending on such factors as differences in axial expansion due to temperature differences between the shaft and housing, differences in radial expansion due to temperature differences between the inner and outer rings, deflection due to load, etc.

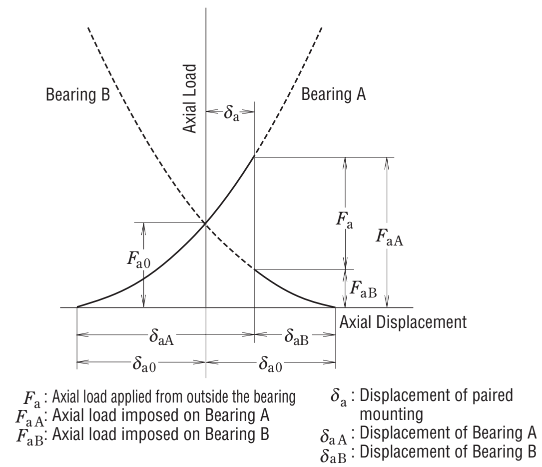


Fig. 9.4 Axial Displacement With Position Preload

Under constant-pressure preload, it is possible to minimize change in the amount of preload because the variation of the spring load with shaft expansion and contraction is negligible. Thus, position preloads are generally preferred for increasing rigidity and constant-pressure preloads are more suitable for high-speed applications, prevention of axial vibration, use with thrust bearings on horizontal shafts, etc.

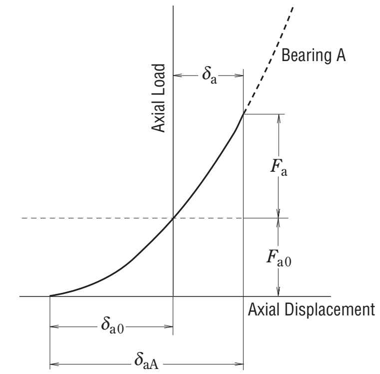


Fig. 9.5 Axial Displacement With Constant-Pressure Preload

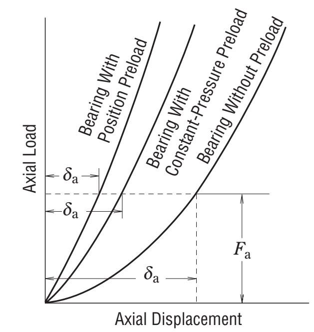


Fig. 9.6 Comparison of Rigidities and Preloading Methods

9.5 Amount of Preload

If preload is larger than necessary, abnormal heat generation, increased frictional torque, reduced fatigue life, or other negative effects may occur. The amount of preload should be carefully determined considering the operating conditions and the purpose of the preload.

9.5.1 Average Preload for Duplex Angular Contact Ball Bearings

Angular contact ball bearings are widely used in spindles for grinding, milling, high-speed turning, and similar applications. NSK divides preloads into four graduated classifications—Extra light, Light, Medium, and Heavy—to allow appropriate preload for the specific application. These four preload classes are expressed by codes EL, L, M, and H respectively when applied to DB and DF bearing sets.

The average preload and measured axial clearance for paired mounting angular contact ball bearing sets with a contact angle of 15° (widely used on machine tool spindles) are given in Tables 9.3 to 9.5.
The measuring load when measuring axial clearance is shown in Table 9.1.
The recommended axial clearance to achieve proper preload was determined for machine-tool spindles and other applications requiring ISO Class 5 and above high-precision bearing sets. The standard values given in Table 9.2 are used for the shaft/inner ring and housing/outer ring fits. Housing fits should be selected from the lower part of the standard clearance range for bearings in fixed-end applications and the higher part of the standard clearance range in free-end applications.

As a general rule, grinding machine spindles or machining center spindles require extra light to light preloads, whereas lathe spindles, which need rigidity, require medium preloads.
If the bearing set is mounted with a tight fit, preloads are greater than those shown in Tables 9.3 to 9.5. Since excessive preload causes bearing temperature rise and increases the risk of seizure among other negative effects, pay careful attention to fitting.
When speeds result in a value of $D_{pw} \times n$ ($d_m n$ value) higher than 500 000, the preload should be very carefully studied and selected. In such a case, please consult with NSK before bearing selection.

Table 9.1 Measuring Load for Axial Clearance

Nominal Outside Diameter D (mm)		Measuring Load
over	incl	(N)
10*	50	24.5
50	120	49
120	200	98
200	—	196

*10 mm is included in this range.

Table 9.2 Target Fitting

Units : μm

Bore or Outside Diameter d or D (mm)		Shaft and Inner Ring	Housing and Outer Ring
over	incl	Target Interference	Target Clearance
—	18	0 to 2	—
18	30	0 to 2.5	2 to 6
30	50	0 to 2.5	2 to 6
50	80	0 to 3	3 to 8
80	120	0 to 4	3 to 9
120	150	—	4 to 12
150	180	—	4 to 12
180	250	—	5 to 15

Table 9.3 Average Preload and Axial Clearance for Series 79C Bearings

Bearing Designation	Extra Light EL		Light L		Medium M		Heavy H	
	Preload	Axial Clearance	Preload	Axial Clearance	Preload	Axial Clearance	Preload	Axial Clearance
	(N)	(μm)	(N)	(μm)	(N)	(μm)	(N)	(μm)
7900C	7	5	16	2	29	−1	58	−6
7901C	8.6	4	16	2	41	−3	77	−8
7902C	12	3	25	0	47	−4	104	−11
7903C	11	3	25	0	56	−5	119	−12
7904C	20	1	42	−3	80	−8	152	−15
7905C	19	1	37	−2	99	−9	203	−17
7906C	25	0	46	−3	95	−8	204	−16
7907C	33	2	67	−2	149	−9	297	−18
7908C	41	1	78	−3	196	−12	384	−22
7909C	49	0	104	−5	192	−11	391	−21
7910C	49	0	95	−4	240	−13	499	−24
7911C	60	−1	111	−5	296	−15	593	−26
7912C	60	−1	113	−5	305	−15	581	−25
7913C	74	−2	151	−7	348	−16	690	−27
7914C	101	−4	205	−10	503	−22	1 004	−36
7915C	103	−4	190	−9	489	−21	997	−35
7916C	104	−4	195	−9	503	−21	986	−34
7917C	138	−6	307	−14	629	−25	1 281	−41
7918C	153	−3	289	−9	740	−23	1 488	−39
7919C	154	−3	294	−9	800	−24	1 588	−40
7920C	191	−5	387	−13	905	−28	1 790	−46

Remark The axial clearance column contains measured values.

Table 9.4 Average Preload and Axial Clearance for Series 70C Bearings

Bearing Designation	Extra Light EL		Light L		Medium M		Heavy H	
	Preload (N)	Axial Clearance (μm)	Preload (N)	Axial Clearance (μm)	Preload (N)	Axial Clearance (μm)	Preload (N)	Axial Clearance (μm)
7000C	13	3	25	0	49	−5	96	−12
7001C	13	3	25	0	57	−6	120	−14
7002C	12	3	29	−1	66	−7	147	−16
7003C	15	2	30	−1	69	−7	156	−16
7004C	25	0	49	−4	119	−12	244	−22
7005C	30	−1	58	−5	148	−14	292	−24
7006C	41	1	75	−3	195	−13	386	−24
7007C	58	−1	121	−7	251	−16	493	−28
7008C	58	−1	114	−6	291	−17	594	−30
7009C	80	−3	144	−8	338	−19	695	−33
7010C	70	−2	152	−8	388	−20	791	−34
7011C	95	−4	200	−11	479	−24	971	−40
7012C	96	−4	189	−10	526	−25	1 092	−42
7013C	130	−6	260	−13	537	−24	1 062	−39
7014C	148	−7	285	−14	732	−30	1 460	−48
7015C	151	−7	294	−14	796	−31	1 573	−49
7016C	202	−6	382	−14	921	−31	1 880	−52
7017C	205	−6	393	−14	995	−32	1 956	−52
7018C	247	−8	502	−18	1 187	−37	2 373	−60
7019C	275	−9	549	−19	1 188	−36	2 348	−58
7020C	282	−9	534	−18	1 278	−37	2 572	−60

Remark The axial clearance column contains measured values.

Table 9.5 Average Preload and Axial Clearance for Series 72C Bearings

Bearing Designation	Extra Light EL		Light L		Medium M		Heavy H	
	Preload (N)	Axial Clearance (μm)	Preload (N)	Axial Clearance (μm)	Preload (N)	Axial Clearance (μm)	Preload (N)	Axial Clearance (μm)
7200C	13	3	29	−1	68	−8	150	−18
7201C	20	1	39	−3	99	−12	197	−22
7202C	20	1	40	−3	97	−11	199	−21
7203C	25	0	46	−4	146	−16	296	−28
7204C	35	−2	68	−7	196	−20	384	−33
7205C	42	1	82	−4	193	−14	402	−27
7206C	57	−1	114	−7	292	−20	591	−35
7207C	75	−3	151	−10	385	−25	794	−43
7208C	98	−5	202	−13	501	−29	985	−47
7209C	123	−7	254	−16	534	−30	1 067	−49
7210C	127	−7	248	−15	590	−31	1 171	−50
7211C	142	−8	289	−17	788	−38	1 554	−60
7212C	190	−11	397	−22	928	−42	1 878	−67
7213C	219	−12	448	−23	1 069	−44	2 175	−70
7214C	243	−9	484	−20	1 164	−42	2 368	−69
7215C	270	−10	530	−21	1 224	−42	2 445	−68
7216C	305	−12	595	−24	1 367	−47	2 752	−76
7217C	355	−14	697	−27	1 658	−53	3 358	−85
7218C	384	−15	771	−29	1 865	−57	3 713	−90
7219C	448	−18	876	−33	2 081	−63	4 153	−99
7220C	503	−20	984	−36	2 337	−68	4 700	−107

Remark The axial clearance column contains measured values.

9.5.2 Preload of Thrust Ball Bearings

When the balls in thrust ball bearings rotate at relatively high speeds, sliding due to gyroscopic moments on the balls may occur. The larger of the two values obtained from Equations (9.1) and (9.2) below should be adopted as the minimum axial load in order to prevent such sliding:

$$F_{a\,min} = \frac{C_{0a}}{100}\left(\frac{n}{N_{max}}\right)^2 \quad \cdots\cdots\cdots (\mathbf{9.1})$$

$$F_{a\,min} = \frac{C_{0a}}{1000} \quad \cdots\cdots\cdots (\mathbf{9.2})$$

where $F_{a\,min}$: Minimum axial load (N), {kgf}
C_{0a} : Basic static load rating (N), {kgf}
n : Speed (min^{-1})
N_{max} : Limiting speed (oil lubrication) (min^{-1})

9.5.3 Preload of Spherical Thrust Roller Bearings

When spherical thrust roller bearings are used, damage such as scoring may occur due to sliding between the rollers and outer ring raceway. The minimum axial load $F_{a\,min}$ necessary to prevent such sliding is obtained from the following equation:

$$F_{a\,min} = \frac{C_{0a}}{1000} \quad \cdots\cdots\cdots (\mathbf{9.3})$$

9.6 Technical Data

9.6.1 Load and Displacement of Position-Preloaded Bearings

Bearing arrangements refer to multiple identical ball or tapered roller bearings mounted side by side as a set. When two bearings are used, they are often referred to as paired mountings or duplex bearings.
In bearing arrangements for machine tool spindles, single-row angular contact ball bearings are most often used for their high rigidity to reduce bearing displacement under load.
Paired mountings fall into three types: back-to-back, with load lines that diverge toward the bearing axis; face-to-face, with load lines that converge toward the bearing axis; and tandem, with parallel load lines. The designations for these arrangements are DB, DF, and DT respectively (Fig. 9.7).

Different arrangements are used depending on the application. DB and DF arrangements can take axial loads in either direction. Since the distances of the load centers of DB arrangements are longer than those of DF arrangements, they are widely used in applications where moment loads occur. DT arrangements can only take axial loads in one direction; however, because the two bearings share load equally between them, this type can be used when load in one direction is large.
By selecting DB or DF arrangements preset with proper preloads, radial and axial displacement of the bearing inner and outer rings can be reduced as much as possible (DT arrangements can not be preloaded in this way). The amount of preload can be adjusted by changing the clearance between bearings δ_{a0}, as shown in Figs. 9.9 to 9.11. Preloads are divided into four graduated classifications: Extra Light (EL), Light (L), Medium (M), and Heavy (H). DB and DF arrangements are often used for applications where shaft misalignments and displacements due to load must be minimized. Arrangements of three bearings (triplex) are designated DBD, DFD, and DTD, as shown in Fig. 9.8. Sets of four or five bearings can also be used depending on application requirements. Paired mountings are often used with a preload. Since preload affects affects various things, such as the rise in bearing temperature during operation, torque, bearing noise, and especially bearing life, it is extremely important to avoid applying excessive preload.

$$\delta_a = c\,F_a^{2/3} \quad \cdots\cdots \quad \textbf{(9.4)}$$

where c: Constant depending on the bearing type and dimensions.

Fig. 9.9 shows the preload curves of a DB arrangement, and Figs. 9.10 and 9.11 show preload curves for a DBD arrangement. If the inner rings of a paired mounting as in Fig. 9.9 are pressed axially, A-side and B-side bearings are deformed by δ_{a0A} and δ_{a0B} respectively and the clearance (between the inner rings) δ_{a0} becomes zero. This condition means that preload F_{a0} is applied on the arrangement. If external axial load F_a is applied on the preloaded arrangement from the A-side, then the A-side bearing will be deformed by an additional δ_{a1} and the displacement of B-side bearing will be reduced to the same amount as the A-side bearing displacement δ_{a1}. Therefore, the displacements of A- and B-side bearings are $\delta_{aA} = \delta_{a0A} + \delta_{a1}$ and $\delta_{aB} = \delta_{a0B} + \delta_{a1}$ respectively. In other words, the load on the A-side bearing including preload is $F_{a0} + F_a - F_a'$ and that on the B-side is $F_{a0} - F_a'$.

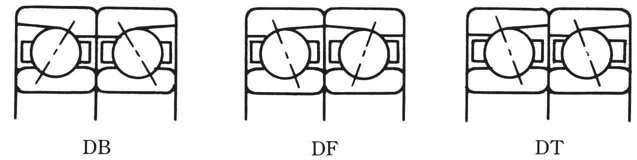


Fig. 9.7 Paired Mounting Bearing Arrangements

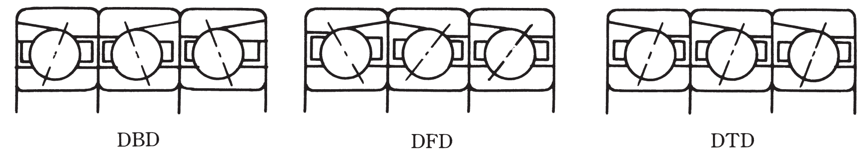


Fig. 9.8 Triplex Bearing Arrangements

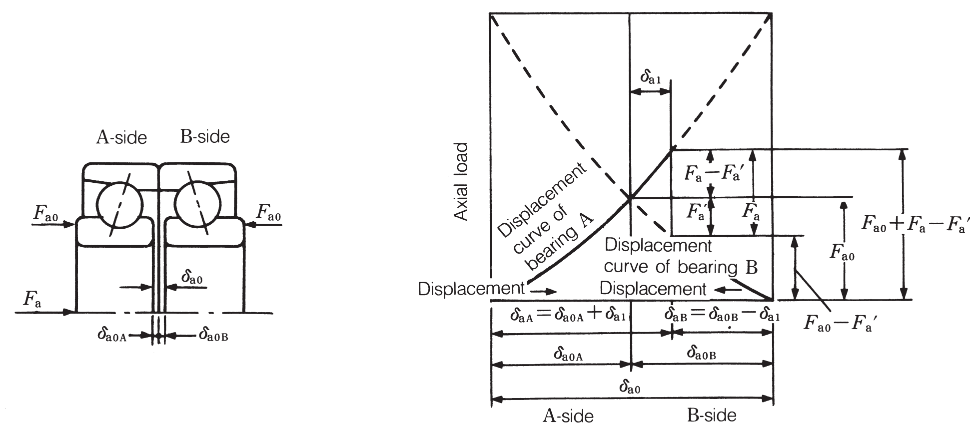


Fig. 9.9 Preload Graph of DB Back-to-Back Arrangement

If the bearing set has an applied preload, the A-side bearing should have sufficient life and load capacity for an axial load ($F_{a0}+F_a-F_a'$) under operating speed conditions. The axial clearance δ_{a0} is shown in Tables 9.3 to 9.5 of Section 9.5.1 (Pages A195 to A197).

Fig. 9.10 shows external axial load F_a applied on AA-side bearings. The axial loads and displacements of AA- and B-side bearings are summarized in Table 9.6.

Fig. 9.11 shows external axial load F_a applied on an A-side bearing. The axial loads and displacements of A- and BB-side bearings are summarized in Table 9.7.

Figs. 9.12 to 9.17 show the relationship of axial load to axial displacement for DB and DBD arrangements of 7018C and 7018A bearings under several preload ranges.

Table 9.6

Direction	Displacement	Axial load
AA-side	$\delta_{a0A}+\delta_{a1}$	$F_{a0}+F_a-F_a'$
B-side	$\delta_{a0B}-\delta_{a1}$	$F_{a0}-F_a'$

Table 9.7

Direction	Displacement	Axial load
A-side	$\delta_{a0A}+\delta_{a1}$	$F_{a0}+F_a-F_a'$
BB-side	$\delta_{a0B}-\delta_{a1}$	$F_{a0}-F_a'$

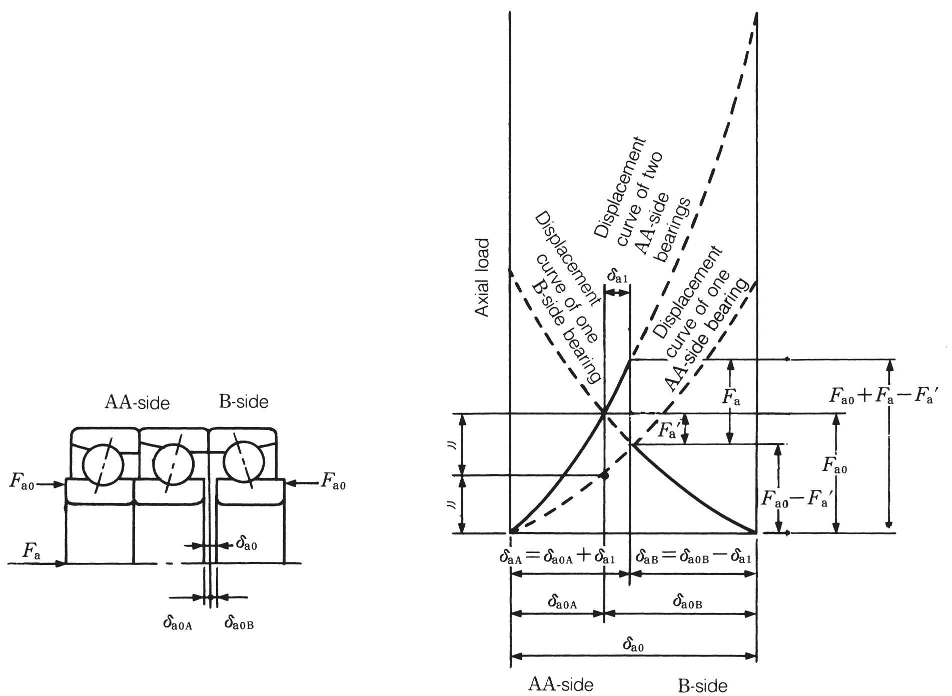


Fig. 9.10 Preload Graph for a DBD Arrangement (Axial Load Applied From the AA-Side)

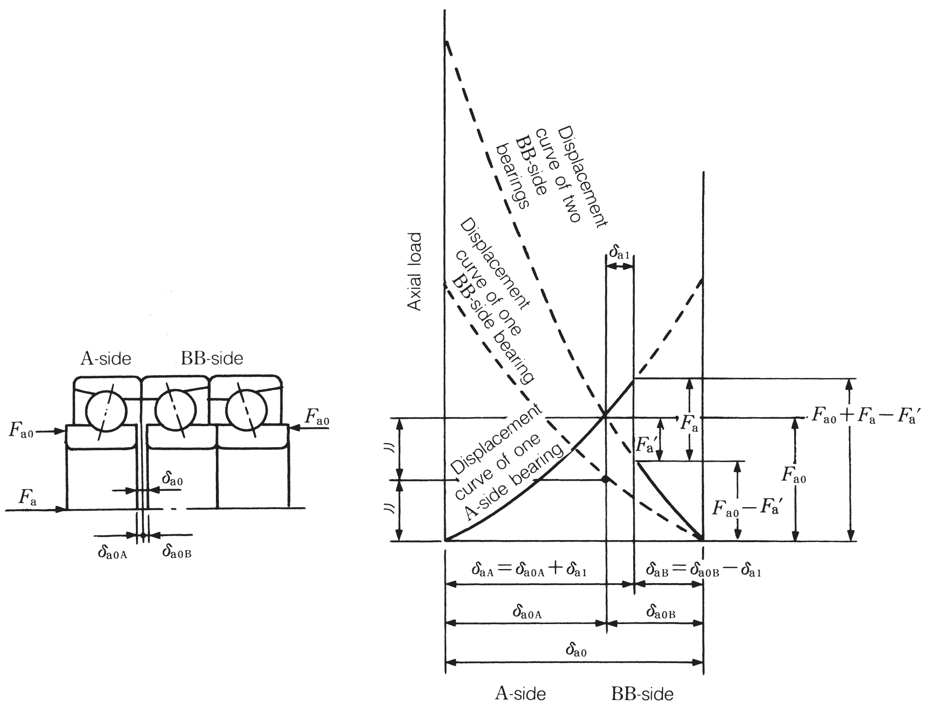


Fig. 9.11 Preload Graph for a DBD Arrangement (Axial Load Applied From A-Side)

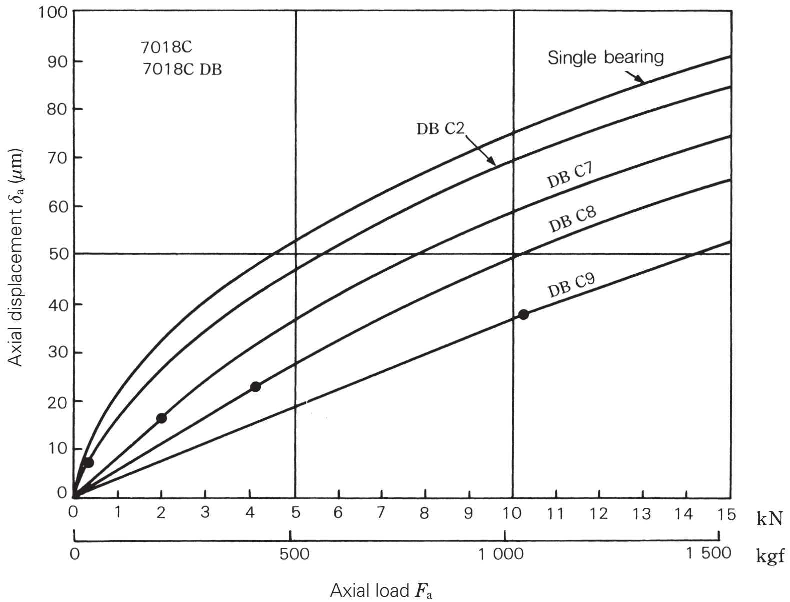


Fig. 9.12

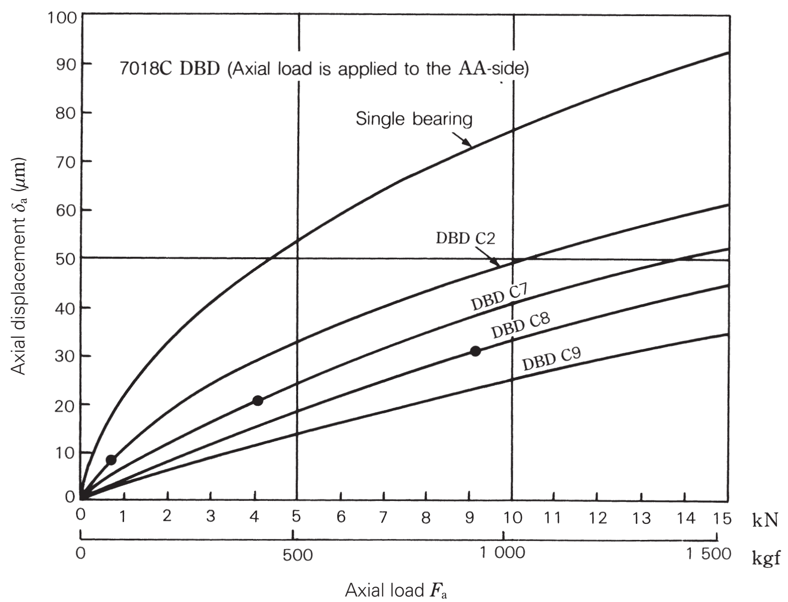


Fig. 9.13

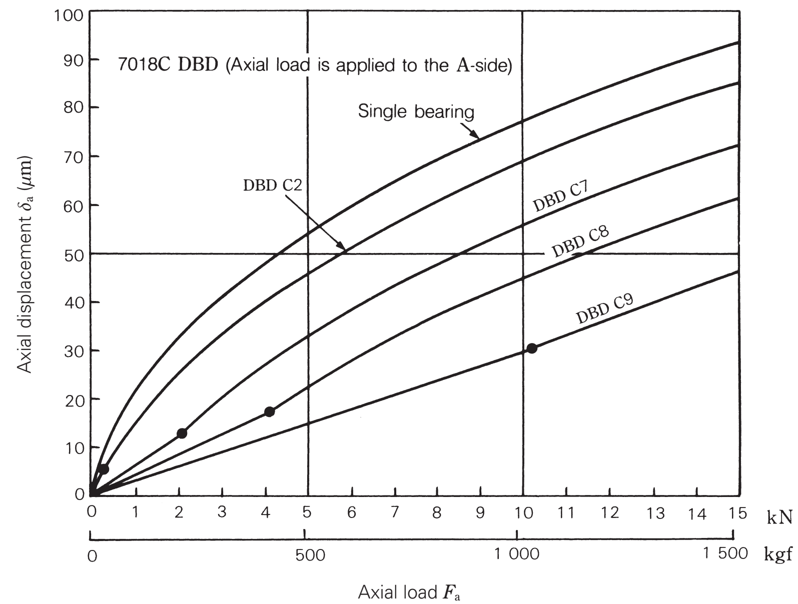


Fig. 9.14

Remark The (•) mark on the axial load or displacement curve indicates the point where the preload is zero. Therefore, if the axial load is larger than this, the opposed bearing does not impose a load.

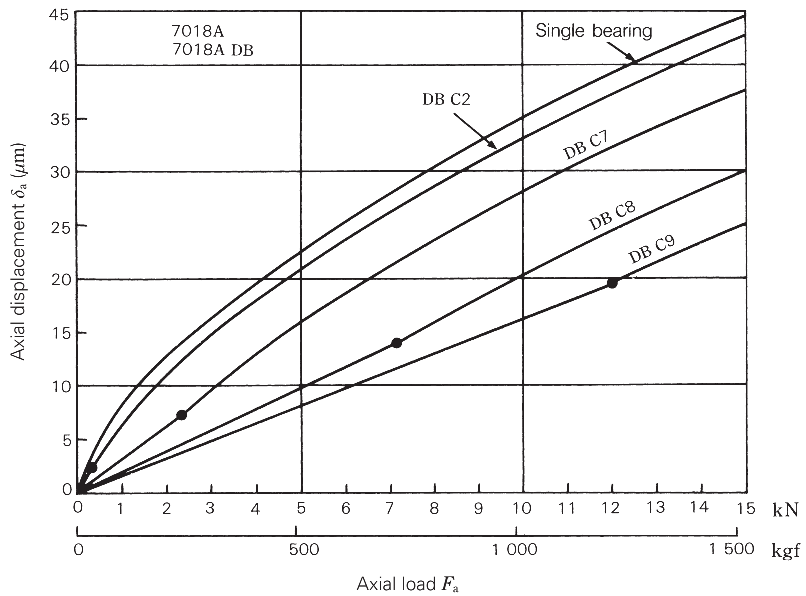


Fig. 9.15

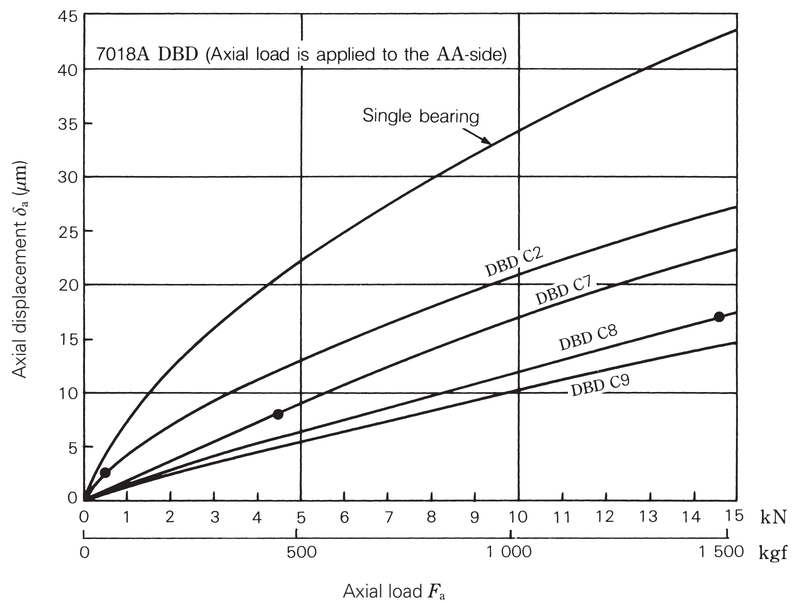


Fig. 9.16

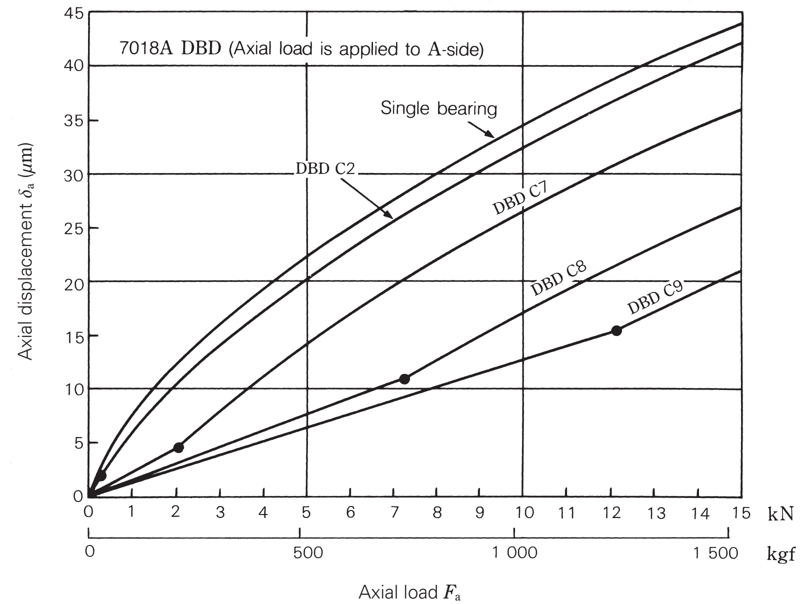


Fig. 9.17

Remark The (•) mark on the axial load or displacement curve indicates the point where the preload is zero. Therefore, if the axial load is larger than this, the opposed bearing does not impose a load.

9.6.2 Axial Displacement of Deep Groove Ball Bearings

When an axial load F_a is applied to a radial bearing with contact angle α_0 and the inner ring is displaced δ_a, the center of the inner ring raceway radius Oi is also moved to Oi' resulting in contact angle α as shown in Fig. 9.18. If δ_N represents the elastic deformation of the raceway and ball in the direction of the rolling element load Q, Equation (9.5) is derived from Fig. 9.18.

$$(m_0+\delta_N)^2=(m_0\cdot\sin\alpha_0+\delta_a)^2+(m_0\cdot\cos\alpha_0)^2$$

$$\therefore\ \delta_N=m_0\left\{\sqrt{\left(\sin\alpha_0+\frac{\delta_a}{m_0}\right)^2+\cos^2\alpha_0}-1\right\} \quad \cdots\cdots\cdots\cdots \textbf{(9.5)}$$

In addition, the following relationship exists between rolling element load Q and elastic deformation δ_N.

$$Q=K_N\cdot\delta_N^{3/2} \quad \cdots\cdots\cdots\cdots \textbf{(9.6)}$$

where K_N: Constant depending on bearing material, type, and dimensions

If we introduce the relation of

$$m_0=\left(\frac{r_e}{D_w}+\frac{r_i}{D_w}-1\right)D_w=B\cdot D_w$$

Equations (9.5) and (9.6) are

$$Q=K_N\ (B\cdot D_w)^{3/2}\left\{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}-1\right\}^{3/2}$$

where $h=\dfrac{\delta_a}{m_0}=\dfrac{\delta_a}{B\cdot D_w}$

If we introduce the relation of $K_N=K\cdot\dfrac{\sqrt{D_w}}{B^{3/2}}$

$$Q=K\cdot D_w^2\left\{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}-1\right\}^{3/2} \quad \cdots\cdots\cdots\cdots \textbf{(9.7)}$$

On the other hand, the relation between bearing axial load and rolling element load is shown in Equation (9.8) using Fig. 9.19:

$$F_a=Z\cdot Q\cdot\sin\alpha \quad \cdots\cdots\cdots\cdots \textbf{(9.8)}$$

Based on Fig. 9.18, we obtain,

$$(m_0+\delta_N)\ \sin\alpha=m_0\cdot\sin\alpha_0+\delta_a$$

$$\therefore\ \sin\alpha=\frac{m_0\cdot\sin\alpha_0+\delta_a}{m_0+\delta_N}=\frac{\sin\alpha_0+h}{1+\dfrac{\delta_N}{m_0}}$$

If we substitute Equation (9.5),

$$\sin\alpha=\frac{\sin\alpha_0+h}{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}} \quad \cdots\cdots\cdots\cdots \textbf{(9.9)}$$

In other words, the relation between the bearing axial load F_a and axial displacement δ_a can be obtained by substituting Equations (9.7) and (9.9) for Equation (9.8).

$$F_a=K\cdot Z\cdot D_w^2\cdot\frac{\left\{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}-1\right\}^{3/2}\times(\sin\alpha_0+h)}{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}} \quad \cdots\cdots\cdots\cdots \textbf{(9.10)}$$

where K: Constant depending on bearing material and design
D_w: Ball diameter
Z: Number of balls
α_0: Initial contact angle
The initial contact angle for single-row deep groove ball bearings can be obtained using Equation (5) on Page C012

Actual axial deformation varies depending on bearing mounting conditions and factors such as the material and thickness of the shaft and housing and bearing fitting. For more details, consult with NSK.

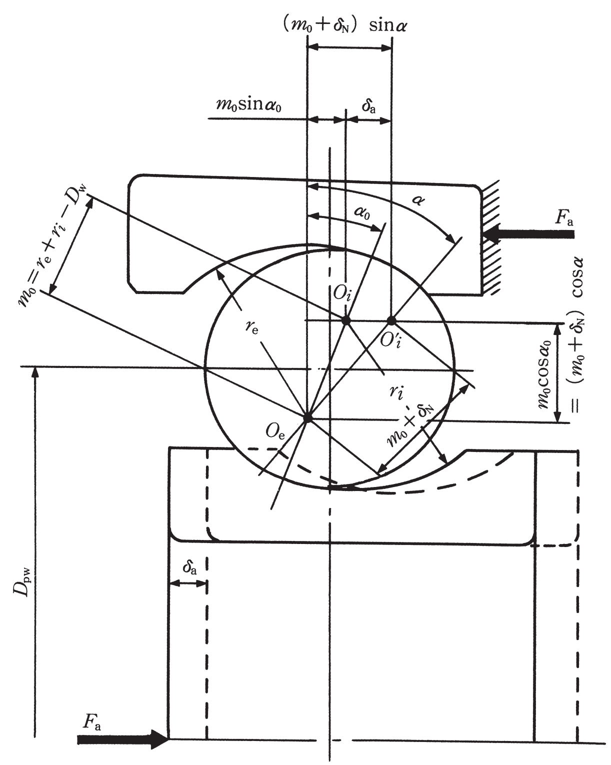


Fig. 9.18

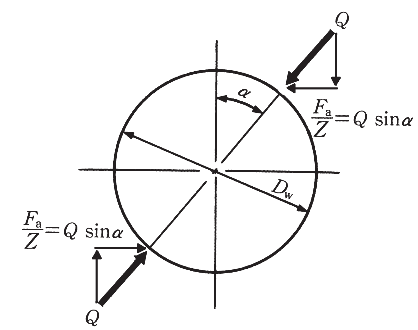


Fig. 9.19

Fig. 9.20 shows the relation between axial load and axial displacement for Series 6210 and 6310 single-row deep groove ball bearings with initial contact angles of 0°, 10°, and 15°. The larger the initial contact angle α_0, the more rigid the bearing will be in the axial direction and the smaller the difference between the axial displacements of 6210 and 6310 under identical axial load. The initial contact angle α_0 depends on the groove radius and the radial clearance.

Fig. 9.21 shows the relation between axial load and axial displacement for Series 72 angular contact ball bearings with initial contact angles of 15° (C), 30° (A), and 40° (B). Series 70 and 73 bearings with identical contact angles and bore diameters can be considered to have almost the same values as Series 72 bearings. Angular contact ball bearings that sustain loads in the axial direction must maintain their running accuracy and reduce elastic deformation from applied loads when used as multiple bearing arrangements with a preload.

To determine the preload to keep the elastic deformation caused by applied loads within the required limits, it is important to know the characteristics of load vs. deformation. The relationship between load and displacement can be expressed by Equation (9.10) as $F_a \propto \delta_a^{3/2}$ or $\delta_a \propto F_a^{2/3}$. In other words, axial displacement δ_a is proportional to axial load F_a to the 2/3 power. When this axial load index is less than one, relative axial displacement will be small, with only a small increase in axial load (Fig. 9.21). The underlying reason for applying a preload is to reduce the amount of displacement.

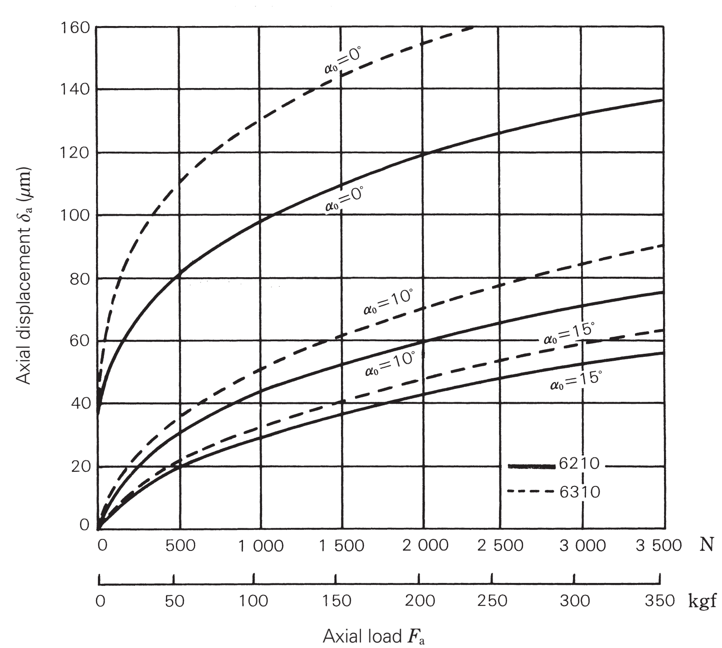


Fig. 9.20 Axial Load and Axial Displacement of Deep Groove Ball Bearings

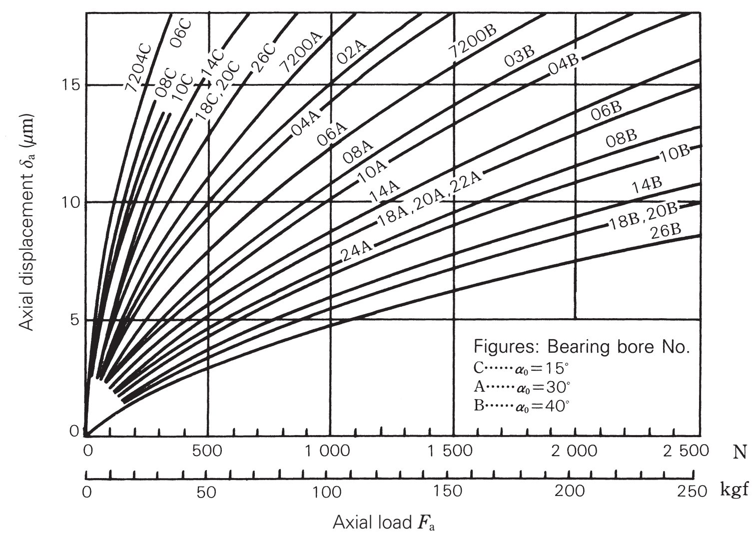


Fig. 9.21 Axial Load and Axial Displacement of Angular Contact Ball Bearings

9.6.3 Axial Displacement of Tapered Roller Bearings

Tapered roller bearings are widely used in pairs like angular contact ball bearings. Take care when selecting appropriate tapered roller bearings.
For example, the bearings of machine-tool head spindles and automobile differential pinions are preloaded to increase shaft rigidity.
When a bearing with an applied preload is to be used, it is essential to have some knowledge of the relationship between axial load and axial displacement. For tapered roller bearings, the axial displacement calculated using Palmgren's method in Equation (9.11) generally agrees well with actual measured values.
Actual axial deformation varies depending on the bearing mounting conditions, such as the material and thickness of the shaft and housing and the bearing fitting. For more details, consult with NSK.

$$\left.\begin{aligned}\delta_a &= \frac{0.000077}{\sin\alpha}\cdot\frac{Q^{0.9}}{L_{we}^{\ 0.8}} && \text{(N)}\\ &= \frac{0.0006}{\sin\alpha}\cdot\frac{Q^{0.9}}{L_{we}^{\ 0.8}} && \{\text{kgf}\}\end{aligned}\right\} \quad \cdots\cdots \ (\mathbf{9.11})$$

where, δ_a: Axial displacement of inner, outer ring (mm)
α: Contact angle...1/2 the outer ring angle (°) (Refer to Fig. 9.22)
Q: Load on rolling elements (N), {kgf}

$$Q = \frac{F_a}{Z\sin\alpha}$$

L_{we}: Length of effective contact on roller (mm)
F_a: Axial load (N), {kgf}
Z: Number of rollers

Equation (9.11) can also be expressed as Equation (9.12).

$$\delta_a = K_a \cdot F_a^{\ 0.9} \quad \cdots\cdots \ (\mathbf{9.12})$$

where,

$$K_a = \frac{0.000077}{(\sin\alpha)^{1.9}\, Z^{0.9}\, L_{we}^{\ 0.8}} \quad \cdots\cdots \ \text{(N)}$$

$$= \frac{0.0006}{(\sin\alpha)^{1.9}\, Z^{0.9}\, L_{we}^{\ 0.8}} \quad \cdots\cdots \ \{\text{kgf}\}$$

Here K_a refers to the coefficient determined by bearing internal design.
Axial loads and axial displacement for tapered roller bearings are plotted in Fig. 9.23.
The amount of axial displacement of tapered roller bearings is proportional to the axial load raised to the 0.9 power. The displacement of ball bearings is proportional to the axial load raised to the 0.67 power; thus, the preload required to control displacement is much greater for ball bearings than for tapered roller bearings.
Take caution not to make the preload indiscriminately large on tapered roller bearings, since too large of a preload can cause excessive heat, seizure, and reduced bearing life.

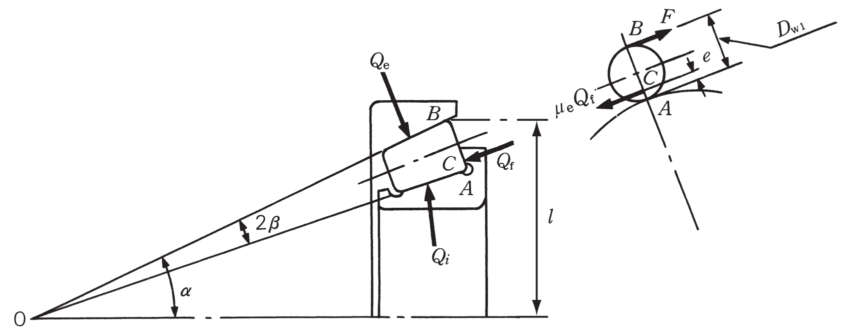


Fig. 9.22

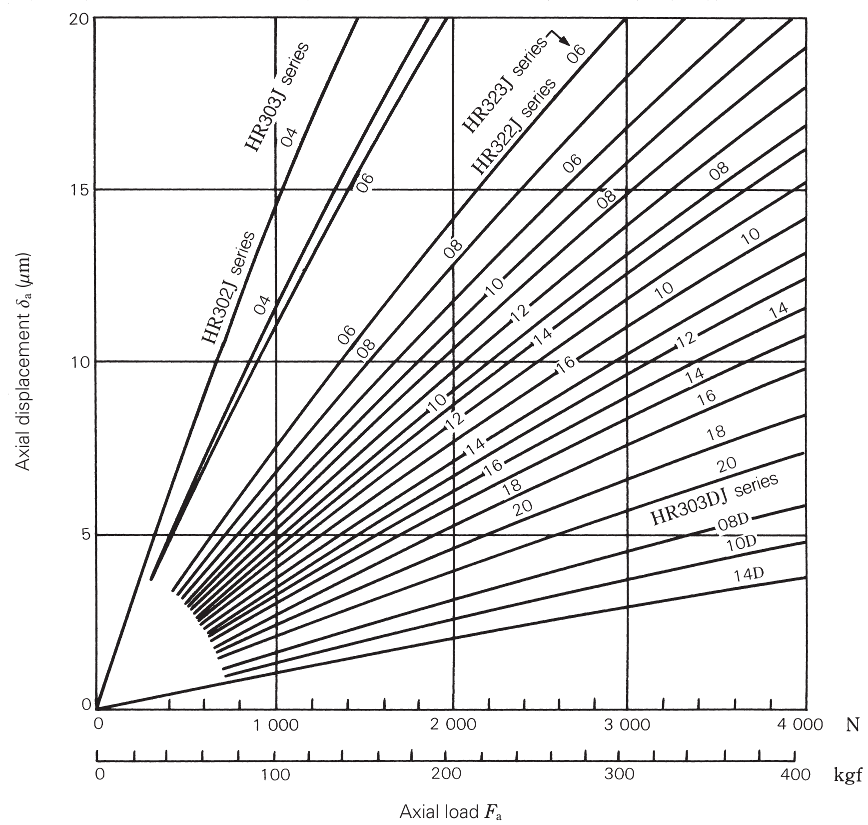
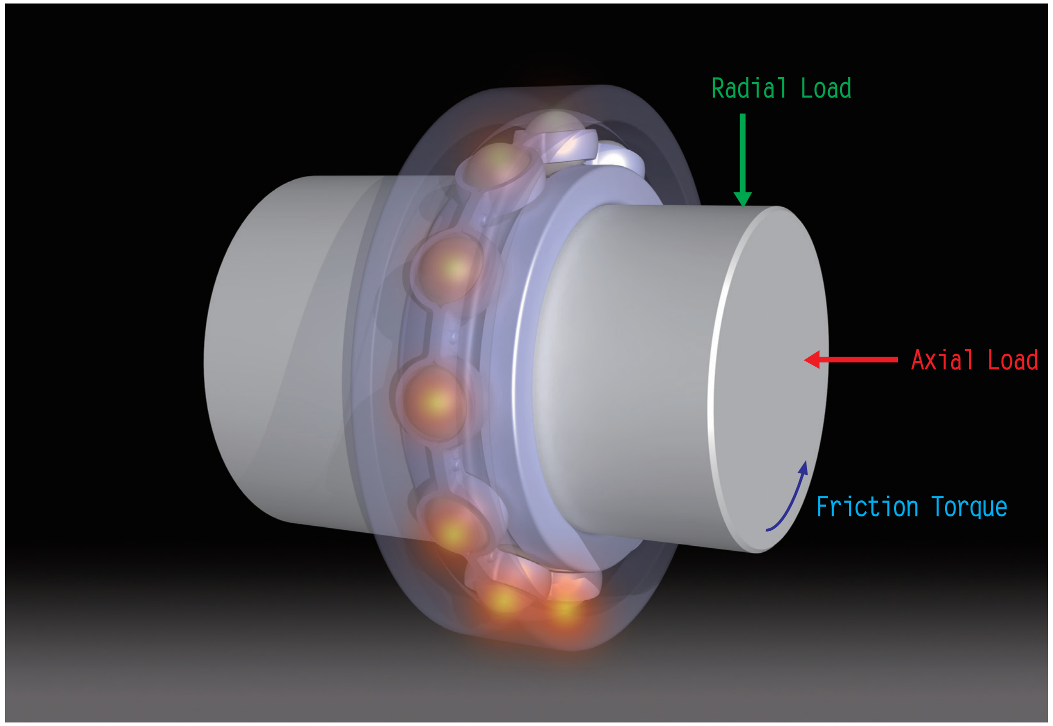


Fig. 9.23 Axial Load and Axial Displacement for Tapered Roller Bearings



10. FRICTION

10. FRICTION

10.1 Coefficients of Dynamic Friction

10.1.1 Bearing Types and Their Coefficients of Dynamic Friction μ

$$\mu=\frac{M}{P\cdot\frac{d}{2}} \quad\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \quad \textbf{(10.1)}$$

M : Dynamic friction torque (N·mm), {kgf·mm}
P : Load on a bearing (dynamic equivalent load) (N), {kgf}
d : Shaft diameter, inner ring bore diameter (mm)

Table 10.1 Coefficients of Dynamic Friction

Bearing Types	Approximate Values of μ
Deep Groove Ball Bearings	0.0013
Angular Contact Ball Bearings	0.0015
Self-Aligning Ball Bearings	0.0010
Thrust Ball Bearings	0.0011
Cylindrical Roller Bearings	0.0010
Tapered Roller Bearings	0.0022
Spherical Roller Bearings	0.0028
Needle Roller Bearings With Cages	0.0015
Full Complement Needle Roller Bearings	0.0025
Spherical Thrust Roller Bearings	0.0028

10.2 Empirical Equations for Running Torque

Dynamic bearing torque (heat generation)
$M=M_l+M_v$

Load term (determined by bearing type and load)
$M_l=f_1 F d_m$
where f_1 : Coefficient determined by bearing type and load
F : Load
d_m : Pitch circle diameter of rolling element

Speed term (determined by oil viscosity, amount, speed)
$M_v=f_0(\nu_0 n)^{2/3} dm^3$
where f_0 : Coefficient determined by bearing and lubricating method
ν_0 : Kinematic viscosity of oil
n : Speed

10.3 Technical Data

10.3.1 Preload and Starting Torque for Angular Contact Ball Bearings

Angular contact ball bearings, like tapered roller bearings, are most often used in pairs rather than alone or in other multiple bearing sets. Back-to-back and face-to-face bearing sets can be preloaded to adjust bearing rigidity. Extra Light (EL), Light (L), Medium (M), and Heavy (H) are standard preloads. Friction torque of the bearing will increase in direct proportion to the preload.
The starting torque of angular contact ball bearings is mainly caused by angular slippage between the balls and contact surfaces on the inner and outer rings. Starting torque for the bearing M due to such spin is given by the following equation:

$$M=M_s\cdot Z\sin\alpha \text{ (mN·m), \{kgf·mm\}} \quad\cdots\cdots\cdots \quad \textbf{(10.2)}$$

where M_s: Spin friction for contact angle α centered on the shaft,

$$M_s=\frac{3}{8}\mu_s\cdot Q\cdot a\cdot E(k) \quad \text{(mN·m), \{kgf·mm\}}$$

μ_s: Contact-surface slip friction coefficient
Q: Load on rolling elements (N), {kgf}
a: (1/2) of contact-ellipse major axis (mm)
$E(k)$: With $k=\sqrt{1-\left(\frac{b}{a}\right)^2}$ as the population parameter, second class complete ellipsoidal integration
b: (1/2) of contact-ellipse minor axis (mm)
Z: Number of balls
α: Contact angle (°)

Actual measurements with 15° angular contact ball bearings correlate well with calculated results using μ_s = 0.15 in Equation (10.2). Fig. 10.1 shows the calculated friction torque for Series 70C and 72C bearings.

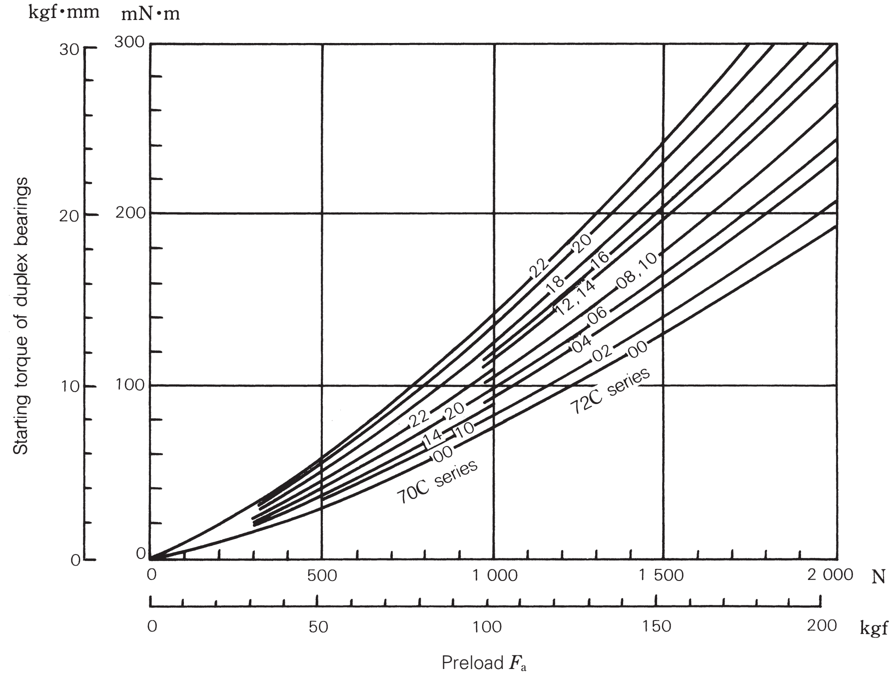


Fig. 10.1 Preload and Starting Torque for DF and DB Angular Contact Ball Bearing (α=15°) Arrangments

10.3.2 Empirical Equations for Running Torque of High-Speed Ball Bearings

Empirical equations are presented below for the running torque of high speed ball bearings subject to axial loading and jet lubrication. These equations are based on the results of tests of angular contact ball bearings with bore diameters of 10 to 30 mm, but they can be extrapolated to larger bearings.

Running torque M can be obtained as the sum of a load term M_l and speed term M_v as follows:

$$M = M_l + M_v \ (\mathrm{mN \cdot m}),\ \{\mathrm{kgf \cdot mm}\} \quad \cdots\cdots\cdots \quad \mathbf{(10.3)}$$

Load term M_l refers to friction with no relation to speed or fluid friction and is expressed by the experimentally-based Equation (10.4).

$$\left.\begin{aligned} M_l &= 0.672\times10^{-3} D_{pw}^{0.7} F_a^{1.2}\ (\mathrm{mN\cdot m}) \\ &= 1.06\times10^{-3} D_{pw}^{0.7} F_a^{1.2}\ \{\mathrm{kgf\cdot mm}\} \end{aligned}\right\}\cdots\cdots \quad \mathbf{(10.4)}$$

where, D_{pw}: Pitch diameter of rolling elements (mm)
F_a: Axial load (N), {kgf}

Speed term M_v refers to fluid friction, which depends on angular speed and is expressed by Equation (10.5).

$$\begin{aligned} M_v &= 3.47\times10^{-10} D_{pw}^{3} n_i^{1.4} Z_B^{a} Q^{b}\ (\mathrm{mN\cdot m}) \\ &= 3.54\times10^{-11} D_{pw}^{3} n_i^{1.4} Z_B^{a} Q^{b}\ \{\mathrm{kgf\cdot mm}\} \end{aligned} \quad \cdots\cdots \quad \mathbf{(10.5)}$$

where, n_i: Inner ring speed (min^{-1})
Z_B: Absolute viscosity of oil at outer ring temperature (mPa·s), {cp}
Q: Oil flow rate (kg/min)

Exponents a and b, which affect oil viscosity and flow rate factors, depend only on angular speed and are given by Equations (10.6) and (10.7) as follows:

$$a = 24 n_i^{-0.37} \quad \cdots\cdots\cdots \quad \mathbf{(10.6)}$$

$$b = 4\times10^{-9} n_i^{1.6} + 0.03 \quad \cdots\cdots\cdots \quad \mathbf{(10.7)}$$

An example estimation for the running torque of high-speed ball bearings is shown in Fig. 10.2. A comparison of values calculated using these equations and actual measurements is shown in Fig. 10.3. When the contact angle exceeds 30°, the influence of spin friction becomes large, so the running torque given by the equations will be low.

Calculation Example

Obtain the running torque of high speed angular contact ball bearing 20BNT02 (ϕ20 × ϕ47 × 14) under the following conditions:

$n_i = 70\,000\ \mathrm{min}^{-1}$
$F_a = 590$ N, {60 kgf}
Lubrication: Jet, oil viscosity:
10 mPa·s {10 cp}
oil flow: 1.5 kg/min

From Equation (10.4),

$$\begin{aligned} M_l &= 0.672\times10^{-3} D_{pw}^{0.7} F_a^{1.2} \\ &= 0.672\times10^{-3}\times33.5^{0.7}\times590^{1.2} \\ &= 16.6\ (\mathrm{mN\cdot m}) \\ M_l &= 1.06\times10^{-3}\times33.5^{0.7}\times60^{1.2} \\ &= 1.7\ \{\mathrm{kgf\cdot mm}\} \end{aligned}$$

From Equations (10.6) and (10.7),

$$\begin{aligned} a &= 24 n_i^{-0.37} \\ &= 24\times70\,000^{-0.37} = 0.39 \\ b &= 4\times10^{-9} n_i^{1.6} + 0.03 \\ &= 4\times10^{-9}\times70\,000^{1.6} + 0.03 = 0.26 \end{aligned}$$

From Equation (10.5),

$$\begin{aligned} M_v &= 3.47\times10^{-10} D_{pw}^{3} n_i^{1.4} Z_B^{a} Q^{b} \\ &= 3.47\times10^{-10}\times33.5^{3}\times70\,000^{1.4}\times10^{0.39}\times1.5^{0.26} \\ &= 216\ (\mathrm{mN\cdot m}) \end{aligned}$$

$$\begin{aligned} M_v &= 3.54\times10^{-11}\times33.5^{3}\times70\,000^{1.4}\times10^{0.39}\times1.5^{0.26} \\ &= 22.0\ \{\mathrm{kgf\cdot mm}\} \end{aligned}$$

$M = M_l + M_v = 16.6 + 216 = 232.6$ (mN·m)
$M = M_l + M_v = 1.7 + 22 = 23.7$ {kgf·mm}

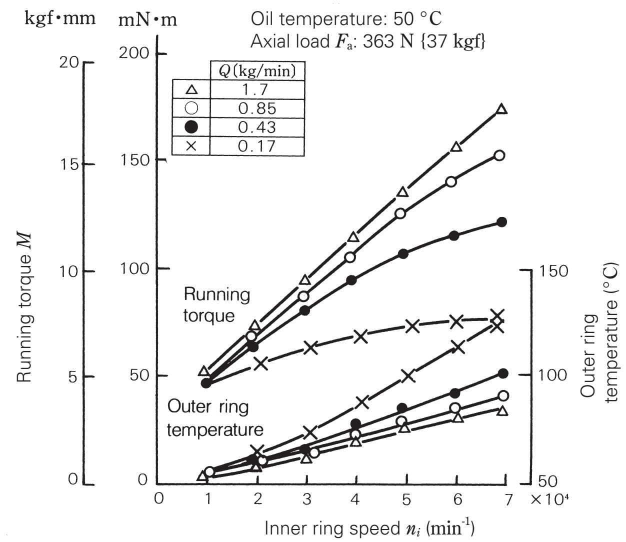


Fig. 10.2 Typical Test Example

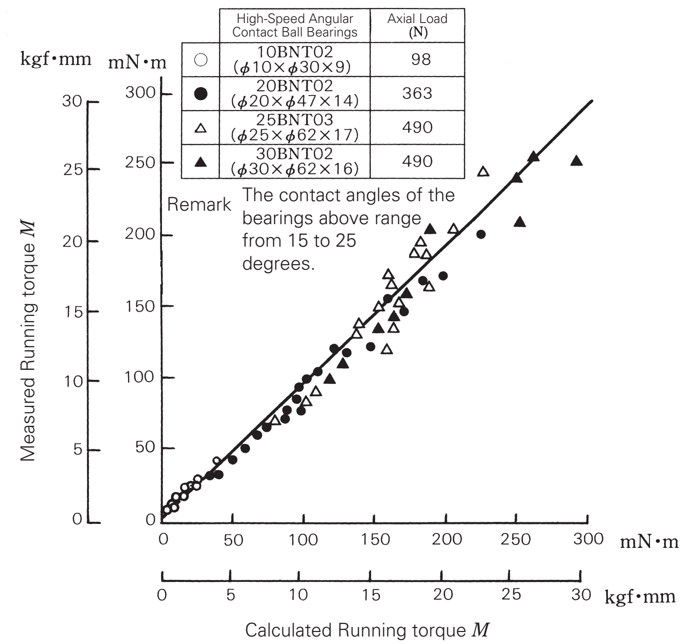


Fig. 10.3 Comparison of Actual Measurements and Calculated Values

10.3.3 Preload and Starting Torque for Tapered Roller Bearings

The balance of loads on the bearing rollers when a tapered roller bearing is subjected to axial load F_a is expressed by the following three Equations:

$$Q_e = \frac{F_a}{Z \sin \alpha} \quad \cdots\cdots \quad (10.8)$$

$$Q_i = Q_e \cos 2\beta = \frac{\cos 2\beta}{Z \sin \alpha} F_a \quad \cdots\cdots \quad (10.9)$$

$$Q_f = Q_e \sin 2\beta = \frac{\sin 2\beta}{Z \sin \alpha} F_a \quad \cdots\cdots \quad (10.10)$$

where Q_e : Rolling element load on outer ring (N), {kgf}
Q_i : Rolling element load on inner ring (N), {kgf}
Q_f : Rolling element load on inner ring large end rib, (N), {kgf} (assume $Q_f \perp Q_i$)
Z : Number of rollers
α : Contact angle...one-half included outer ring angle (°)
β : One-half the tapered roller angle (°)
D_{w1} : Roller large end diameter (mm) (Fig. 10.4)
e : Contact point between roller end and rib (Fig. 10.4)

As represented in Fig. 10.4, when circumferential load F is applied to the bearing outer ring and the roller turns in the direction of the applied load, the starting torque for contact point C relative to instantaneous center A becomes $e\,\mu_e Q_f$.

Therefore, the balance of frictional torque is

$$D_{w1} F = e\,\mu_e Q_f \ (\mathrm{mN \cdot m}),\ \{\mathrm{kgf \cdot mm}\} \quad \cdots\cdots \quad (10.11)$$

where μ_e: Friction coefficient between inner ring large rib and roller end face

Starting torque M for one bearing is given by

$$M = F Z l$$

$$= \frac{e\,\mu_e\,l \sin 2\beta}{D_{w1} \sin \alpha} F_a \quad (\mathrm{mN \cdot m}),\ \{\mathrm{kgf \cdot mm}\} \quad \cdots\cdots \quad (10.12)$$

because $D_{w1} = 2\,\overline{OB} \sin \beta$, and $l = \overline{OB} \sin \alpha$.

If we substitute these into Equation (10.12), we obtain

$$M = e\,\mu_e \cos\beta\, F_a \ (\mathrm{mN \cdot m}),\ \{\mathrm{kgf \cdot mm}\} \quad \cdots\cdots \quad (10.13)$$

Starting torque M is sought considering only the slip friction between the roller end and the inner ring large-end rib. However, when the load on a tapered roller bearing reaches or exceeds a certain level (around the preload), the slip friction in the space between the roller end and inner ring large end rib becomes the decisive factor for bearing starting torque, and the torque caused by other factors can be ignored. Values for e and β in Equation (10.12) are determined by bearing design. Consequently, by assuming a value for μ_e, the starting torque can be calculated.

The values for μ_e and for e must be considered as a dispersion; thus, even bearings with the same number can have quite diverse individual starting torques. When using a value for e determined by the bearing design, the average value for the bearing starting torque can be estimated using $\mu_e = 0.20$, which is the average value determined from various test results.

Fig. 10.5 shows the results of calculations for various tapered roller bearing series.

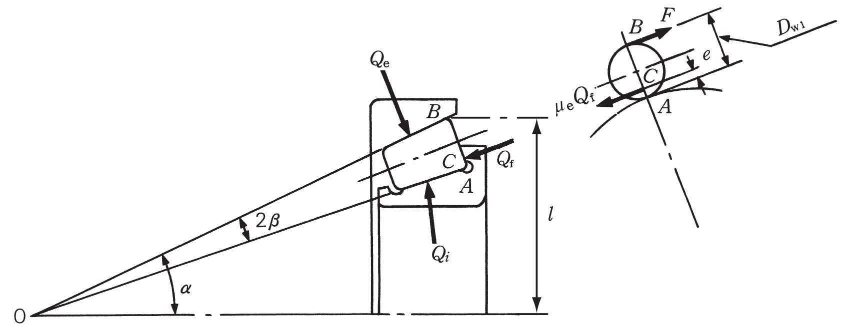


Fig. 10.4

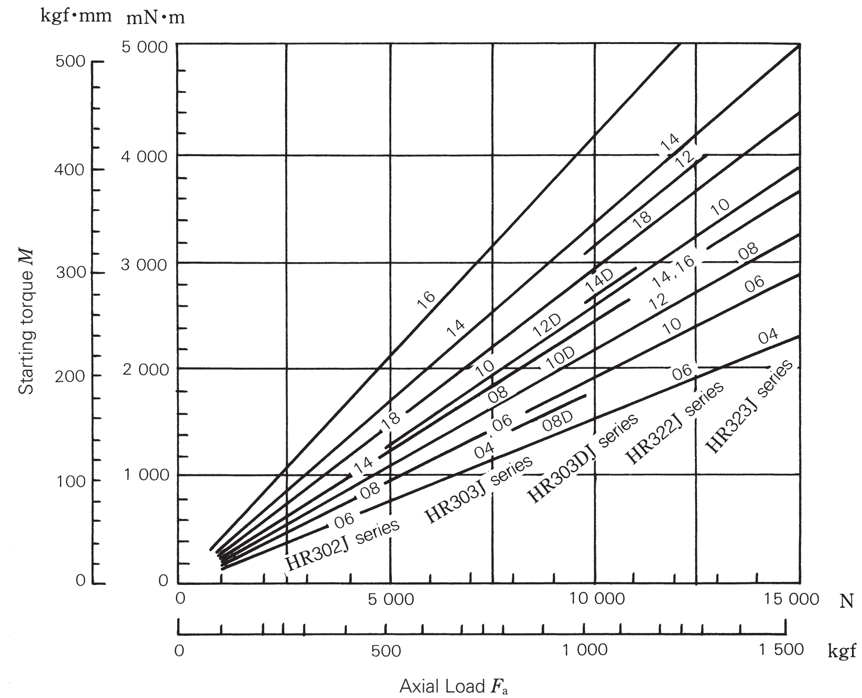


Fig. 10.5 Axial Load and Starting Torque for Tapered Roller Bearings

10.3.4 Empirical Equations for Running Torque of Tapered Roller Bearings

When tapered roller bearings operate under axial load, the torque of tapered roller bearings is based on the following two kinds of resistance, which are the major components of friction:

(1) Rolling resistance (friction) of rollers with outer or inner ring raceways—elastic hysteresis and viscous rolling resistance of EHL

(2) Sliding friction between inner ring ribs and roller ends.

When an axial load F_a is applied on tapered roller bearings, the loads shown in Fig. 10.6 are applied on the rollers.

$$Q_e \fallingdotseq Q_i = \frac{F_a}{Z\sin\alpha} \quad \cdots\cdots (10.14)$$

$$Q_f = \frac{F_a \sin 2\beta}{Z\sin\alpha} \quad \cdots\cdots (10.15)$$

where Q_e: Rolling element load on outer ring
Q_i: Rolling element load on inner ring
Q_f: Rolling element load on inner ring large end rib
Z: Number of rollers
α: Contact angle…One-half included outer ring angle
β: One-half tapered roller angle

For simplicity, Fig. 10.7 shows a model using the average diameter D_{we} along with the following variables:

M_i, M_e: Rolling resistance (moment)
F_{si}, F_{se}, F_{sf}: Sliding friction
R_i, R_e: Radii at center of inner and outer ring raceways
e: Contact height of roller end face with rib

When the balance of sliding friction and moments on the rollers are considered as represented in Fig 10.7, the following equations are obtained:

$$F_{se} - F_{si} = F_{sf} \quad \cdots\cdots (10.16)$$

$$M_i + M_e = \frac{D_w}{2}F_{se} + \frac{D_w}{2}F_{si} + \left(\frac{D_w}{2} - e\right)F_{sf} \quad \cdots\cdots (10.17)$$

When the running torque M applied on the outer (inner) ring is calculated using Equations (10.16) and (10.17) and multiplied by the number of rollers Z, the following is obtained:

$$\begin{aligned} M &= Z\,(R_e F_{se} - M_e) \\ &= \frac{Z}{D_w}(R_e M_i + R_i M_e) + \frac{Z}{D_w} R_e\, e\, F_{sf} \\ &= M_R + M_S \end{aligned}$$

Therefore, the rolling friction on the raceway surface M_R and sliding friction on the ribs M_S are separately obtained.

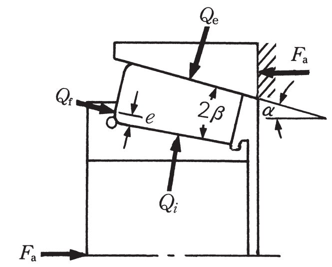


Fig. 10.6 Loads Applied on Roller

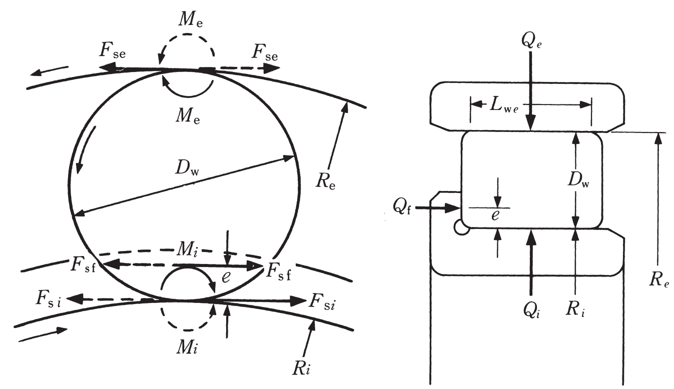


Fig. 10.7 Parts Where Friction is Generated

The running torque M of a tapered roller bearing can be obtained from the rolling friction on the raceway M_R and sliding friction on the ribs M_S.

$$M=M_R+M_S=\frac{Z}{D_w}(R_e M_i+R_i M_e)+\frac{Z}{D_w}R_e\, e\, F_{sf} \quad \textbf{(10.18)}$$

Sliding Friction on Rib M_S
As a part of M_S, F_{sf} refers to the tangential load caused by sliding, so we can write $F_{sf}=\mu Q_f$ using the coefficient of dynamic friction μ. Further, by substituting in axial load F_a, the following equation is obtained:

$$M_S=e\,\mu\cos\beta\, F_a \quad \textbf{(10.19)}$$

This is identical to the equation for starting torque, but μ is not constant and decreases depending on operating conditions and running-in. For this reason, Equation (10.19) can be rewritten as follows:

$$M_S=e\,\mu_0\cos\beta\, F_a f'(\Lambda, t, \sigma) \quad \textbf{(10.20)}$$

where μ_0 is approximately 0.2 and $f'(\Lambda, t, \sigma)$ is a function that decreases with running in and oil film formation but is set equal to one at starting.

Rolling Friction on Raceway Surface M_R
Most of the rolling friction on the raceway is from viscous oil resistance (EHL rolling resistance) corresponding to M_i and M_e in Equation (10.18). A theoretical equation exists but should be corrected based on the results of experiments. The following equation includes corrective terms:

$$M_{i,e}=\left[f(w)\left(\frac{1}{1+0.29L^{0.78}}\right)\frac{4.318}{\alpha_0}(G\cdot U)^{0.658}W^{0.0126}R^2L_{we}\right]_{i,e} \quad \textbf{(10.21)}$$

$$f(w)=\left(\frac{kF_a}{E'\,D_w L_{we} Z\sin\alpha}\right)^{0.3} \quad \textbf{(10.22)}$$

Therefore, M_R can be obtained using Equations (10.21) and (10.22) together with the following equation:

$$M_R=\frac{Z}{D_w}(R_e M_i+R_i M_e)$$

Running Torque of Bearings M
From these, the running torque of tapered roller bearings M is given by Equation (10.23).

$$M=\frac{Z}{D_w}(R_e M_i+R_i M_e)+e\,\mu_0\cos\beta\, F_a f'(\Lambda, t, \sigma) \quad \textbf{(10.23)}$$

As shown in Figs. 10.8 and 10.9, the values obtained using Equation (10.23) correlate rather well with actual measurements; therefore, estimation of running torque with good accuracy is possible. Please consult NSK with any questions or concerns.

[Symbol Definitions]
- G, W, U: EHL dimensionless parameters
- L: Coefficient of thermal load
- α_0: Pressure coefficient of lubricating oil viscosity
- R: Equivalent radius
- k: Constant
- E': Equivalent elastic modulus
- α: Contact angle (One-half included outer ring angle)
- R_i, R_e: Inner and outer ring raceway radii (center)
- β: Half angle of roller
- i, e: Indication of inner ring or outer ring respectively
- L_{we}: Effective roller length

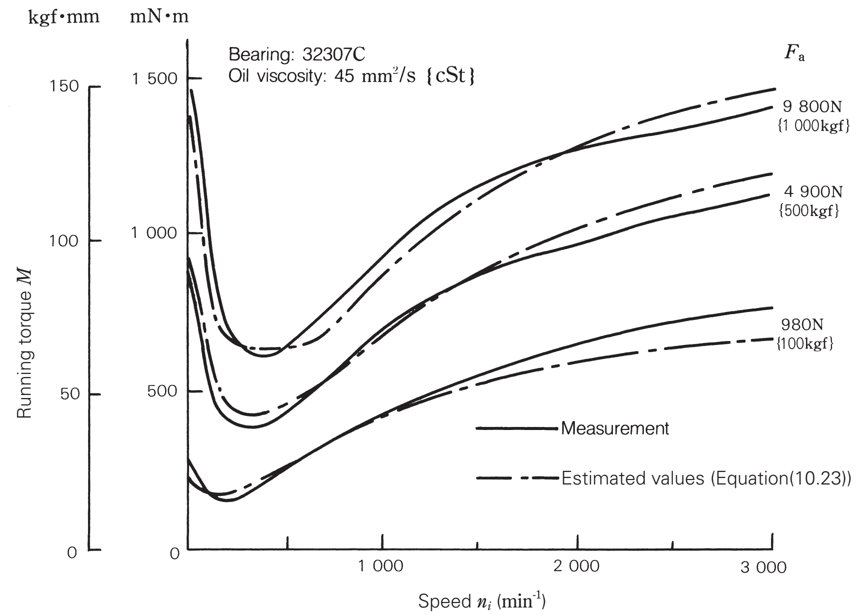


Fig. 10.8 Comparison of Estimated Values With Actual Measurements

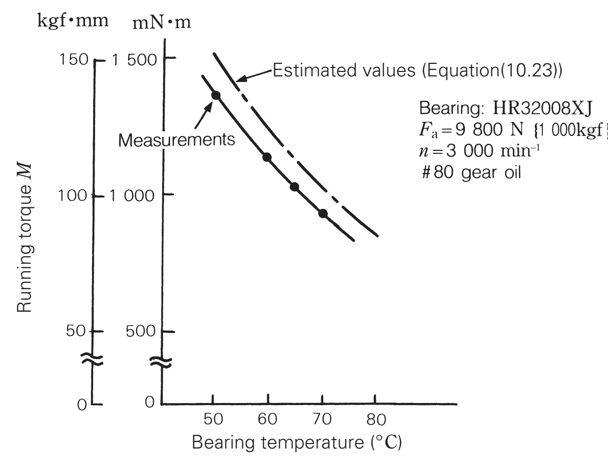


Fig. 10.9 Viscosity Variation and Running Torque

11. LUBRICATION

11. LUBRICATION

11.1 Purposes of Lubrication

The main purposes of lubrication are to reduce friction and wear inside the bearings that may cause premature failure. The effects of lubrication may be briefly explained as follows:

(1) Reduction of Friction and Wear
Direct metallic contact between the basic components of the bearing (bearing rings, rolling elements and cage) is prevented by an oil film that reduces friction and wear in contact areas.

(2) Extension of Fatigue Life
The rolling fatigue life of bearings depends greatly upon the viscosity and film thickness between the rolling contact surfaces. An oil film with the proper thickness during rotation prolongs bearing life. Conversely, an insufficient oil film or excessively low viscosity lubricant will shorten life.

(3) Dissipation of Frictional Heat and Cooling
Circulating lubrication is used to carry away frictional or absorbed heat, prevent the bearing from overheating, and prevent oil from deteriorating.

(4) Others
Adequate lubrication also helps to prevent foreign material from entering the bearings and guards against corrosion and rusting.

11.2 Lubricating Methods

Lubricating methods are first divided into either grease or oil lubrication. Satisfactory bearing performance can be achieved by adopting the lubricating method most suitable for the particular application and operating condition.
In general, oil offers superior lubrication; however, grease lubrication allows for a simpler structure around the bearings. A comparison of grease and oil lubrication is given in Table 11.1.

Table 11.1 Comparison of Grease and Oil Lubrication

Item	Grease Lubrication	Oil Lubrication
Housing Structure/ Seal Configuration	Can be simple	Somewhat complex. Requires careful maintenance
Speed	Supports 65-80% the permissible speeds of oil lubrication	Supports higher speeds than grease
Cooling Effect	None	Can effectively dissipate heat (with circulating lubrication, etc.)
Flowability	Poor	Good
Full Replacement	Somewhat complex	Relatively easy
Contaminant Filtration	Difficult	Easy
External Contamination	Little effect from leaks	Unsuitable in areas where contamination must be avoided

11.2.1 Grease Lubrication

(1) *Grease Quantity*
The quantity of grease to be packed in a housing depends on the housing design and free space, grease characteristics, and ambient temperature. For example, bearings for main shafts of machine tools where accuracy may be impaired by a small temperature rise require only a small amount of grease.
Sufficient grease must be packed inside the bearing including the cage guide face. The amount to be packed inside the the housing depends on the speed of the application:

Fill **1/2 to 2/3** of the free internal space with grease when the speed is **50% of the limiting speed or less**.

Fill **1/3 to 1/2** of the free internal space with grease when the speed is **50% of the limiting speed or more**.

(2) *Replacement of Grease*
Grease, once packed, usually need not be replenished for a long time; however, under severe operating conditions, grease should be frequently replenished or replaced. In such cases, the bearing housing should be designed to facilitate grease replenishment and replacement.
When replenishment intervals are short, provide replenishment and discharge ports at appropriate positions so that deteriorated grease is replaced by fresh grease. For example, the housing space on the grease-supply side can be divided into several sections with partitions. The grease on the partitioned side gradually passes through the bearings and old grease forced from the bearing is discharged through a grease valve (Fig. 11.1). If a grease valve is not used, the space on the discharge side must be larger than the partitioned side so that it can retain old grease, which can be removed periodically by removing the cover.

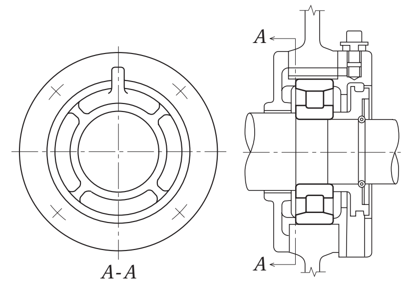
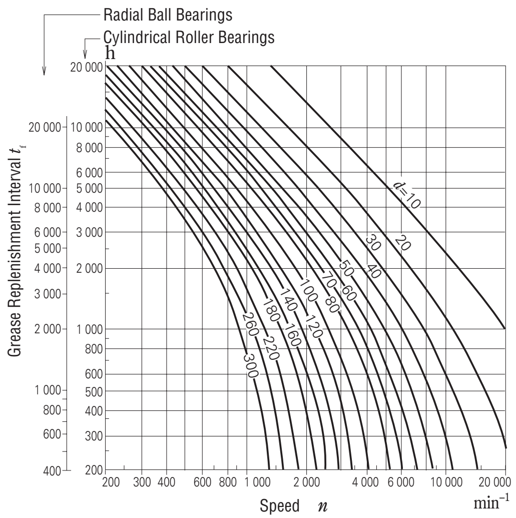
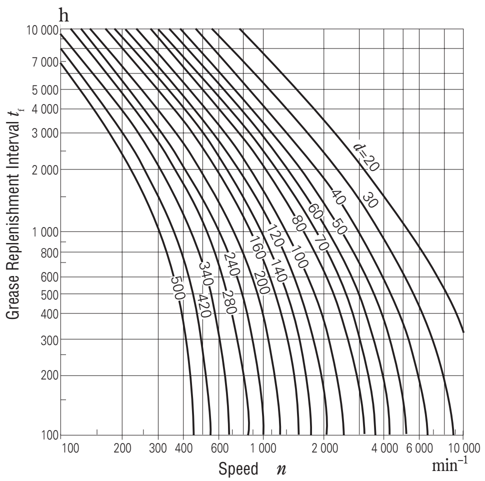


Fig. 11.1 Combination of Partitioned Grease Reservoir and Grease Valve

(3) *Replenishment Interval*
Even if high-quality grease is used, grease deterioration occurs with time; therefore, periodic replenishment is required. Graphs (1) and (2) in Fig 11.2 show the replenishment intervals for various bearing types running at different speeds. These graphs show high-quality, lithium-soap mineral oil grease at a bearing temperature of 70 °C under normal load (P/C=0.1).
· Temperature
If the bearing temperature exceeds 70 °C, reduce the replenishment interval by half for every 15 °C increase.
· Grease
Replenishment intervals can be extended depending on the grease used, especially for ball bearings. For example, high-quality lithium-soap synthetic oil grease roughly doubles the replenishment interval shown in Fig.11.2 (1). If the temperature of the bearings is less than 70 °C, the usage of lithium-soap mineral oil grease or lithium-soap synthetic oil grease is appropriate.
· Load
The replenishment interval depends on the magnitude of the bearing load.
Please refer to Fig.11.2 (3) for details.
If P/C exceeds 0.16, please consult NSK.



(1) Radial Ball Bearings, Cylindrical Roller Bearings



(2) Tapered Roller Bearings, Spherical Roller Bearings

(3) Load factor

P/C	≦0.06	0.1	0.13	0.16
Load factor	1.5	1	0.65	0.45

Fig. 11.2 Grease Replenishment Intervals

(4) *Grease Life of Sealed Ball Bearings*

When grease is packed into single-row deep groove ball bearings, the grease life may be estimated using Equation (11.1), Equation (11.2), or Fig. 11.3:

(General purpose grease ([1]))

$$\log t = 6.54 - 2.6\frac{n}{N_{max}} - \left(0.025 - 0.012\frac{n}{N_{max}}\right)T \quad \text{......................(11.1)}$$

(Wide-range grease ([2]))

$$\log t = 6.12 - 1.4\frac{n}{N_{max}} - \left(0.018 - 0.006\frac{n}{N_{max}}\right)T \quad \text{......................(11.2)}$$

where t : Average grease life (h)
n : Speed (min^{-1})
N_{max} : Limiting speed with grease lubrication (min^{-1}) (values for ZZ and VV types are listed in the bearing tables)
T : Operating temperature °C

Equation (11.1), Equation (11.2), and Fig. 11.3 apply under the following conditions:

(a) Speed n

$$0.25 \leqq \frac{n}{N_{max}} \leqq 1$$

when $\frac{n}{N_{max}} < 0.25$, assume $\frac{n}{N_{max}} = 0.25$

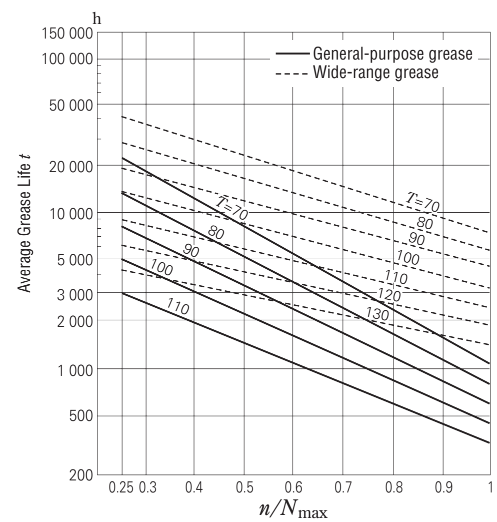


Fig. 11.3 Grease Life of Sealed Ball Bearings

(b) Operating Temperature T
For general-purpose grease ([1])

$$70\ °C \leqq T \leqq 110\ °C$$

For wide-range grease ([2])

$$70\ °C \leqq T \leqq 130\ °C$$

When $T < 70$ °C assume $T = 70$ °C

(c) Bearing Loads
The bearing loads should be about 1/10 or less the basic load rating C_r.

Notes ([1]) Mineral-oil base greases (e.g. lithium-soap base grease) often used around – 10 to 110 °C.
([2]) Synthetic-oil base greases are usable over a wide temperature range around –40 to 130 °C.

11.2.2 Oil Lubrication

(1) *Oil-Bath Lubrication*

Oil-bath lubrication is widely used with low or medium speeds. The oil level should be at the center of the lowest rolling element. Ideally, provide a sight gauge so the proper oil level may be maintained (Fig. 11.4)

(2) *Drip-Feed Lubrication*

Drip-feed lubrication is widely used for small ball bearings operated at relatively high speeds. As shown in Fig. 11.5, oil is stored in a visible oiler. The oil drip rate is controlled with a screw in the top.

(3) *Splash Lubrication*

With this method, oil is splashed onto the bearings by gears or a simple rotating disc installed near bearings without submerging the bearings in oil.
Splash lubrication is commonly used in automobile transmissions and final drive gears. Fig. 11.6 shows this lubricating method used on a reduction gear.

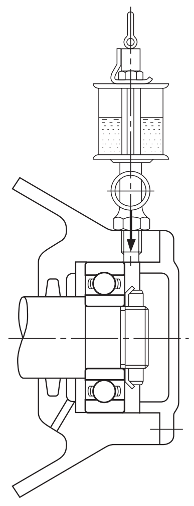


Fig. 11.5 Drip-Feed Lubrication

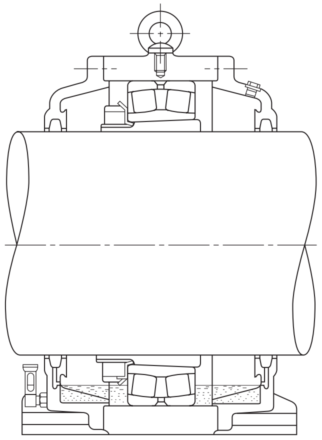


Fig. 11.4 Oil-Bath Lubrication

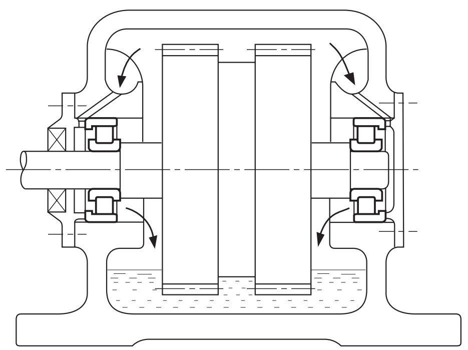


Fig. 11.6 Splash Lubrication

(4) *Circulating Lubrication*

Circulating lubrication is commonly used for high-speed operation requiring bearing cooling and for bearings used at high temperatures. In Fig. 11.7(a), oil is supplied to a specified level by a pipe on the right side. Once this level is reached, oil flows out the discharge pipe on the left.
In Figs. 11.7(b) and (c), the oil does not accumulate within the housing, but rather the oil is returned to a tank. Once cooled, the oil passes through a pump and filter and returns to the bearing.
Oil discharge pipes should be larger than the supply pipe so that oil does not over-accumulate in the housing.

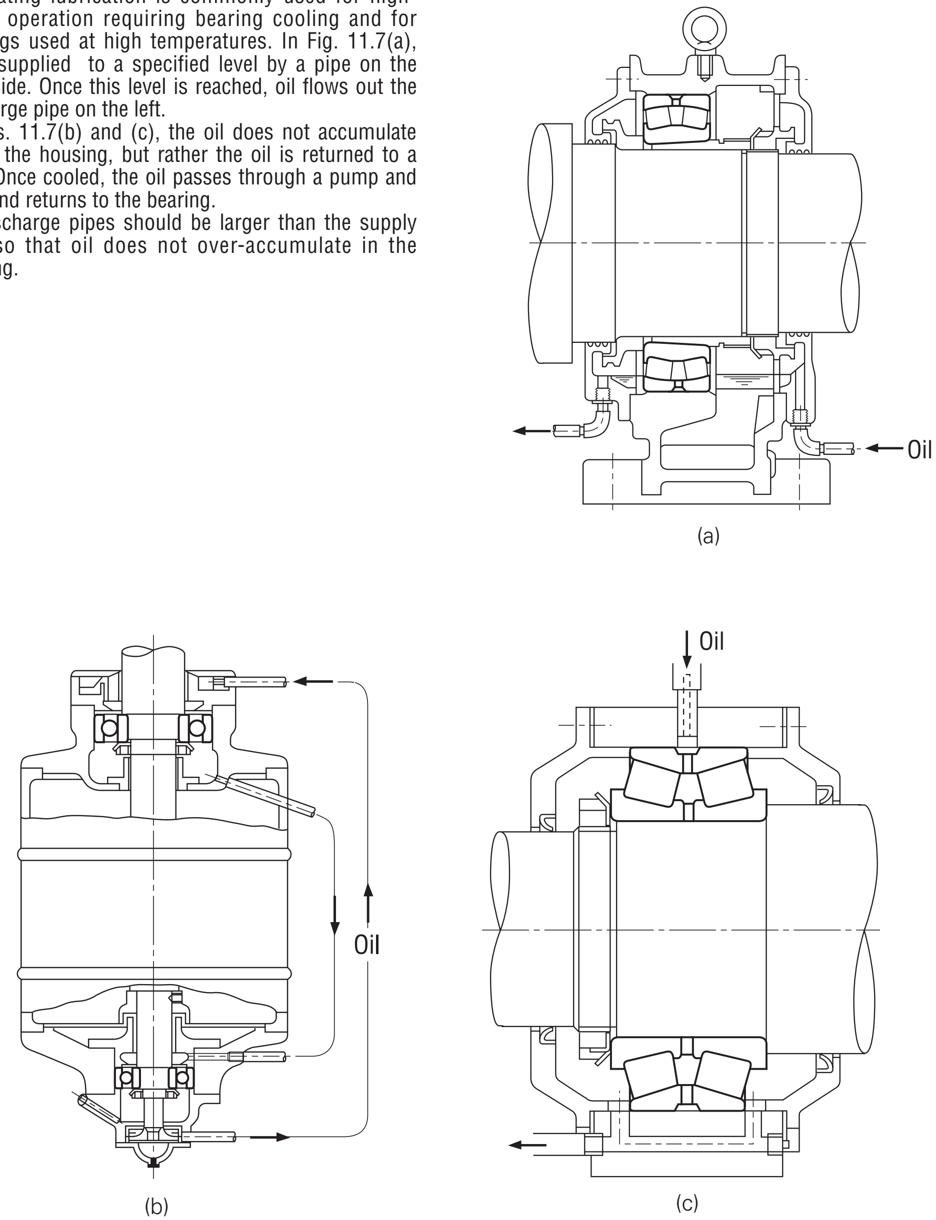


Fig. 11.7 Circulating Lubrication

(5) ***Jet Lubrication***

Jet lubrication is often used for ultra-high-speed bearings with a $d_m n$ valve (d_m: pitch diameter of rolling element set in mm; n: rotational speed in min^{-1}) exceeding one million, such as the bearings in jet engines. Lubricating oil is sprayed under pressure from one or more nozzles directly into the bearing.
Fig. 11.8 shows an example of ordinary jet lubrication. The lubricating oil is sprayed on the inner ring and cage guide face. In high-speed operation, the air surrounding the bearing rotates with it, causing the oil jet to be deflected. The jetting speed of the oil from the nozzle should be more than 20 % of the circumferential speed of the inner ring outer surface, or cage guide face.
More uniform cooling and a better temperature distribution is achieved using more nozzles for a given amount of oil. The oil should be forcibly discharged so that the agitating resistance of the lubricant can be reduced and the oil can effectively carry away the heat.

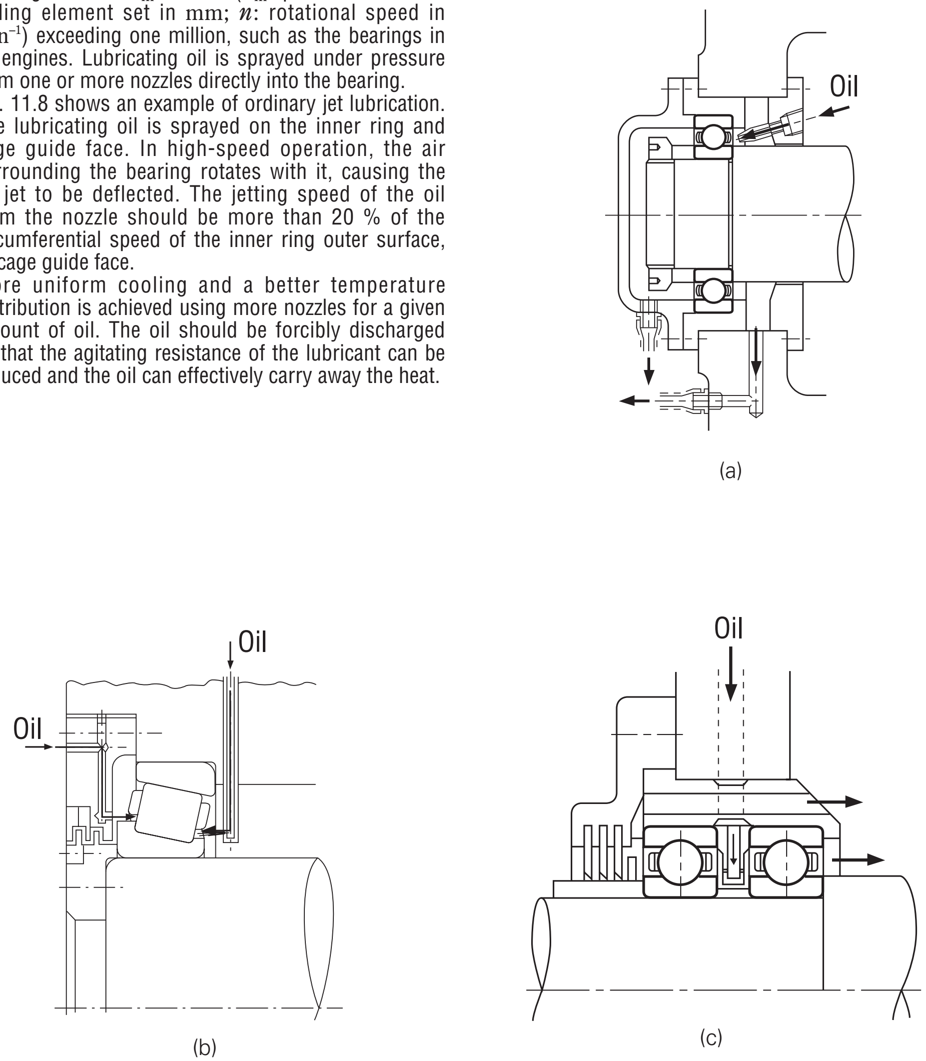


Fig. 11.8 Jet Lubrication

(6) ***Oil-Mist Lubrication***

Oil-mist lubrication, or oil fog lubrication, utilizes an oil mist sprayed into a bearing. This method has the following advantages:
(a) Because of the small quantity of oil required, the oil agitation resistance is small, and higher speeds are possible.
(b) Contamination of the vicinity around the bearing is limited because the oil leakage is small.
(c) It is relatively easy to continuously supply fresh oil; therefore, bearing life is extended.
This lubricating method is used for bearings in the high-speed spindles of machine tools, high-speed pumps, roll necks of rolling mills, and so on (see example in Fig. 11.9).
Please consult NSK regarding oil-mist lubrication of large bearings.

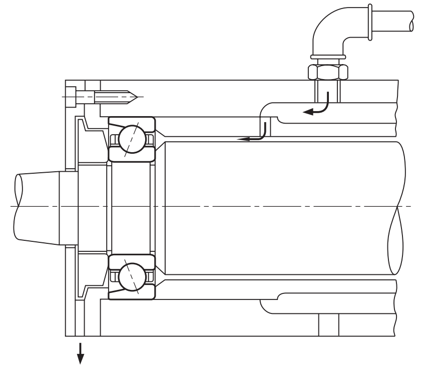


Fig. 11.9 Oil-Mist Lubrication

(7) ***Oil-Air Lubricating Method***

With the oil-air lubricating method, a very small amount of oil is discharged intermittently by a constant-quantity piston into a pipe carrying a constant flow of compressed air. The oil flows along the wall of the pipe and approaches a constant flow rate.
The major advantages of oil-air lubrication are as follows:
(a) The minimum necessary amount of oil is supplied, making this method suitable for high speeds because less heat is generated.
(b) Since oil is fed continuously, bearing temperature remains stable. Also, because of the small amount of oil, there is almost no atmospheric pollution.
(c) Only fresh oil is fed to the bearings, so oil deterioration need not be considered.
(d) Compressed air is constantly fed to the bearings and keeps internal pressure high. This prevents the entry of dust, cutting fluid, etc.
For these reasons, this method is used in the main spindles of machine tools and other high speed-applications (see example in Fig. 11.10).

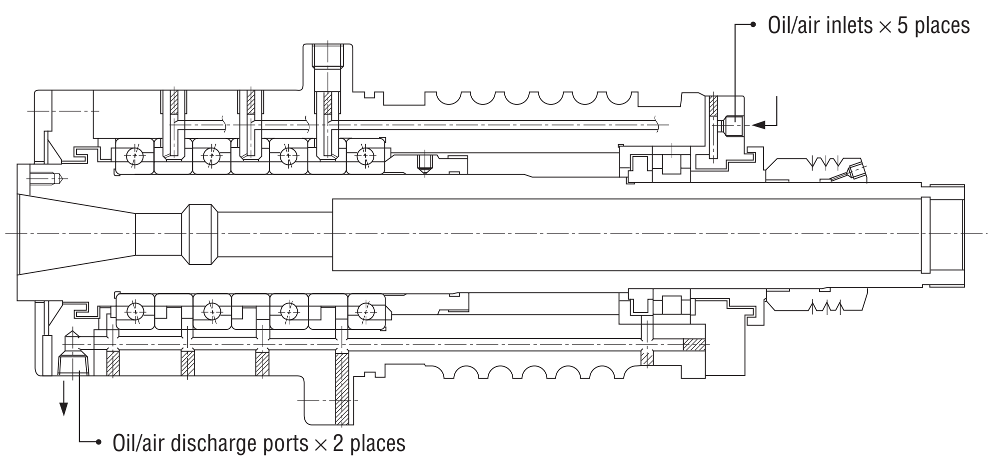


Fig. 11.10 Oil-Air Lubrication

11.3 Lubricants

11.3.1 Lubricating Grease

Grease is a semisolid lubricant consisting of base oil, a thickener, and additives. The main types and general properties of grease are shown in Table 11.2. Note that different brands of the same type of grease may have different properties.

(1) ***Base Oil***

Mineral oils or synthetic oils, such as silicone or diester oil, are often used as the base oil for grease. The lubricating properties of grease depend mainly on the characteristics of its base oil. Therefore, the viscosity of the base oil is just as important when selecting grease as when selecting an oil. Usually, grease made with low viscosity base oils is more suitable for high speeds and low temperatures, while grease made with high viscosity base oils is more suited for high temperatures and heavy loads.
However, the thickener also influences the lubricating properties of grease; therefore, the selection criteria for grease is not the same as for lubricating oil. Moreover, please be aware that ester-based grease will cause acrylic rubber material to swell, and that silicone-based grease will cause silicone-based material to swell.

(2) ***Thickener***

Thickeners for lubricating grease include several types of metallic soaps, inorganic thickeners such as silica gel and bentonite, and heat resisting organic thickeners such as polyurea and fluoric compounds.
The type of thickener is closely related to the grease dropping point [1]; generally, grease with a high dropping point also has a high temperature capability during operation. However, this type of grease does not have a high working temperature unless the base oil is heat-resistant. The highest possible working temperature for grease should be determined considering the heat resistance of the base oil.
The water resistance of grease depends upon the type of thickener. Sodium-soap grease or compound grease containing sodium soap emulsifies when exposed to water or high humidity and therefore cannot be used where moisture is prevalent. Moreover, note that urea-based grease will cause fluorine-based material to deteriorate.

Note [1] The grease dropping point is that temperature at which a grease heated in a specified small container becomes sufficiently fluid to drip.

Table 11.2 Grease Properties

Name (Popular Name)	Lithium Grease			Sodium Grease (Fiber Grease)	Calcium Grease (Cup Grease)	Mixed Base Grease	Complex Base Grease (Complex Grease)	Non-Soap Base Grease (Non-Soap Grease)	
Thickener	Li Soap			Na Soap	Ca Soap	Na + Ca Soap, Li + Ca Soap, etc.	Ca Complex Soap, Al Complex Soap, Li Complex Soap, etc.	Urea, Bentonite, Carbon Black, Fluoric Compounds, Heat Resistant Organic Compound, etc.	
Base Oil / Properties	Mineral Oil	Diester Oil, Polyatomic Ester Oil	Silicone Oil	Mineral Oil	Mineral Oil	Mineral Oil	Mineral Oil	Mineral Oil	Synthetic Oil (Ester Oil, Polyatomic Ester Oil, Synthetic Hydrocarbon Oil, Silicone Oil, Fluoric Base Oil)
Dropping Point, ℃	170 to 195	170 to 195	200 to 210	170 to 210	70 to 90	160 to 190	180 to 300	> 230	> 230
Working Temperatures, ℃	−20 to +110	−50 to +130	−50 to +160	−20 to +130	−20 to +60	−20 to +80	−20 to +130	−10 to +130	< +220
Working Speed, %[1]	70	100	60	70	40	70	70	70	40 to 100
Mechanical Stability	Good	Good	Good	Good	Poor	Good	Good	Good	Good
Pressure Resistance	Fair	Fair	Poor	Fair	Poor	Fair to Good	Fair to Good	Fair	Fair
Water Resistance	Good	Good	Good	Poor	Good	Poor for Na Soap Grease	Good	Good	Good
Rust Prevention	Good	Good	Poor	Poor to Good	Good	Fair to Good	Fair to Good	Fair to Good	Fair to Good
Remarks	General-purpose grease used for numerous applications	Good low-temperature and torque characteristics. Often used for small motors and instrument bearings. Monitor for rust caused by insulation varnish.	Mainly used for high temperature applications. Unsuitable for bearings at high and low speeds, under heavy loads, or with numerous sliding-contact areas (roller bearings, etc.)	Long-and-short-fiber types are available. Long-fiber grease is unsuitable for high speeds. Monitor for water and high temperatures.	Extreme-pressure grease containing high viscosity mineral oil and extreme pressure additives (Pb soap, etc.) with high pressure resistance.	Often used for roller bearings and large ball bearings.	Suitable for extreme pressures mechanically stable	Mineral oil base grease is used for medium to high temperatures. Synthetic-oil base grease is recommended for low or high temperatures. Some silicone and fluoric oil base grease have poor rust prevention and noise characteristics.	

Note [1] The values listed are percentages of the limiting speeds given in the bearing tables.

Remark The grease properties shown here can vary between brands.

(3) ***Additives***

Grease often contains various additives such as antioxidants, corrosion inhibitors, and extreme pressure additives to give it special properties. Extreme pressure additives are recommended for heavy load applications. For long use without replenishment, an antioxidant should be added.

(4) ***Consistency***

Consistency indicates the "softness" of grease. Table 11.3 shows the relation between consistency and operating conditions.

Table 11.3 Consistency and Working Conditions

Consistency Number	0	1	2	3	4
Consistency[1] 1/10 mm	355 to 385	310 to 340	265 to 295	220 to 250	175 to 205
Working Conditions (Application)	·For centralized oiling ·When fretting is likely to occur	·For centralized oiling ·When fretting is likely to occur ·For low temperatures	·For general use ·For sealed ball bearings	·For general use ·For sealed ball bearings ·For high temperatures	·For high temperatures ·For grease seals

Note [1] Consistency: The depth to which a cone descends into grease when a specified weight is applied, indicated in units of 1/10 mm. The larger the value, the softer the grease.

(5) ***Mixing Different Types of Grease***

In general, different brands of grease must not be mixed. Mixing grease with different types of thickeners may destroy its composition and physical properties. Even if thickeners are of the same type, possible differences in the additive may cause detrimental effects.

11.3.2 Lubricating Oil

The lubricating oils used for rolling bearings are usually highly refined mineral oils or synthetic oils that have a high oil-film strength and superior oxidation and corrosion resistance. When selecting a lubricating oil, the viscosity during operation is important. If the viscosity is too low, a proper oil film does not form and abnormal wear and seizure may occur. On the other hand, if the viscosity is too high, excessive viscous resistance may cause heating or energy loss. In general, low viscosity oils should be used at high speed; however, the viscosity should increase with increasing bearing load and size.

Table 11.4 gives generally recommended viscosities for bearings under normal operating conditions.

Fig. 11.11 shows the relationship between oil temperature and viscosity, and selected lubricating oils are shown in Table 11.5.

Table 11. 4 Bearing Types and Proper Viscosity of Lubricating Oils

Bearing Type	Proper Viscosity at Operating Temperature
Ball Bearings and Cylindrical Roller Bearings	Higher than 13mm²/s
Tapered Roller Bearings and Spherical Roller Bearings	Higher than 20mm²/s
Spherical Thrust Roller Bearings	Higher than 32mm²/s

Remark 1mm²/s=1cSt (centistokes)

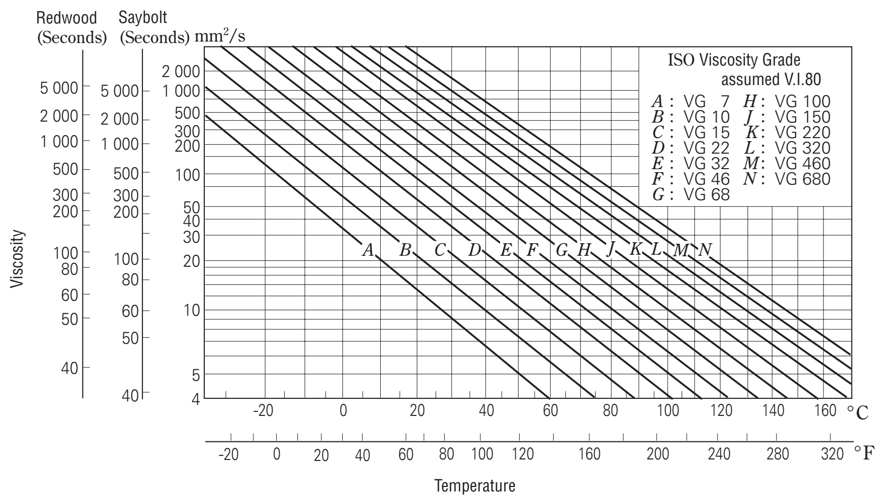


Fig. 11.11 Temperature-Viscosity Chart

Oil Replacement Intervals

Oil replacement intervals depend on operating conditions and oil quantity.

In cases where the operating temperature is less than 50 °C and environmental conditions are good with little dust, the oil should be replaced approximately once a year. However, in cases where the oil temperature is near 100 °C, the oil must be changed at least once every three months.

If moisture may enter or if foreign matter may be mixed in the oil, then the oil replacement interval must be shortened.

Do not mix different brands of oil as, like grease, their composition and physical properties may be negatively affected.

Table 11. 5 Example Lubricating Oils for Bearing Operating Conditions

Operating Temperature	Speed	Light or Normal Load	Heavy or Shock Load
−30 to 0 °C	Less than limiting speed	ISO VG 15, 22, 32 (refrigerating machine oil)	–
0 to 50 °C	Less than 50% of limiting speed	ISO VG 32, 46, 68 (bearing oil, turbine oil)	ISO VG 46, 68, 100 (bearing oil, turbine oil)
	50 to 100% of limiting speed	ISO VG 15, 22, 32 (bearing oil, turbine oil)	ISO VG 22, 32, 46 (bearing oil, turbine oil)
	More than limiting speed	ISO VG 10, 15, 22 (bearing oil)	–
50 to 80 °C	Less than 50% of limiting speed	ISO VG 100, 150, 220 (bearings oil)	ISO VG 150, 220, 320 (bearing oil)
	50 to 100% of limiting speed	ISO VG 46, 68, 100 (bearing oil, turbine oil)	ISO VG 68, 100, 150 (bearing oil, turbine oil)
	More than limiting speed	ISO VG 32, 46, 68 (bearing oil, turbine oil)	–
80 to 110 °C	Less than 50% of limiting speed	ISO VG 320, 460 (bearing oil)	ISO VG 460, 680 (bearing oil, gear oil)
	50 to 100% of limiting speed	ISO VG 150, 220 (bearing oil)	ISO VG 220, 320 (bearing oil)
	More than limiting speed	ISO VG 68, 100 (bearing oil, turbine oil)	–

Remarks
1. Use the values listed in the bearing tables for limiting speeds.
2. Refer to Refrigerating Machine Oils (JIS K 2211), Bearing Oils (JIS K 2239), Turbine Oils (JIS K 2213), and Gear Oils (JIS K 2219) for more information.
3. If the operating temperature is near the high end of the temperature range listed in the left column, select a high-viscosity oil.
4. If the operating temperature is lower than -30 °C or higher than 110 °C, please consult NSK.

11.4 Technical Data

11. 4. 1 Brands and Properties of Lubricating Grease

Table 11. 6 Brands of Lubricating Grease

Brand	Thickener	Base Oil(s)	Dropping Point (°C)	Consistency	Working Temperature Range(1)(°C)	Pressure Resistance	Usable Limit Compared to Listed Limiting Speed (Grease)(2)(%)
EA3 GREASE	Urea (3)	Poly-α-olefin oil	≧260	230	−40 to +150	Fair	100
EA5 GREASE	Urea (3)	Poly-α-olefin oil	≧260	251	−40 to +160	Good	60
EA6 GREASE	Urea (3)	Poly-α-olefin oil	≧260	220	−40 to +160	Fair	70
EA7 GREASE	Urea (3)	Poly-α-olefin oil	≧260	243	−40 to +160	Fair	100
EA9 GREASE	Urea (3)	Poly-α-olefin oil	≧260	314	−40 to +140	Fair	100
ENS GREASE	Urea (3)	Polyol ester oil (4)	≧260	264	−40 to +160	Poor	100
ECE GREASE	Lithium	Poly-α-olefin oil	≧260	235	−10 to +120	Poor	100
DOW CORNING(R) SH 44 M GREASE	Lithium	Silicone oil (5)	210	260	−30 to +130	Poor	60
NS HI-LUBE	Lithium	Ester oil + Diester oil (4)	192	250	−40 to +130	Poor	100
LG2 GREASE	Lithium	Poly-α-olefin oil + Mineral oil	201	199	−20 to +70	Poor	100
LGU GREASE	Urea (3)	Poly-α-olefin oil	≧260	201	−40 to +120	Fair	70
EMALUBE 8030	Urea (3)	Mineral oil	≧260	280	0 to +130	Good	60
KP1 GREASE	PTFE	Perfluoropolyether oil	Not applicable	290	−30 to +200	Fair	60
SHELL ALVANIA GREASE S2	Lithium	Mineral oil	181	275	−10 to +110	Fair	70
SHELL ALVANIA GREASE S3	Lithium	Mineral oil	182	242	−10 to +110	Fair	70
SHELL SUNLIGHT GREASE 2	Lithium	Mineral oil	200	274	−10 to +110	Fair	70
WPH GREASE	Urea (3)	Poly-α-olefin oil	259	240	−40 to +150	Fair	70
NIGLUBE RSH	Sodium Complex	Glycol oil	≧260	270	−20 to +140	Fair	60
PALMAX RBG	Lithium Complex	Mineral oil	216	300	−10 to +130	Good	70
MULTEMP PS No.2	Lithium	Poly-α-olefin oil + Diester oil (4)	190	275	−50 to +110	Poor	100
MOLYKOTE(R) FS-3451GREASE	PTFE	Fluorosilicone oil (5)	Not applicable	285	0 to +180	Fair	70
UME GREASE	Urea (3)	Mineral oil	≧260	272	−10 to +130	Fair	70
RW1 GREASE	Urea (3)	Mineral oil	≧260	300	−10 to +130	Fair	70
HA1 GREASE	Urea (3)	Ether oil	≧260	290	−40 to +160	Fair	70
HA2 GREASE	Urea (3)	Ether + Poly-α-olefin oil	≧260	295	−30 to +170	Fair	70
KLUBERSYNTH HB 72-52	Urea (3)	Ester oil (4)	250	295	−30 to +160	Fair	70
NOXLUB KF0921	PTFE	Perfluoropolyether oil	Not applicable	280	−40 to +200	Fair	70
ECH GREASE	Carbon Black	Perfluoropolyether oil	Not applicable	205	−30 to +260	Fair	60
FWG GREASE	Urea (3)	Mineral oil + Poly-α-olefin oil	≧260	268	−30 to +150	Fair	70
HT1 GREASE	Urea (3)	Poly-α-olefin oil	≧260	236	−40 to +150	Fair	100
ARAPEN RB320	Lithium-Calcium	Mineral oil	177	305	−10 to +80	Fair	70
SHELL GADUSRAIL S4 HIGH SPEED EUFR	Lithium	Mineral oil	188	266	−10 to +110	Fair	100

Notes (1) If grease will be used at the upper or lower limit of the temperature range or in a special environment such as a vacuum, please consult NSK.
(2) For short-term operation or when adequate cooling is provided, grease may be used at speeds exceeding the above limits provided the supply of grease is appropriate.
(3) Urea-based grease causes fluorine-based material to deteriorate.
(4) Ester-based grease causes acrylic rubber material to swell.
(5) Silicone-based grease causes silicone-based material to swell.

12. BEARING MATERIALS

12. BEARING MATERIALS

Bearing rings and rolling elements of rolling bearings are subjected to repetitive high pressure with a small amount of sliding. Tension, compression, and sliding contact with the rolling elements and either or both of the bearing rings impact the cage.

Therefore, the materials used for the rings, rolling elements, and cages require the following characteristics:

Material characteristics required for bearing rings and rolling elements:
- High rolling contact fatigue strength
- High hardness
- High wear resistance
- High dimensional stability
- High mechanical strength

Characteristics required for cage material:
- High wear resistance
- High dimensional stability
- High mechanical strength

Other necessary characteristics, such as ease of production, shock and heat resistance, and corrosion resistance, are required depending on individual applications.

12.1 Materials for Bearing Rings and Rolling Elements

Primarily, high-carbon-chromium bearing steel (Table 12.1) is used for the bearing rings and rolling elements. Most NSK bearings are made of SUJ2 steel, while larger bearings generally use SUJ3 (additional types are listed in Table 12.1). The chemical composition of SUJ2 is approximately the same as AISI 52100, DIN 100 Cr6, and BS 535A99.

Bearings subjected to very severe shock loads often utilize carburized low-carbon alloy steels such as chrome steel, chrome-molybdenum steel, nickel-chrome-molybdenum steel, etc. Such steels, when they are carburized to the proper depth and have sufficient surface hardness, are more shock-resistant than normal, through-hardened bearing steels because of their softer energy-absorbing core. The chemical composition of common carburized bearing steels is listed in Table 12.2.

NSK uses highly pure vacuum-degassed bearing steel containing minimal oxygen, nitrogen, and hydrogen compound impurities. The rolling fatigue life of bearings has been remarkably improved using this material combined with the appropriate heat treatment.

For special-purpose bearings, high-temperature bearing steel, which has superior heat resistance, and stainless steel with good corrosion resistance may be used. The chemical composition of these special materials are given in Tables 12.3 and 12.4.

12.2 Cage Materials

The main types of low carbon steels used for pressed cages are shown in Table 12.5. Depending on the application, brass or stainless steel may be used. For machined cages, high-strength brass (Table 12.6) or carbon steel (Table 12.5) is used. Synthetic resin is also sometimes used.

Table 12.1 Chemical Composition of High-Carbon-Chromium Bearing Steel (Major Elements)

Standard	Designation	Chemical Composition (%)						
		C	Si	Mn	P	S	Cr	Mo
JIS G 4805	SUJ 2	0.95 to 1.10	0.15 to 0.35	Less than 0.50	Less than 0.025	Less than 0.025	1.30 to 1.60	—
	SUJ 3	0.95 to 1.10	0.40 to 0.70	0.90 to 1.15	Less than 0.025	Less than 0.025	0.90 to 1.20	—
	SUJ 4	0.95 to 1.10	0.15 to 0.35	Less than 0.50	Less than 0.025	Less than 0.025	1.30 to 1.60	0.10 to 0.25
ASTM A 295	52100	0.93 to 1.05	0.15 to 0.35	0.25 to 0.45	Less than 0.025	Less than 0.015	1.35 to 1.60	Less than 0.10

Table 12.2 Chemical Composition of Carburizing Bearing Steels (Major Elements)

Standard	Designation	Chemical Composition (%)							
		C	Si	Mn	P	S	Ni	Cr	Mo
JIS G 4052	SCr 420 H	0.17 to 0.23	0.15 to 0.35	0.55 to 0.95	Less than 0.030	Less than 0.030	Less than 0.25	0.85 to 1.25	—
	SCM 420 H	0.17 to 0.23	0.15 to 0.35	0.55 to 0.95	Less than 0.030	Less than 0.030	Less than 0.25	0.85 to 1.25	0.15 to 0.35
	SNCM 220 H	0.17 to 0.23	0.15 to 0.35	0.60 to 0.95	Less than 0.030	Less than 0.030	0.35 to 0.75	0.35 to 0.65	0.15 to 0.30
	SNCM 420 H	0.17 to 0.23	0.15 to 0.35	0.40 to 0.70	Less than 0.030	Less than 0.030	1.55 to 2.00	0.35 to 0.65	0.15 to 0.30
JIS G 4053	SNCM 815	0.12 to 0.18	0.15 to 0.35	0.30 to 0.60	Less than 0.030	Less than 0.030	4.00 to 4.50	0.70 to 1.00	0.15 to 0.30
ASTM A 534	8620 H	0.17 to 0.23	0.15 to 0.35	0.60 to 0.95	Less than 0.025	Less than 0.015	0.35 to 0.75	0.35 to 0.65	0.15 to 0.25
	4320 H	0.17 to 0.23	0.15 to 0.35	0.40 to 0.70	Less than 0.025	Less than 0.015	1.55 to 2.00	0.35 to 0.65	0.20 to 0.30
	9310 H	0.07 to 0.13	0.15 to 0.35	0.40 to 0.70	Less than 0.025	Less than 0.015	2.95 to 3.55	1.00 to 1.40	0.08 to 0.15

Table 12.3 Chemical Composition of High-Speed Steel for Bearings Used at High Temperatures

Standard	Designation	Chemical Composition (%)											
		C	Si	Mn	P	S	Cr	Mo	V	Ni	Cu	Co	W
AISI	M50	0.77 to 0.85	Less than 0.25	Less than 0.35	Less than 0.015	Less than 0.015	3.75 to 4.25	4.00 to 4.50	0.90 to 1.10	Less than 0.10	Less than 0.10	Less than 0.25	Less than 0.25

Table 12.4 Chemical Composition of Stainless Steel for Rolling Bearing (Major Elements)

Standard	Designation	Chemical Composition (%)						
		C	Si	Mn	P	S	Cr	Mo
JIS G 4303	SUS 440 C	0.95 to 1.20	Less than 1.00	Less than 1.00	Less than 0.040	Less than 0.030	16.00 to 18.00	Less than 0.75
SAE J 405	51440 C	0.95 to 1.20	Less than 1.00	Less than 1.00	Less than 0.040	Less than 0.030	16.00 to 18.00	Less than 0.75

Table 12.5 Chemical Composition of Steel Sheet and Carbon Steel for Cages (Major Elements)

Classification	Standard	Designation	Chemical Composition (%)				
			C	Si	Mn	P	S
Steel sheet and strip for pressed cages	JIS G 3141	SPCC	Less than 0.12	—	Less than 0.50	Less than 0.04	Less than 0.045
	BAS 361	SPB 2	0.13 to 0.20	Less than 0.30	0.25 to 0.60	Less than 0.03	Less than 0.030
	JIS G 3311	S 50 CM	0.47 to 0.53	0.15 to 0.35	0.60 to 0.90	Less than 0.03	Less than 0.035
Carbon steel for machined cages	JIS G 4051	S 25 C	0.22 to 0.28	0.15 to 0.35	0.30 to 0.60	Less than 0.03	Less than 0.035

Remark BAS refers to the Japanese Bearing Association Standard.

Table 12.6 Chemical Composition of High-Strength Brass for Machined Cages

Standard	Designation	Chemical Composition (%)								
		Cu	Zn	Mn	Fe	Al	Sn	Ni	Impurities	
									Pb	Si
JIS H 5120	CAC301 (HBsC 1)	55.0 to 60.0	33.0 to 42.0	0.1 to 1.5	0.5 to 1.5	0.5 to 1.5	Less than 1.0	Less than 1.0	Less than 0.4	Less than 0.1
JIS H 3250	C 6782	56.0 to 60.5	Residual	0.5 to 2.5	0.1 to 1.0	0.2 to 2.0	—	—	Less than 0.5	—

Remark Improved HBsC 1 is also used.

12.3 Characteristics of Bearing, Shaft, and Housing Materials

Rolling bearings must be able to support high loads, run at high speeds, and endure long periods of operation. It is also important to know the material characteristics of the shaft and housing to maximize bearing performance. Physical and mechanical properties of typical bearing, shaft, and housing materials are shown in Table 12.7.

Table 12.7 Physical and Mechanical Properties of Bearing, Shaft, and Housing Materials

	Material	Heat Treatment	Density g/cm³	Specific Heat kJ/(kg·K)	Thermal Conductivity W/(m·K)	Electric Resistance μΩ·cm	Linear Expansion Coeff. (0 to 100 ℃) ×10⁻⁶/℃	Young's Modulus MPa {kgf/mm²}	Yield Point MPa {kgf/mm²}	Tensile Strength MPa {kgf/mm²}	Elongation %	Hardness HB	Remarks
Bearing	SUJ2	Quenching, tempering	7.83	0.47	46	22	12.5	208 000 {21 200}	1 370 {140}	1 570 to 1 960 {160 to 200}	0.5 Max.	650 to 740	High-carbon-chrome bearing steel No. 2
	SUJ2	Spheroidizing annealing	7.86				11.9		420 {43}	647 {66}	27	180	
	SCr420	Quenching, low temp tempering	7.83		48	21	12.8		882 {90}	1 225 {125}	15	370	Chrome steel
	SAE4320 (SNCM420)	Quenching, low temp tempering			44	20	11.7		902 {92}	1 009 {103}	16	**293 to 375	Nickel-chrome-molybdenum steel
	SNCM815	Quenching, low temp tempering	7.89		40	35	—		—	*1 080 {110} Min.	*12 Min.	*311 to 375	
	SUS440C	Quenching, low temp tempering	7.68	0.46	24	60	10.1	200 000 {20 400}	1 860 {190}	1 960 {200}	—	**580	Martensitic stainless steel
	SPCC	Annealing	7.86	0.47	59	15	11.6	206 000 {21 000}	—	*275 {28} Min.	*32 Min.	—	Cold rolled steel plate
	S25C	Annealing		0.48	50	17	11.8		323 {33}	431 {44}	33	120	Carbon steel for machine structures
	CAC301 (HBsC1)	—	8.5	0.38	123	6.2	19.1	103 000 {10 500}	—	*431 {44} Min.	*20 Min.	—	High-tension brass
Shaft	S45C	Quenching, 650 ℃ tempering	7.83	0.48	47	18	12.8	207 000 {21 100}	440 {45}	735 {75}	25	217	Carbon steel for machine structures
	SCr430	Quenching, 520 to 620 ℃ fast cooling				22	12.5	208 000 {21 100}	*637 {65} Min.	*784 {80} Min.	*18 Min.	*229 to 293	Chrome steel
	SCr440	Quenching, 520 to 620 ℃ fast cooling			45	23			*784 {80} Min.	*930 {95} Min.	*13 Min.	*269 to 331	
	SCM420	Quenching, 150 to 200 ℃ air cooling		0.47	48	21	12.8		—	*930 {95} Min.	*14 Min.	*262 to 352	Chrome-molybdenum steel
	SNCM439	Quenching, 650 ℃ tempering			38	30	11.3	207 000 {21 100}	920 {94}	1 030 {105}	18	320	Nickel-chrome-molybdenum steel
	SC46	Normalizing	—	—	—	—	—	206 000 {21 000}	294 {30}	520 {53}	27	143	Low-carbon cast steel
	SUS420J2	1 038 ℃ oil cooling, 400 ℃ air cooling	7.75	0.46	22	55	10.4	200 000 {20 400}	1 440 {147}	1 650 {168}	10	400	Martensitic stainless steel
Housing	FC200	Casting	7.3	0.50	43	—		98 000 {10 000}	—	*200 {20} Min.	—	*217 Max.	Gray cast iron
	FCD400	Casting	7.0	0.48	20	—	11.7	169 000 {17 200}	*250 {26} Min.	*400 {41} Min.	*12 Min.	*201 Max.	Spheroidal graphite cast iron
	A1100	Annealing	2.69	0.90	222	3.0	23.7	70 600 {7 200}	34 {3.5}	78 {8}	35	—	Pure aluminum
	AC4C	Casting	2.68	0.88	151	4.2	21.5	72 000 {7 350}	88 {9}	167 {17}	7	—	Aluminum alloy for sand casting
	ADC10	Casting	2.74	0.96	96	7.5	22.0	71 000 {7 240}	167 {17}	323 {33}	4	—	Aluminum alloy for die casting
	SUS304	Annealing	8.03	0.50	15	72	15.7 to 16.8	193 000 {19 700}	245 {25}	588 {60}	60	150	Austenitic stainless steel

Note * JIS standard or reference value.
** Though the Rockwell C scale is generally used, Brinel hardness is shown for comparison.

Remark Proportional limits of SUJ2 and SCr420 are 833 MPa {85 kgf/mm²} and 440 MPa {45 kgf/mm²} respectively.

12.4 Technical Data

12.4.1 Comparison of National Standards of Rolling Bearing Steel

The Dimension Series of rolling bearings as mechanical elements have been standardized internationally, and the material to be used for them specified in ISO 683/17 *Heat-treated steels, alloy steels, and free-cutting steels--Part 17: Ball and roller bearing steels*. However, materials are also standardized according to the standards of individual countries and, in some cases, manufacturer modifications.
As internationalization continues, more references to these steel standards will be made. Some applicable standards and their features are described and compared in Tables 12.8 and 12.9.

Table 12.8 Applicable National Standards and Chemical Composition of High-Carbon-Chrome Bearing Steel

JIS G 4805	ASTM	Other Major National Standards	Chemical Composition (%)						Application	Remarks
			C	Si	Mn	Cr	Mo	Others		
SUJ2	—	—	0.95 to 1.10	0.15 to 0.35	≦0.50	1.30 to 1.60	≦0.08	*1	Typical steel for small- and medium-size bearings	Equivalent to each other though there are slight differences in the ranges.
—	A 295-89 52100	—	0.93 to 1.05	0.15 to 0.35	0.25 to 0.45	1.35 to 1.60	≦0.10	P≦0.025 S≦0.015		
—	—	100Cr6 (DIN)	0.90 to 1.05	0.15 to 0.35	0.25 to 0.40	1.40 to 1.65	—	—		
—	—	100C6 (NF)	0.95 to 1.10	0.15 to 0.35	0.20 to 0.40	1.35 to 1.60	≦0.08	P≦0.030 S≦0.025		
—	—	535A99 (BS)	0.95 to 1.10	0.10 to 0.35	0.40 to 0.70	1.20 to 1.60	—	*1		
SUJ3	—	—	0.95 to 1.10	0.40 to 0.70	0.90 to 1.15	0.90 to 1.20	≦0.08	*1	For large-size bearings	SUJ3 is equivalent to Grade 1. Grade 2 has better quenching capability
—	A 485-03 Grade 1	—	0.90 to 1.05	0.45 to 0.75	0.90 to 1.20	0.90 to 1.20	≦0.10	P≦0.025 S≦0.015		
—	A 485-03 Grade 2	—	0.85 to 1.00	0.50 to 0.80	1.40 to 1.70	1.40 to 1.80	≦0.10	P≦0.025 S≦0.015		
SUJ4	—	—	0.95 to 1.10	0.15 to 0.35	≦0.50	1.30 to 1.60	0.10 to 0.25	*1	Rarely used	Better quenching capability than SUJ2
SUJ5	—	—	0.95 to 1.10	0.40 to 0.70	0.90 to 1.15	0.90 to 1.20	0.10 to 0.25	*1	For extra large bearings and thin-walled bearings	Though Grade 3 is equivalent to SUJ5, quenching capability of Grade 3 is better than SUJ5.
—	A 485-03 Grade 3	—	0.95 to 1.10	0.15 to 0.35	0.65 to 0.90	1.10 to 1.50	0.20 to 0.30	P≦0.025 S≦0.015		

Note *1: P ≦ 0.025, S ≦ 0.025
Remark ASTM: Standard of American Society of Testing Materials, DIN: German Standard, NF: French Standard, BS: British Standard

Table 12.9 JIS and ASTM Standards and Chemical Composition of Carburizing Bearing Steel

JIS G 4052 G 4053	ASTM A 534-90	Chemical Composition (%)							Application	Remarks
		C	Si	Mn	Ni	Cr	Mo	Others		
SCr420H	—	0.17 to 0.23	0.15 to 0.35	0.55 to 0.95	≦0.25	0.85 to 1.25	—	*2	For small bearings	Similar steel type
—	5120H	0.17 to 0.23	0.15 to 0.35	0.60 to 1.00	—	0.60 to 1.00	—	*3		
SCM420H	—	0.17 to 0.23	0.15 to 0.35	0.55 to 0.95	≦0.25	0.85 to 1.25	0.15 to 0.35	*2	For small bearings	Similar steel type, though quenching capability of 4118H is inferior to SCM420H
—	4118H	0.17 to 0.23	0.15 to 0.35	0.60 to 1.00	—	0.30 to 0.70	0.08 to 0.15	*3		
SNCM220H	—	0.17 to 0.23	0.15 to 0.35	0.60 to 0.95	0.35 to 0.75	0.35 to 0.65	0.15 to 0.30	*2	For small bearings	Equivalent, though there are slight differences
—	8620H	0.17 to 0.23	0.15 to 0.35	0.60 to 0.95	0.35 to 0.75	0.35 to 0.65	0.15 to 0.25	*3		
SNCM420H	—	0.17 to 0.23	0.15 to 0.35	0.40 to 0.70	1.55 to 2.00	0.35 to 0.65	0.15 to 0.30	*2	For medium bearings	Equivalent, though there are slight differences
—	4320H	0.17 to 0.23	0.15 to 0.35	0.40 to 0.70	1.55 to 2.00	0.35 to 0.65	0.20 to 0.30	*3		
SNCM815	—	0.12 to 0.18	0.15 to 0.35	0.30 to 0.60	4.00 to 4.50	0.70 to 1.00	0.15 to 0.30	*2	For large bearings	Similar steel type
—	9310H	0.07 to 0.13	0.15 to 0.35	0.40 to 0.70	2.95 to 3.55	1.00 to 1.45	0.08 to 0.15	*3		

Note *2: P ≦ 0.030, S ≦ 0.030 *3: P ≦ 0.025, S ≦ 0.015

12.4.2 Long-Life Bearing Steel (NSK Z Steel)

The rolling fatigue life of high-carbon chrome bearing steel (SUJ2, SAE52100) used for rolling bearings is greatly affected by non-metallic inclusions.
Non-metallic inclusions are roughly divided into three-types: sulfide, oxide, and nitride. A long-term life test showed that oxide non-metallic inclusions exert a particularly adverse effect on rolling fatigue life.

Fig. 12.1 shows the parameter (oxygen content) indicating the amount of oxide non-metallic inclusions as it relates to life.
The oxygen amount in steel was minimized as much as possible by reducing impurities (Ti, S) substantially, thereby achieving a decrease in oxide non-metallic inclusions. The resulting long-life steel is known as Z steel. Z steel is an achievement of improved steelmaking facilities and operating conditions made possible by cooperation with steel makers using data from numerous life tests. A graph of the oxygen content in steel over the last 25 years is shown in Fig. 12.2.

The results of a life test with the sample materials in Fig. 12.2 are shown in Fig. 12.3. Life tends to become longer with decreasing oxygen content in steel. High-quality Z steel has a life span which is about 1.8 times longer than that of conventional degassed steel.

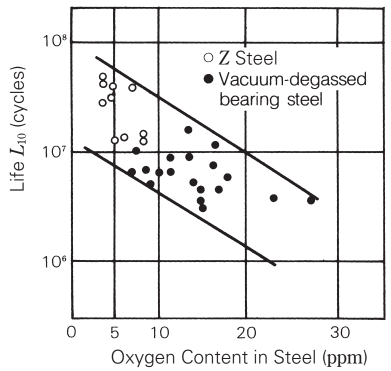


Fig. 12.1 Oxygen Content in Steel and Life of Bearing Steel

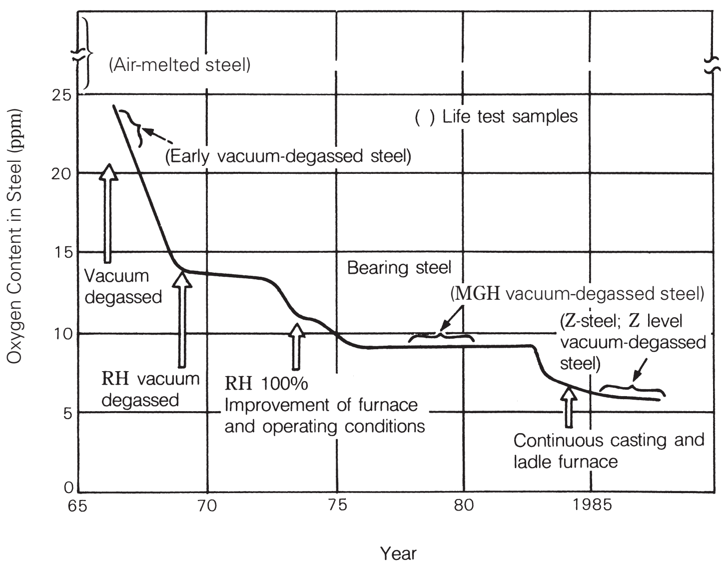
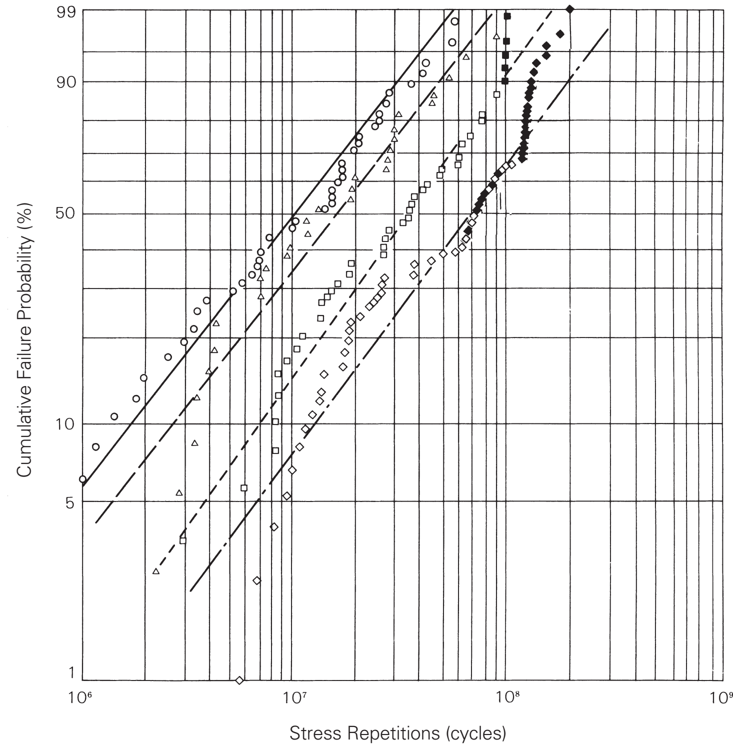


Fig. 12.2 Change in Oxygen Content of NSK Bearing Steels



Classification	Test quantity	Failured quantity	Weibull slope	L_{10}	L_{50}
○ Air-melted steel	44	44	1.02	1.67×10^6	1.06×10^7
△ Vacuum-degassed steel	30	30	1.10	2.82×10^6	1.55×10^7
□ MGH vacuum-degassed steel	46	41	1.16	6.92×10^6	3.47×10^7
◇ Z steel	70	39	1.11	1.26×10^7	6.89×10^7

Remark Testing of bearings marked with ■ and ◆ had not yet completed.

Fig. 12.3 Bearing Steel Life Test Results

12.4.3 Dimensional Stability of Bearing Steel

Sectional changes or changes in the dimensions of rolling bearings over time is called aging deformation. When the inner ring develops expansion due to such deformation, there is a resulting decrease in the interference between the shaft and inner ring. This becomes one of the causes of inner ring creep. Creep (a phenomenon where the shaft and inner ring slip mutually) causes the bearing to generate excess heat that may result in seizure, resulting in critical damage to the entire machine. Consequently, appropriate measures must be taken against aging deformation of the bearing depending on the application.
Aging deformation of bearings may be attributed to secular thermal decomposition of retained austenite in steel after heat treatment. The bearing develops gradual expansion along with phase transformation.

The dimensional stability of the bearings, therefore, varies in accordance with the relative relationship between tempering during heat treatment and the bearing's operating temperature. Dimensional stability increases with rising tempering temperature while retained austenite decomposition gradually expands as the bearing's operating temperature rises.

Fig. 12.4 shows how temperature influences the bearing's dimensional stability. The right side of the figure shows the interference between the inner ring and shaft for various shaft tolerance classes as percentages of the shaft diameter. As is evident from Fig. 12.4, dimensional stability becomes more unfavorable as the bearing's temperature rises. Under these conditions, the interference between the shaft and inner ring of a general bearing is expected to decrease gradually. In this view, loosening of the fit surface must be prevented by using a bearing that has undergone a dimensional stabilization treatment.

When bearing temperature is high, there is a possibility of inner ring creep. Please contact NSK beforehand in order to select an appropriate bearing.

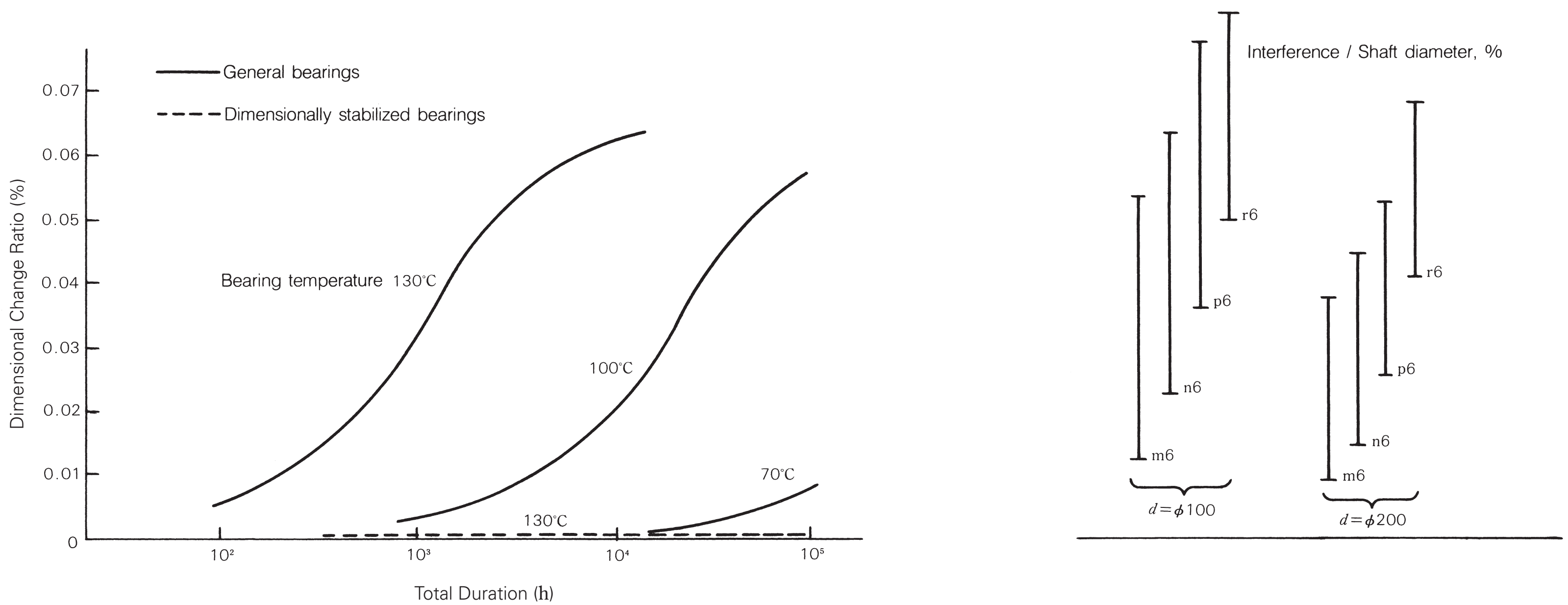


Fig. 12.4 Bearing Temperature and Dimensional Change Ratio

12.4.4 Fatigue Analysis

In order to fully predict the fatigue life of rolling bearings and estimate residual life, knowledge of all fatigue breakdown phenomena of bearings is essential. Unfortunately, this means it will take some time before we reach a stage enabling perfect prediction and estimation. However, rolling fatigue proceeds under compressive stress at the contact point and is known to cause material changes until breakdown occurs. In many cases, it is possible to estimate the degree of fatigue in bearings by detecting this material change. However, this estimation method is not effective in cases where defects in the raceway surface cause premature cracking or where chemical corrosion occurs on the raceway. In these two cases, flaking progresses ahead of material change.

(1) ***Measurement of Fatigue Degree***

The progress of fatigue in a bearing can be determined by using an X-ray to measure changes in residual stress, diffraction half-value width, and amount of retained austenite.

These values change as fatigue progresses as shown in Fig. 12.5. As residual stress grows early and approaches a saturation value, it can be used to detect extremely small fatigue. For large fatigue, change of the diffraction half-value width and retained austenite amount correlate with the progress of fatigue. These measurements are consolidated into one parameter (fatigue index) to determine a relationship with the endurance test period of a bearing.

Measured values were collected by carrying out endurance tests with many ball, tapered roller, and cylindrical roller bearings under various loads and lubrication conditions. Simultaneously, measurements were made on bearings used in actual machines.

Fig. 12.6 summarizes the data. Variance is considerable, reflecting the complexity of fatigue. Nevertheless, there is a correlation between the fatigue index and the endurance test period or operating hours. The degree of fatigue can be handled quantitatively, albeit with some uncertainty.

The description of "subsurface fatigue" in Fig. 12.6 applies when fatigue is governed by internal shearing stress. On the other hand, "surface fatigue" is correlated with earlier and more severe fatigue resulting from contamination or breakdown of the lubricating oil film.

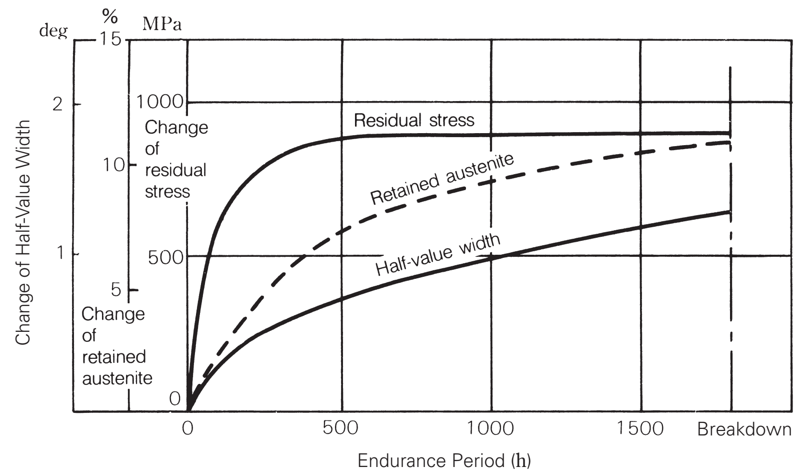


Fig. 12.5 Change in X-Ray Measurements

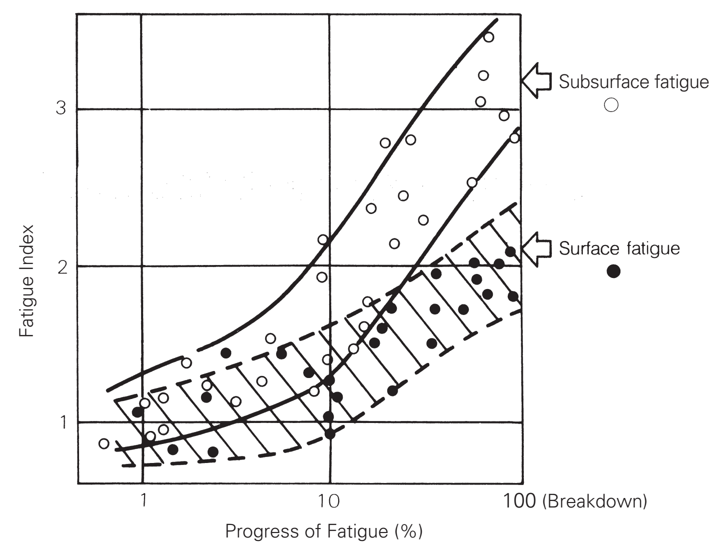


Fig. 12.6 Fatigue Progress and Fatigue Index

(2) ***Surface and Subsurface Fatigue***

Rolling bearings have an extremely smooth finish surface and enjoy relatively satisfactory lubrication conditions. Generally, internal shearing stress below the rolling surface governs the failure of a bearing.

When shearing stress caused by rolling contact reaches a maximum level at a certain depth below the surface, a crack (the origin of a breakdown) occurs under the surface. When the raceway is broken due to such subsurface fatigue, the fatigue index measured by depth increases according to the theoretical calculation of shearing stress, as evident from the example ball bearing shown in Fig. 12.7. The fatigue pattern shown in Fig. 12.7 usually occurs when lubrication conditions are satisfactory and a sufficiently thick oil film is present at rolling contact points. The basic dynamic load rating described in this catalog is determined using data from bearing failures according to the above internal fatigue pattern. Fig. 12.8 shows an example cylindrical roller bearing with an unsatisfactory oil film subjected to an endurance test. It is evident that the amount of surface fatigue increases much earlier than indicated by the calculated life.

In this test, all bearings failed before subsurface fatigue became apparent. Thus, bearing failure due to surface fatigue is mostly attributed to lubrication conditions such as insufficient oil film due to excessively low oil viscosity or entry of foreign material or moisture into the lubricant.

Therefore, bearing failure induced by surface fatigue occurs before that of subsurface fatigue in most bearing applications, even though subsurface fatigue is the metric used to determine the original life.

Fatigue analysis has shown that bearings used in actual machines overwhelmingly show the surface fatigue pattern of failure instead of the subsurface pattern.

Knowing the distribution of the fatigue index of bearings in use leads to an increased understanding of residual life, lubrication, and load conditions.

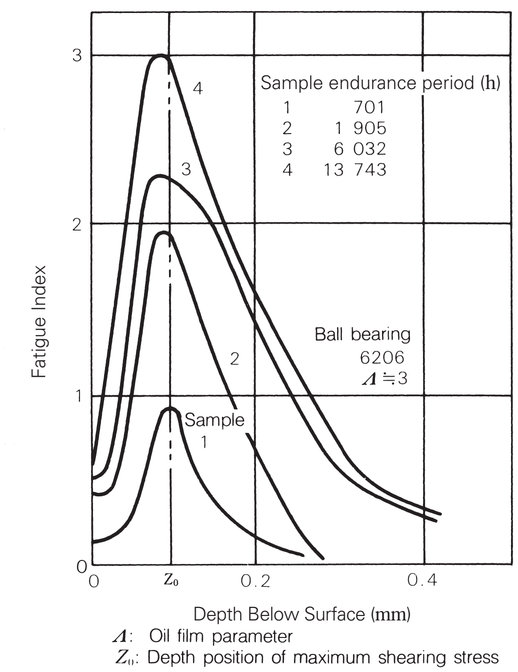


Fig. 12.7 Progress of Subsurface Fatigue

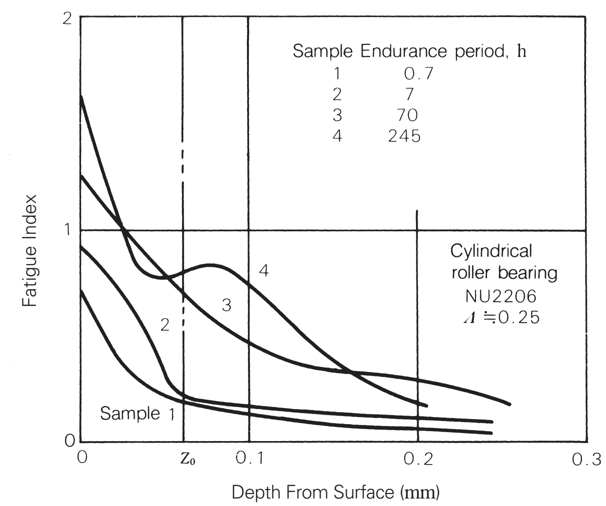


Fig. 12.8 Progress of Surface Fatigue

12.4.5 Hi-TF Bearings and Super-TF Bearings

(1) *Hi-TF Bearings, Super-TF Bearings, and TF Technology*

In its quest for longer bearing service life, NSK has spent many years analyzing the mechanisms of fatigue in bearings, researching and developing materials and heat treatment processes, and optimizing for operating conditions. The range of approaches to achieving longer service life taken by our research team is shown in Fig. 12.9. Technology incorporated in our Hi-TF bearings and Super-TF bearings is designed to maximize service life under conditions where bearings are subject to surface-originated flaking.

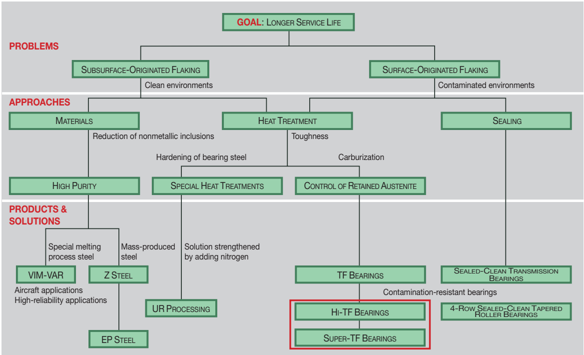


Fig. 12.9 Approaches to Achieving Longer Service Life in Bearings

(2) *TF Technology*

Bearings may be required to operate under clean or dirty conditions; under dirty conditions, their lubricating oil is easily contaminated. Metal particles or casting sand in the lubricating oil make dents in the contact surfaces. As shown in Fig. 12.10, stress is concentrated around these dents and eventually leads to cracking and to surface-originated flaking. The concentration of stress around a dent is expressed by the equation $[P/P_0 \propto (r/c)^{-0.24}]$, where "$r$" is the radius at the shoulder of the dent and "$2c$" is the shoulder-to-shoulder width of the dent. The greater the value of "r/c", the smaller the stress concentration and the longer the service life of the bearing.

NSK is a world leader in the research and development of materials that reduce the concentration of stress around surface dents. As shown in Fig. 12.11, our work has revealed that a high level of retained austenite is an extremely effective means of maximizing the r/c value around surface dents in the bearing material. TF technology is a unique heat treatment process developed by NSK to optimize the level of retained austenite in bearing materials.

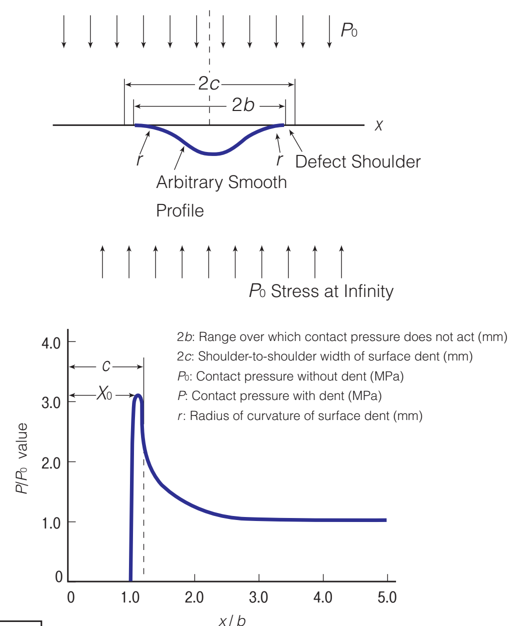


Fig. 12.10 Concentration of Stress around a Surface Dent

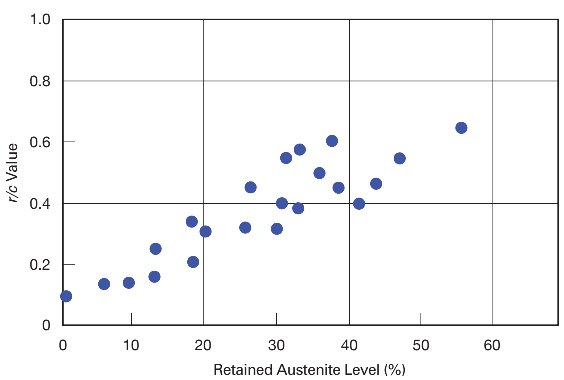


Fig. 12.11 Relationship of r/c Value to Retained Austenite Level

(3) *Material Properties of Hi-TF Bearings and Super-TF Bearings*

NSK has developed Hi-TF and Super-TF bearings as two Series of bearings that offer longer service life exceeding that of TF Bearings. As mentioned, the approach to achieving long service life taken by NSK is to minimize the concentration of stress around the shoulders of surface dents. A high level of retained austenite helps achieve this and maximizes r/c values. However, austenite itself has a soft microstructure and reduces the hardness of the bearing material. In order to meet the seemingly conflicting needs for greater hardness of the bearing material and a higher level of retained austenite, NSK adopted a technique that both promotes uniform distribution and reduces the diameter of carbide and carbonitride particles in the bearing material.

To this end, our researchers developed a new type of steel that has the proper quantity of elements used in the formation of carbides and developed a proprietary process to impart minute carbides and nitride into the material surface. Hi-TF bearings contain a specific amount of added chrome, while Super-TF bearings contain a specific amount of added chrome and molybdenum. Figures 12.12 and 12.13 illustrate the image analysis results of carbide distribution in the structures of Super-TF bearings and ordinary carburized steel bearings. It is clear that Super-TF bearings have a greater amount of fine-size carbide and carbonitride particles. Fig. 12.14 shows that the formations of finer carbide and carbonitride particles give Hi-TF bearings and Super-TF bearings a greater degree of hardness and higher retained austenite levels than those of TF Bearings. As a result, Hi-TF bearings and Super-TF bearings achieve a higher r/c value. (Fig. 12.15)

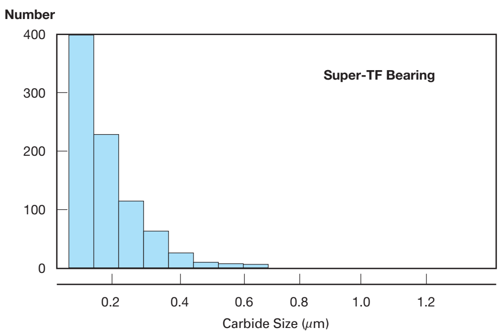


Fig. 12.12 Average Diameter of Carbide and Carbonitride Particles in a Super-TF Bearing

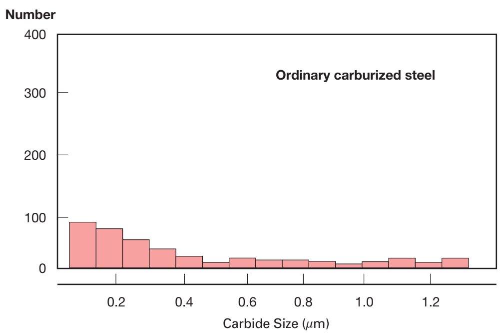


Fig. 12.13 Average Diameter of Carbide Particles in an Ordinary Carburized Steel Bearing

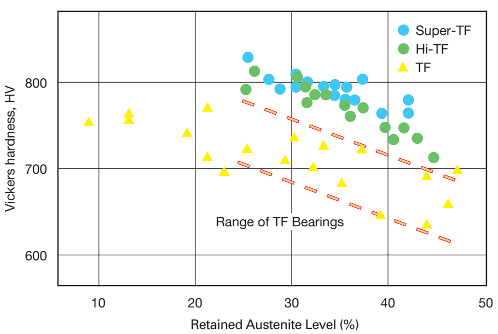


Fig. 12.14 Relationship of Material Hardness and Retained Austenite Level

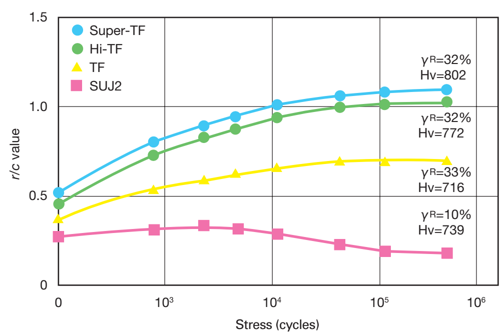


Fig. 12.15 Change of r/c Value Under Repeated Stress

(4) *Service Life Under Contaminated Lubrication Conditions*

Table 12.10 and Fig. 12.17 show the results of service life tests conducted under contaminated lubrication conditions with L44649/10 tapered roller bearings. If the service life of an ordinary carburized steel bearing of this type is taken as 1, then the L_{10} life of TF, Hi-TF, and Super-TF bearings would be 4.5, 7.1, and 10.2 respectively (Table 12.10). Hi-TF bearings and Super-TF bearings thus offer over seven to ten times the service life of ordinary carburized steel bearings. Service life is generally affected both by the conditions in which the bearing is used and by the amount of contamination in the lubricant. Under contaminated lubrication conditions, service life may fall to as little as 1/5 the catalog life.
Hi-TF bearings and Super-TF bearings can achieve for the first time service life that exceeds the catalog life of existing products under contaminated lubrication.

Ordinary Carburized Steel	TF	Hi-TF	Super-TF
1	4.5	7.1	10.2

Table 12.10 Comparison of Service Life of L44649/10 Tapered Roller Bearings

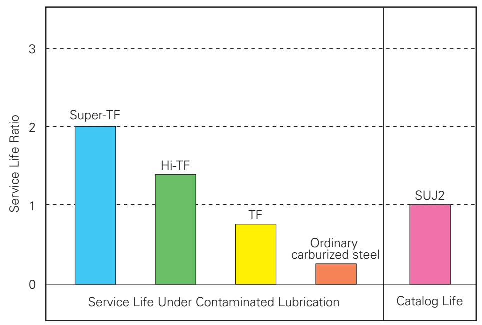


Fig. 12.16 Comparison of Service Life Under Contaminated Lubrication

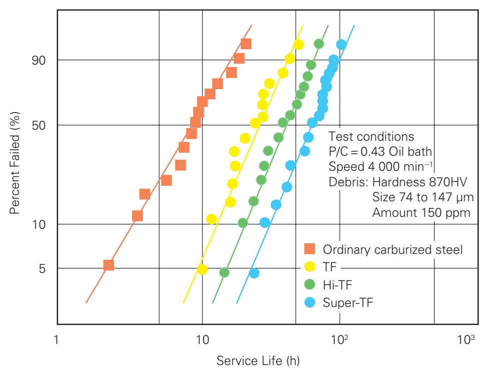


Fig. 12.17 Service Life of L44649/10 Bearings Under Contaminated Lubrication

(5) *Service Life under Clean Lubrication Conditions*

Fig. 12.18 shows the results of service life tests under clean lubrication conditions using Series 6206 deep groove ball bearings. Under clean lubrication, Hi-TF Bearings and Super-TF Bearings show a slightly longer service life than those made with SUJ2 steel. The most important factor is the cleanliness of the steel from which the bearing is made. Material with a greater degree of purity offers a greater degree of long-life performance.

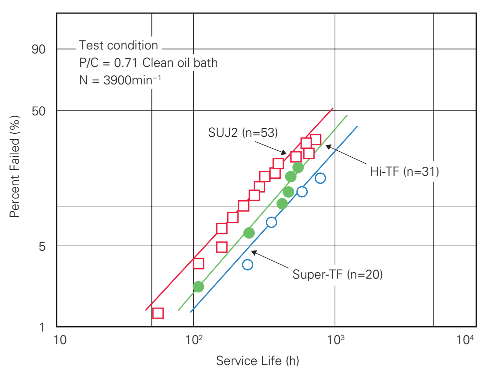


Fig. 12.18 Service Life Tests of 6206 Bearings Under Clean Lubrication

(6) *Service Life Under Boundary Lubrication Conditions*

Under boundary lubrication conditions where there is an insufficient EHL film, metal-to-metal contact occurs, thus reducing bearing life. Fig. 12.19 shows the results of service life tests conducted under conditions where oil film parameter Λ, which represents the ratio of the thickness of the oil film to the roughness of the surface, is very small (Λ=0.3). At this very small ratio, peeling damage occurs (Fig. 12.20), but in Hi-TF bearings and Super-TF bearings, the concentration of stress around the projections of the contact area is reduced, giving a service life approximately 4.7 times and 5.5 times greater than that of ordinary carburized steel bearings.

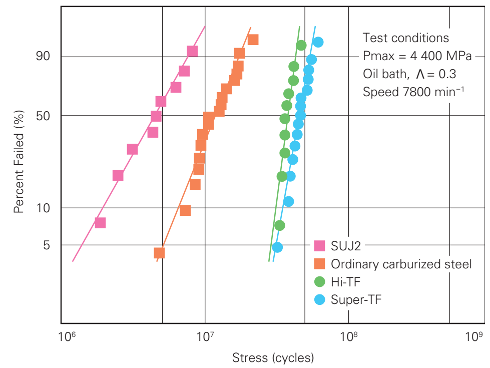


Fig. 12.19 Service Life Tests Under Boundary Lubrication Conditions

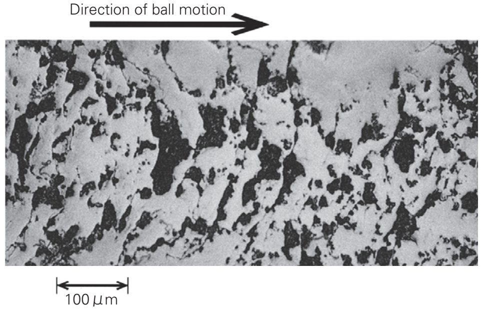


Fig. 12.20 Peeling Damage

(7) *Wear and Seizure Resistance*

Besides extending service life under contaminated lubrication conditions, another goal is to increase the bearing's resistance to wear and seizure by ensuring the dispersion of a large number of fine carbides and nitrides in the bearing material. Fig. 12.21 presents the results of a Sawin wear test that shows the degree of wear and the seizure limit for different types of bearing material. The test reveals that Hi-TF bearings and Super-TF bearings have superior wear resistance to both SUJ2 steel and TF bearings. Hi-TF bearings and Super-TF bearings are also 20 % to 40 % more resistant to seizure than both SUJ2 steel and TF bearings.

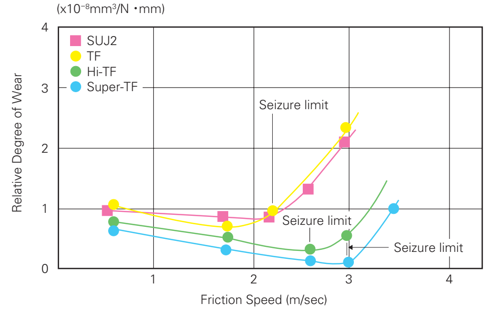


Fig. 12.21 Comparison of Wear Resistance

(8) *Heat Resistance*

Fig. 12.22 shows the results of service life tests conducted with Series 6206 ball bearings at 160 ℃ under clean lubrication conditions. Test results reveal that Super-TF bearings (heat-resistant specifications) have approximately 4 times the service life of SUJ2X26 steel bearings.

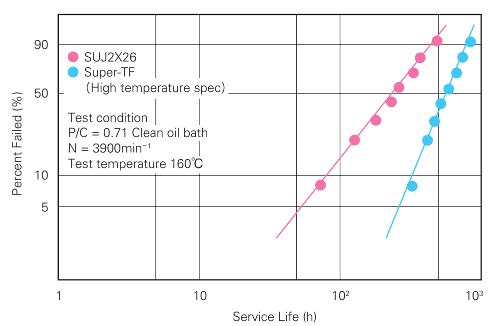


Fig. 12.22 Service Life Test of 6206 Under High Temperature Clean Lubrication

12.4.6 Physical Properties of Representative Polymers Used as Bearing Material

Because of their light weight, easy formability, and high corrosion resistance, polymers are widely used as cage materials. Polymers may be used independently, but they are usually combined with functional fillers to form a composite material. Composites can be customized to have specific properties and thus be used as bearing materials. For example, fillers can be used to impart attributes such as low friction, low wear, non-stick slip characteristics, high limit *PV* values, non-scrubbing of counterpart material, mechanical properties, heat resistance, and so on.

Table 12.11 shows characteristics of representative polymer materials used for bearings.

Table 12.11 Characteristics of Representative Polymers

Plastics	Elastic Modulus (GPa) (1)	Strength GPa (1)	Density g/cm^3	Specific Elastic Modulus $\times 10^4$mm	Specific Strength $\times 10^4$mm	Melting point ℃	Glass Transition Temp ℃	Thermal Deformation Temperature ℃ (2)	Continuous Operating Temperature ℃	Remarks
Polyethylene HDPE UHMWPE	 0.115 0.5	 0.03 0.025	 0.96 0.94	 12.6 53.2	 3.3 2.7	 132 136	 −20 −20	 75/50 75/50	 — —	High creep and toughness, softening
Polyamide Nylon 6 Nylon 66	 2.5 3.0	 0.07 0.08	 1.13 1.14	 221.2 263.2	 6.2 7.0	 215 264	 50 60	 150/57 180/60	 80 to 120 80 to 120	High water absorption and toughness
Nylon 11	1.25	0.04	1.04	120.2	3.8	180	—	150/55	Lower than nylon 6 or 66	Low water absorption
Polytetra fluoroethylene PTFE	0.40	0.028	2.16	18.5	1.3	327	115	120/—	260	High creep, sintering, low friction, low adhesion, inert. Stable at 290 ℃
Poly buthylene terephthalate PBT	2.7	0.06	1.31	206.1	4.6	225	30	230/215	155	
Polyacetal POM Homo-polymer Co-polymer	 3.2 2.9	 0.07 0.06	 1.42 1.41	 225.3 205.7	 4.9 4.3	 175 165	 −13 —	 170/120 155/110	 — 104	High hardness and toughness, low water absorption
Polyether sulfon PES	2.46	0.086	1.37	179.6	6.3	—	225	210/203	180	Usable up to 200 ℃. Chemically stable
Polysulfon PSf	2.5	0.07	1.24	201.6	5.6	—	190	181/175	150	
Polyallylate (Aromatic polyester)	1.3 3.0	0.07 0.075	1.35 1.40	96.3 214.3	5.2 5.4	350 350	— —	293 293	300 260 to 300	Inert, high hardness, used as filler for PTFE. Stable up to 320 ℃
Polyphenylene sulfide PPS (GF 40%)	4.2	0.14	1.64	256.1	8.5	275	94	>260	220	Hot cured at 360 ℃
Polyether ether keton PEEK	1.7	0.093	1.30	130.8	7.2	335	144	152	240	
Poly-meta-phenylene isophthalic amide	10 (fiber) 7.7 (mold)	0.7 0.18	1.38 1.33	724.6 579	50.7 13.5	375 415 (decomposition)	>230 >230	280 280	220 220	Fire retardant, heat-resistant fiber
Polypromellitic imide (Aromatic polyimide) PI	3 (film)	0.17	1.43	203	7.0	Heat de-composition	417 decomposition	360/250	300 (3)	No change in inert gas up to 350 ℃
	2.5 to 3.2 (mold)	0.1	1.43	203	7.0	Heat de-composition	417 decomposition	360/250	260	Usable up to 300 ℃ for bearings. Sintering, no fusion (molded products)
Polyamide imide PAI	4.7	0.2	1.41	333.3	14.2	—	280	260	210	Usable up to 290 ℃ as an adhesive or enamel. Improved polyimide for melt molding
Polyether imide (Aromatic polyimide) PI	3.6	0.107	1.27	240.9	—	—	215	210/200	170	Improved polyimide for melt molding
Polyamino bis-maleimide	—	0.35	1.6	—	21.9	—	—	330 (3)	260	

Notes (1) GPa≒10^4 kgf/cm^2=10^2 kgf/mm^2
(2) If there is a slash mark "/" in the thermal deformation temperature column, the left value applies to 451 kPa, while all other values apply to 1.82 MPa.
(3) Reference value

12.4.7 Characteristics of Nylon Cage Material

Recently, plastic cages are increasingly used in place of metal cages in bearings. Advantages of using plastic cages include:

(1) Light weight and favorable for use with high-speed rotation
(2) Self-lubricating functionality and low wear. Abrasion powders are usually not produced when plastic cages are used, so a highly clean internal state can be maintained.
(3) Low noise appropriate for silent environments
(4) Highly corrosion resistant, no rusting
(5) Highly shock resistant, proving durable under high moment loading
(6) Easy molding of complicated shapes ensures high freedom of cage shape selection. Thus, better cage performance can be obtained.

Disadvantages when compared with metal cages include low heat resistance and a limited operating temperature range (normally 120 °C). Care is also necessary for use because plastic cages are sensitive to certain chemicals. Polyamide resin is a representative plastic cage material. Among polyamide resins, nylon-66 is often used because of its high heat resistance and mechanical properties.

Polyamide resin contains the amide coupling (-NHCO-) with hydrogen bonding capability in its molecular chain and is characterized by its regulation of mechanical properties and water absorption according to concentration and hydrogen bonding state. High water absorption (Fig. 12.23) of nylon 66 is generally regarded as a shortcoming because it causes dimensional distortion and deterioration of rigidity. On the other hand, water absorption helps enhance flexibility and prevents cage damage during bearing assembly when a cage is required to have a substantial holding interference for the rolling elements. This also improves toughness which is effective for shock absorption during operation. As such, this so-called shortcoming may be considered as an advantage under certain conditions.

Nylon can be improved substantially in strength and heat resistance by adding a small amount of fiber. Therefore, materials reinforced by glass fiber may be used depending on the cage type and application. To maintain deformation of the cage during bearing assembly, a relatively small amount of glass fiber is commonly used to reinforce the cage. (Table 12.12)

Nylon-66 demonstrates vastly superior performance under light operating conditions and has wide application possibilities as a mainstream plastic cage material. However, it often develops sudden deterioration under severe conditions (in high-temperature oil, etc.). Therefore, carefully monitor this material during practical operation.

As an example, Table 12.13 shows the time necessary for the endurance performance of various nylon-66 materials to drop to 50% of the initial value under several different cases. Material deterioration in oil varies depending on the kind of oil. Deterioration is excessive if the oil contains an extreme-pressure agent. Sulfurous extreme-pressure agents accelerate deterioration more than phosphorous extreme-pressure agents and such deterioration occurs more rapidly with rising temperatures.

On the other hand, material deteriorates less in grease or air than in oil. In addition, materials reinforced with glass fiber can suppress deterioration by the reinforcement effect of glass fibers, thereby helping to extend the durability period.

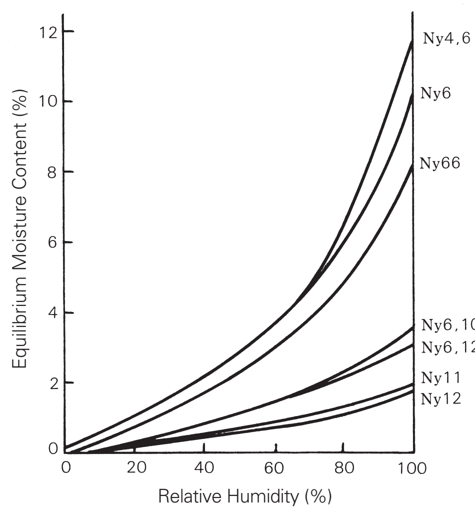


Fig. 12.23 Equilibrium Moisture Content and Relative Humidity of Various Nylons

Table 12.12 Example Applications With Fiber Reinforced Nylon Cages

	Bearing Type	Main Applications	Cage Material
Ball Bearing	Miniature ball bearings	VCR, IC cooling fans	Nylon 66 (Glass fiber content: 0 to 10%)
	Deep groove ball bearings	Alternators, fan motors for air conditioners	
	Angular contact ball bearings	Magnetic clutches, automotive wheels	
Roller Bearing	Needle roller bearings	Automotive transmissions	Nylon 66 (Glass fiber content: 10 to 25%)
	Tapered roller bearings	Automotive wheels	
	ET-type cylindrical roller bearings	General	
	H-type spherical roller bearings	General	

Table 12.13 Environmental Resistance of Nylon-66 Resin

Environment		Temperature, °C	Glass Content	Time Until Physical Property Value Drops to 50% (h) 500 1000 1500 2000	Remarks
Oil	Gear oil	120	0, D		Contains an extreme pressure additive
		140	0, A, D		
	Synthetic lubricating oil	100	A		Contains an extreme pressure additive
		120	A		
		130	0, A, C		
		150	0, B, D		
	Hydraulic oil	80	0, D		Contains an extreme pressure additive
		120	0, D		
		150	0, D		
	ATF oil	120	0, D		
		140	0, A, D		
	Engine oil	120	0, D		
Grease		80	0, D		
		120	0, D		
		130	A, D		
Air		160	0, A, C		
		180	0, B		

Remarks: Glass content: A<B<C<D

12.4.8 Heat-Resistant Resin Materials for Cages

Currently, polyamide resin shows superior performance under medium-intensity operating environmental conditions. This feature plus its relative inexpensiveness has led to its use in increasing quantities. However, contact with acids or oils containing an extreme pressure agent or continuous use at or above 120 °C deteriorates and ages the material over time.

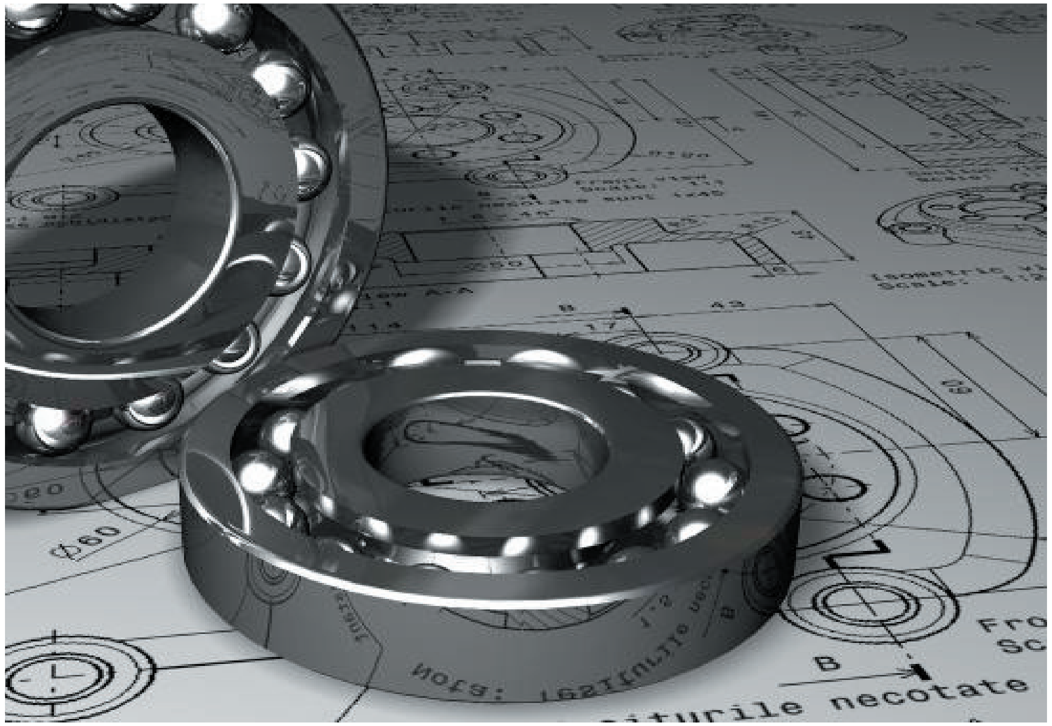
Super-engineering plastics should be used for the cage materials of bearings running in severe environments, such as temperatures over 150 °C or with corrosive chemicals present. Though super-engineering plastics have good material heat resistance, chemical resistance, rigidity at high temperature, and mechanical strength, they have problems with characteristics required of cage materials, such as toughness during molding or bearing assembly, weld strength, and fatigue resistance. Furthermore, the costs of these materials are high. Table 12.14 shows the properties of typical super-engineering plastics that can be injection molded into cage shapes.

Among the materials in Table 12.14, though branch-type polyphenylene sulfide (PPS) is often used, the cage design is restricted since forced-removal from the die is difficult due to poor toughness and brittleness. Moreover, PPS is not ideal as a cage material since the claw, stay, ring, or flange of the cage can be easily broken on the bearing assembly line. On the other hand, the heat-resistant plastic cage developed by NSK, is made of linear-chain high molecules which have been polymerized from molecular chains. These molecular chains do not contain branches or crosslinking, so they have high toughness compared to the former material (branch PPS). Linear PPS is not only superior in heat resistance, oil resistance, and chemical resistance, but also has good mechanical characteristics such as the potential for snap fitting (an important characteristic for cages), and high-temperature rigidity.

NSK has reduced the disadvantages associated with linear PPS, chiefly difficulty of removal from the die and slow crystallization speed, thereby establishing it as a material suitable for cages. Thus, linear PPS satisfies the required capabilities for a heat-resistant cage material that considers cost and performance.

Table 12.14 Properties of Typical Super-Engineering Plastic Materials for Cages

Classification	Polyether Sulfone (PES)	Polyether Imide (PEI)	Polyamide Imide (PAI)	Polyether Etherketon (PEEK)	Branch Polyphenylene Sulfide (PPS)	Linear Polyphenylene Sulfide (L-PPS)
Resin	Amorphous resin	Amorphous resin	Amorphous resin	Crystalline resin	Crystalline resin	Crystalline resin
Continuous Temp.	180 ℃	170 ℃	210 ℃	240 ℃	220 ℃	220 ℃
Physical Properties	•Poor toughness (Care necessary regarding cage shape) •Low weld strength •Small fatigue resistance	•Poor toughness •Low weld strength •Low fatigue resistance	•Very brittle (No forced-removal molding) •Special heat treatment before use •High rigidity, after heat treatment	•Excellent toughness, wear, and fatigue resistance •Small weld strength	•Excellent mechanical properties •Slightly low toughness	•Excellent mechanical properties •Good toughness •Good dimensional stability (No water absorption)
Environmental Properties	•Water absorption (Poor dimensional stability) •Good aging resistance •Poor stress cracking resistance	•Good aging resistance •Poor stress cracking resistance	•Good environmental resistance	•Good environmental resistance	•Good environmental resistance	•Good environmental resistance (not affected by most chemicals, doesn't deteriorate in high temperature oil with extreme pressure additives).
Material Cost (Superiority)	3	2	5	4	1	1
Cage Application	•Many performance problems •High material price	•Many performance problems •High material cost	•Good performance •High material and molding cost (For special applications)	•Excellent performance •High material cost (For special applications)	•Problems with toughness •Cost is high compared to performance	•Reasonable cost for performance (For general applications)



13. DESIGN OF SHAFTS AND HOUSINGS

13. DESIGN OF SHAFTS AND HOUSINGS

13.1 Accuracy and Surface Finish of Shafts and Housings

If the accuracy of a shaft or housing does not meet the specification, the performance of the bearings will be affected and they will not perform to their full capability. For example, inaccuracy in the squareness of the shaft shoulder may cause misalignment of the bearing inner and outer rings, which may reduce the bearing fatigue life by adding an edge load in addition to the normal load. Cage fracture and seizure sometimes occur for this reason. Housings should be rigid in order to provide firm bearing support and are also advantageous in regards to noise, load distribution, etc.

For normal operating conditions, a turned finish or smooth bored finish is sufficient for the fitting surface; however, a ground finish is necessary for applications where vibration and noise must be low or where heavy loads are applied.

In cases where two or more bearings are mounted in a single-piece housing, the fitting surfaces of the housing bore should be designed so both bearing seats may be finished together in one operation, such as inline boring. In the case of split housings, take care in the fabrication of the housing so that the outer ring does not become deformed during installation. The tolerance and surface finish of shafts and housings for normal operating conditions are listed in Table 13.1.

Table 13. 1 Tolerance and Roughness of Shaft and Housing

Item	Class of Bearings	Shaft	Housing Bore
Tolerance for Out-of-Roundness	Normal, Class 6	$\frac{IT3}{2}$ to $\frac{IT4}{2}$	$\frac{IT4}{2}$ to $\frac{IT5}{2}$
	Class 5, Class 4	$\frac{IT2}{2}$ to $\frac{IT3}{2}$	$\frac{IT2}{2}$ to $\frac{IT3}{2}$
Tolerance for Cylindricality	Normal, Class 6	$\frac{IT3}{2}$ to $\frac{IT4}{2}$	$\frac{IT4}{2}$ to $\frac{IT5}{2}$
	Class 5, Class 4	$\frac{IT2}{2}$ to $\frac{IT3}{2}$	$\frac{IT2}{2}$ to $\frac{IT3}{2}$
Tolerance for Shoulder Runout	Normal, Class 6	IT3	IT3 to IT4
	Class 5, Class 4	IT3	IT3
Roughness of Fitting Surfaces R_a	Small Bearings	0.8	1.6
	Large Bearings	1.6	3.2

Remarks This table gives general recommendation using the radius measuring method. The basic tolerance (IT) class should be selected in accordance with the bearing tolerance class. Please refer to the Appendix Table 11 (Page E016) for IT values.
If the outer ring is mounted in the housing bore with interference or a thin cross-section bearing is mounted on a shaft and housing, the tolerance of the shaft and housing should be tighter since this affects the bearing raceway directly.

13.2 Shoulder and Fillet Dimensions

The shoulders of the shaft or housing in contact with the face of a bearing must be perpendicular to the shaft center line (refer to Table 13.1). The front face side shoulder bore of the housing for a tapered roller bearing should be parallel with the bearing axis in order to avoid interference with the cage.

The fillets of the shaft and housing should not come in contact with the bearing chamfer; therefore, the fillet radius r_a must be smaller than the minimum bearing chamfer dimension r or r_1 .

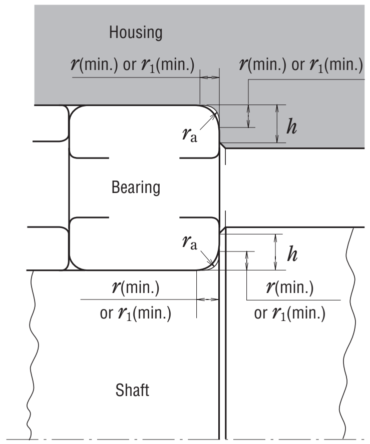


Fig. 13.1 Chamfer Dimensions, Fillet Radius of Shaft and Housing, and Shoulder Height

The shoulder heights for both shafts and housings for radial bearings should be sufficient to provide good support over the face of the bearings, but enough face should extend beyond the shoulder to permit use of special dismounting tools. The recommended minimum shoulder heights for Metric Series radial bearings are listed in Table 13.2

Nominal dimensions associated with bearing mounting, including the proper shoulder diameters, are listed in the bearing tables. Sufficient shoulder height is particularly important for supporting the side ribs of tapered roller bearings and for cylindrical roller bearings subjected to high axial loads.

The values of h and r_a in Table 13.2 should be adopted in those cases where the fillet radius of the shaft or housing is as shown in Fig. 13.2 (a), while the values in Table 13.3 are generally used with an undercut fillet radius produced when grinding the shaft as shown in Fig. 13.2 (b).

Table 13. 2 Recommended Minimum Shoulder Heights for Use With Metric Series Radial Bearings

Units : mm

Nominal Chamfer Dimensions r (min.) or r_1 (min.)	Shaft or Housing		
	Fillet Radius r_a (max.)	Minimun Shoulder Heights h (min.)	
		Deep Groove Ball Bearings ([1]), Self-Aligning Ball Bearings, Cylindrical Roller Bearings ([1]), Solid Needle Roller Bearings	Angular Contact Ball Bearings, Tapered Roller Bearings ([2]), Spherical Roller Bearings
0.05	0.05	0.2	—
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	1.25
0.6	0.6	2	2.5
1	1	2.5	3
1.1	1	3.25	3.5
1.5	1.5	4	4.5
2	2	4.5	5
2.1	2	5.5	6
2.5	2	—	6
3	2.5	6.5	7
4	3	8	9
5	4	10	11
6	5	13	14
7.5	6	16	18
9.5	8	20	22
12	10	24	27
15	12	29	32
19	15	38	42

Note ([1]) When axial loads are applied, the shoulder height must be sufficiently higher than these values.
([2]) When heavy axial loads are applied, the shoulder height must be sufficiently higher than these values.

Remarks 1. The corner fillet radius is also applicable to thrust bearings.
2. The shoulder diameter is listed instead of shoulder height in the bearing tables.

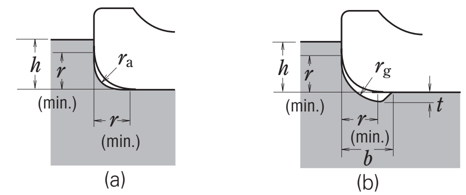


Fig. 13. 2 Chamfer Dimensions, Fillet Radius, and Shoulder Height

Table 13. 3 Shaft Undercut

Units : mm

Chamfer Dimensions of Inner and Outer Rings r (min.) or r_1 (min.)	Undercut Dimensions		
	t	r_g	b
1	0.2	1.3	2
1.1	0.3	1.5	2.4
1.5	0.4	2	3.2
2	0.5	2.5	4
2.1	0.5	2.5	4
2.5	0.5	2.5	4
3	0.5	3	4.7
4	0.5	4	5.9
5	0.6	5	7.4
6	0.6	6	8.6
7.5	0.6	7	10

For thrust bearings, the squareness and contact area of the supporting face for the bearing rings must be adequate. In the case of thrust ball bearings, the housing shoulder diameter D_a should be less than the pitch circle diameter of the balls, and the shaft shoulder diameter d_a should be greater than the pitch circle diameter of the balls (Fig. 13.3).
For thrust roller bearings, the full contact length between rollers and rings should be supported by the shaft and housing shoulder (Fig. 13.4).
These diameters, d_a and D_a, are listed in the bearing tables.

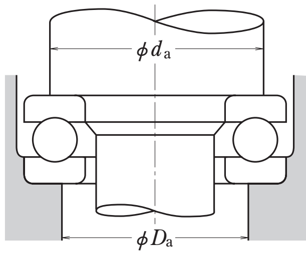


Fig. 13.3 Face-Supporting Diameters for Thrust Ball Bearings

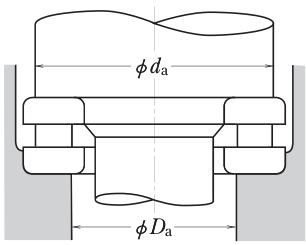


Fig. 13.4 Face-Supporting Diameters for Thrust Roller Bearings

13.3 Bearing Seals

To insure the longest possible life of a bearing, seals may be necessary to prevent leakage of lubricant or entry of dust, water, or other harmful material such as metallic particles. The seals must be free from excessive running friction and chance of seizure. They should also be easy to assemble and disassemble. Be sure to select a suitable seal for individual applications that considers the lubricating method.

13.3.1 Non-Contact Seals

Various sealing devices that do not contact the shaft are available, such as oil grooves, flingers, and labyrinths. Satisfactory sealing can usually be obtained with such seals because of their close running clearance. Centrifugal force may also assist in preventing internal contamination and leakage of the lubricant.

(1) *Oil Groove Seals*

Oil groove seals function by the use of a small gap between the shaft and housing cover in combination with multiple grooves in the housing cover and/or shaft surface (Fig. 13.5 (a), (b)). Since the use of oil grooves alone is not completely effective except at low speeds, a flinger or labyrinth seal is often combined with an oil groove seal (Fig. 13.5 (c)). The entry of dust can be impeded by packing a grease with a consistency of about 200 (NLGI Grade 4) into the grooves of the shaft and/or housing.
The smaller the gap between the shaft and housing, the greater the sealing effect; however, the shaft and housing must not come in contact while running. The recommended gaps are given in Table 13.4.
The recommended groove width is approximately 3 to 5 mm, with a depth of about 4 to 5 mm. When sealing using grooves only, there should be three or more grooves.

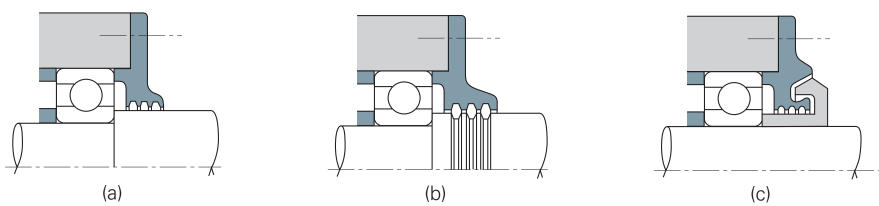


Fig. 13.5 Example Oil Grooves

(2) *Flinger (Slinger) Seals*

A flinger is designed to force water and dust away by centrifugal force acting on any contaminants on the shaft. Sealing mechanisms with flingers inside the housing, as shown in Fig. 13.6 (a) and (b), are mainly intended to prevent oil leakage and are used in environments with relatively little dust. Dust and moisture cannot enter due to the centrifugal force of flingers as shown in Fig. 13.6 (c) and (d).

Table 13. 4 Gaps Between Shafts and Housings for Oil-Groove Type Seals

Units : mm

Nominal Shaft Diameter	Radial Gap
Under 50	0.25 to 0.4
50-200	0.5 to 1.5

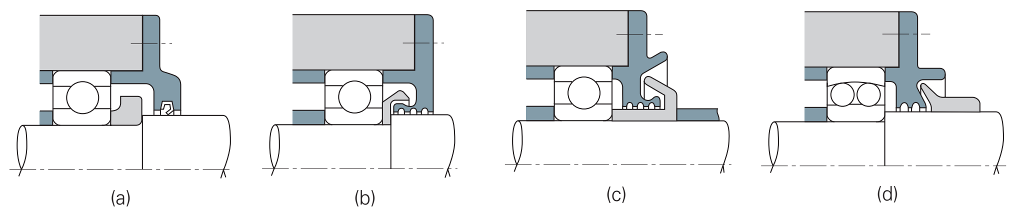


Fig. 13.6 Example Flinger Configurations

(3) *Labyrinth Seals*

Labyrinth seals are formed by interdigitated segments attached to the shaft and housing that are separated by a very small gap. They are particularly suitable for preventing oil leakage from the shaft at high speeds.
The type shown in Fig. 13.7 (a) is widely used because of its ease of assembly, but those shown in Fig. 13.7 (b) and (c) have better seal effectiveness.
Normal radial and axial labyrinth seal gaps are shown in Table 13.5.

Table 13. 5 Labyrinth Seal Gaps

Units : mm

Nominal Shaft Diameter	Labyrinth Gaps	
	Radial Gap	Axiall Gap
Under 50	0.25 to 0.4	1 to 2
50-200	0.5 to 1.5	2 to 5

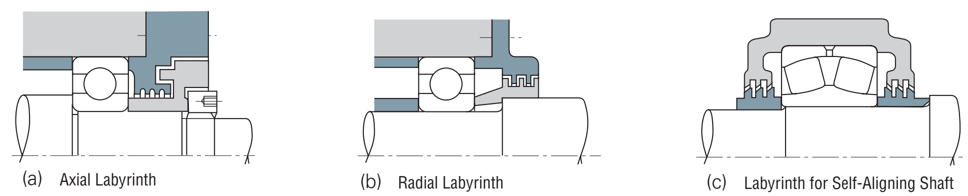


Fig. 13.7 Example Labyrinth Designs

13.3.2 Contact Seals

Contact seals function by physical contact between the shaft and seal, which may be made of synthetic rubber, synthetic resin, felt, etc. Oil seals with synthetic rubber lips are most frequently used.

(1) *Oil Seals*

Many types of oil seals are used to prevent lubricant from leaking while also preventing dust, water, and other foreign matter from entry (Figs. 13.8 and 13.9)

In Japan, such oil seals are standardized (refer to JIS B 2402) by size and type. Since many oil seals are equipped with circumferential springs to maintain adequate contact force, oil seals can follow the non-uniform rotational movement of a shaft to some degree.

Synthetic rubber seal lip materials are often used, including nitrile, acrylate, silicone, fluorine, and tetrafluoride ethylene. The maximum allowable operating temperature for each material increases in this same order.

Synthetic rubber oil seals may cause troubles such as overheating, wear, and seizure unless there is an oil film between the seal lip and shaft. Therefore, some lubricant should be applied to the seal lip when the seals are installed. Furthermore, the lubricant inside the housing should spread slightly between the sliding surfaces. However, please be aware that ester-based grease will cause acrylic rubber material to swell while low aniline point mineral oil, silicone-based grease, and silicon-based oil will cause silicone-based material to swell. Moreover, urea-based grease will cause fluorine-based material to deteriorate.

The permissible circumferential speed for oil seals varies depending on type, finish of the shaft surface, liquid to be sealed, temperature, shaft eccentricity, etc. The temperature range for oil seals is restricted by the lip material. Approximate circumferential surface speeds and permitted temperatures under favorable conditions are listed in Table 13.6.

When oil seals are used at high circumferential surface speed or under high internal pressure, the contact surface of the shaft must be smoothly finished, and shaft eccentricity should be under 0.02 to 0.05 mm.

The hardness of the shaft's contact surface should be over HRC40 by heat treatment or hard-chrome plating in order to gain abrasion resistance. If possible, a hardness over HRC 55 is recommended.

The approximate level of contact surface finish required for several shaft circumferential surface speeds is given in Table 13.7.

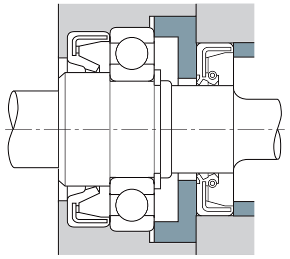


Fig. 13.8 Example of Application of Oil Seal (1)

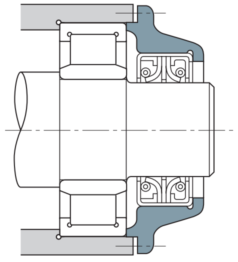


Fig. 13.9 Example of Application of Oil Seal (2)

Table 13. 6 Permissible Circumferential Surface Speeds and Temperature Ranges for Oil Seals

Seal Materials		Permissible Circumferential Speeds (m/sec)	Operating Temperature Range (℃) ([1])
Synthetic Rubber	Nitrile Rubber	Under 16	−25 to +100
	Acrylic Rubber	Under 25	−15 to +130
	Silicone Rubber	Under 32	−70 to +200
	Fluorine-containes Rubber	Under 32	−30 to +200
Tetrafluoride Ethylene Resin		Under 15	−50 to +220

Note ([1]) The upper limit of the temperature range may be raised about 20 ℃ for operation at short intervais.

Table 13. 7 Shaft Circumferential Surface Speeds and Finish of Contact Surfaces

Circumferential Surface Speeds (m/s)	Surface Finish R_a (μm)
Under 5	0.8
5 to 10	0.4
Over 10	0.2

(2) *Felt Seals*

Felt seals are one of the simplest and most common seals used in transmission shafts and other applications.

However, since oil permeation and leakage are unavoidable if oil is used, this type of seal is used only for grease lubrication, primarily to prevent dust and other foreign matter from entry. Felt seals are not suitable for circumferential surface speeds exceeding 4m/sec; therefore, they should be replaced with synthetic rubber seals if the application allows.