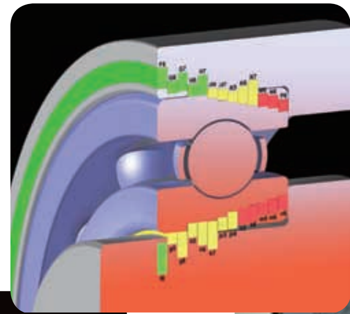
TECHNICAL INFORMATION



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 two rings, rolling elements, and a cage, and 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 they are further segregated by differences in their design or specific purpose.
The most common bearing types and name of bearing parts 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 the structure surrounding rolling bearings is simple.
(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).

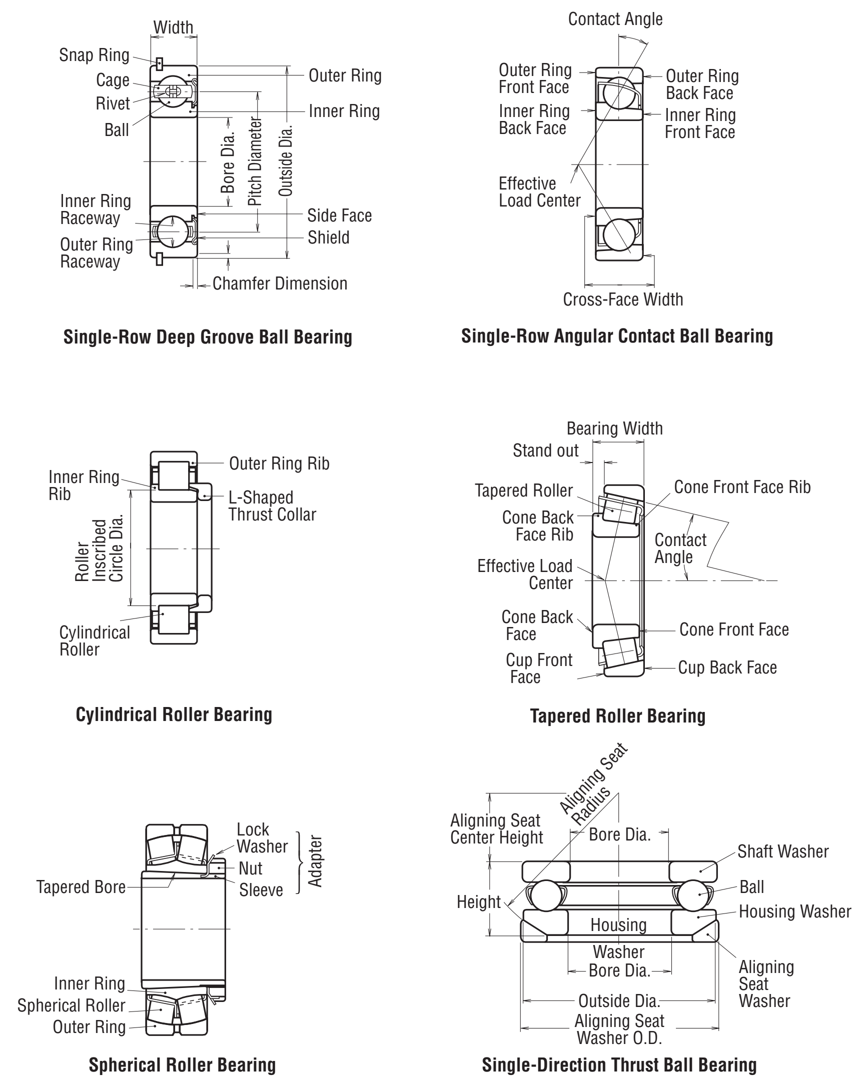


Fig. 1.1 Name of Bearing Parts

TYPES AND FEATURES OF ROLLING BEARINGS

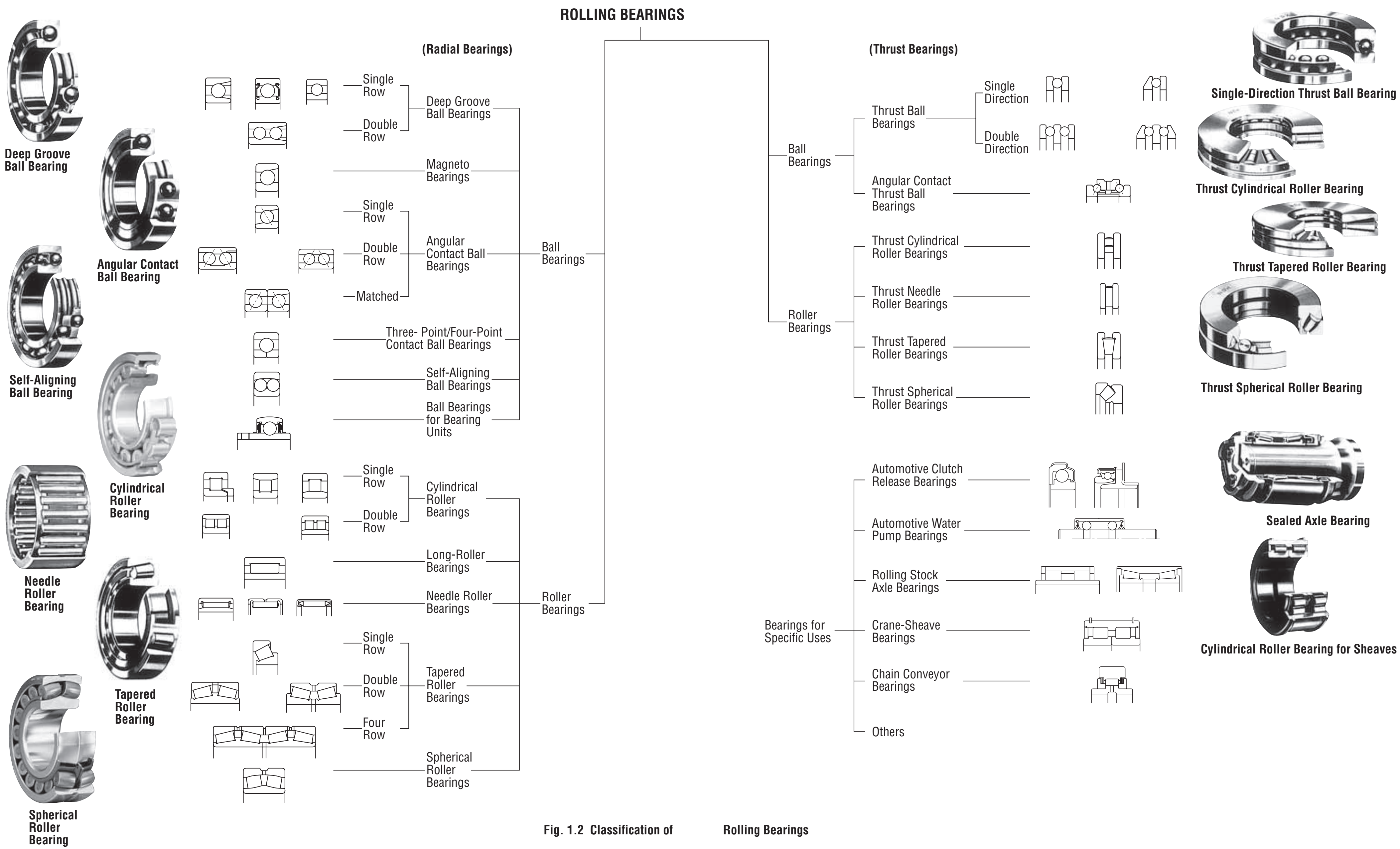
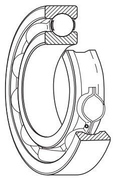


Fig. 1.2 Classification of Rolling Bearings

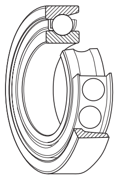
TYPES AND FEATURES OF ROLLING BEARINGS

Single-Row Deep Groove Ball Bearings



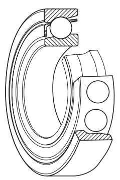
Single-row deep groove ball bearings are the most common type of rolling bearings. Their use is very widespread. The raceway grooves on both the inner and outer rings have circular arcs of slightly larger radius than that of the balls. In addition to radial loads, axial loads can be imposed in either direction. Because of their low torque, they are highly suitable for applications where high speeds and low power loss are required.
In addition to open type bearings, these bearings often have steel shields or rubber seals installed on one or both sides and are prelubricated with grease. Also, snap rings are sometimes used on the periphery. As to cages, pressed steel ones are the most common.

Magneto Bearings



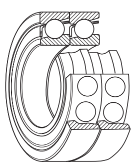
The inner groove of magneto bearings is a little shallower than that of deep groove bearings. Since the outer ring has a shoulder on only one side, the outer ring may be removed. This is often advantageous for mounting. In general, two such bearings are used in duplex pairs. Magneto bearings are small 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



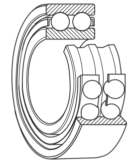
Individual bearings of this type are capable of taking radial loads and also axial loads in one direction. Four contact angles of 15°, 25°, 30°, and 40° are available. The larger the contact angle, the higher the axial load capacity. For high speed operation, however, the smaller contact angles are preferred.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

Duplex Bearings



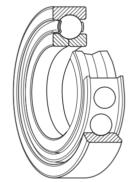
A combination of two radial bearings is called a duplex pair. Usually, they are formed using angular contact ball bearings or tapered roller bearings. Possible combinations include face-to-face, which have the outer ring faces together (type DF), back-to-back (type DB), or both front faces in the same direction (type DT). DF and DB duplex bearings are capable of taking radial loads and axial loads in both direction. Type DT is used when there is a strong axial load in one direction and it is necessary to impose the load equally on each bearing.

Double-Row Angular Contact Ball Bearings



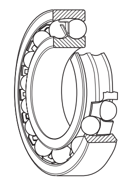
Double-row angular contact ball bearings are basically two single-row angular contact ball bearings mounted back-to-back except that they have only one inner ring and one outer ring, each having raceways. They can take axial loads in both direction.

Four-Point Contact Ball Bearings



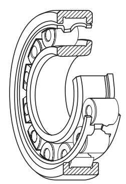
The inner and outer rings of four-point contact ball bearings are separable because the inner ring is split in a radial plane. They can take axial loads from either direction. The balls have a contact angle of 35° with each ring. Just one bearing of this type can replace a combination of face-to-face 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.

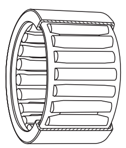
Cylindrical Roller Bearings



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.
There are different types designated NU, NJ, NUP, N, NF for single-row bearings, and NNU, NN for double-row bearings 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 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 rings 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 also used.

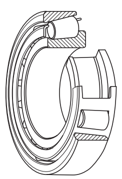
TYPES AND FEATURES OF ROLLING BEARINGS

Needle Roller Bearings



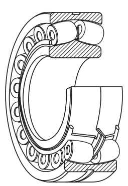
Needle roller bearings contain many slender rollers with a length 3 to 10 times their diameter. As a result, the ratio of the bearing outside diameter to the inscribed circle diameter is small, and they have a rather high radial load capacity.
There are numerous types available, and many have no inner rings. 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 are without cages.

Tapered Roller Bearings



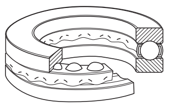
Bearings of this type use conical rollers guided by a back-face rib on the cone. These bearings are capable of taking high radial loads and also axial loads in one direction. In the HR series, the rollers are increased in both size and number giving it an even higher load capacity.
They are generally 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 cones or cups of the two opposed bearings. Since they are separable, the cone assemblies and cups can be mounted independently.
Depending upon the contact angle, tapered roller bearings are divided into three types called the 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

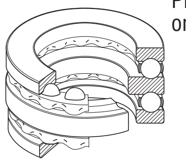


These bearings have barrel-shaped rollers between the inner ring, which has two raceways, and the outer ring which has one spherical raceway. Since the center of curvature of the outer ring raceway surface coincides with the bearing axis, they are self-aligning in a manner similar to that of self-aligning ball bearings. Therefore, if there is deflection of the shaft or housing or misalignment of their axes, it is automatically corrected so 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

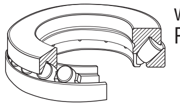


Double-Direction Thrust Ball Bearings



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 that attached to the housing is called the housing washer(or outer ring).
In double-direction thrust ball bearings, there are three rings with the middle one (center ring) being fixed to the shaft.
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 the smaller bearings and machined cages in the larger ones.

Spherical Thrust 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 in 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.

TYPES AND FEATURES OF ROLLING BEARINGS

Table 1. 1 Types and Characteristics of Rolling Bearings

Features \ Bearing Types		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	◯	○	⊙	⊙	⊙	○	◯	⊙	◎	⊙
	Axial Loads	↔ ◯	← ○	← ⊙	↔ ⊙	↔ ⊙	↔ ⊙	↔ ○	×	×	← ◯
	Combined Loads	◯	○	⊙	⊙	⊙	◯	○	×	×	◯
High Speeds		◎	⊙	◎	◯	⊙	⊙	⊙	◎	⊙	⊙
High Accuracy		◎		◎		◎	⊙		◎	◎	
Low Noise and Torque		◎							⊙		
Rigidity						⊙			⊙	◎	⊙
Angular Misalignment		⊙	○	○	○	○	○	◎	◯	○	◯
Self-Aligning Capability								☆			
Ring Separability			☆				☆		☆	☆	☆
Fixed-End Bearing		☆			☆	☆	☆	☆			
Free-End Bearing		★			★	★	★	★	☆	☆	
Tapered Bore in Inner Ring								☆		☆	
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 use on free-end is not possible.	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 Types		Cylindrical Roller Bearings with Thrust Collars	Needle Roller Bearings	Tapered Roller Bearings	Double-and Multiple-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	⊙	⊙	⊙	◎	◎	×	×	×	×	×	○	—
	Axial Loads	↔ ◯	×	← ⊙	↔ ⊙	↔ ◯	← ⊙	← ⊙	↔ ⊙	← ◎	← ◎	← ◎	—
	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
Tapered Bore in Inner Ring						☆							A150 B008 B012
Remarks		Including NUP type		Two bearings are usually mounted in opposition. Clearance adjustment is necessary.	KH, KV types are also available but use on free-end is impossible.					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 ○ Poor × Impossible ← One direction only ↔ Two directions

☆ Applicable ★ Applicable, but it is necessary to allow shaft contraction/elongation at fitting surfaces of bearings.

1.3 Contact Angle and Bearing Types

The 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 loading direction on the bearing.

Radial bearing α: Less than 45°
(A primarily radial load is applied.)

Thrust bearing α: Over 45°
(A primarily axial load is applied.)

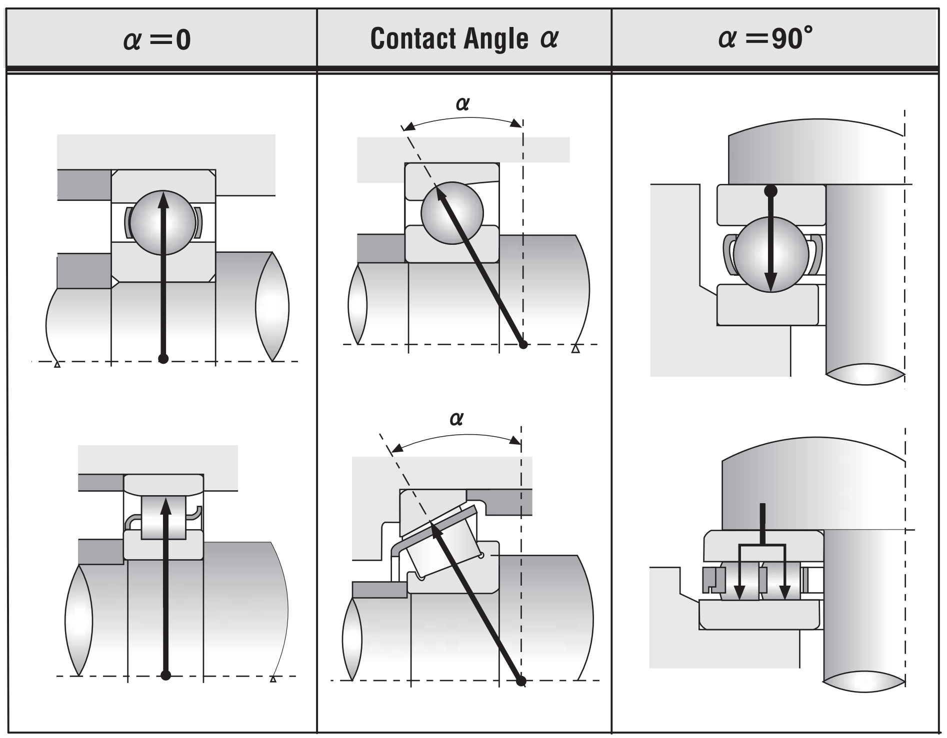


Fig. 1.3 Contact Angle α

1.4 Types of Load on Bearings

An example deep groove ball bearing is shown.
Figure 1.4 shows the types of the load applied to a rolling bearing.

(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.

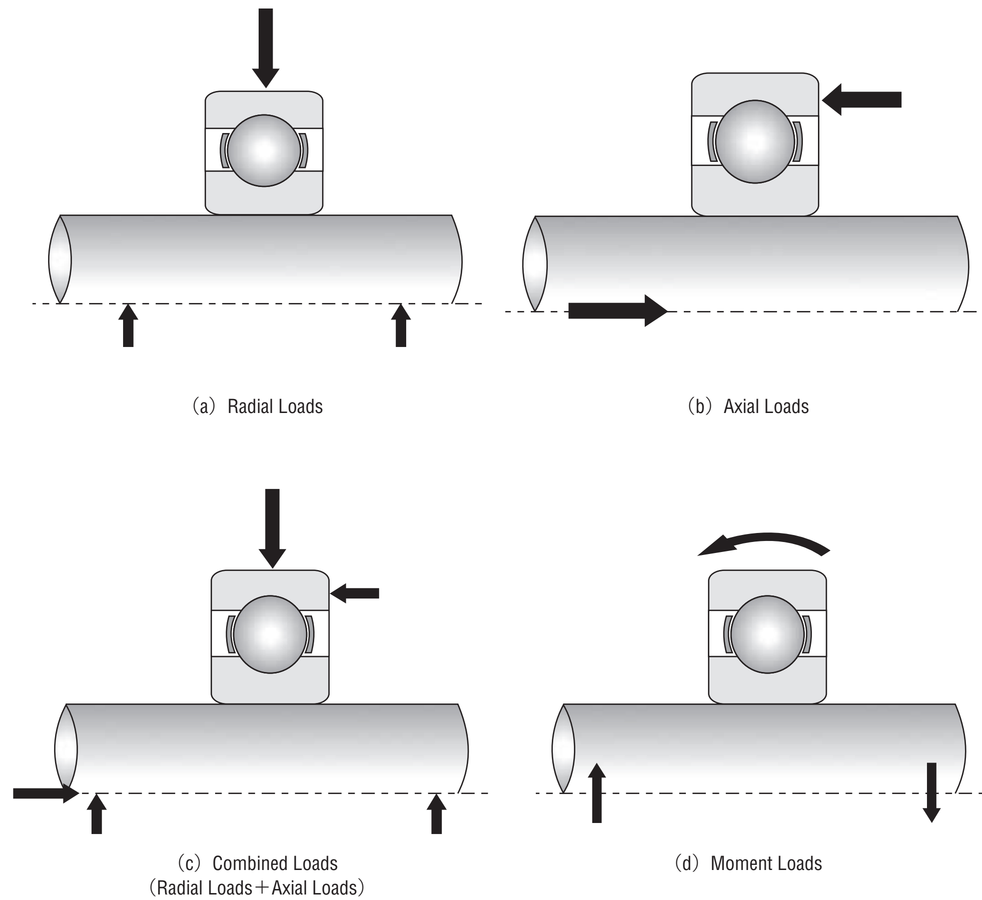


Fig. 1.4 Types of Load

2. SELECTION OF BEARING TYPES

2. SELECTION OF BEARING TYPES

2.1 Bearing Selection Procedure

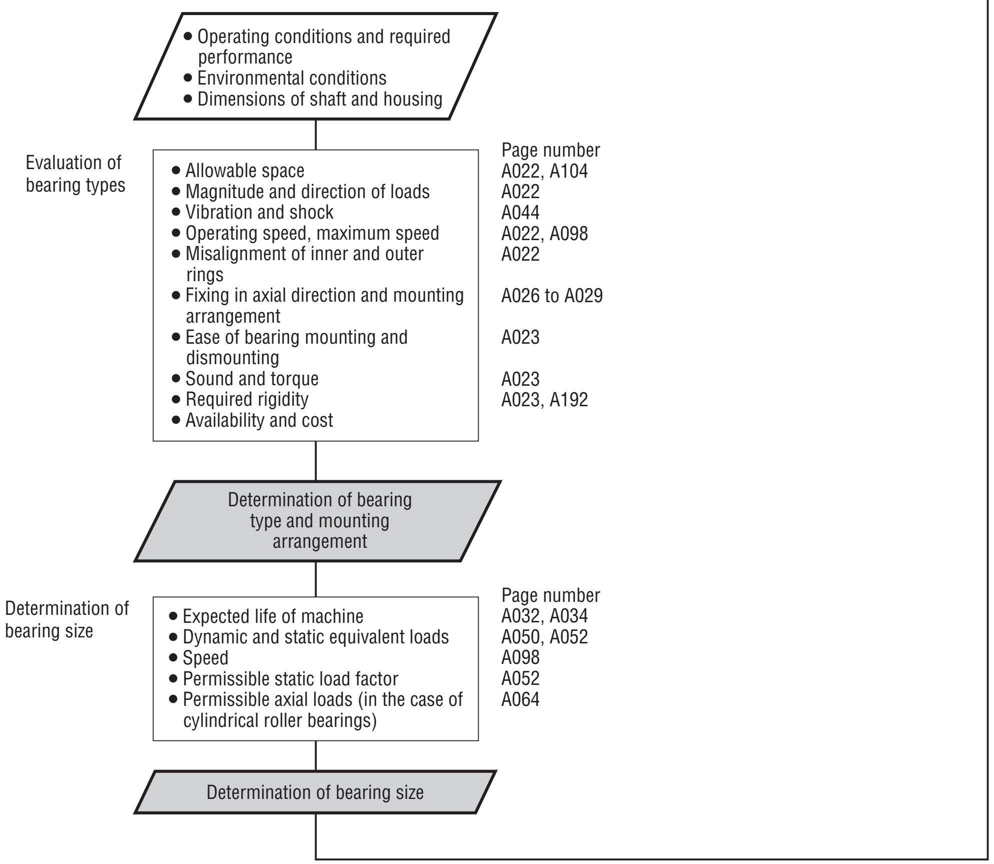
The number of applications for rolling bearings is almost countless and the operating conditions and environments also vary greatly. In addition, the diversity of operating conditions and bearing requirements continue to grow with the rapid advancement of technology. Therefore, it is necessary to study bearings carefully from many angles to select the best one from the thousands of types and sizes available.

Usually, a bearing type is provisionally chosen considering the 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, it is necessary 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 your 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 of the bearing selection procedure.



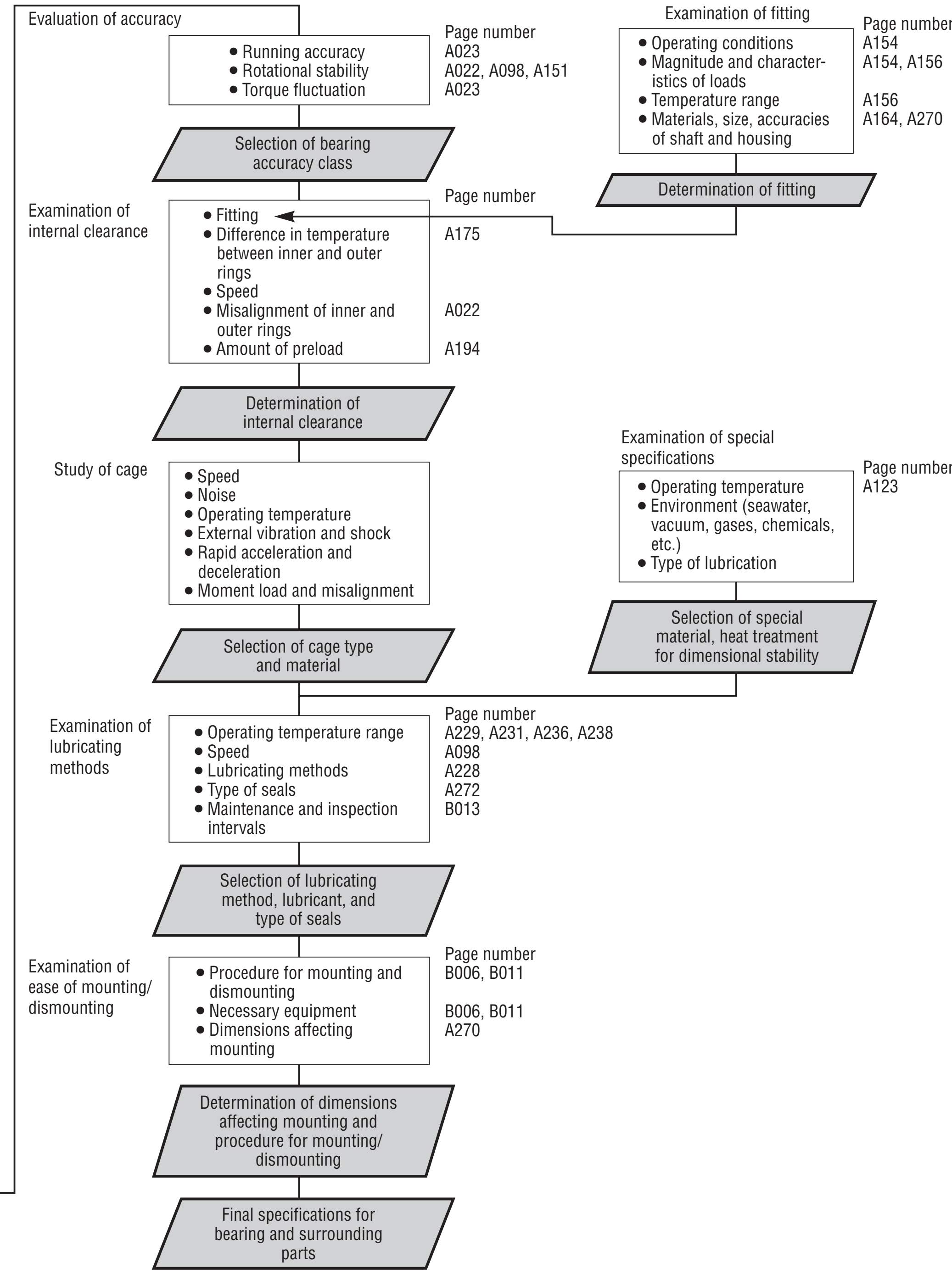


Fig. 2.1 Flow Chart for Selection of Rolling Bearings

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 its bore size. For rolling bearings, there are numerous standardized dimension series and types, and the selection of the optimum bearing from among them is necessary. 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 the radial load capacity (see Page A032) in a manner that depends on the bearing design as shown in Fig. 2.3. This figure makes it clear 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 depending, not only the type of bearing, but also its size, type of cage, loads, lubricating method, heat dissipation, etc. Assuming the common oil bath lubrication method, the bearing types are roughly ranked from higher speed to lower as shown in Fig. 2.4.

2.5 Misalignment of Inner/Outer Rings and Bearing Types

Because of deflection of a shaft caused by applied loads, dimensional error 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 a small angle less than 0.0012 radian (4').

When a large misalignment is expected, bearings having a 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).

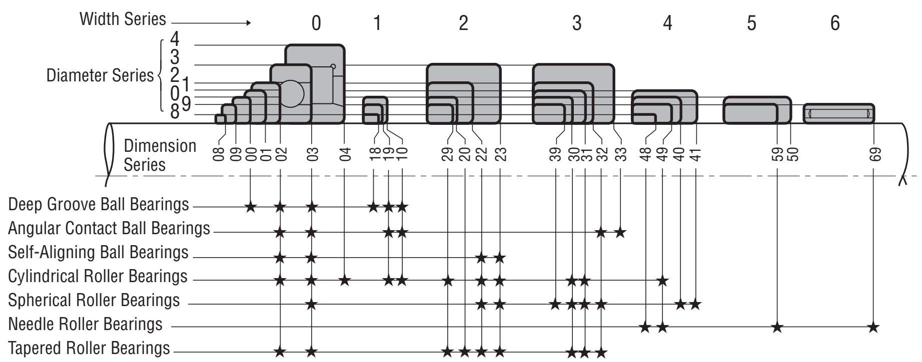
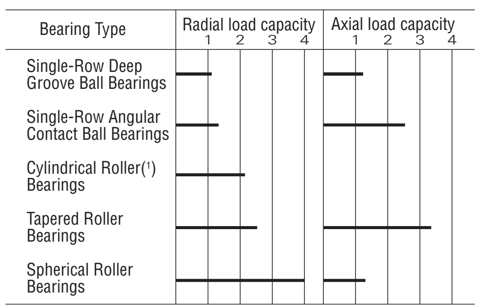


Fig. 2.2 Dimension Series of Radial Bearings



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

Fig. 2.3 Relative Load Capacities of Various Bearing Types

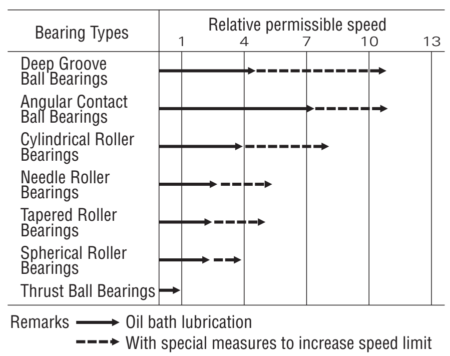


Fig. 2.4 Relative Permissible Speeds of Various Bearing Types

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

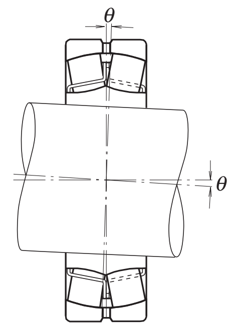


Fig. 2.5 Permissible Misalignment of Spherical Roller Bearings

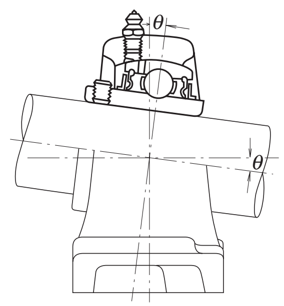


Fig. 2.6 Permissible Misalignment of Ball Bearing Units

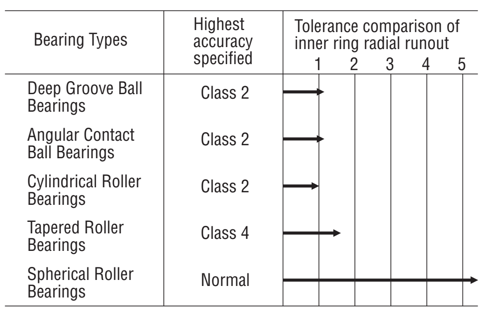


Fig. 2.7 Relative Inner Ring Radial Runout of Highest Accuracy Class for Various Bearing Types

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. For the main spindles of machine tools, it is necessary to have high rigidity of the bearings together with the rest of the spindle. Consequently, since roller bearings are deformed less by load, they are more often selected than ball bearings. When extra high rigidity is required, bearings are given a preload, which means that 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 particularly, the noise level is sometimes specified depending on their purpose. For high precision miniature ball bearings, the starting torque is specified. Deep groove ball bearings are recommended for applications in which low noise and torque are required, such as motors and instruments.

2.8 Running Accuracy and Bearing Types

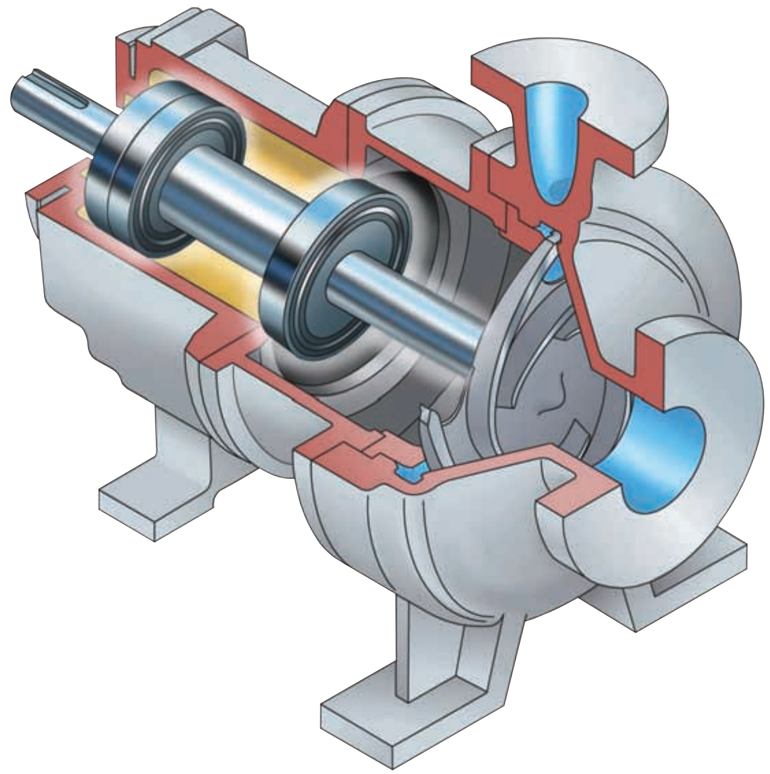
For the main spindles of machine tools that require high running accuracy or high speed applications like superchargers, high precision bearings of Class 5, 4 or 2 are usually used.

The running accuracy of rolling bearings is specified in various ways, and the 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.

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

2.9 Mounting and Dismounting of Various Bearing Types

Separable types of bearings like cylindrical roller bearings, needle roller bearings and tapered roller bearings are convenient for mounting and dismounting. For machines in which bearings are mounted and dismounted rather often for periodic inspection, these types of bearings are recommended. Also, self-aligning ball bearings and spherical roller bearings (small ones) with tapered bores can be mounted and dismounted relatively easily using sleeves.



3. SELECTION OF BEARING ARRANGEMENT

3. SELECTION OF BEARING ARRANGEMENT

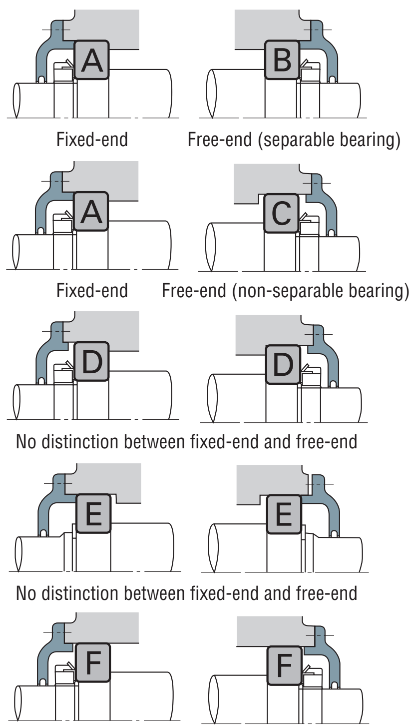
In general, shafts are supported by only two bearings. When considering the bearing mounting arrangement, the following items must be investigated:

(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 the loads at their proper positions and to transmit them.

3.1 Fixed-End and Free-End Bearings

Among the bearings on a shaft, only one can be a "fixed-end" bearing that is used to fix the shaft axially. For this fixed-end bearing, a type which can carry both radial and axial loads must be selected.
Bearings other than the fixed-end one 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 are 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. When these types are used, mounting and dismounting are also easier.
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 fitting 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 it is done at the bore.
([2]) For each type, two bearings are used in opposition.

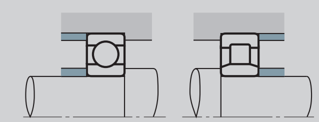
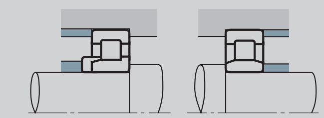
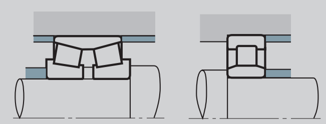
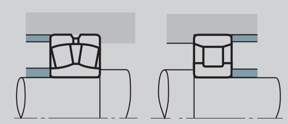
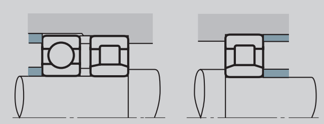
Fig. 3.1 Bearing Mounting Arrangements and Bearing Types

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 of 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.

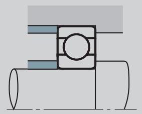
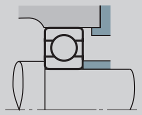
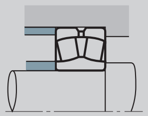
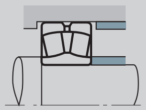
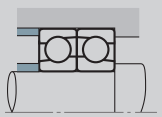
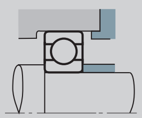
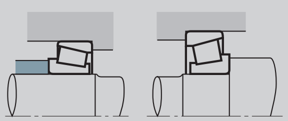
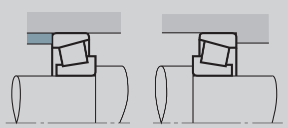
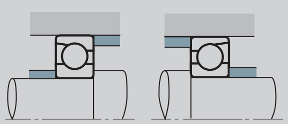
Table 3. 1 Representative Bearing Mounting Arrangements and Application Examples

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 the mounting error is small, this is suitable for high speeds.	Medium size electric motors, blowers
		◯This can withstand heavy loads and shock loads and can take some axial load. ◯Every type of cylindrical roller bearing is separable. This is helpful when interference is necessary for both the inner and outer rings.	Traction motors for rolling stock
		◯This is used when loads are relatively heavy. ◯For maximum rigidity of the fixed-end bearing, it is a back-to-back type. ◯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 is also 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 is suitable for high speeds and heavy radial loads. Moderate axial loads can also be applied. ◯It is necessary to provide some clearance 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

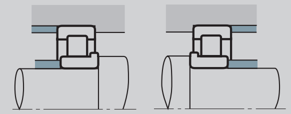
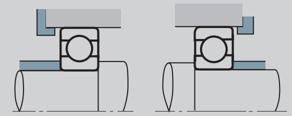
Continued on next page

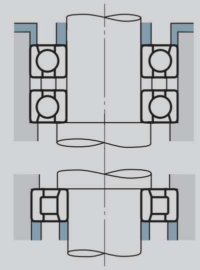
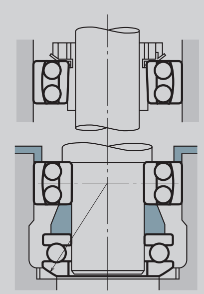
SELECTION OF BEARING ARRANGEMENT

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 moderate axial loads also.	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 a 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, attention must be paid to the 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 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

Continued on next page

When there is no distinction between fixed-end and free-end	Remarks	Application Examples
 NJ + NJ mounting	◯This can withstand heavy loads and shock loads. ◯It can be used if interference is necessary for both the inner and outer rings. ◯Care must be taken so the axial clearance doesn't become too small during running. ◯NF type + NF type mounting is also possible.	Final reduction gears of construction machines
	◯Sometimes a spring is used at the side of the outer ring of one bearing.	Small electric motors, small speed reducers, small pumps

Vertical arrangements	Remarks	Application Examples
	◯Matched angular contact ball bearings are on the fixed end. ◯Cylindrical roller bearing is 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 various functions required of rolling bearings vary according to the bearing application. These functions must be performed for a prolonged period. 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 the broad sense of the term, is the period during which bearings continue to operate and to satisfy their required functions. This bearing life may be defined as noise life, abrasion life, grease life, or rolling fatigue life, depending on which one 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 of the seals or the 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 "flaking". Rolling fatigue life is represented by the total number of revolutions at which time the bearing surface will start flaking due to stress. This is called fatigue life. As shown in Fig. 4.2, even for seemingly identical bearings, which are of the same type, size, and material and 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-prelubricated 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 Load Rating and Fatigue Life

4.2.1 Basic Load Rating

The basic 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 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. The load ratings are listed under C_r for radial bearings and C_a for thrust bearings in the dimension tables.

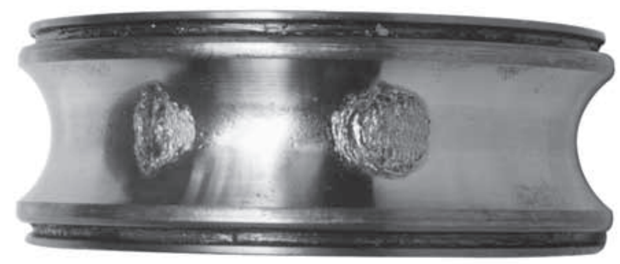


Fig. 4.1 Example of Flaking

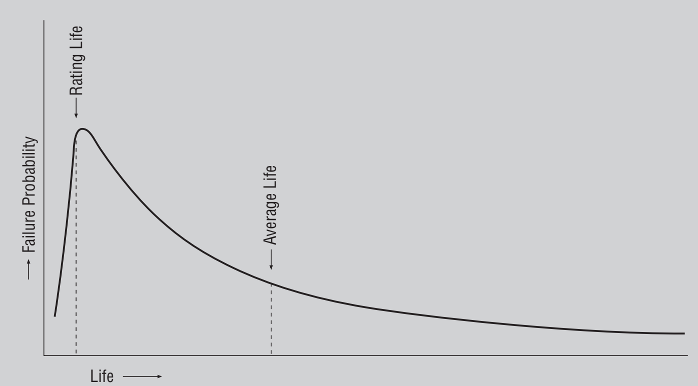


Fig. 4.2 Failure Probability and Bearing Life

4.2.2 Machinery in which Bearings are Used and Projected Life

It is not advisable to select bearings with unnecessarily high load ratings, for such bearings may be too large and uneconomical. In addition, the 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 are to be mounted should also be considered. Bearings are used in a wide range of applications and the design life varies with specific applications and operating conditions. Table 4.1 gives an empirical fatigue life factor derived from customary operating experience for various machines. Also refer to Table 4.2.

Table 4.1 Fatigue Life Factor f_h for Various Bearing Applications

Operating Periods	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 like vacuum cleaners and washing machines ・Hand 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 Parameters	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 A30)
C : Basic load rating (N), {kgf}
For radial bearings, C is written C_r
For thrust bearings, C is written C_a

In the case of bearings that run at a constant speed, it is convenient to express the fatigue life in terms of hours. 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, determine a fatigue life factor f_h appropriate for the projected life of the machine and then calculate the basic load rating C by means of the following equation.

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

A bearing which 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 temperature, 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 load rating should be adjusted for the higher temperature using the following equation:

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

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

If large bearings are used at higher than 120°C, they must be given special dimensional stability heat treatment to prevent excessive dimensional changes. The basic load rating of bearings given such special dimensional stability heat treatment may become lower than the basic load rating 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

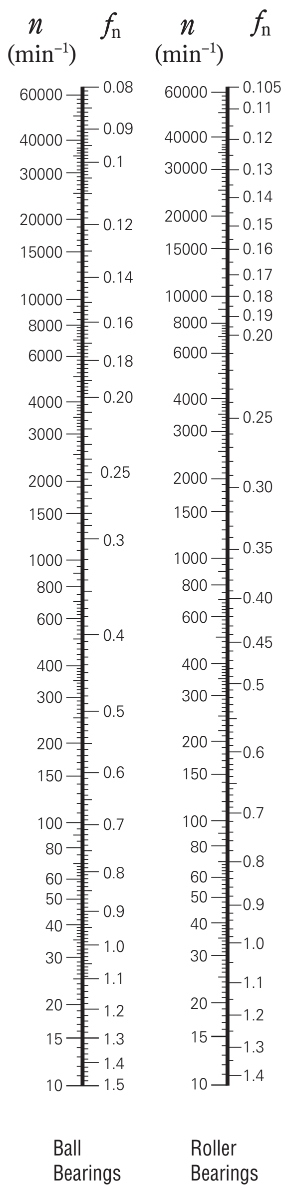


Fig. 4.3 Bearing Speed and Speed Factor

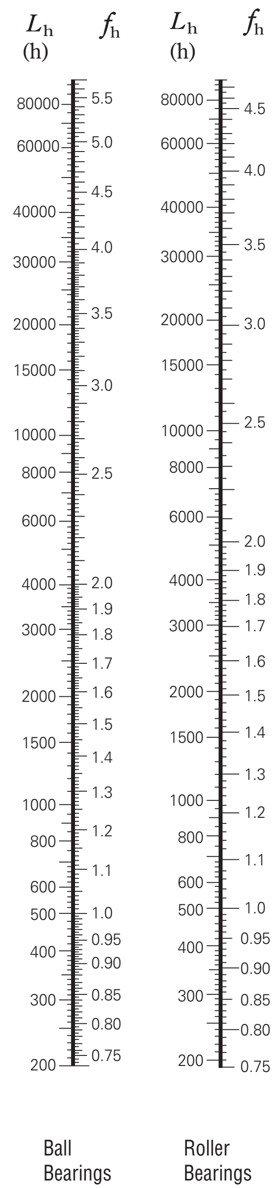


Fig. 4.4 Fatigue Life Factor and Fatigue Life

4.2.5 Correction of Basic Rating Life

As described previously, the basic equations for calculating the 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**)

The 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 the fatigue life. In addition, the developent 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 the results of tests by NSK show that life is greatly improved when 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), it is sufficient to assume that 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 the bearing temperature is high.
- When the 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 for 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 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. Generally, however, the machine 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 revolution or hours to which 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 below equation 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 with ease by using Fig. 4.5.

Take the value L_1 of 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 the intersection with the L scale. In this way, the value L_A of

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

is determined. Take this value L_A on the L_1 scale and the value L_3 on the L_2 scale, connect them with a straight line, and read an intersection with the L scale. In this way, the value L of

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

can be determined.

Example

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

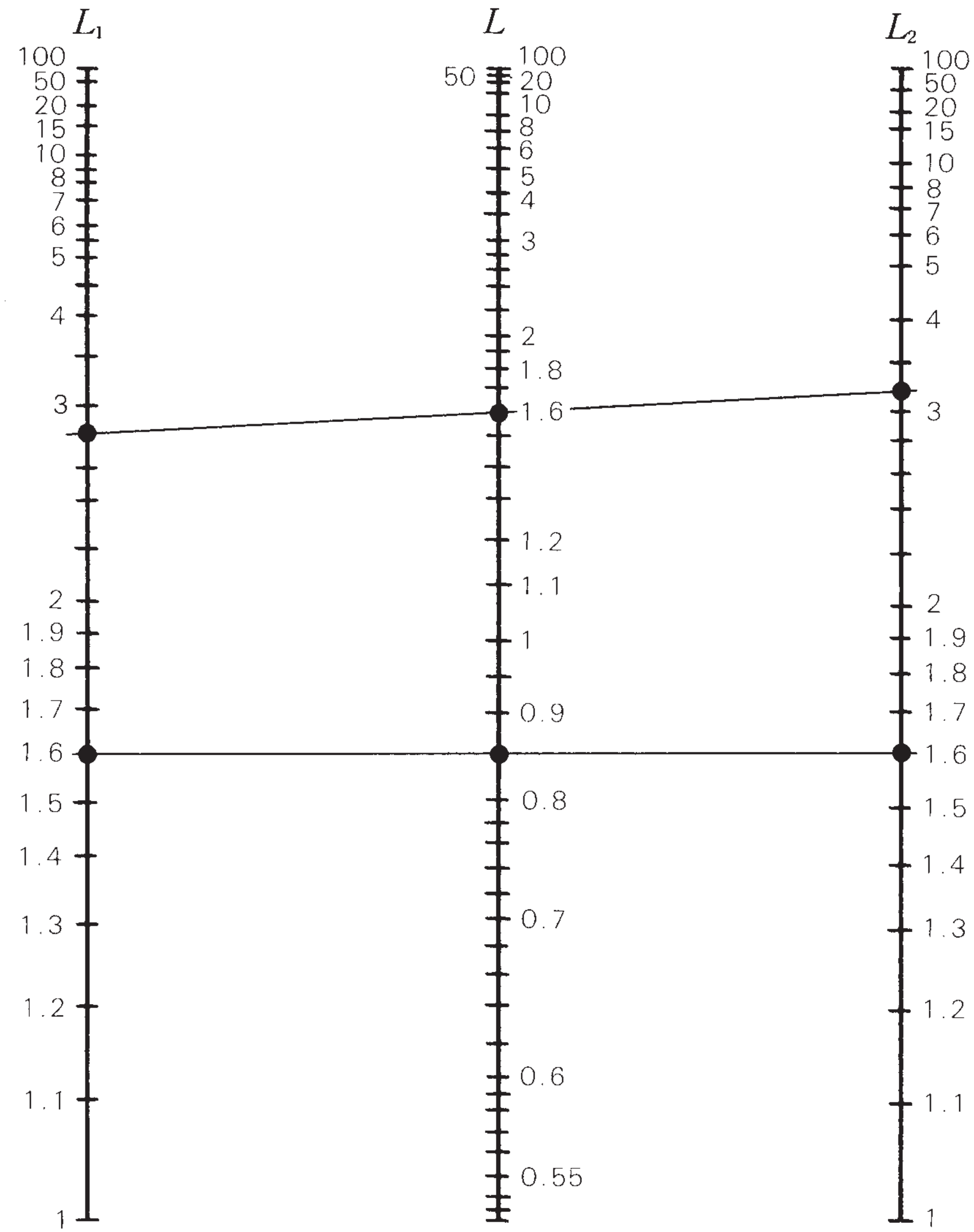


Fig. 4.5 Chart for Life Calculation

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 can now have a longer rolling fatigue life in a cleaner environment, 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, based on the theories of G. Lundberg and A. Palmgren (L-P theory, hereafter) addresses only sub-surface originated flaking. This is the phenomenon in which cracks initially occur due to dynamic shear stress immediately below the rolling surface then progressively reach the surface in the form of flaking.

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

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

NSK New Life Calculation Formula

(1) Sub-surface originated flaking
A pre-condition of sub-surface originated flaking of 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 applied on the ordinate and the abscissa, respectively.
In the figure, line L_{10} theoretical is the theoretical line obtained using the conventional life calculation formula. As maximum surface contact pressure decreases, the actual life line separates from the line created by using conventional theoretical calculation and moves towards longer life. This separation suggests the presence of fatigue load limit P_u below which no rolling fatigue occurs. This is better illustrated in Fig. 4.7.

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

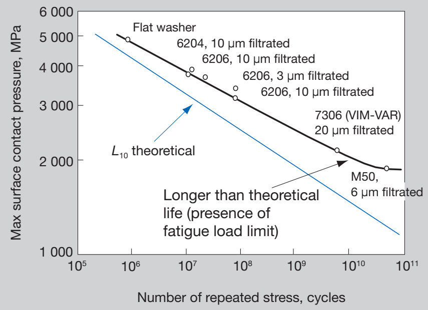


Fig. 4.6 Life Test Result under Clean Lubrication Condition

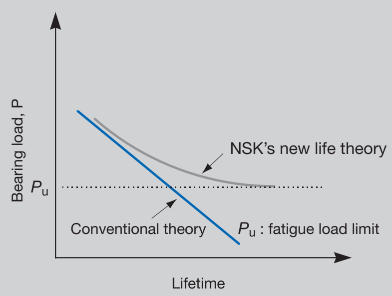


Fig. 4.7 NSK's New Life Theory That Considers Fatigue Limit

(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 the foreign particles are mixed in the lubricant, the particles are pressed onto the raceways by the rolling elements and dents occur on the surfaces of the raceways and rolling elements. Stress concentration occurs at the edges of the dents, generating fine cracks, which over time, propagate into flaking of the raceways and rolling elements.
As shown in Fig. 4.8, the actual life is shorter than conventional calculated life, under conditions of contaminated lubrication at low max surface pressure. The actual life line separates from the line created by theoretical life calculations and moves towards a shorter life. 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 Value 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 bearing for electrical appliances and information technology equipment, etc.	Sealed grease lubricated bearing for electric motors Sealed grease bearing for railway axle boxes and machine tools, etc.	Normal usage Automotive hub unit bearing, etc.	Bearing for automotive transmission; Bearing for industrial gearbox; Bearing for construction machine, etc.	—

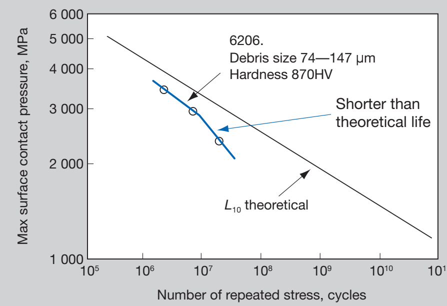


Fig. 4.8 Life Test Result under Contaminated Lubrication Condition

Therefore, the NSK new life calculation formula considers the trend in the results of the life test under conditions of clean environment and at low load zone. Based on these results, the new life equation is a function of $(P\text{-}P_u)/C$, which is affected by specific lubrication conditions identified by the lubrication parameter. Also, it is assumed that effects of different types and shapes of foreign particles are strongly influenced by the bearing load and lubrication conditions present, and that such a relationship can be expressed as a function of the load parameter. This relationship of the new life calculation formula is defined by $(P\text{-}P_u)/C \cdot 1/a_c$.
Calculation formula for surface originated flaking, based on the above concept, is as follows:

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

(3) Calculation of Contamination Coefficient a_c
The contamination coefficient in terms of lubrication cleanliness is shown in Table 4.5. Test results on ball and roller bearings with grease lubrication and clean filtration show the life as being a number of times longer than that of the contaminated calculation. Yet when the foreign object is harder than Hv350, hardness becomes a factor and a dent appears on the raceway. Fatigue damage from these dents, can progress to flaking in a short time. Test results on ball and roller bearings under conditions of foreign object contamination show from 1/3 to 1/10 the life when compared with conventionally calculated life.
Based on these test results, the contamination coefficient a_c is classified into five steps for NSK's new life theory.

(4) New life calculation formula L_{able}

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

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

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

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:

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

NSK's new life theory considers the life extending affect of improved material and heat treatment by correcting the contamination factor a_c. The theory also utilizes viscosity ratio K ($K = \nu/\nu_1$ 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 K) the longer the life.

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

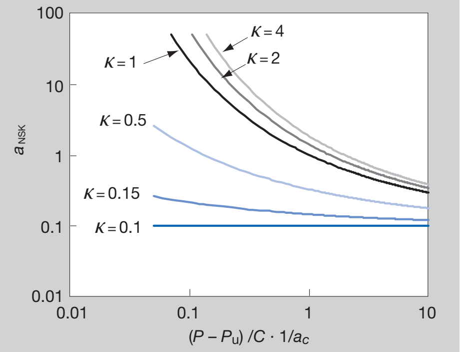


Fig. 4.9 New Life Calculation Diagram for Ball Bearings

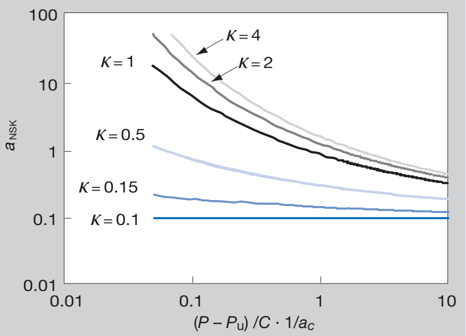


Fig. 4.10 New Life Calculation Diagram for Roller Bearings

To Access the NSK Calculation Tools

Visit our website at http://www.nsk.com

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 revolving elements themselves, the transmission power of gears and belting, the load produced by the operation of the machine in which the bearings are used, etc. These loads can be theoretically calculated, but some of them are difficult to estimate. Therefore, it becomes necessary to correct the estimated 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 operation of the machine. 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.

4.3.2 Bearing Loads in Belt or Chain Transmission Applications

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

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

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

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

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

In the case of a chain transmission, the values corresponding to f_b should be 1.25 to 1.5.

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:

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

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

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

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

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 using the gear factor f_g by multiplying the theoretically calculated load by this factor.
The values of f_g should generally be those in Table 4.8. When vibration from other sources accompanies gear operation, the actual load is obtained by multiplying the load factor by this gear factor.

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:

$$F_{CI} = \frac{b}{c}K \quad \cdots\cdots (4.23)$$

$$F_{CII} = \frac{a}{c}K \quad \cdots\cdots (4.24)$$

where F_{CI} : Radial load applied on bearing I (N), {kgf}
F_{CII} : 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, and then, the sum of the vectors may be calculated according to the load direction.

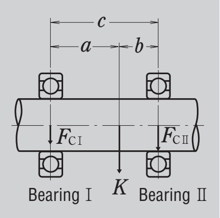


Fig. 4.11 Radial Load Distribution (1)

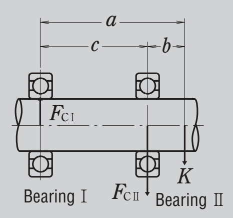


Fig. 4.12 Radial Load Distribution (2)

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

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

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

4.3.5 Average of Fluctuating Load

When the load applied on bearings fluctuates, an average load which 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
⋮
Load F_n : Speed n_n ; Operating time t_n

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

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

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:

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

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

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

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

(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)

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

In the case of Fig. 4.15 (b)

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

(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 $F_R \geqq F_S$

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

b) Where $F_R < F_S$

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

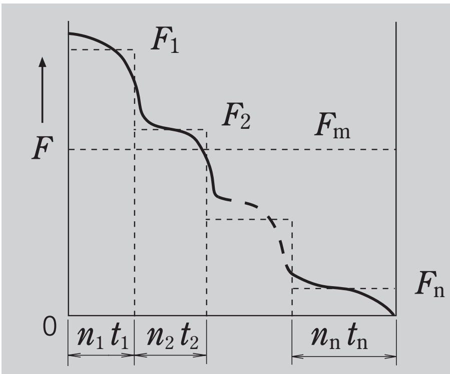


Fig. 4.13 Incremental Load Variation

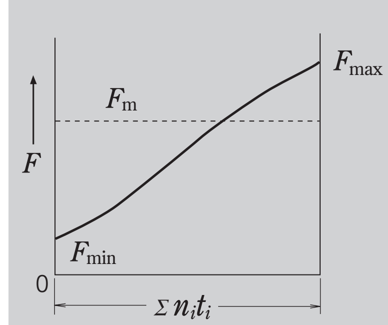


Fig. 4.14 Simple Load Fluctuation

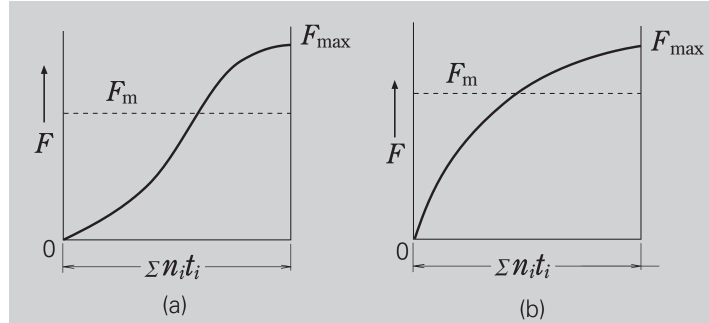


Fig. 4.15 Sinusoidal Load Variation

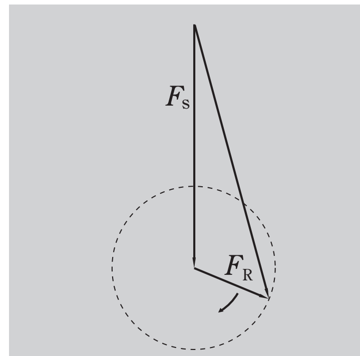


Fig. 4.16 Rotating Load and Stationary Load

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 an unbalanced or a vibration weight, and the stationary load, which is caused by gravity or power transmission, may act simultaneously. The combined mean effective load when the indeterminate load caused by rotating and static loads can be calculated as follows. There are two kinds of combined loads; rotating and stationary which are classified depending on the magnitude of these loads, as shown in Fig. 4.17.

Namely, the combined load becomes a running load with its 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:

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

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

The value F can be approximated by Load Equations (4.33) and (4.34) which vary sinusoidally depending on the magnitude of F_R and F_S, that is, in such a manner that 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 $F_R \ll F_S$.

$$F_R \gg F_S,\ F=F_R-F_S\cos\theta \quad \cdots\cdots \quad (4.33)$$

$$F_R \ll F_S,\ F=F_S-F_R\cos\theta \quad \cdots\cdots \quad (4.34)$$

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

$F_R > F_S$,

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

$F_R < F_S$,

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

Curves of Equations (4.32), (4.33), (4.35), and (4.36) are as shown in Fig. 2.

The mean value F_m of the load varying 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).

$F_m=F_{min}+0.65\ (F_{max}-F_{min})$

$$F_R \geqq F_S,\ F_m=F_R+0.3F_S \quad \cdots\cdots \quad (4.37)$$

$$F_R \leqq F_S,\ F_m=F_S+0.3F_R \quad \cdots\cdots \quad (4.38)$$

$F_m=F_{min}+0.75\ (F_{max}-F_{min})$

$$F_R \geqq F_S,\ F_m=F_R+0.5F_S \quad \cdots\cdots \quad (4.39)$$

$$F_R \leqq F_S,\ F_m=F_S+0.5F_R \quad \cdots\cdots \quad (4.40)$$

Generally, as the value F exists somewhere among Equations (4.37), (4.38), (4.39), and (4.40), the factor 0.3 or 0.5 of the second terms of Equations (4.37) and (4.38) as well as (4.39) and (4.40) is assumed to change linearly along with F_S/F_R or F_R/F_S. Then, 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:

$F_R \geqq F_S$,

$$F_m=F_R+(0.3+0.2\frac{F_S}{F_R})\ F_S$$

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

$F_R \leqq F_S$,

$$F_m=F_S+(0.3+0.2\frac{F_R}{F_S})\ F_R$$

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

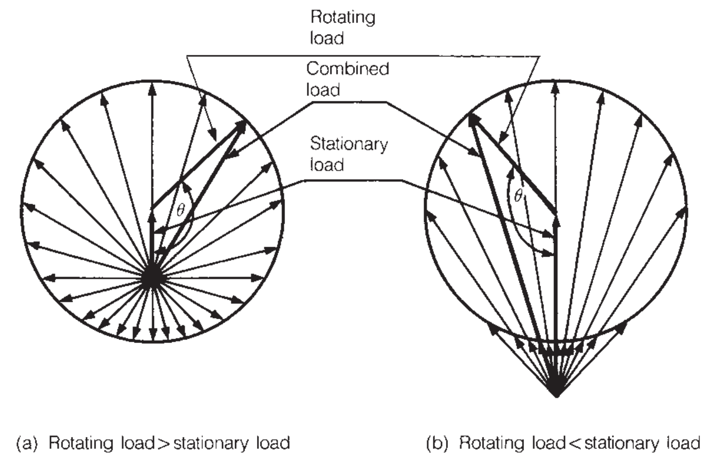


Fig. 4.17 Combined Load of Rotating and Stationary Loads

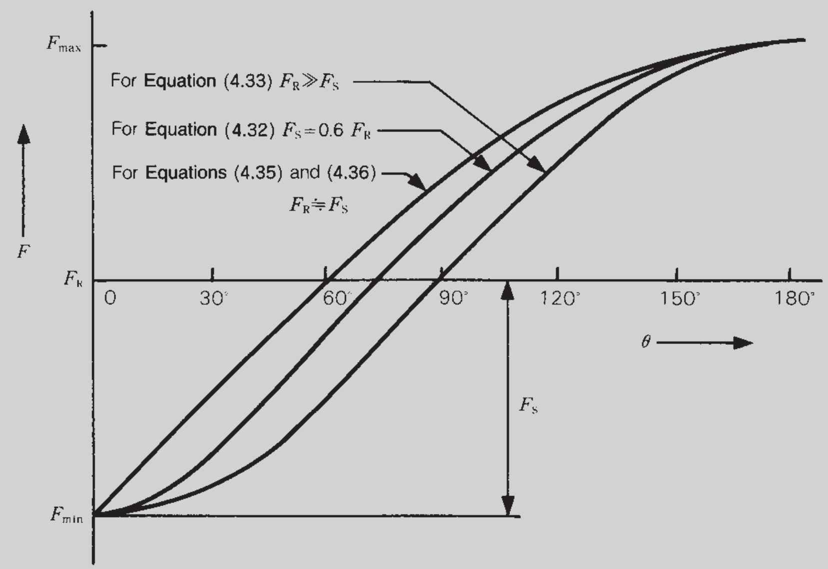


Fig. 4.18 Chart of Combined Loads

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 that has a constant magnitude and passes through the center of the bearing, and will give the same bearing life that the bearing would attain under actual conditions of load and rotation should be estimated. 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:

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

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:

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

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

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 of 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 of 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:

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

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, Y_{II} and the radial load factor is X, then the equivalent loads P_I , P_{II} may be calculated as follows:

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

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

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

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

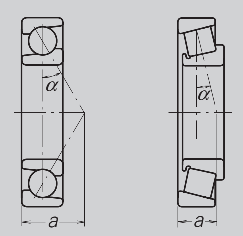


Fig. 4.19 Effective Load Centers

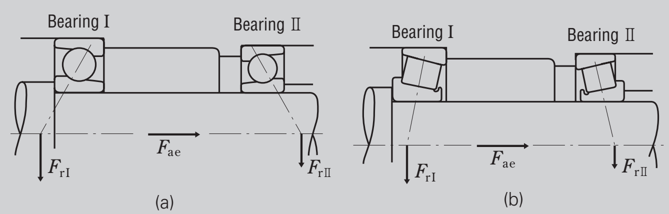


Fig. 4.20 Loads in Opposed Duplex Arrangement

4.5 Static Load Ratings and Static Equivalent Loads

4.5.1 Static Load Ratings

When subjected to an excessive load or a strong shock load, 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. The nonelastic deformation increases in area and depth as the load increases, and when the load exceeds a certain limit, the smooth running of the bearing is impeded.

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

For self-aligning ball bearings	4 600MPa {469kgf/mm^2}
For other ball bearings	4 200MPa {428kgf/mm^2}
For roller bearings	4 000MPa {408kgf/mm^2}

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's diameter. The basic static load rating C_o is written C_{or} for radial bearings and C_{oa} for thrust bearings in the bearing tables.

In addition, following the modification of the criteria for basic static load rating by ISO, the new C_o values for NSK's ball bearings became about 0.8 to 1.3 times the past values and those for roller bearings about 1.5 to 1.9 times. Consequently, the values of permissible static load factor f_s have also changed, so please pay attention to this.

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 under actual conditions, while the bearing is stationary (including very slow rotation or oscillation), in the area of contact between the most heavily stressed rolling element and bearing raceway. 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.

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

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

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

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

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 with $\alpha = 90°$ 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 and also their 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. Conforming to the modification of the static load rating, the values of f_s were revised, especially for bearings for which the values of C_o were increased, please keep this in mind when selecting bearings.

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

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

For spherical thrust roller bearings, the values 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	2
Standard operating conditions	1	1.5

4.6 Examples of Bearing Calculations

(Example1)
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, {255kgf} and speed n =900 min^{-1}.

The basic load rating C_r of **6208** is 29 100N, {2 970kgf} (Bearing Table, Page C024). Since only a radial load is applied, the equivalent load P may be obtained as follows:

$P=F_r=2\ 500\text{N}$, {255kgf}

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 being regularly used, etc., and according to the equation in Table 4.2 or Fig. 4.4 (Page A036), it corresponds approximately to 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 000N, {306kgf}
Speed n =1 900 min^{-1}
Basic rating life L_h≧10 000h

The fatigue life factor f_h of ball bearings with a rating fatigue life longer than 10 000 hours is f_h≧2.72. Because f_n = 0.26, $P=F_r$ = 3 000N. {306kgf}

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

therefore, $C_r\geqq 2.72\times\frac{3\ 000}{0.26}=31\ 380\text{N}$, {3 200kgf}

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

(Example3)
Obtain C_r/P or fatigue life factor f_h when an axial load F_a=1 000N, {102kgf} 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 obtainable, depending on the magnitude of f_0F_a/C_{or}, from the table above the single-row deep groove ball bearing table.
The basic static load rating C_{or} of ball bearing **6208** is 17 900N, {1 820kgf} (Page C024)

$f_0F_a/C_{or}=14.0\times 1\ 000/17\ 900=0.782$

$e\fallingdotseq 0.26$

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

$P=XF_r+YF_a$

$=0.56\times 2\ 500+1.67\times 1\ 000$

$=3\ 070\text{N}$, {313kgf}

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

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

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

(Example 4)
Select a spherical roller bearing of series 231 satisfying the following conditions:
Radial load F_r = 45 000N
Axial load F_a = 8 000N
Speed n = 500min^{-1}
Basic rating life L_h≧30 000h

The value of the fatigue life factor f_h which makes L_h≧30 000h is bigger than 3.45 from Fig. 4.4 (Page A036).

The dynamic equivalent load P of spherical roller bearings is given by:

when $F_a/F_r\leqq e$

$P=XF_r+YX_a=F_r+Y_3F_a$

when $F_a/F_r>e$

$P=XF_r+YF_a=0.67\ F_r+Y_2F_a$

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

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 bearings of series 231:

Therefore, $P=XF_r+YF_a=F_r+Y_3F_a$

$=45\ 000+2.2\times 8\ 000$

$=62\ 600\text{N}$

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, $C_r\geqq 490\ 000\text{N}$
Among spherical roller bearings of series 231 satisfying this value of C_r, the smallest is **23126CE4** (C_r= 505 000N)
Once the bearing is determined, substitude the value of Y_3 in the equation and obtain the value of P.

$P=F_r+Y_3F_a=45\ 000+2.4\times 8\ 000$

$=64\ 200\text{N}$

$$L_h=500\left(f_n\frac{C_r}{P}\right)^{\frac{10}{3}}$$

$$=500\left(0.444\times\frac{505\ 000}{64\ 200}\right)^{\frac{10}{3}}$$

$$=500\times 3.49^{\frac{10}{3}}\fallingdotseq 32\ 000\ \text{h}$$

(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 cup back faces is 50 mm.
Calculate the basic rating life of each bearing when beside the radial load F_r= 5 500N, {561kgf}, axial load F_{ae}=2 000N,{204kgf} are applied to **HR30305DJ** as shown in Fig. 4.21. The speed is 600 min^{-1}.

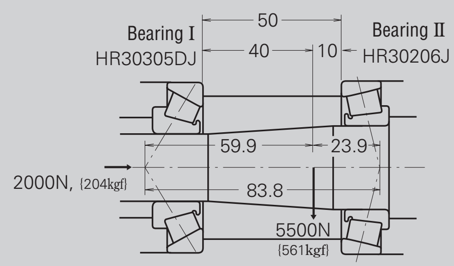


Fig. 4.21 Loads on Tapered Roller Bearings

To distribute the radial load F_r on bearings I and II, the effective load centers must be located for tapered roller bearings. 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:

$$F_{rI}=5\ 500\times\frac{23.9}{83.8}=1\ 569\text{N},\ \{160\text{kgf}\}$$

$$F_{rII}=5\ 500\times\frac{59.9}{83.8}=3\ 931\text{N},\ \{401\text{kgf}\}$$

From the data in the bearing table, the following values are obtained;

Bearings	Basic dynamic load rating C_r (N) {kgf}	Axial load factor Y_1	Constant e
Bearing I (**HR30305DJ**)	38 000 {3 900}	$Y_I=0.73$	0.83
Bearing II (**HR30206J**)	43 000 {4 400}	$Y_{II}=1.6$	0.38

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

$$F_{ae}+\frac{0.6}{Y_{II}}F_{rII}=2\,000+\frac{0.6}{1.6}\times 3\,931$$
$$=3\,474\text{N}, \{354\text{kgf}\}$$
$$\frac{0.6}{Y_I}F_{rI}=\frac{0.6}{0.73}\times 1\,569=1\,290\text{N}, \{132\text{kgf}\}$$

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

For bearing I

$F_{rI}=1\,569\text{N}$, {160kgf}

$F_{aI}=3\,474\text{N}$, {354kgf}

since $F_{aI}/F_{rI}=2.2>e=0.83$

the dynamic equivalent load $P_I=XF_{rI}+Y_IF_{aI}$

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

$=3\,164\text{N}$, {323kgf}

The fatigue life factor $f_h=f_n\frac{C_r}{P_I}$

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

and the rating fatigue life $L_h=500\times 5.04^{\frac{10}{3}}=109\,750\text{h}$

For bearing II

since $F_{rII}=3\,931\text{N}$, {401kgf}, $F_{aII}=0$

the dynamic equivalent load

$P_{II}=F_{rII}=3\,931\text{N}$, {401kgf}

the fatigue life factor

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

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

Remarks 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 $F_r=245\,000\text{N}$

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

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

Size limitation

Shaft diameter: 300mm

Bore of housing: Less than 500mm

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 limitation (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 CAE4**	1 540 000	0.19	3.5
	460	118	**23060 CAE4**	2 400 000	0.24	2.8
	460	160	**24060 CAE4**	2 890 000	0.32	2.1
	500	160	**23160 CAE4**	3 350 000	0.31	2.2
	500	200	**24160 CAE4**	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 examples of applications (refer to Page A034), a value of f_h, between 3 and 5 seems 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

$$C_r=\frac{(F_r+Y_3F_a)\times(3\text{ to }5)}{0.444}$$
$$=\frac{(245\,000+2.1\times 49\,000)\times(3\text{ to }5)}{0.444}$$
$$=2\,350\,000\text{ to }3\,900\,000\text{ N}$$

The bearings which satisfy this range are **23060CAE4**, **24060CAE4**, **23160CAE4**, and **24160CAE4**.

4.7 Bearing Type and Allowable Axial Load

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

(1) Change of 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 the contact angle and width. When heat generation or seizure has occurred, the bearing should be disassembled and checked for running trace to discover whether there has been a change in the contact angle during operation. In this way, it is possible to see whether an abnormal axial load has been sustained.

The relation shown below can be established among the axial load F_a on a bearing, the load of rolling element Q, and the contact angle α when the load is applied. (See Equations (9.8), (9.9), and (9.10) in Section 9.6.2)

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

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

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

Namely, δ_a is the change in Equation (4.52) to determine 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 a bearing are necessary in this case for calculation, the contact angle α was approximated from the axial load. The basic static load rating C_{0r} is expressed by Equation (4.53) for the case of a single row radial ball bearing.

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

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):

$$\frac{f_0}{C_{0r}}F_a = A\,F_a$$
$$= K\,\{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}-1\}^{3/2}\cdot\frac{\sin\alpha}{\cos\alpha_0} \quad \cdots\cdots \quad \textbf{(4.54)}$$

where, K: Constant determined from material and design of bearing

In other words, "h" is assumed and 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 A for each bore number of an angular contact ball bearing in Table 4.14. The relationship between $A\,F_a$ and α is shown in Fig. 4.22.

Example 1

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

A=0.212 is calculated from Table 4.10 and $A\,F_a$=0.212×35.0=7.42 and α≒26° are obtained from Fig. 4.22. An initial contact angle of 15° has changed to 26° under the axial load.

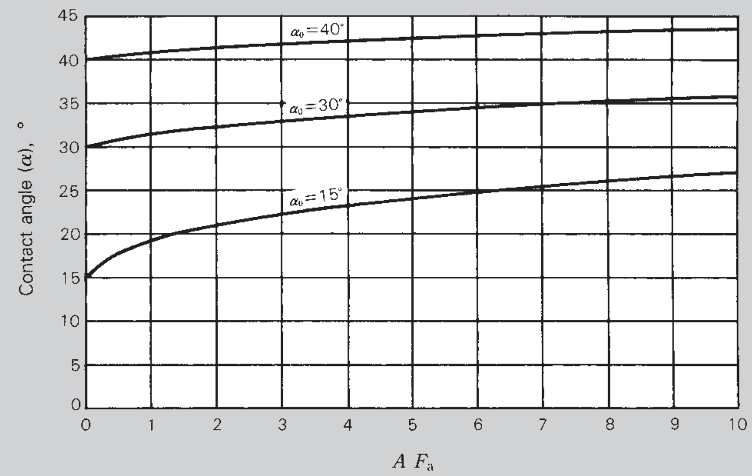


Fig. 4.22 Change of the Contact Angle of Angular Contact Ball Bearing under Axial Load

Table 4.10 Constant A Value of Angular Contact Ball Bearing

Units: kN^{-1}

Bearing bore No.	Bearing series 70			Bearing series 72			Bearing series 73		
	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 a deep groove ball bearing are similarly shown in Table 4.11 and Fig. 4.23.

Example 2
Change in the contact angle is calculated when the 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 10° is obtained from Fig. 3, Page C014. A=0.303 is determined from Table 4.11 and $A\ F_a$=0.303×24.75≑7.5 and α≑24° from Fig. 4.23.

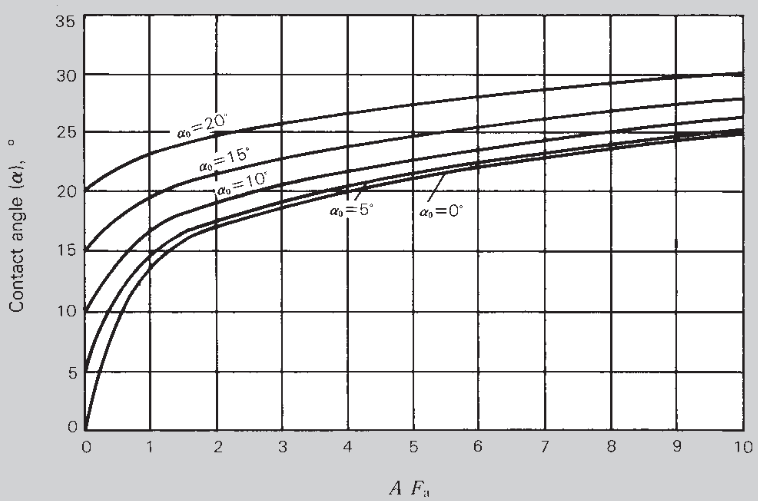


Fig. 4.23 Change in the Contact Angle of the Deep Groove Ball Bearing under Axial Load

Table 4.11 Contact A Value of Deep Groove Ball Bearing

Units: kN^{-1}

Bearing bore No.	Bearing series 62				
	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 means 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. Note also 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 is determined as follows. The contact angle α for F_a is determined from the right term of Equation (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 also 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 internal specifications of a bearing are known, Fig. 4.25 shows the result of a calculation to determine the allowable axial load for a deep groove radial ball bearing.

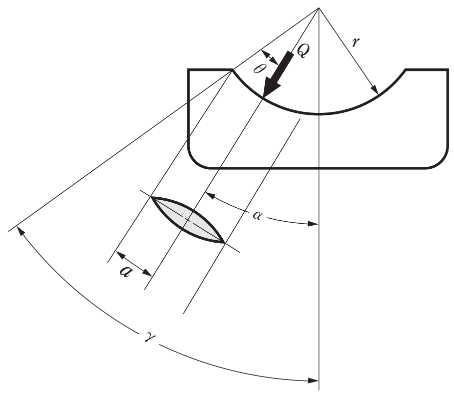


Fig. 4.24

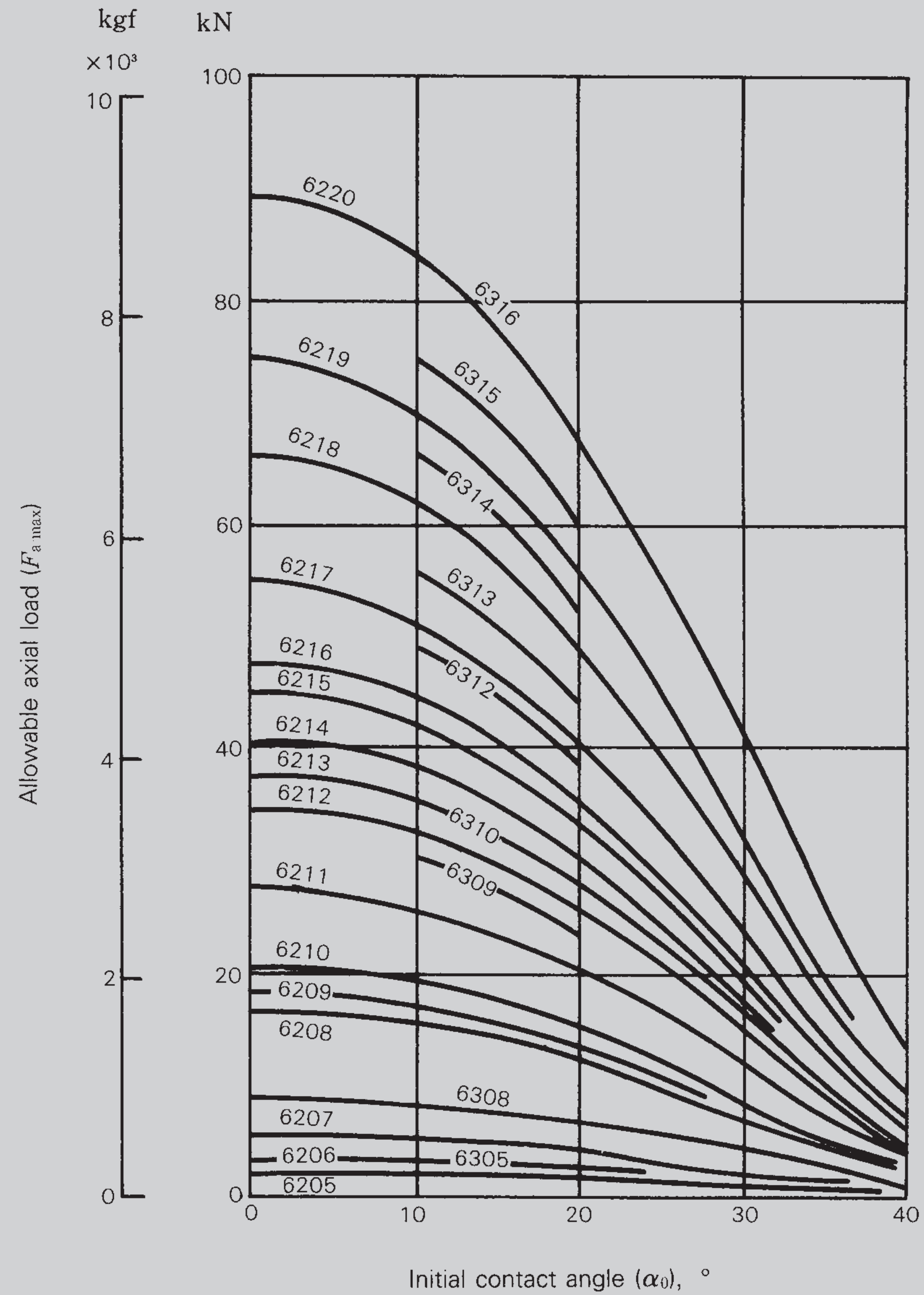


Fig. 4.25 Allowable Axial Load for a Deep Groove Ball Bearing

4.7.2 Allowable Axial Load (Break Down Strength of The Ribs) for a Cylindrical Roller Bearings

Both the inner and outer rings may be exposed to an axial load to a certain extent during rotation in a cylindrical roller bearing with ribs. The axial load capacity is limited by heat generation, seizure, etc. at the slip surface between the roller end surface and rib, or the rib strength.
The allowable axial load (the load considered the heat generation between the end face of rollers and the rib face) for the cylindrical roller bearing of the diameter series 3, which is applied continuously under grease or oil lubrication, is shown in Fig. 4.26.

Grease lubrication (Empirical equation)

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

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

Oil lubrication (Empirical equation)

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

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

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

In the equations (4.55) and (4.56), the examination for the rib strength is excluded. Concerning the rib strength, please consult with NSK.
To enable the cylindrical roller bearing to sustain the axial load capacity stably, it is necessary to take into account the following points concerning the bearing and its surroundings.

- ○ Radial load must be applied and the magnitude of radial load should be larger than that of axial load by 2.5 times or more.
- ○ There should be sufficient lubricant between the roller end face and rib.
- ○ Use a lubricant with an additive for extreme pressures.
- ○ Running-in-time should be sufficient.
- ○ Bearing mounting accuracy should be good.
- ○ Don't use a bearing with an unnecessarily large internal clearance.

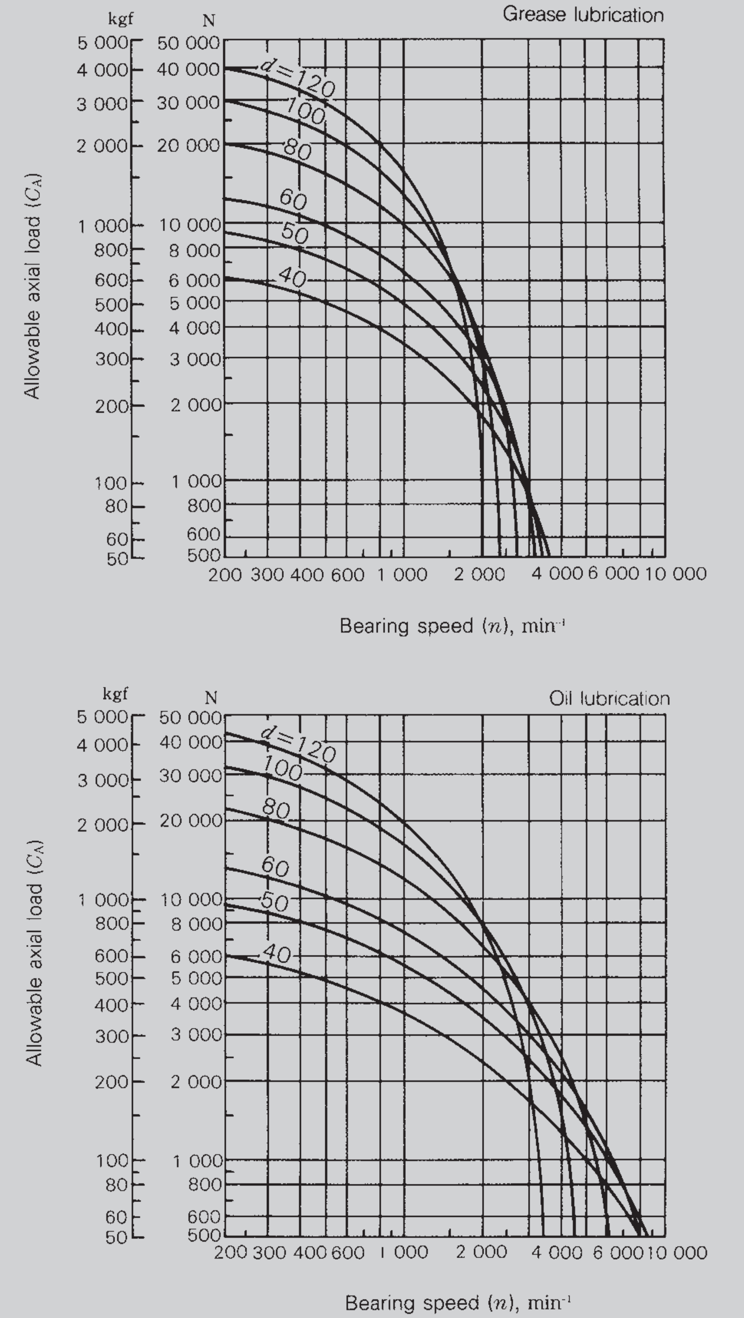
Moreover, if the bearing speed is very slow or exceeds 50% of the allowable speed in the bearing catalog, or if the bearing bore diameter exceeds 200 mm, it is required for each bearing to be precisely checked for lubrication, cooling method, etc. Please contact NSK in such cases.

f: Load factor

	f value
Continuous loading	1
Intermittent loading	2
Short time loading	3

k : Dimensional factor

	k value
Bearing diameter series 2	0.75
Bearing diameter series 3	1
Bearing diameter series 4	1.2



Conditions are continuous loading (f = 1) and bearing diameter series 3 (k = 1.0)

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 applications such as aircraft, satellites, or rockets, greatly increased reliability is demanded of each component. This concept is being applied 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 for which 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 life. For example, the average value is employed frequently to describe the life span of human beings. However, if the average value were used for bearings, then too many bearings would fail before the average life value is reached. On the other hand, if a low or minimum value is used as a criterion, then too many bearings would have a life much longer than the set value. In this view, the value 90% was chosen for common practice. The value 95% could have been taken as the statistical reliability, but nevertheless, the slightly looser reliability of 90% was taken for bearings empirically from the practical and economical viewpoint. A 90% reliability however is not acceptable for parts of aircraft or electronic computers or communication systems these days, and a 99% or 99.9% reliability is demanded 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%). Below the damage ratio of 10% (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 the 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 as shown in the table below 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=0.33×L_{10}=3 300 hours. In this manner, the reliability of the bearing life can be matched to the degree of reliability required of the equipment and difficulty of overhaul and inspection.

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.

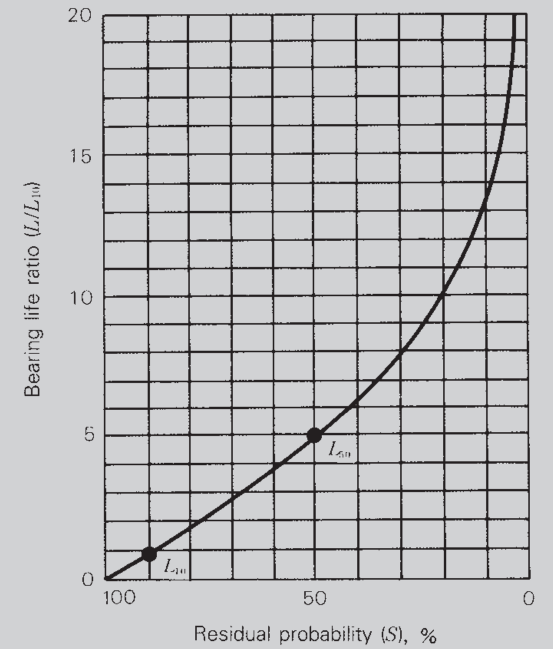


Fig. 4.27 Bearing Life and Residual Probability

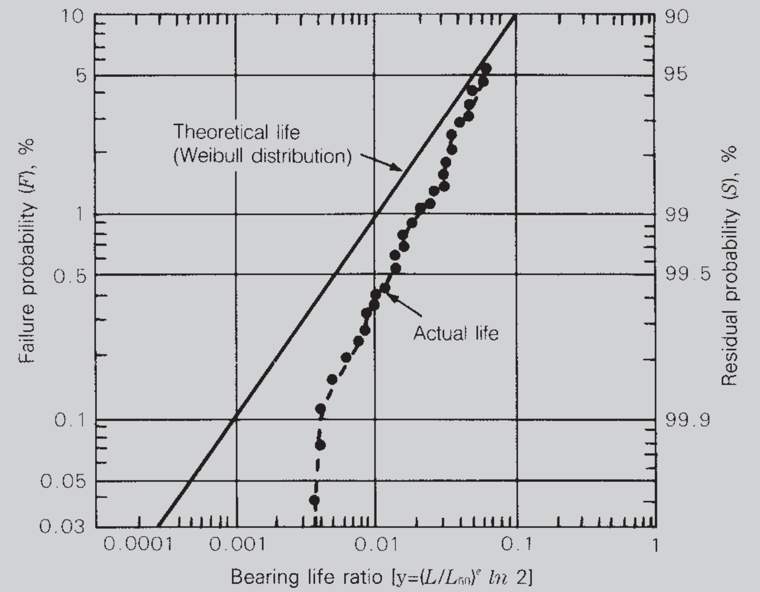


Fig. 4.28 Life Distribution in the Low Failure Ratio Range

4.8.2 Radial Clearance and Fatigue Life

As shown in the catalog, etc., the fatigue life calculation equation of rolling bearings is Equation (4.57):

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

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 is based on a prerequisite that 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 internal clearance is zero. In this sense, the normal fatigue life calculation is intended to obtain the value when the clearance is zero. When the effect of the radial clearance is taken into account, the bearing fatigue life can be calculated as follows. 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 bearing

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

For cylindrical roller bearing

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

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 the radial internal clearance is Δ_r can also be obtained as shown in Table 4.13.

Fig. 4.30 shows the relationship between the radial clearance and bearing fatigue life while taking 6208 and NU208 as examples.

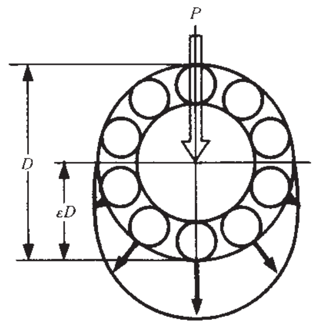


Fig. 4.29 Load Distribution with ε=0.5

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

ε	Deep groove ball bearing $f(\varepsilon)$	Deep groove ball bearing $\frac{L_\varepsilon}{L}$	Cylindrical roller bearing $f(\varepsilon)$	Cylindrical roller bearing $\frac{L_\varepsilon}{L}$
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

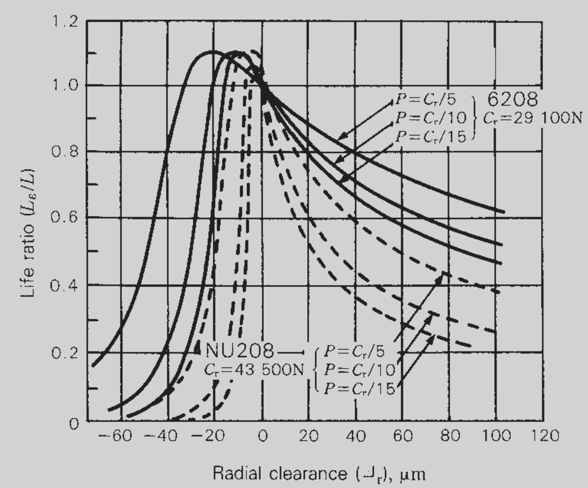


Fig. 4.30 Radial Clearance and Bearing Life Ratio

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 it is essential to take utmost care with machining and assembly accuracies of surrounding shafts and housing if 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 different sizes of bearings are selected as examples from the 62 and 63 series deep-groove ball bearings.

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 $L_\theta/L_{\theta=0}$. The result is shown in Figs. 4.31 to 4.34.

As an example of ordinary running conditions, the radial load F_r (N) {kgf} and axial load F_a (N) {kgf} were assumed respectively to be approximately 10% (normal load) and 1% (light preload) of the dynamic load rating C_r (N) {kgf} of a bearing and were used as load conditions for the calculation. Normal radial clearance was used and the shaft fit was set to around j5. Also taken into account was the decrease of the internal clearance due to expansion of the inner ring. 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 problem.

When the misalignment exceeds a certain limit, however, the fatigue life degrades rapidly as shown in the figure. Attention is therefore necessary in this respect.

When the clearance is small, not much effect is observed as long as the misalignment is small, as shown in the figure. But the life decreases substantially when the misalignment increases. As previously mentioned, it is essential to minimize the mounting error as much as possible when a bearing is to be used.

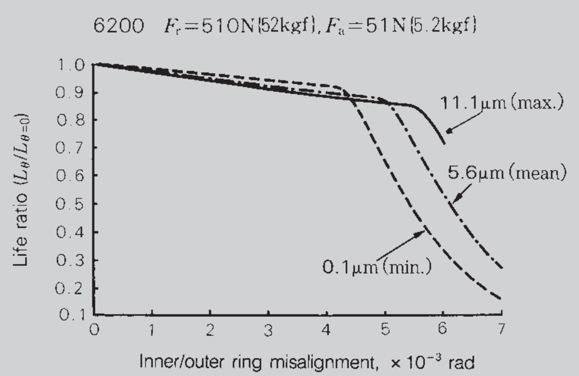


Fig. 4.31

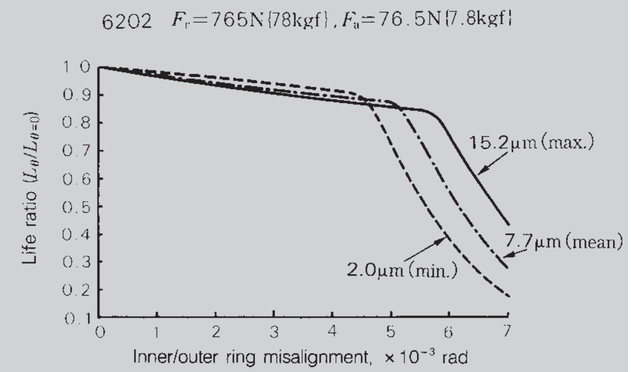


Fig. 4.32

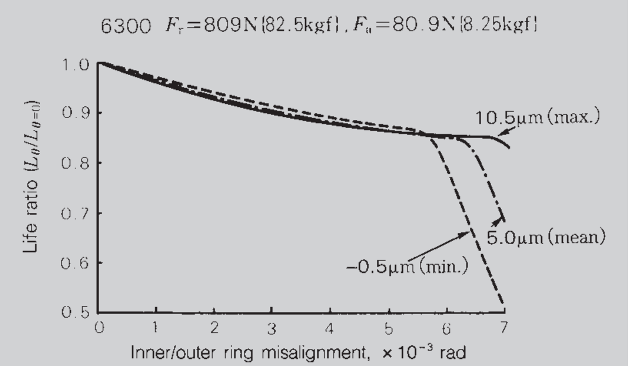


Fig. 4.33

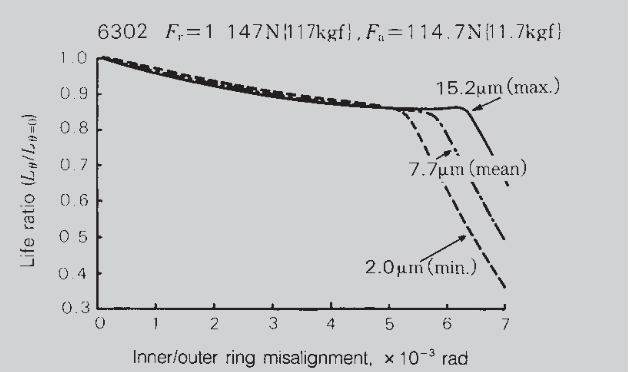


Fig. 4.34

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, there arises misalignment between the inner and outer rings of the bearings, thereby lowering their fatigue life. The degree of life degradation depends on the bearing type and interior design but also varies depending on the radial internal clearance and the magnitude of load during operation.

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

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 that with misalignment is L_θ.

Figs. 4.35 and 4.36 show the case with constant load (10% of basic dynamic load rating C_r of a bearing) for each case when the internal clearance is a normal, C3 clearance, or C4 clearance. Figs. 4.37 and 4.38 show the case 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. But the life ratio itself differs among bearing series and dimensions, with life degradation rapid in 22 and 23 series bearings (wide type). It is advisable to use a bearing of special design when considerable misalignment is expected during application.

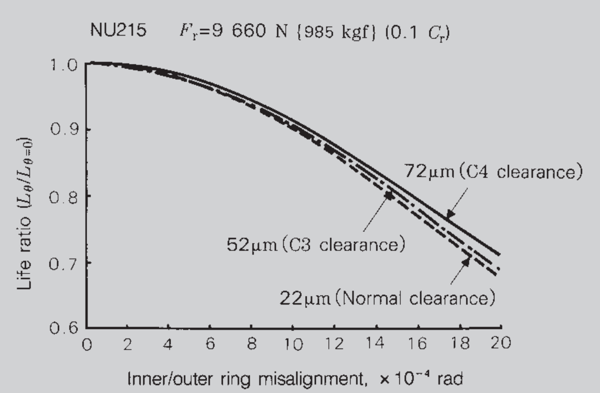


Fig. 4.35

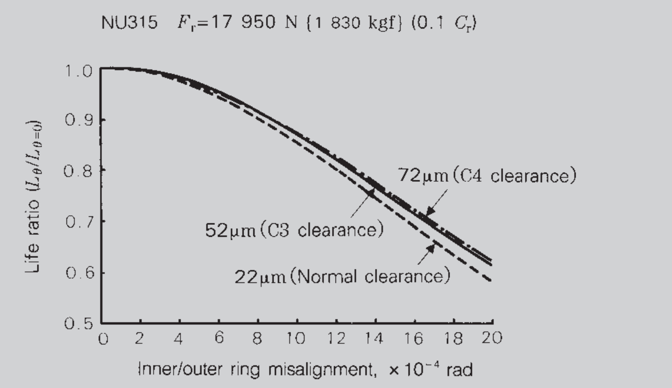


Fig. 4.36

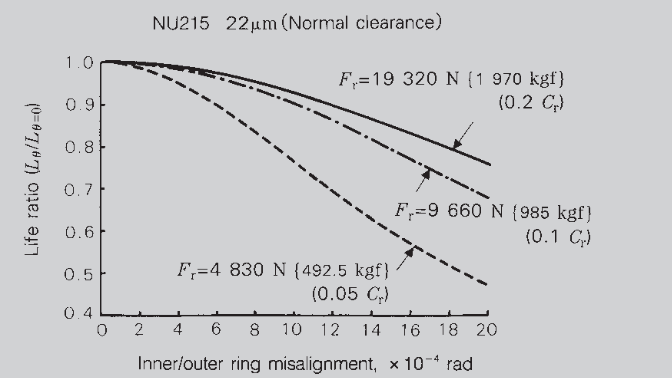


Fig. 4.37

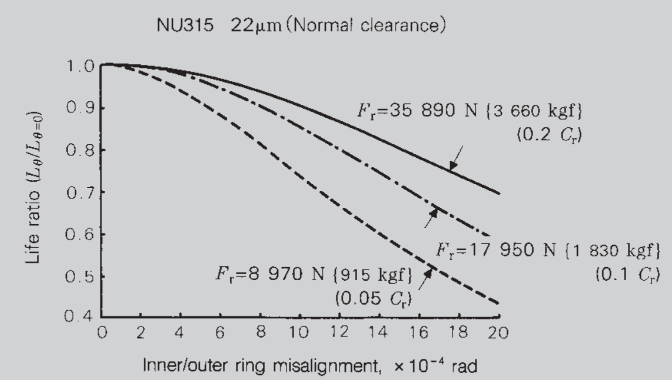


Fig. 4.38

4.8.5 Oil Film Parameters and Rolling Fatigue Life

Based on numerous experiments and experiences, the rolling fatigue life of rolling bearings can be shown to be closely related to the lubrication.
The rolling fatigue life is expressed by the maximum number of rotations, which a bearing can endure, until the raceway or rolling surface of a bearing develops fatigue in the material, resulting in flaking of the surface, under action of cyclic stress by the bearing. Such flaking begins with either microscopic non-uniform portions (such as non-metallic inclusions, cavities) in the material or with microscopic defect in the material's surface (such as extremely small cracks or surface damage or dents caused by contact between extremely small projections in the raceway or rolling surface). The former flaking is called sub-surface 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 state of 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 (flaw, dent, etc.), the life is determined mainly by sub-surface originating flaking. On the other hand, a decrease in Λ tends to develop 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 using different lubricants and bearing materials (● and ▲ in Fig. 4.40). Fig. 4.40 shows a summary of the principal experiments selected from among those reported up to now. As is evident, the life decreases rapidly at around $\Lambda \fallingdotseq 1$ when compared with the life values at around Λ=3 ~ 4 where life changes at a slower rate. The life becomes about 1/10 or less at $\Lambda \leqq 0.5$. This is a result of severe surface-originating flaking.
Accordingly, it is advisable for extension of the fatigue life of rolling bearings to increase the oil film parameter (ideally to a value above 3) by improving lubrication conditions.

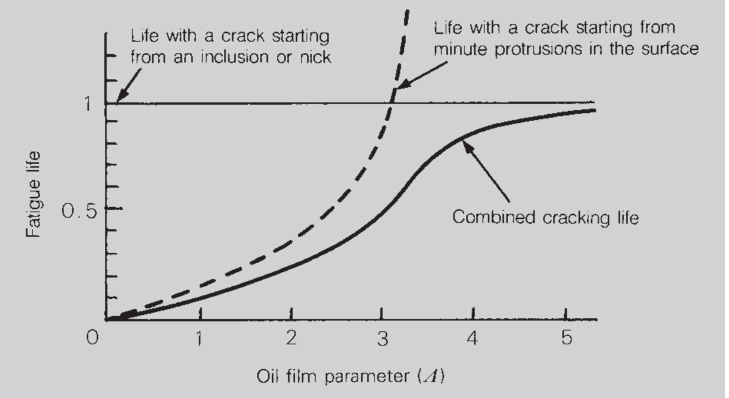


Fig. 4.39 Expression of Life According to Λ (Tallian, et al.)

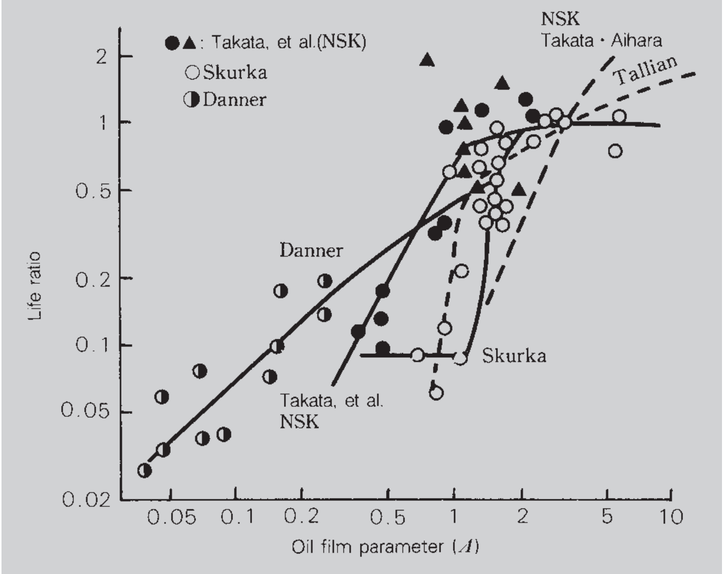


Fig. 4.40 Typical Experiment with Λ and Rolling Fatigue Life
(Expressed with reference to the life at Λ =3)

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 – surface roughness ratio), the most critical among the EHL qualities.

(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, lubricating conditions cannot be discussed without considering this surface roughness. For example, given a particular mean oil film thickness, there are two conditions which 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 an oil film parameter in the study and application of EHL.

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

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

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

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

(2) Oil Film Parameter Calculation Diagram

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

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

The oil film thickness to be used is that of the inner ring under the maximum rolling element load (at which the thickness becomes minimum).
Equation (4.61) can be expressed as follows by grouping into terms (R) for speed, (A) for viscosity, (F) for load, and (J) for bearing technical specifications. t is a constant.

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

R and A may be quantities not dependent on a 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 determined roughly from the bearing size, however, such change may be limited to 20 to 30%. As a result, F is handled as a lump with the term J of bearing specifications [F=F(J)]. Traditional Equation (4.62) can therefore be grouped as shown below:

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

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

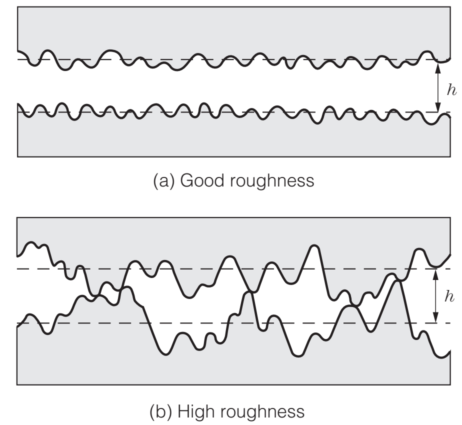


Fig. 4.41 Oil Film and Surface Roughness

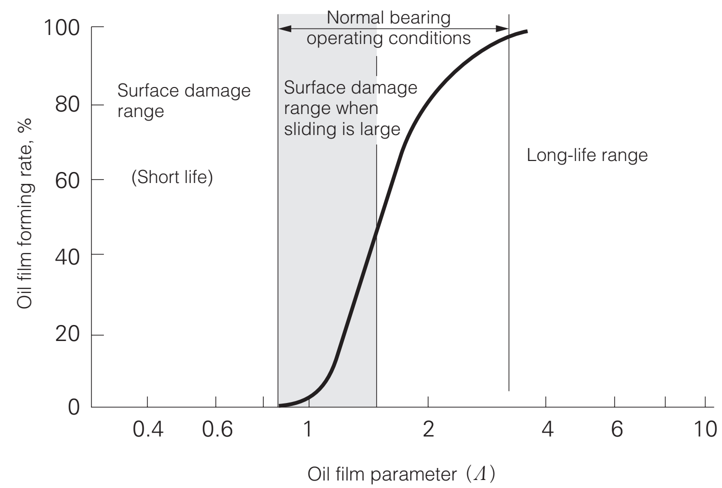


Fig. 4.42 Effect of Oil Film on Bearing Performance

The oil film parameter Λ, which is most vital among quantities related to EHL, is expressed by a simplified equation shown below. The fatigue life of rolling bearings becomes shorter when Λ is smaller.
In the equation $\Lambda = T \cdot R \cdot A \cdot D$ terms include A for oil viscosity η_0 (mPa·s, {cp}), R for the speed n (min^{-1}), and D for bearing bore diameter d (mm). The calculation procedure is described below.
(i) Determine the value 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 oil kind 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 is the density (g/cm^3) and uses the approximate value as shown below:
Mineral oil $\rho = 0.85$
Silicon oil $\rho = 1.0$
Diester oil $\rho = 0.9$

When it is not known whether the mineral oil is naphthene or paraffin, use the paraffin curve shown in Fig. 4.44.
(iv) Determine the D value from the diameter series and bore diameter d (mm) in Fig. 4.45.
(v) The product of the above values is used as an oil film parameter.

Table 4.14 Value T

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

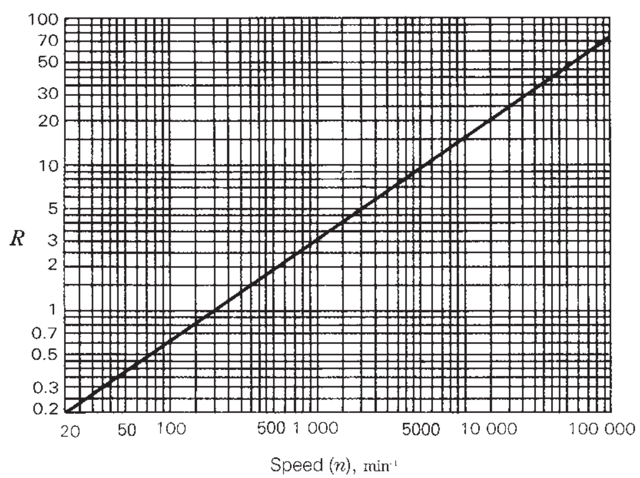


Fig. 4.43 Speed Term, R

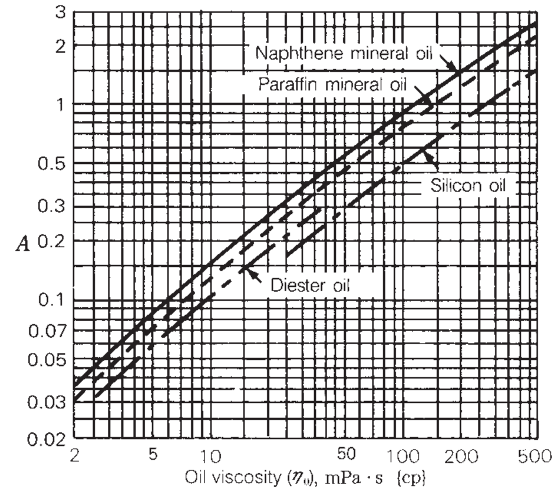


Fig. 4.44 Term Related to Lubricant Viscosity, A

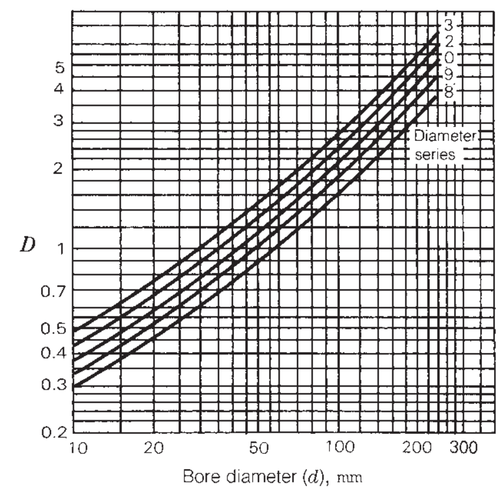


Fig. 4.45 Term Related to Bearing Specifications, D

Examples of EHL oil film parameter calculation are described below.

(**Example 1**)
The oil film parameter is determined when a deep groove ball bearing 6312 is operated with paraffin mineral oil (η_0=30 mPa·s, {cp}) at the speed 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**)
The oil film parameter is determined when a cylindrical roller bearing NU240 is operated with paraffin mineral oil (η_0=10 mPa·s, {cp}) at the speed 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) Effect of Oil Shortage and Shearing Heat Generation
The oil film parameter obtained above is the value when the requirements, that is, the contact inlet fully flooded with oil and isothermal inlet are satisfied. However, these conditions may not be satisfied depending on lubrication and operating conditions.
One such condition is called starvation, and the actual oil film paramerer value may become smaller than determined by Equation (4.64). Starvation might occur if lubrication becomes limited. In this condition, a guideline for adjusting the oil film parameter is 50 to 70% of the value obtained from Equation (4.64).

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

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

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 rolling element set.

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

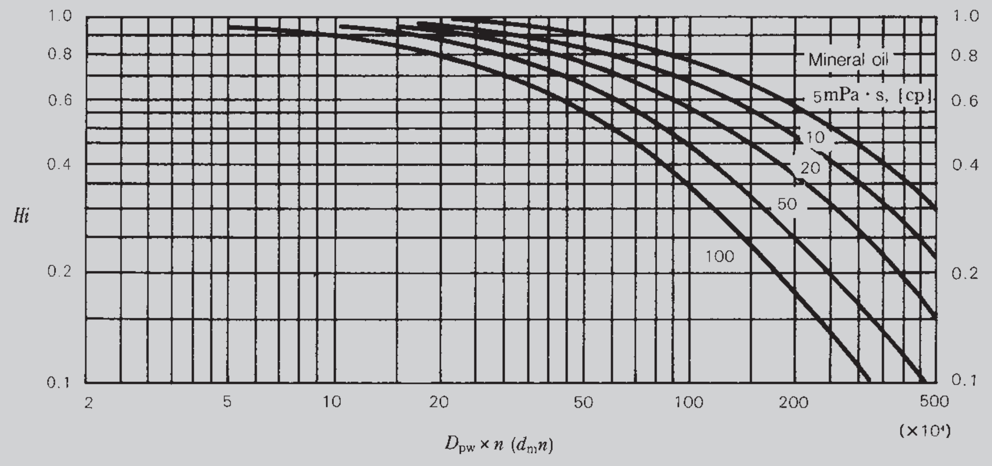


Fig. 4.46 Oil Film Thickness Decrease Factor Hi Due to Shearing Heat Generation

4.8.7 Load Calculation of Gears

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

There is an extremely close relationship among the two mechanical elements, 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:

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

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

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

The magnitudes of the 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:

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

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

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

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

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:

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

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

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

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.

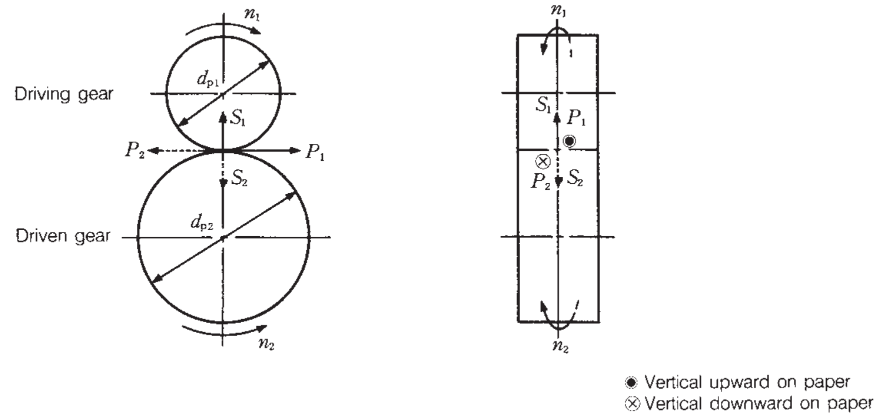


Fig. 4.47 Spur Gear

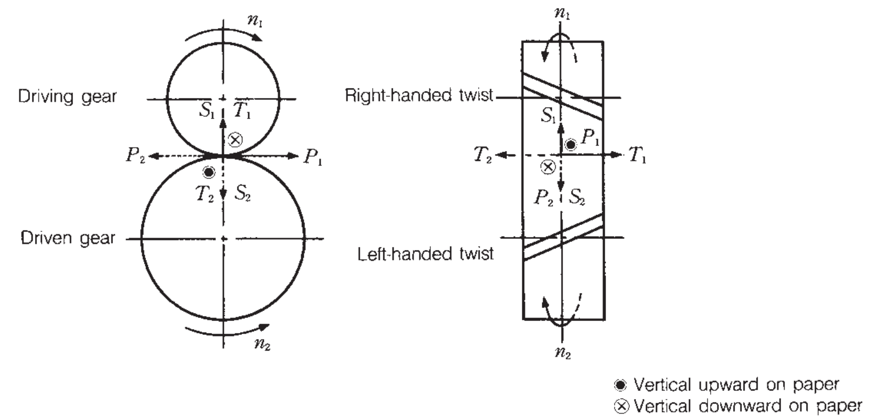


Fig. 4.48 Helical Gear

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:

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

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

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	$P_A = \dfrac{b}{a+b} P_1$ ⊗	$P_B = \dfrac{a}{a+b} P_1$ ⊗
	From S_1	$S_A = \dfrac{b}{a+b} S_1$ ↑	$S_B = \dfrac{a}{a+b} S_1$ ↑
	From T_1	$U_A = \dfrac{d_{p1}/2}{a+b} T_1$ ↑	$U_B = \dfrac{d_{p1}/2}{a+b} T_1$ ↓
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 direction is shown referring to left side of Fig. 4.49.

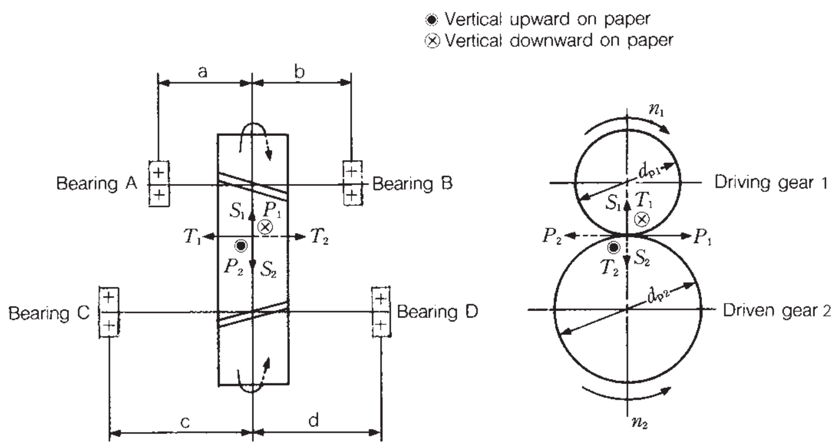


Fig. 4.49

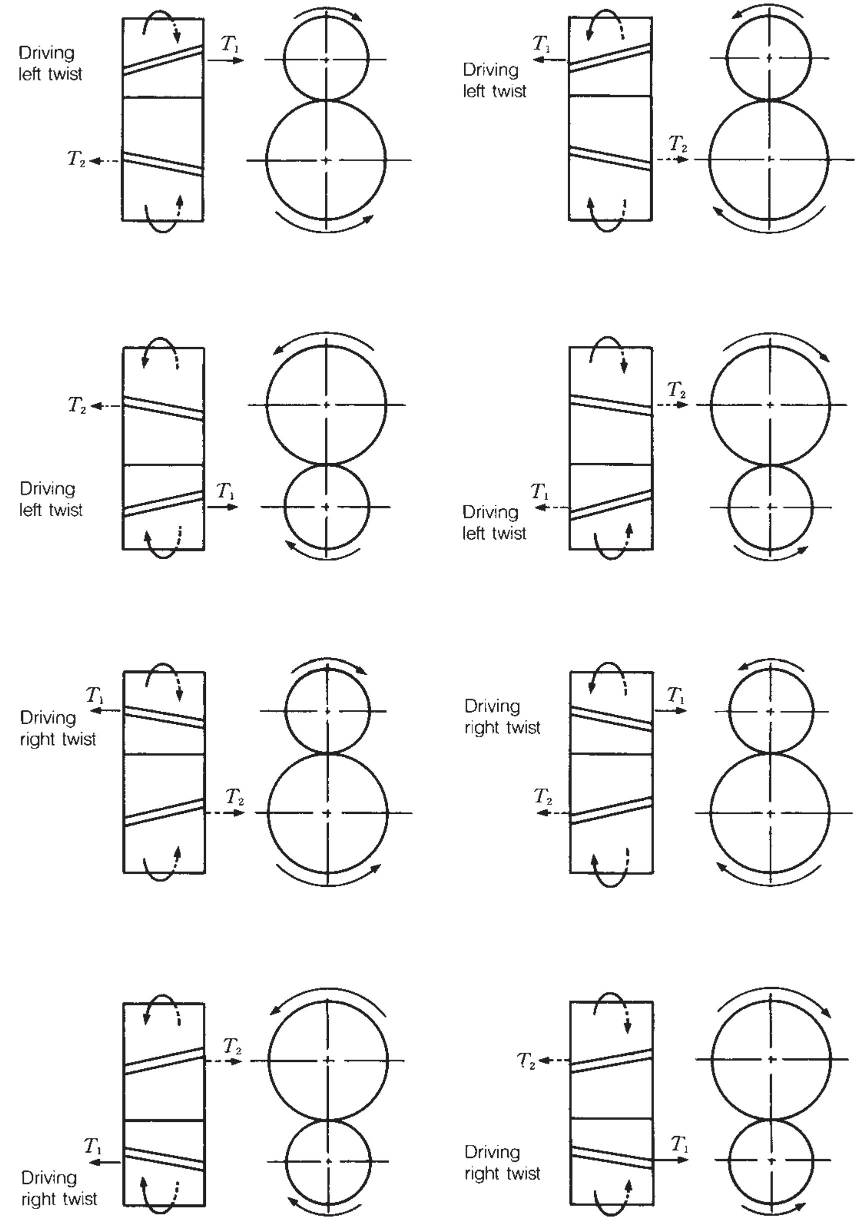


Fig. 4.50 Thrust Direction

(2) Calculation of Load Acting on Straight Bevel Gears

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

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

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

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

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

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

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

◉ Vertical upward on paper
⊗ Vertical downward on paper

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 direction is shown referring to Fig. 4.52.

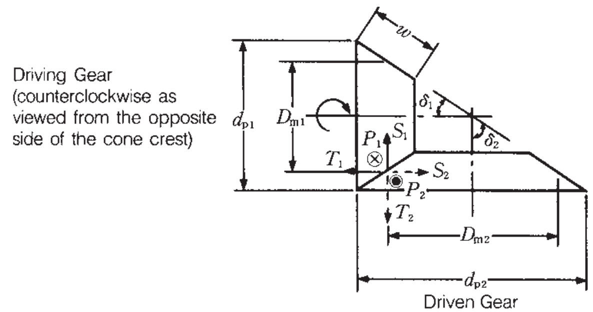


Fig. 4.51

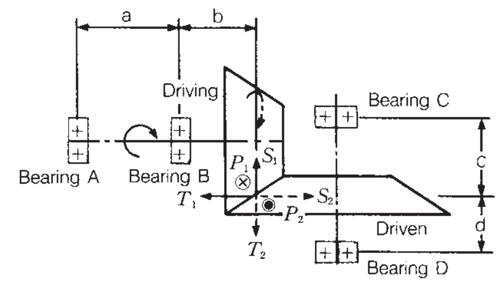


Fig. 4.52

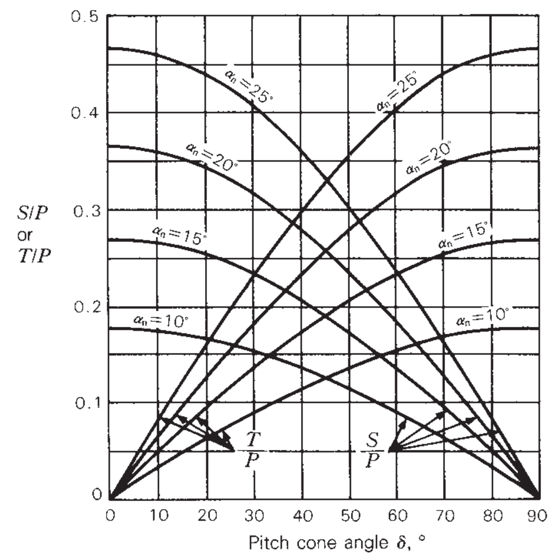


Fig. 4.53

(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 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:

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

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

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 T are as follows depending on the running direction and gear twist direction:

(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

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

The positive (plus) calculation result means that the load is acting in a direction to separate the gears while a negative (minus) one means that the load is acting in a direction to bring the gears nearer.

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 Acting on Straight Bevel Gears."

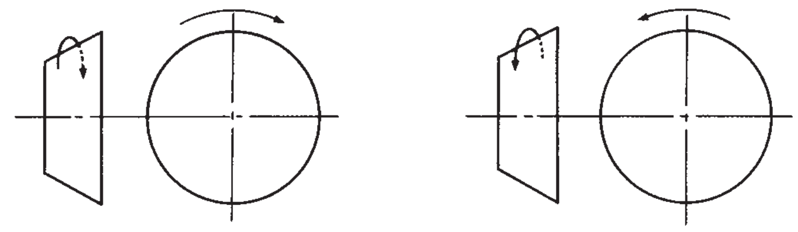


Fig. 4.54

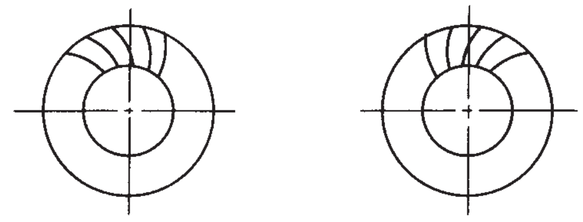


Fig. 4.55

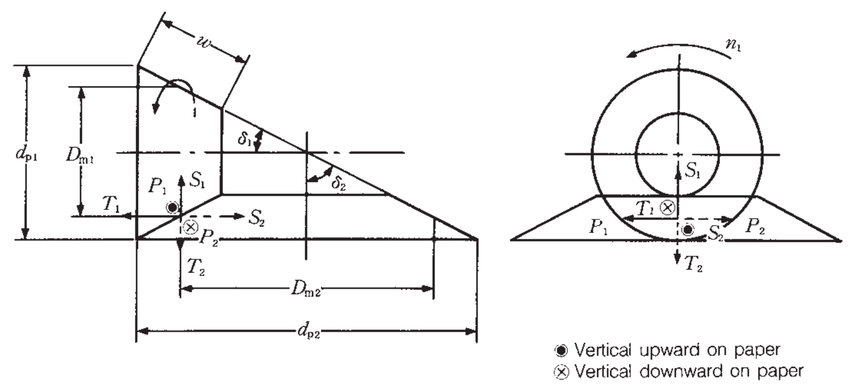


Fig. 4.56

(4) Calculation of Load Acting on Hypoid Gears

The force acting at the meshing point of Hypoid Gears is calculated as follows:

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

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

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

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

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

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

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 T are as follows depending on the running direction and gear twist direction:

(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

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

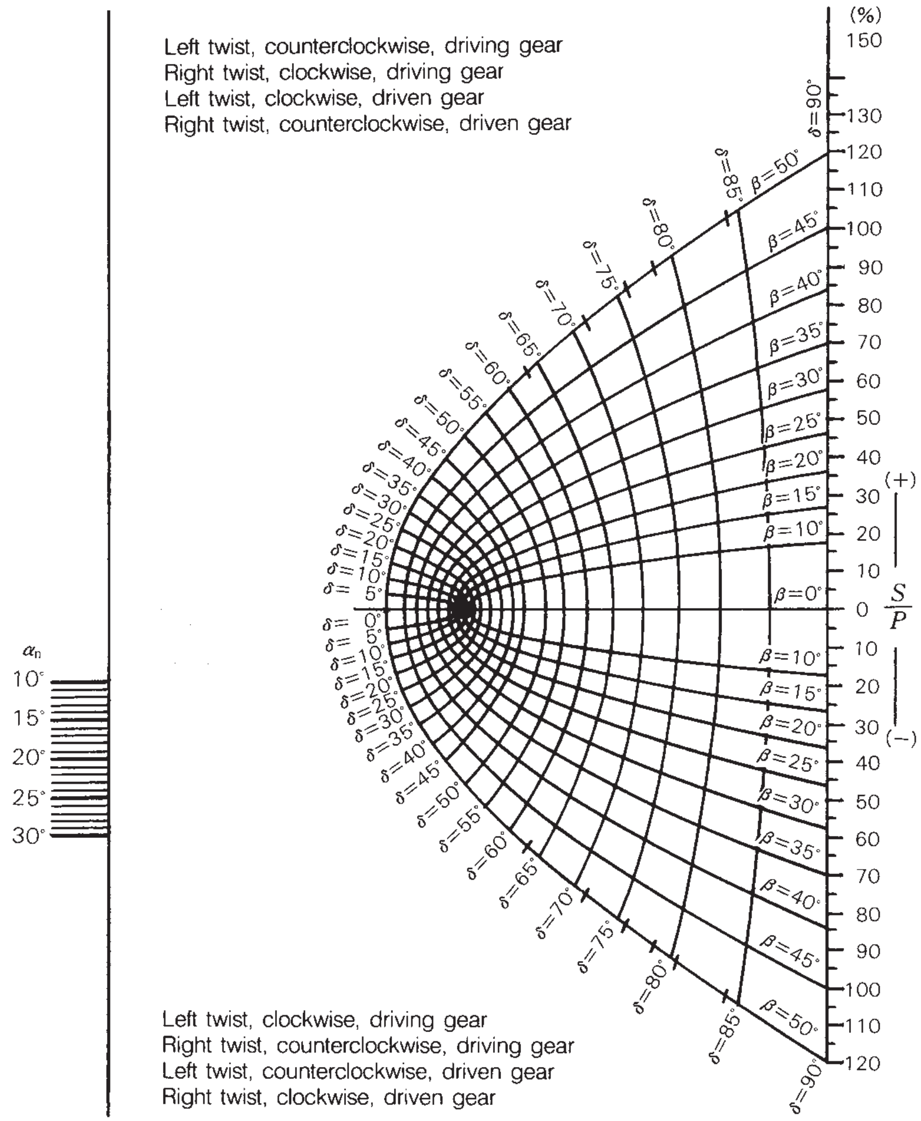
The positive (plus) calculation result means that the load is acting in a direction to separate the gears while a negative (minus) one means that the load is acting in a direction to bring the gears nearer.

For the 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.”

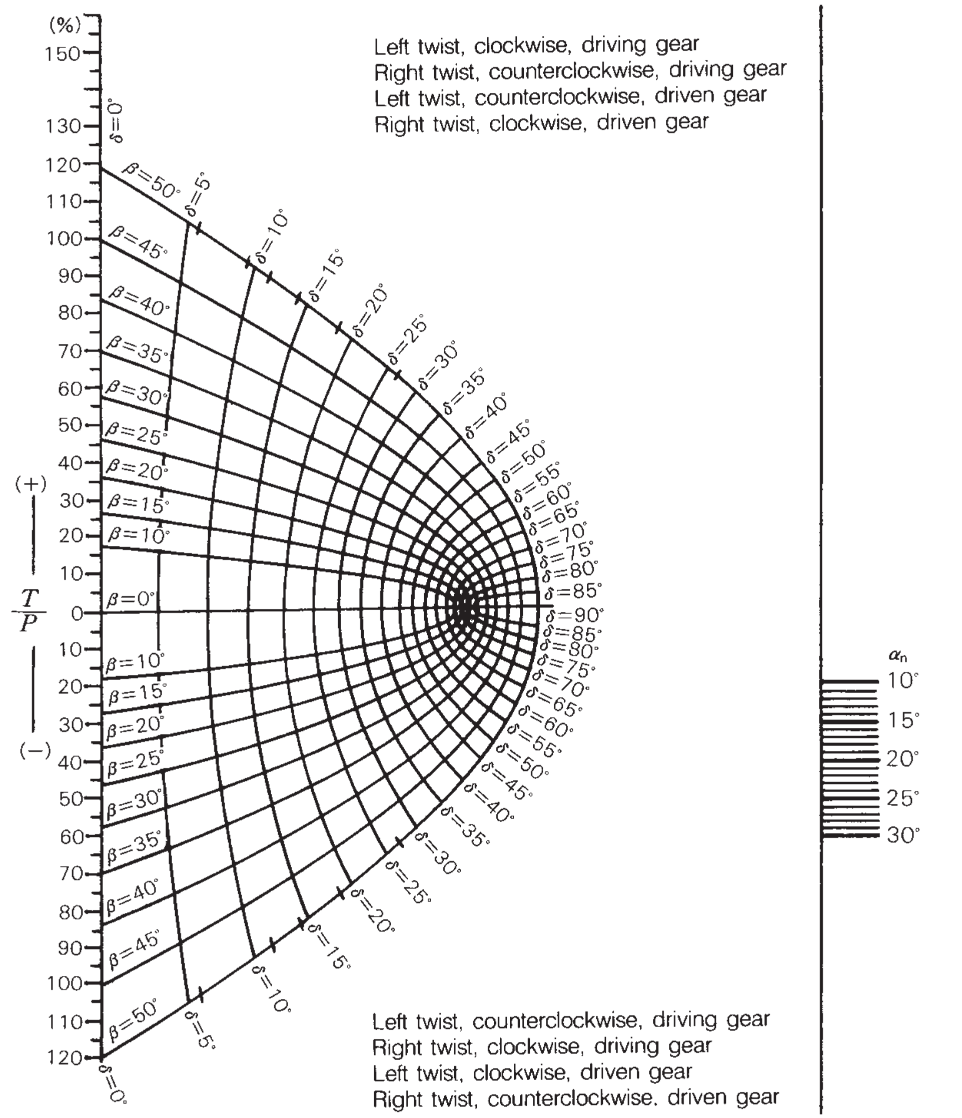
The next calculation diagram is used to determine the approximate value and direction of separating force S and thrust T.

[How To Use]
The method of determining the separating force S is shown. The thrust T can also be determined in a similar manner.

1. Take the gear normal pressure angle α_n from the vertical scale on the left side of the diagram.
2. Determine the intersection between the pitch cone angle δ and the twist angle β. Determine one point which is either above or below the β=0 line according to the rotating direction and gear twist direction.
3. Draw a line connecting the two points and read the point at which the line cuts through the right vertical scale. This reading gives the ratio (S/P, %) of the separating force S to the tangential force P in percentage.



Calculation Diagram of Separating Force S



Calculation Diagram of Thrust T

(5) Calculation of Load on Worm Gear

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

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

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

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

ψ: For the frictional angle, the value obtained

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

as shown in Fig. 4.57 is used.

When V_R is 0.2 m/s or less, then use ψ=8°.
When V_R exceeds 6 m/s, use ψ=1°4′.

α_n : Gear normal pressure angle
α_a : Shaft plane pressure angle
Z_w : No. of threads (No. of teeth of worm gear)
Z_2 : No. of teeth of worm wheel
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.

The load on the bearing is obtained from the magnitude and direction of each component at the meshing point of the worm gears according to the method shown in Table 4.15 of Section 4.8.7 (1), Calculation of loads on spur, helical, and double-helical gears.

Table 4.17

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

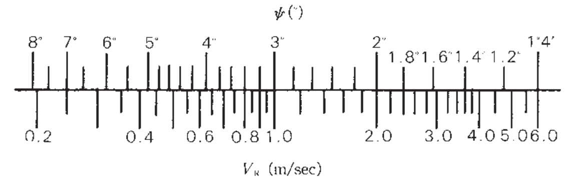


Fig. 4.57

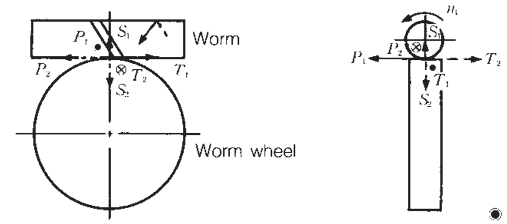


Fig. 4.58 Right Twist Worm Gear

◉ Vertical upward on paper
⊗ Vertical downward on paper

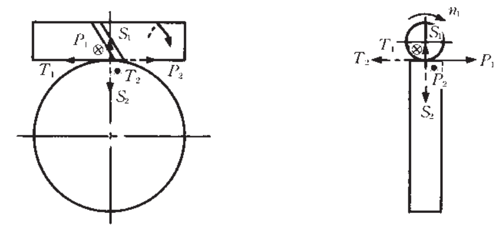


Fig. 4.59 Right Twist Worm Gear (Worm Rotation is Opposite that of Fig. 4.58)

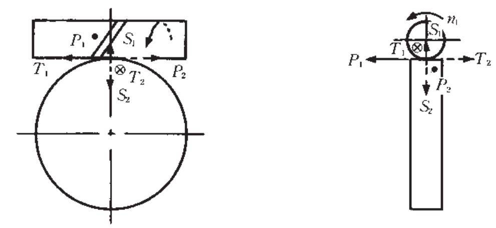


Fig. 4.60 Left Twist Worm Gear

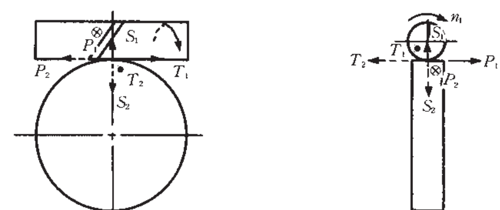
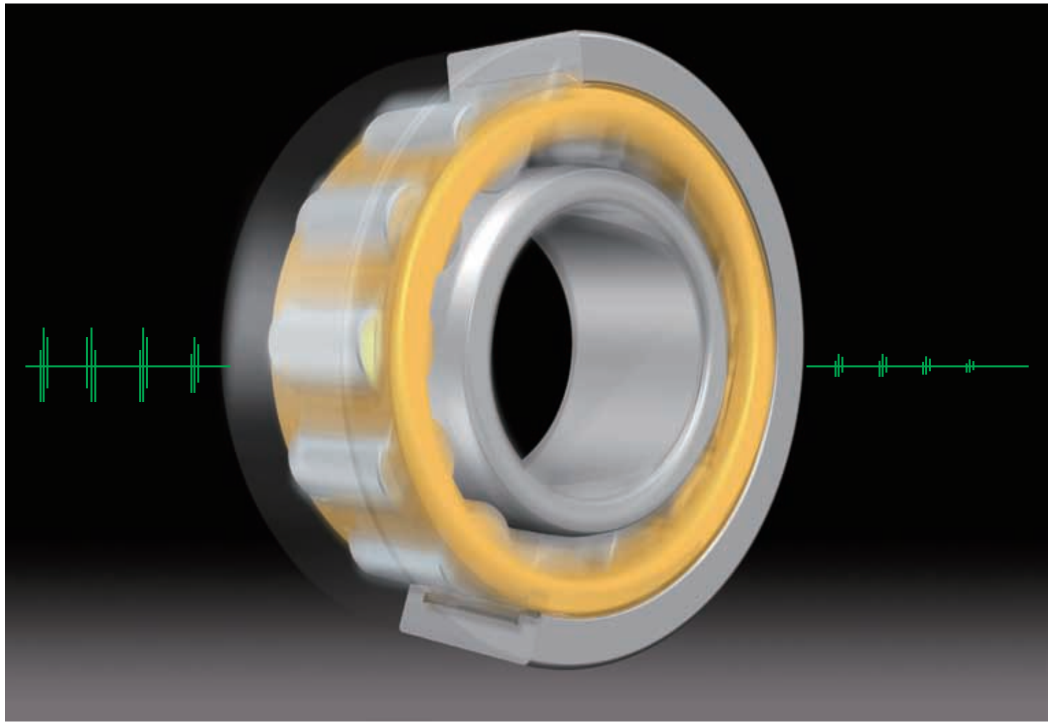


Fig. 4.61 Left Twist Worm Gear (Worm Rotation is Opposite that of Fig. 4.60)



5. SPEEDS

5. SPEEDS

In this catalog, NSK uses four definitions of speed shown in Table 5.1.

Table 5.1 Overview of Speeds

Speeds	Overview	Applicable lubrication methods
Limiting Speed (Grease)	Empirically obtained and comprehensive bearing limiting speed in grease lubrication.	Grease lubrication
Limiting Speed (Oil)	Empirically obtained and comprehensive bearing limiting speed in oil bath lubrication.	Oil bath lubrication
Thermal Reference Speed(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 the reference conditions defined by ISO 15312. One among various criteria showing the suitability for operation at high speed.	Oil bath lubrication when subject to reference conditions outlined in ISO 15312
Limiting Speed (Mechanical)(1)	Mechanical and kinematic limiting speed achievable under ideal conditions for lubrication, heat dissipation and temperature.	e.g. Properly designed and controlled forced-circulation oil lubrication

Note (1) Thermal reference speeds and limiting speed (mechanical) are listed only in the tables of single row cylindrical roller bearings and spherical roller bearings.

5.1 Limiting Speed (Grease/Oil)

When bearings are operating, 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 form and material, load, lubricating method, and heat dissipating method including the design of the bearing's surroundings.

The limiting speed (grease) and limiting speed (oil) in the bearing tables are applicable to bearings of standard design and subjected to normal loads, i.e. $C/P \geqq 12$ and $F_a/F_r \leqq 0.2$ approximately. 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 speed (grease) or limiting speed (oil), it is necessary to select a grease or oil which has good high speed characteristics.

(Refer to)
Table 11.2 Grease Properties (Pages A236 and 237)
Table 11.5 Example of Selection of Lubricant for Bearing Operating Conditions (Page A239)
Table 11.6 Bands and Properties of Lubricating Grease (Pages A240 and A241)

5.1.1 Correction of Limiting Speed (Grease/Oil)

When the 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 limiting speed 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. It is recommended that NSK be consulted regarding high speed applications.

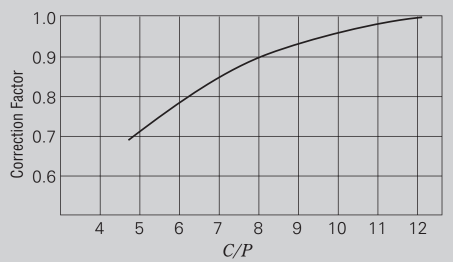


Fig. 5.1 Limiting Speed Correction Factor Variation with Load Ratio

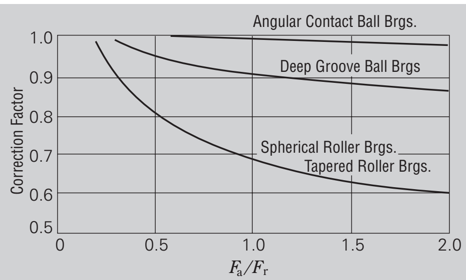


Fig. 5.2 Limiting Speed Correction Factor for Combined Radial and Axial Loads

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

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

The maximum permissible speed for contact rubber 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 Reference Speed

The thermal reference speed 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 the reference conditions defined by ISO 15312. It is one among various criteria showing the suitability for operation at high speed.
The below reference conditions are defined by ISO 15312.
-Outer-ring fixed, Inner-ring rotating
-Mean ambient temperature 20 degrees C
-Mean bearing temperature at the outer ring 70 degrees C
-Load on radial bearings 0.05 Cor
-Oil bath lubrication
-Lubricant ISO VG32 (radial bearings)
-Normal bearing internal clearance

The heat dissipation through the housing and shaft can be obtained from Fig.5.3. In Fig.5.3, A_r (mm^2) is 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 (radial bearings), and q_r (W/mm^2) as the heat flow density. The heat dissipation is calculated by multiplying the bearing seating surface area (A_r) by the heat flow density (q_r).

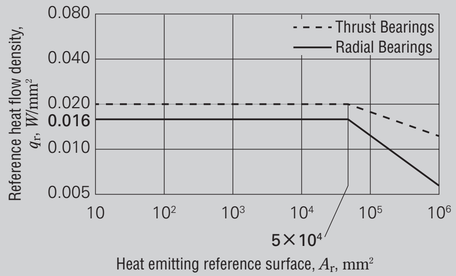


Fig. 5.3 Reference Heat Flow Density

5.3 Limiting Speed (Mechanical)

Limiting speed (mechanical) is the mechanical and kinematic limiting speed of bearings achievable under ideal conditions for lubrication, heat dissipation and temperature, 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, the centrifugal and gyratory forces, etc. The values in the tables are applicable to bearings of standard design and subjected to normal loads (C/P = 12 approximately).

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

5.4 Technical Data

5.4.1 Rotation and Revolution Speed of Rolling Element

When the rolling element rotates without slip between bearing rings, the distance which the rolling element rolls on the inner ring raceway is equal to that on the outer ring raceway. This fact allows establishment of a relationship among rolling speed n_i and n_e of the inner and outer rings and the number of rotation n_a of rolling elements.

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 the inner or outer ring being stationary). The rotation and revolution of the rolling element can be related as expressed by Equations (5.1) through (5.4).

No. of rotation

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

Rotational circumferential speed

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

No. of revolutions (No. of cage rotation)

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

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

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

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})

The rotation and revolution of the rolling element is 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 α=90°.

As an example, Table 5.4 shows the rotation speed n_a and revolution speed n_c of the rolling element during rotating of the inner ring of ball bearings 6210 and 6310.

Table 5.4 n_a and 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 $\gamma=\dfrac{D_w\cos\alpha}{D_{pw}}$

Table 5.3 Rolling Element's 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 rolling (n_e=0)	Outer ring rolling (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. ±: The "+" symbol indicates clockwise rotation while the "−" symbol indicates counterclockwise rotation.
2. $\gamma=\dfrac{D_w\cos\alpha}{D_{pw}}$ ($0°\leqq\alpha<90°$), $\gamma=\dfrac{D_w}{D_{pw}}$ (α=90°)

6. BOUNDARY DIMENSIONS AND IDENTIFYING NUMBERS FOR BEARINGS

6. BOUNDARY DIMENSIONS AND IDENTIFYING NUMBERS FOR BEARINGS

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, are the dimensions that define their external geometry. They include bore diameter d, outside diameter D, width B, bearing width(or height) T, chamfer dimension r, etc. It is necessary to know all of these dimensions 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 Table 6.1 to 6.3 (Pages A106 to A115).

In these boundary dimension tables, for each bore number, which prescribes the bore diameter, other boundary dimensions are listed for each diameter series and dimension series. A very large number of series are possible; however, not all of them are commercially available so more can be added in the future. Across the top of each bearing table (6.1 to 6.3), representative bearing types and series symbols are shown (refer to Table 6.5, Bearing Series Symbols, Page A121).

The relative cross-sectional dimensions of radial bearings (except tapered roller bearings) and thrust bearings for the various series classifications 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 are specified by ISO 464. Also, 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).

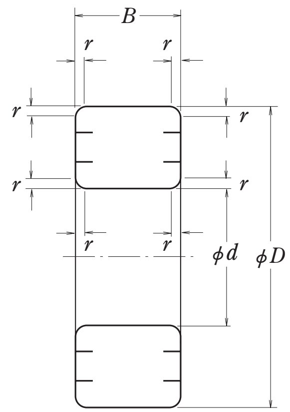


Fig. 6.1 Boundary Dimensions of Radial Ball and Roller Bearings

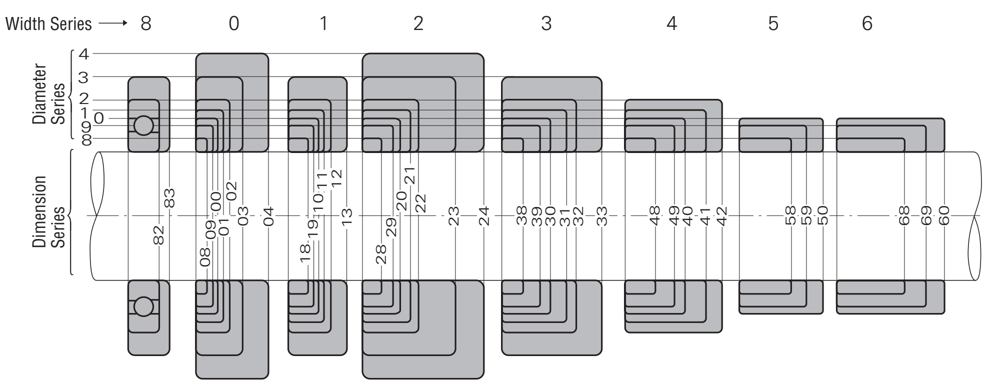


Fig. 6.6 Comparison of Cross Sections of Radial Bearings (except Tapered Roller Bearings) for various Dimensional Series

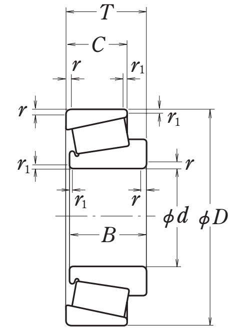


Fig. 6.2 Tapered Roller Bearings

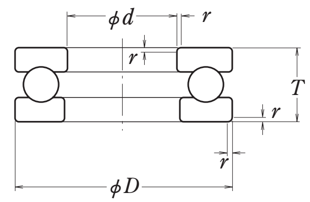


Fig. 6.3 Single-Direction Thrust Ball Bearings

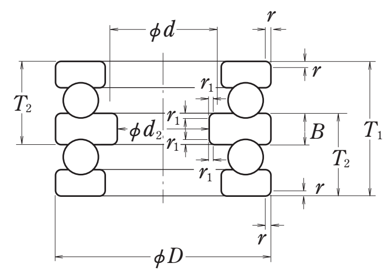


Fig. 6.4 Double-Direction Thrust Ball Bearings

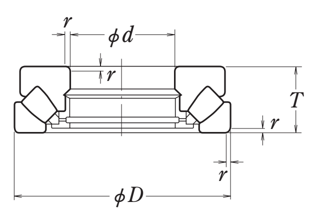


Fig. 6.5 Spherical Thrust Roller Bearings

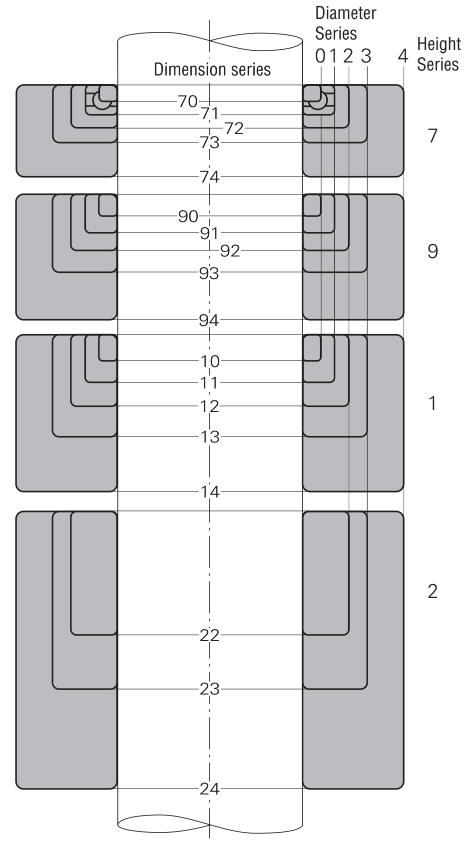


Fig. 6.7 Comparison of Cross Sections of Thrust Bearings (except Diameter Series 5) for Various Dimension Series

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

Units: mm

Bearing type	Diameter Series 8					Diameter Series 9							Diameter Series 0				
	18	28	38	48		19	29	39	49	59	69		00	10	20	30	40
Single-Row Ball Brgs.	68					69 79							160	60 70			
Double-Row Ball Brgs.																	
Cylindrical Roller Brgs.		N28	NN38	NN48		N19	N29	NN39	NN49					N10	N20	NN30	NN40
Needle Roller Brgs.				NA48					NA49	NA59	NA69						
Spherical Roller Brgs.								239								230	240

Diameter Series 7

Bore Number	d	D	B 17	B 27	B 37	r (min.) 17~37
—	0.6	2	0.8	—	—	0.05
1	1	2.5	1	—	—	0.05
—	1.5	3	1	—	1.8	0.05
2	2	4	1.2	—	2	0.05
—	2.5	5	1.5	—	2.3	0.08
3	3	6	2	2.5	3	0.08
4	4	7	2	2.5	3	0.08
5	5	8	2	2.5	3	0.08
6	6	10	2.5	3	3.5	0.1
7	7	11	2.5	3	3.5	0.1
8	8	12	2.5	—	3.5	0.1
9	9	14	3	—	4.5	0.1
00	10	15	3	—	4.5	0.1
01	12	18	4	—	5	0.2
02	15	21	4	—	5	0.2
03	17	23	4	—	5	0.2
04	20	27	4	—	5	0.2
/22	22	—	—	—	—	—
05	25	32	4	—	5	0.2
/28	28	—	—	—	—	—
06	30	37	4	—	5	0.2
/32	32	—	—	—	—	—
07	35	—	—	—	—	—
08	40	—	—	—	—	—
09	45	—	—	—	—	—
10	50	—	—	—	—	—
11	55	—	—	—	—	—
12	60	—	—	—	—	—
13	65	—	—	—	—	—
14	70	—	—	—	—	—
15	75	—	—	—	—	—
16	80	—	—	—	—	—
17	85	—	—	—	—	—
18	90	—	—	—	—	—
19	95	—	—	—	—	—
20	100	—	—	—	—	—
21	105	—	—	—	—	—
22	110	—	—	—	—	—
24	120	—	—	—	—	—
26	130	—	—	—	—	—
28	140	—	—	—	—	—
30	150	—	—	—	—	—
32	160	—	—	—	—	—
34	170	—	—	—	—	—
36	180	—	—	—	—	—
38	190	—	—	—	—	—
40	200	—	—	—	—	—
44	220	—	—	—	—	—
48	240	—	—	—	—	—
52	260	—	—	—	—	—
56	280	—	—	—	—	—
60	300	—	—	—	—	—
64	320	—	—	—	—	—
68	340	—	—	—	—	—
72	360	—	—	—	—	—
76	380	—	—	—	—	—
80	400	—	—	—	—	—
84	420	—	—	—	—	—
88	440	—	—	—	—	—
92	460	—	—	—	—	—
96	480	—	—	—	—	—
/500	500	—	—	—	—	—
/530	530	—	—	—	—	—
/560	560	—	—	—	—	—
/600	600	—	—	—	—	—
/630	630	—	—	—	—	—
/670	670	—	—	—	—	—
/710	710	—	—	—	—	—
/750	750	—	—	—	—	—
/800	800	—	—	—	—	—
/850	850	—	—	—	—	—
/900	900	—	—	—	—	—
/950	950	—	—	—	—	—
/1000	1000	—	—	—	—	—
/1060	1060	—	—	—	—	—
/1120	1120	—	—	—	—	—
/1180	1180	—	—	—	—	—
/1250	1250	—	—	—	—	—
/1320	1320	—	—	—	—	—
/1400	1400	—	—	—	—	—
/1500	1500	—	—	—	—	—
/1600	1600	—	—	—	—	—
/1700	1700	—	—	—	—	—
/1800	1800	—	—	—	—	—
/1900	1900	—	—	—	—	—
/2000	2000	—	—	—	—	—

Diameter Series 8

Bore Number	d	D	B 08	B 18	B 28	B 38	B 48	B 58	B 68	r (min.) 08	r (min.) 18~68
—	0.6	2.5	—	1	—	1.4	—	—	—	—	0.05
1	1	3	—	1	—	1.5	—	—	—	—	0.05
—	1.5	4	—	1.2	—	2	—	—	—	—	0.05
2	2	5	—	1.5	—	2.3	—	—	—	—	0.08
—	2.5	6	—	1.8	—	2.6	—	—	—	—	0.08
3	3	7	—	2	—	3	—	—	—	—	0.1
4	4	9	—	2.5	3.5	4	—	—	—	—	0.1
5	5	11	—	3	4	5	—	—	—	—	0.15
6	6	13	—	3.5	5	6	—	—	—	—	0.15
7	7	14	—	3.5	5	6	—	—	—	—	0.15
8	8	16	—	4	5	6	8	—	—	—	0.2
9	9	17	—	4	5	6	8	—	—	—	0.2
00	10	19	—	5	6	7	9	—	—	—	0.3
01	12	21	—	5	6	7	9	—	—	—	0.3
02	15	24	—	5	6	7	9	—	—	—	0.3
03	17	26	—	5	6	7	9	—	—	—	0.3
04	20	32	4	7	8	10	12	16	22	0.3	0.3
/22	22	34	4	7	—	10	—	16	22	0.3	0.3
05	25	37	4	7	8	10	12	16	22	0.3	0.3
/28	28	40	4	7	—	10	—	16	22	0.3	0.3
06	30	42	4	7	8	10	12	16	22	0.3	0.3
/32	32	44	4	7	—	10	—	16	22	0.3	0.3
07	35	47	4	7	8	10	12	16	22	0.3	0.3
08	40	52	4	7	8	10	12	16	22	0.3	0.3
09	45	58	4	7	8	10	13	18	23	0.3	0.3
10	50	65	5	7	10	12	15	20	27	0.3	0.3
11	55	72	7	9	11	13	17	23	30	0.3	0.3
12	60	78	7	10	12	14	18	24	32	0.3	0.3
13	65	85	7	10	13	15	20	27	36	0.3	0.6
14	70	90	8	10	13	15	20	27	36	0.3	0.6
15	75	95	8	10	13	15	20	27	36	0.3	0.6
16	80	100	8	10	13	15	20	27	36	0.3	0.6
17	85	110	9	13	16	19	25	34	45	0.3	1
18	90	115	9	13	16	19	25	34	45	0.3	1
19	95	120	9	13	16	19	25	34	45	0.3	1
20	100	125	9	13	16	19	25	34	45	0.3	1
21	105	130	9	13	16	19	25	34	45	0.3	1
22	110	140	10	16	19	23	30	40	54	0.6	1
24	120	150	10	16	19	23	30	40	54	0.6	1
26	130	165	11	18	22	26	35	46	63	0.6	1.1
28	140	175	11	18	22	26	35	46	63	0.6	1.1
30	150	190	13	20	24	30	40	54	71	0.6	1.1
32	160	200	13	20	24	30	40	54	71	0.6	1.1
34	170	215	14	22	27	34	45	60	80	0.6	1.1
36	180	225	14	22	27	34	45	60	80	0.6	1.1
38	190	240	16	24	30	37	50	67	90	1	1.5
40	200	250	16	24	30	37	50	67	90	1	1.5
44	220	270	16	24	30	37	50	67	90	1	1.5
48	240	300	19	28	36	45	60	80	109	1	2
52	260	320	19	28	36	45	60	80	109	1	2
56	280	350	22	33	42	52	69	95	125	1.1	2
60	300	380	25	38	48	60	80	109	145	1.5	2.1
64	320	400	25	38	48	60	80	109	145	1.5	2.1
68	340	420	25	38	48	60	80	109	145	1.5	2.1
72	360	440	25	38	48	60	80	109	145	1.5	2.1
76	380	480	31	46	60	75	100	136	180	2	2.1
80	400	500	31	46	60	75	100	136	180	2	2.1
84	420	520	31	46	60	75	100	136	180	2	2.1
88	440	540	31	46	60	75	100	136	180	2	2.1
92	460	580	37	56	72	90	118	160	218	2.1	3
96	480	600	37	56	72	90	118	160	218	2.1	3
/500	500	620	37	56	72	90	118	160	218	2.1	3
/530	530	650	37	56	72	90	118	160	218	2.1	3
/560	560	680	37	56	72	90	118	160	218	2.1	3
/600	600	730	42	60	78	98	128	175	236	3	3
/630	630	780	48	69	88	112	150	200	272	3	4
/670	670	820	48	69	88	112	150	200	272	3	4
/710	710	870	50	74	95	118	160	218	290	4	4
/750	750	920	54	78	100	128	170	230	308	4	5
/800	800	980	57	82	106	136	180	243	325	4	5
/850	850	1030	57	82	106	136	180	243	325	4	5
/900	900	1090	60	85	112	140	190	258	345	5	5
/950	950	1150	63	90	118	150	200	272	355	5	5
/1000	1000	1220	71	100	128	165	218	300	400	5	6
/1060	1060	1280	71	100	128	165	218	300	400	5	6
/1120	1120	1360	78	106	140	180	243	325	438	5	6
/1180	1180	1420	78	106	140	180	243	325	438	5	6
/1250	1250	1500	80	112	145	185	250	335	450	6	6
/1320	1320	1600	88	122	165	206	280	375	500	6	6
/1400	1400	1700	95	132	175	224	300	400	545	6	7.5
/1500	1500	1820	—	140	185	243	315	—	—	—	7.5
/1600	1600	1950	—	155	200	265	345	—	—	—	7.5
/1700	1700	2060	—	160	206	272	355	—	—	—	7.5
/1800	1800	2180	—	165	218	290	375	—	—	—	9.5
/1900	1900	2300	—	175	230	300	400	—	—	—	9.5
/2000	2000	2430	—	190	250	325	425	—	—	—	9.5

Diameter Series 9

Bore Number	d	D	B 09	B 19	B 29	B 39	B 49	B 59	B 69	r (min.) 09	r (min.) 19~39	r (min.) 49~69
—	0.6	—	—	—	—	—	—	—	—	—	—	—
1	1	4	—	1.6	—	2.3	—	—	—	—	0.1	—
—	1.5	5	—	2	—	2.6	—	—	—	—	0.15	—
2	2	6	—	2.3	—	3	—	—	—	—	0.15	—
—	2.5	7	—	2.5	—	3.5	—	—	—	—	0.15	—
3	3	8	—	3	—	4	—	—	—	—	0.15	—
4	4	11	—	4	—	5	—	—	—	—	0.15	—
5	5	13	—	4	—	6	10	—	—	—	0.2	0.15
6	6	15	—	5	—	7	10	—	—	—	0.2	0.15
7	7	17	—	5	—	7	10	—	—	—	0.3	0.15
8	8	19	—	6	—	9	11	—	—	—	0.3	0.2
9	9	20	—	6	—	9	11	—	—	—	0.3	0.3
00	10	22	—	6	8	10	13	16	22	—	0.3	0.3
01	12	24	—	6	8	10	13	16	22	—	0.3	0.3
02	15	28	—	7	8.5	10	13	18	23	—	0.3	0.3
03	17	30	—	7	8.5	10	13	18	23	—	0.3	0.3
04	20	37	7	9	11	13	17	23	30	0.3	0.3	0.3
/22	22	39	7	9	11	13	17	23	30	0.3	0.3	0.3
05	25	42	7	9	11	13	17	23	30	0.3	0.3	0.3
/28	28	45	7	9	11	13	17	23	30	0.3	0.3	0.3
06	30	47	7	9	11	13	17	23	30	0.3	0.3	0.3
/32	32	52	7	10	13	15	20	27	36	0.3	0.6	0.6
07	35	55	7	10	13	15	20	27	36	0.3	0.6	0.6
08	40	62	8	12	14	16	22	30	40	0.3	0.6	0.6
09	45	68	8	12	14	16	22	30	40	0.3	0.6	0.6
10	50	72	8	12	14	16	22	30	40	0.3	0.6	0.6
11	55	80	9	13	16	19	25	34	45	0.3	1	1
12	60	85	9	13	16	19	25	34	45	0.3	1	1
13	65	90	9	13	16	19	25	34	45	0.3	1	1
14	70	100	10	16	19	23	30	40	54	0.6	1	1
15	75	105	10	16	19	23	30	40	54	0.6	1	1
16	80	110	10	16	19	23	30	40	54	0.6	1	1
17	85	120	11	18	22	26	35	46	63	0.6	1.1	1.1
18	90	125	11	18	22	26	35	46	63	0.6	1.1	1.1
19	95	130	11	18	22	26	35	46	63	0.6	1.1	1.1
20	100	140	13	20	24	30	40	54	71	0.6	1.1	1.1
21	105	145	13	20	24	30	40	54	71	0.6	1.1	1.1
22	110	150	13	20	24	30	40	54	71	0.6	1.1	1.1
24	120	165	14	22	27	34	45	60	80	0.6	1.1	1.1
26	130	180	16	24	30	37	50	67	90	1	1.5	1.5
28	140	190	16	24	30	37	50	67	90	1	1.5	1.5
30	150	210	19	28	36	45	60	80	109	1	2	2
32	160	220	19	28	36	45	60	80	109	1	2	2
34	170	230	19	28	36	45	60	80	109	1	2	2
36	180	250	22	33	42	52	69	95	125	1.1	2	2
38	190	260	22	33	42	52	69	95	125	1.1	2	2
40	200	280	25	38	48	60	80	109	145	1.5	2.1	2.1
44	220	300	25	38	48	60	80	109	145	1.5	2.1	2.1
48	240	320	25	38	48	60	80	109	145	1.5	2.1	2.1
52	260	360	31	46	60	75	100	136	180	2	2.1	2.1
56	280	380	31	46	60	75	100	136	180	2	2.1	2.1
60	300	420	37	56	72	90	118	160	218	2.1	3	3
64	320	440	37	56	72	90	118	160	218	2.1	3	3
68	340	460	37	56	72	90	118	160	218	2.1	3	3
72	360	480	37	56	72	90	118	160	218	2.1	3	3
76	380	520	44	65	82	106	140	190	250	3	4	4
80	400	540	44	65	82	106	140	190	250	3	4	4
84	420	560	44	65	82	106	140	190	250	3	4	4
88	440	600	50	74	95	118	160	218	290	4	4	4
92	460	620	50	74	95	118	160	218	290	4	4	4
96	480	650	54	78	100	128	170	230	308	4	5	5
/500	500	670	54	78	100	128	170	230	308	4	5	5
/530	530	710	57	82	106	136	180	243	325	4	5	5
/560	560	750	60	85	112	140	190	258	345	5	5	5
/600	600	800	63	90	118	150	200	272	355	5	5	5
/630	630	850	71	100	128	165	218	300	400	5	6	6
/670	670	900	73	103	136	170	230	308	412	5	6	6
/710	710	950	78	106	140	180	243	325	438	5	6	6
/750	750	1000	80	112	145	185	250	335	450	6	6	6
/800	800	1060	82	115	150	195	258	355	462	6	6	6
/850	850	1120	85	118	155	200	272	365	488	6	6	6
/900	900	1180	88	122	165	206	280	375	500	6	6	6
/950	950	1250	95	132	175	224	300	400	545	6	7.5	7.5
/1000	1000	1320	103	140	185	236	315	438	580	6	7.5	7.5
/1060	1060	1400	109	150	195	250	335	462	615	7.5	7.5	7.5
/1120	1120	1460	109	150	195	250	335	462	615	7.5	7.5	7.5
/1180	1180	1540	115	160	206	272	355	488	650	7.5	7.5	7.5
/1250	1250	1630	122	170	218	280	375	515	690	7.5	7.5	7.5
/1320	1320	1720	128	175	230	300	400	545	710	7.5	7.5	7.5
/1400	1400	1820	—	185	243	315	425	—	—	—	9.5	9.5
/1500	1500	1950	—	195	258	335	450	—	—	—	9.5	9.5
/1600	1600	2060	—	200	265	345	462	—	—	—	9.5	9.5
/1700	1700	2180	—	212	280	355	475	—	—	—	9.5	9.5
/1800	1800	2300	—	218	290	375	500	—	—	—	12	12
/1900	1900	2430	—	230	308	400	530	—	—	—	12	12
/2000	2000	—	—	—	—	—	—	—	—	—	—	—

Diameter Series 0

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

Remarks The chamfer dimensions listed in this table do not necessarily apply to the following chamfers: (a) Chamfers of the grooves in outer rings that have snap ring grooves.
(b) For thin section cylindrical roller bearings, the chamfers on side without rib and bearing bore (in case of an inner ring) or outer surface (in case of an outer ring).
(c) For angular contact ball bearings, the chamfers between the front face and bore (in case of an inner ring) or outer surface (in case of an outer ring).
(d) Chamfers on inner rings of bearings with tapered bores.

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

Units: mm

Single-Row Ball Brgs.												62 72		622	632						63 73		623	633				64 74		
Double-Row Ball Brgs.												12		42 22	52 53						13		43 23	53 33				104		
Cylindrical Roller Brgs.						NN 31						N 2		N 22	N 32						N 3		N 23	N 33				N 4		
Needle Roller Brgs.																														
Spherical Roller Brgs.						231	241							222	232						213		223							
Bore Number	*d*	Diameter Series 1								Diameter Series 2									Diameter Series 3								Diameter Series 4			
		D	Dimension Series *B* 01	11	21	31	41	Dimension Series *r* (min.) 01	11~41	*D*	Dimension Series *B* 82	02	12	22	32	42	Dimension Series *r* (min.) 82	02~42	*D*	Dimension Series *B* 83	03	13	23	33	Dimension Series *r* (min.) 83	03~33	*D*	Dimension Series *B* 04	24	Dimension Series *r* (min.) 04~24
—	0.6	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1	1	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
—	1.5	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2	2	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
—	2.5	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
3	3	—	—	—	—	—	—	—	—	10	2.5	4	—	—	5	—	0.1	0.15	13	—	5	—	—	7	—	0.2	—	—	—	—
4	4	—	—	—	—	—	—	—	—	13	3	5	—	—	7	—	0.15	0.2	16	—	5	—	—	9	—	0.3	—	—	—	—
5	5	—	—	—	—	7	—	—	—	16	3.5	5	—	—	8	—	0.15	0.3	19	—	6	—	—	10	—	0.3	—	—	—	—
6	6	—	—	—	8	10	—	—	—	19	4	6	—	—	10	—	0.2	0.3	22	—	7	—	11	13	—	0.3	—	—	—	—
7	7	—	—	—	9	11	14	—	—	22	5	7	—	—	11	—	0.3	0.3	26	—	9	—	13	15	—	0.3	—	—	—	—
8	8	—	—	—	10	12	15	—	—	24	5	8	—	—	12	—	0.3	0.3	28	—	9	—	13	15	—	0.3	30	10	14	0.6
9	9	—	—	—	10	12	16	—	—	26	6	8	—	—	13	—	0.3	0.3	30	—	10	—	14	16	—	0.6	32	11	15	0.6
00	10	—	—	—	12	14	18	—	—	30	7	9	—	14	14.3	—	0.3	0.6	35	9	11	—	17	19	0.3	0.6	37	12	16	0.6
01	12	—	—	—	12	14	18	—	—	32	7	10	—	14	15.9	—	0.3	0.6	37	9	12	—	17	19	0.3	1	42	13	19	1
02	15	—	—	—	12	14	18	—	—	35	8	11	—	14	15.9	20	0.3	0.6	42	9	13	—	17	19	0.3	1	52	15	24	1.1
03	17	—	—	—	13	15	20	—	—	40	8	12	—	16	17.5	22	0.3	0.6	47	10	14	—	19	22.2	0.6	1	62	17	29	1.1
04	20	—	—	—	15	18	24	—	—	47	9	14	—	18	20.6	27	0.3	1	52	10	15	—	21	22.2	0.6	1.1	72	19	33	1.1
/22	22	—	—	—	16	19	25	—	—	50	9	14	—	18	20.6	27	0.3	1	56	11	16	—	21	25	0.6	1.1	—	—	—	—
05	25	—	—	—	16	19	25	—	—	52	10	15	—	18	20.6	27	0.3	1	62	12	17	—	24	25.4	0.6	1.1	80	21	36	1.5
/28	28	—	—	—	17	20	27	—	—	58	10	16	—	19	23	30	0.6	1	68	13	18	—	24	30	0.6	1.1	—	—	—	—
06	30	—	—	—	18	21	28	—	—	62	10	16	—	20	23.8	32	0.6	1	72	13	19	—	27	30.2	0.6	1.1	90	23	40	1.5
/32	32	—	—	—	19	23	30	—	—	65	11	17	—	21	25	33	0.6	1	75	14	20	—	28	32	0.6	1.1	—	—	—	—
07	35	—	—	—	21	25	33	—	—	72	12	17	—	23	27	37	0.6	1.1	80	14	21	—	31	34.9	0.6	1.5	100	25	43	1.5
08	40	—	—	—	22	26	35	—	—	80	13	18	—	23	30.2	40	0.6	1.1	90	16	23	—	33	36.5	1	1.5	110	27	46	2
09	45	—	—	—	22	26	35	—	—	85	13	19	—	23	30.2	40	0.6	1.1	100	17	25	—	36	39.7	1	1.5	120	29	50	2
10	50	—	—	—	22	26	35	—	—	90	13	20	—	23	30.2	40	0.6	1.1	110	19	27	—	40	44.4	1	2	130	31	53	2.1
11	55	—	—	—	24	30	40	—	—	100	14	21	—	25	33.3	45	0.6	1.5	120	21	29	—	43	49.2	1.1	2	140	33	57	2.1
12	60	—	—	—	24	30	40	—	—	110	16	22	—	28	36.5	50	1	1.5	130	22	31	—	46	54	1.1	2.1	150	35	60	2.1
13	65	—	—	—	27	34	45	—	—	120	18	23	—	31	38.1	56	1	1.5	140	24	33	—	48	58.7	1.1	2.1	160	37	64	2.1
14	70	—	—	—	27	34	45	—	—	125	18	24	—	31	39.7	56	1	1.5	150	25	35	—	51	63.5	1.5	2.1	180	42	74	3
15	75	—	—	—	30	37	50	—	—	130	18	25	—	31	41.3	56	1	1.5	160	27	37	—	55	68.3	1.5	2.1	190	45	77	3
16	80	—	—	—	30	37	50	—	—	140	19	26	—	33	44.4	60	1	2	170	28	39	—	58	68.3	1.5	2.1	200	48	80	3
17	85	—	—	—	31	41	56	—	—	150	21	28	—	36	49.2	65	1.1	2	180	30	41	—	60	73	2	3	210	52	86	4
18	90	150	—	—	33	45	60	—	2	160	22	30	—	40	52.4	69	1.1	2	190	30	43	—	64	73	2	3	225	54	90	4
19	95	160	—	—	39	52	65	—	2	170	24	32	—	43	55.6	75	1.1	2.1	200	33	45	—	67	77.8	2	3	240	55	95	4
20	100	165	21	30	39	52	65	1.1	2	180	25	34	—	46	60.3	80	1.5	2.1	215	36	47	51	73	82.6	2.1	3	250	58	98	4
21	105	175	22	33	42	56	69	1.1	2	190	27	36	—	50	65.1	85	1.5	2.1	225	37	49	53	77	87.3	2.1	3	260	60	100	4
22	110	180	22	33	42	56	69	1.1	2	200	28	38	—	53	69.8	90	1.5	2.1	240	42	50	57	80	92.1	3	3	280	65	108	4
24	120	200	25	38	48	62	80	1.5	2	215	—	40	42	58	76	95	—	2.1	260	44	55	62	86	106	3	3	310	72	118	5
26	130	210	25	38	48	64	80	1.5	2	230	—	40	46	64	80	100	—	3	280	48	58	66	93	112	3	4	340	78	128	5
28	140	225	27	40	50	68	85	1.5	2.1	250	—	42	50	68	88	109	—	3	300	50	62	70	102	118	4	4	360	82	132	5
30	150	250	31	46	60	80	100	2	2.1	270	—	45	54	73	96	118	—	3	320	—	65	75	108	128	—	4	380	85	138	5
32	160	270	34	51	66	86	109	2	2.1	290	—	48	58	80	104	128	—	3	340	—	68	79	114	136	—	4	400	88	142	5
34	170	280	34	51	66	88	109	2	2.1	310	—	52	62	86	110	140	—	4	360	—	72	84	120	140	—	4	420	92	145	5
36	180	300	37	56	72	96	118	2.1	3	320	—	52	62	86	112	140	—	4	380	—	75	88	126	150	—	4	440	95	150	6
38	190	320	42	60	78	104	128	3	3	340	—	55	65	92	120	150	—	4	400	—	78	92	132	155	—	5	460	98	155	6
40	200	340	44	65	82	112	140	3	3	360	—	58	70	98	128	160	—	4	420	—	80	97	138	165	—	5	480	102	160	6
44	220	370	48	69	88	120	150	3	4	400	—	65	78	108	144	180	—	4	460	—	88	106	145	180	—	5	540	115	180	6
48	240	400	50	74	95	128	160	4	4	440	—	72	85	120	160	200	—	4	500	—	95	114	155	195	—	5	580	122	190	6
52	260	440	57	82	106	144	180	4	4	480	—	80	90	130	174	218	—	5	540	—	102	123	165	206	—	6	620	132	206	7.5
56	280	460	57	82	106	146	180	4	5	500	—	80	90	130	176	218	—	5	580	—	108	132	175	224	—	6	670	140	224	7.5
60	300	500	63	90	118	160	200	5	5	540	—	85	98	140	192	243	—	5	620	—	109	140	185	236	—	7.5	710	150	236	7.5
64	320	540	71	100	128	176	218	5	5	580	—	92	105	150	208	258	—	5	670	—	112	155	200	258	—	7.5	750	155	250	9.5
68	340	580	78	106	140	190	243	5	5	620	—	92	118	165	224	280	—	6	710	—	118	165	212	272	—	7.5	800	165	265	9.5
72	360	600	78	106	140	192	243	5	5	650	—	95	122	170	232	290	—	6	750	—	125	170	224	290	—	7.5	850	180	280	9.5
76	380	620	78	106	140	194	243	5	5	680	—	95	132	175	240	300	—	6	780	—	128	175	230	300	—	7.5	900	190	300	9.5
80	400	650	80	112	145	200	250	6	6	720	—	103	140	185	256	315	—	6	820	—	136	185	243	308	—	7.5	950	200	315	12
84	420	700	88	122	165	224	280	6	6	760	—	109	150	195	272	335	—	7.5	850	—	136	190	250	315	—	9.5	980	206	325	12
88	440	720	88	122	165	226	280	6	6	790	—	112	155	200	280	345	—	7.5	900	—	145	200	265	345	—	9.5	1030	212	335	12
92	460	760	95	132	175	240	300	6	7.5	830	—	118	165	212	296	365	—	7.5	950	—	155	212	280	365	—	9.5	1060	218	345	12
96	480	790	100	136	180	248	308	6	7.5	870	—	125	170	224	310	388	—	7.5	980	—	160	218	290	375	—	9.5	1120	230	365	15
/500	500	830	106	145	190	264	325	7.5	7.5	920	—	136	185	243	336	412	—	7.5	1030	—	170	230	300	388	—	12	1150	236	375	15
/530	530	870	109	150	195	272	335	7.5	7.5	980	—	145	200	258	355	450	—	9.5	1090	—	180	243	325	412	—	12	1220	250	400	15
/560	560	920	115	160	206	280	355	7.5	7.5	1030	—	150	206	272	365	475	—	9.5	1150	—	190	258	335	438	—	12	1280	258	412	15
/600	600	980	122	170	218	300	375	7.5	7.5	1090	—	155	212	280	388	488	—	9.5	1220	—	200	272	355	462	—	15	1360	272	438	15
/630	630	1030	128	175	230	315	400	7.5	7.5	1150	—	165	230	300	412	515	—	12	1280	—	206	280	375	488	—	15	1420	280	450	15
/670	670	1090	136	185	243	336	412	7.5	7.5	1220	—	175	243	315	438	545	—	12	1360	—	218	300	400	515	—	15	1500	290	475	15
/710	710	1150	140	195	250	345	438	9.5	9.5	1280	—	180	250	325	450	560	—	12	1420	—	224	308	412	530	—	15	—	—	—	—
/750	750	1220	150	206	272	365	475	9.5	9.5	1360	—	195	265	345	475	615	—	15	1500	—	236	325	438	560	—	15	—	—	—	—
/800	800	1280	155	212	272	375	475	9.5	9.5	1420	—	200	272	355	488	615	—	15	1600	—	258	355	462	600	—	15	—	—	—	—
/850	850	1360	165	224	290	400	500	12	12	1500	—	206	280	375	515	650	—	15	1700	—	272	375	488	630	—	19	—	—	—	—
/900	900	1420	165	230	300	412	515	12	12	1580	—	218	300	388	515	670	—	15	1780	—	280	388	500	650	—	19	—	—	—	—
/950	950	1500	175	243	315	438	545	12	12	1660	—	230	315	412	530	710	—	15	1850	—	290	400	515	670	—	19	—	—	—	—
/1000	1000	1580	185	258	335	462	580	12	12	1750	—	243	330	425	560	750	—	15	1950	—	300	412	545	710	—	19	—	—	—	—
/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	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

Remarks The chamfer dimensions listed in this table do not necessarily apply to the following chamfers: (a) Chamfers of the grooves in outer rings that have snap ring grooves.
(b) For thin section cylindrical roller bearings, the chamfers on side without rib and bearing bore (in case of an inner ring) or outer surface (in case of an outer ring).
(c) For angular contact ball bearings, the chamfers between the front face and bore (in case of an inner ring) or outer surface (in case of an outer ring).
(d) Chamfers on inner rings of bearings with tapered bores.

■BOUNDARY DIMENSIONS AND IDENTIFYING NUMBERS FOR BEARINGS

Table 6. 2 Boundary Dimensions of Tapered Roller Bearings

Units: mm

Tapered Roller Brgs.		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			Cone	Cup								Cone	Cup					Cone	Cup
Bore Number	*d*	*D*	*B*	*C*	*T*	*B*	*C*	*T*	*r* (min.)		*D*	*B*	*C*	*T*	*B*	*C*	*T*	*r* (min.)		*D*	*B*	*C*	*T*	*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 the Dimension Series of Diameter Series 9, Classification I is those specified by the old standard, Classification II is those specified by the ISO.
   Dimension Series not classified 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 Brgs.	
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			
										Cone	Cup												Cone	Cup		
D	*B*	*C*	*T*	*B*	*C*	*T*	*B*	*C*	*T*	*r* (min.)		*D*	*B*	*C*	*C* (1)	*T*	*B*	*C*	*T*	*B*	*C*	*T*	*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) Regarding steep-slope bearing 303D, in DIN, the one corresponding to 303D of JIS is numbered 313. For bearings with bore diameters larger than 100 mm, those of dimension series 13 are numbered 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	T	T	r (min.)	D	T	T	T	r (min.)	D	T	T	T	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	—

		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	T	T	T	d_2	B	r (min.)	r_1 (min.)	D	T	T	T	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

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).

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

Units: mm

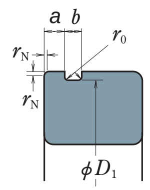
Thrust Ball Brgs.										511					512	522				
Spherical Thrust Roller Brgs.														292						
		Diameter Series 0					Diameter Series 1					Diameter Series 2								
			Dimension Series					Dimension Series					Dimension Series							
			70	90	10			71	91	11			72	92	12	22	22 Central Washer			
Bore Number	d	D	T			r (min.)	D	T			r (min.)	D	T				d_2	B	r (min.)	r_1 (min.)
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	—	—	—	—	—	—	—	—	—

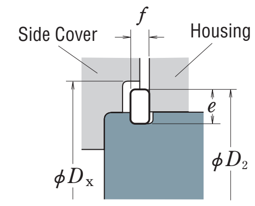
			513	523								514	524									Thrust Ball Brgs.		
		293									294											Spherical Thrust Roller Brgs.		
Diameter Series 3									Diameter Series 4									Diameter Series 5						
	Dimension Series									Dimension Series									Dimension Series					
	73	93	13	23	23 Central Washer					74	94	14	24	24 Central Washer					95					
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		
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		

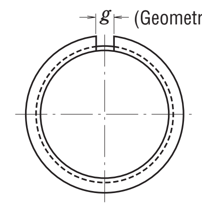
Remarks 1. Dimension Series 22, 23, and 24 are double direction bearings.
2. The maximum permissible outside diameter of shaft and central washers and minimum permissible bore diameter of housing washers are omitted here. (Refer to the bearings tables for Thrust Bearings).

BOUNDARY DIMENSIONS AND IDENTIFYING NUMBERS FOR BEARINGS

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







Units: mm

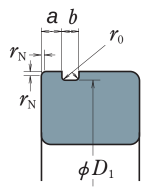
Applicable Bearings			Snap Ring Groove								
d		D	Snap Ring Groove Diameter D_1		Snap Ring Groove Position a				Snap Ring Groove Width b		Radius of Bottom Corners r_0
Dimension Series					Bearing Dimension Series						
					18		19				
18	19		max.	min.	max.	min.	max.	min.	max.	min.	max.
—	10	**22**	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

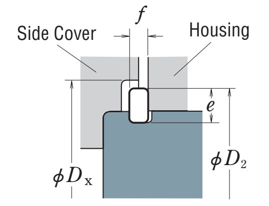
Remarks The minimum permissible chamfer dimensions r_N on the snap-ring-groove side of the outer rings are as follows:
Dimension series 18 : For outside diameters of 78mm and less, use 0.3mm chamfer.
For all others exceeding 78mm, use 0.5mm chamfer.
Dimension series 19 : For outside diameters of 24mm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

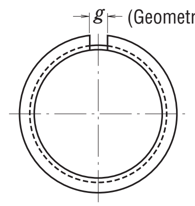
Locating Snap Ring							Side Cover
Locating Snap Ring Number	Cross Sectional Height e		Thickness f		Geometry of snap ring fitted in groove (Reference)		Stepped Bore Diameter (Reference) D_X
					Slit Width g	Snap Ring Outside Diameter D_2	
	max.	min.	max.	min.	approx.	max.	min.
NR 1022	2.0	1.85	0.7	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 IDENTIFYING NUMBERS FOR BEARINGS

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







Units: mm

Applicable Bearings: d, Diameter Series 0	2	3	4	D	Snap Ring Groove Diameter D_1 max.	D_1 min.	Snap Ring Groove Position a, Bearing Diameter Series 0 max.	0 min.	2, 3, 4 max.	2, 3, 4 min.	Snap Ring Groove Width b max.	b min.	Radius of Bottom Corners r_0 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 Number	Cross Sectional Height e max.	e min.	Thickness f max.	f min.	Geometry of snap ring fitted in groove (Reference): Slit Width g approx.	Snap Ring Outside Diameter D_2 max.	Side Cover: Stepped Bore Diameter (Reference) D_X 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 0.5mm. However, for bearings of diameter series 0 having outside diameters 35mm and below, it is 0.3mm.

6.2 Formulation of Bearing Numbers

Bearing numbers are alphanumeric combinations that indicate the bearing type, boundary dimensions, dimensional and running accuracies, internal clearance, and other related specifications. They consist of basic numbers and supplementary symbols. 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 (Bearing Numbers for Rolling Bearings). Due to a need for more detailed classification, NSK uses auxiliary symbols other than those specified by JIS.

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

(Example 1) **6 3 0 8 ZZ C3**

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

(Example 2) **7 2 2 0 A DB C3**

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

(Example 3) **1 2 0 6 K +H206X**

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

(Example 4) **NU 3 1 8 M CM**

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

(Example 5) **NN 3 0 1 7 K CC1 P4**

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

(Example 6) **HR 3 0 2 0 7 J**

- HR — High Capacity Bearing
- 3 — Tapered Roller Bearing
- 0 — Width Series **0**
- 2 — Diameter Series **2**
- 07 — Bearing Bore **35mm**
- J — Small Diameter of Cup Raceway and Contact Angle Conform to ISO

(Example 7) **2 4 0 /1000 CA M K30 E4 C3**

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

(Example 8) **5 1 2 1 5**

- 5 — Thrust Ball Bearing
- 1 — Height Series **1**
- 2 — Diameter Series **2**
- 15 — Bearing Bore **75mm**

Table 6. 5 Bearing Series Symbols

Bearing Type	Bearing Series Symbols	Type Symbols	Dimension Symbols: Width Symbols	Dimension Symbols: Diameter Symbols
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 Symbols	Type Symbols	Dimension Symbols: Width Symbols or Height Symbols	Dimension Symbols: Diameter Symbols
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 Symbol 213 should logically be 203, but customarily it is numbered 213.
Remark Numbers in ( ) in the column of width symbols are usually omitted from the bearing number.

Table 6. 6 Formulation of Bearing Numbers

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

Bearing Series Symbols (1)

Symbol	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

Symbol	Meaning
1	Bearing Bore 1mm
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 Symbol

Symbol	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	Contact Angle about 20°
D	Contact Angle about 28°

NSK Symbol (Marked on Bearings)

Internal Design Symbol

Symbol	Meaning
A	Internal Design Differs from Standard One
J	Smaller Diameter of Outer Ring Raceway, Contact Angle, and Outer Ring Width of Tapered Roller Bearings Conform 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 Symbol

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

NSK Symbol (Not Marked on Bearings)

Cage Symbol

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

External Features — Seals, Shields Symbol (NSK Symbol)

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

Auxiliary Symbols

Symbol for Design of Rings (Partially the same as JIS(5); In Principle, Marked on Bearings)

Symbol	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 Symbol (Same as JIS(5); In Principle, Marked on Bearings)

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

Internal Clearance Symbol / Preload Symbol (NSK Symbol; Partially the same as JIS(5)/BAS(6); In Principle, Marked on Bearings)

Symbol	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 Symbol (Same as JIS(5); In Principle, Marked on Bearings)

Symbol	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 Symbol (NSK Symbol, Partially the same as JIS(5); In Principle, Marked on Bearings)

Symbol	Meaning
	(Bearings treated for 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 or Sleeve Symbol (NSK Symbol, Partially the same as JIS(5); Not Marked on Bearings)

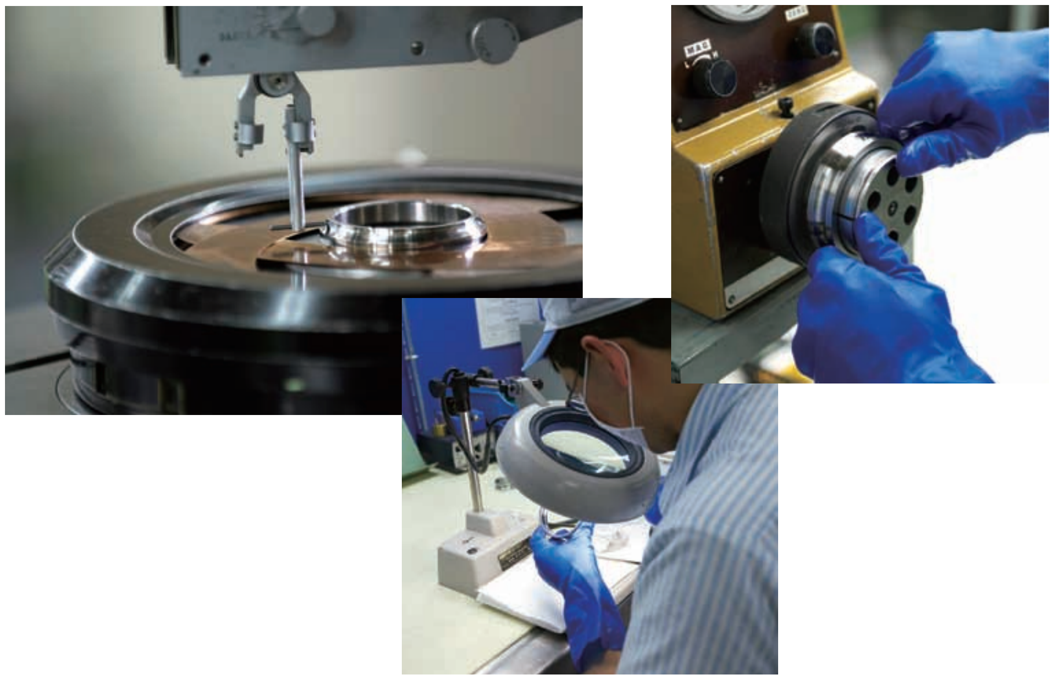
Symbol	Meaning
+K	Bearings with Outer Ring Spacers
+L	Bearings with Inner Ring Spacers
+KL	Bearings with Both Inner and Outer Ring Spacers
H	Adapter Designation
AH	Withdrawal Sleeve Designation
HJ	Thrust Collar Designation

Grease Symbol (NSK Symbol, Partially the same as JIS(5); Not Marked on Bearings)

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

Notes (1) Bearing Series Symbols conform to Table 6.5.
(2) For basic numbers of tapered roller bearings in ISO's new series, refer to Page C182.
(3) For Bearing Bore Numbers 04 through 96, five times the bore number gives the bore size (mm) (except double-direction thrust ball bearings).
(4) HR is prefix to bearing series symbols and it is NSK's original prefix.
(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 492/199/582 (Accuracies of Rolling Bearings). Tolerances are specified for the following items:

Regarding bearing accuracy classes, besides ISO normal accuracy, as the accuracy improves there are Class 6X (for tapered roller bearings), Class 6, Class 5, Class 4, and Class 2, with Class 2 being the highest in ISO. The applicable accuracy classes for each bearing type and the correspondence of these classes are shown in Table 7.1.

Accuracy of Rolling Bearings

- Tolerances for Dimensions (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 inner ring face runout with bore
  - · Permissible outer ring variation of outside surface generatrix inclination with face
  - · Permissible raceway to back face thickness variation of thrust bearings

Table 7. 1 Bearing Types and Tolerance Classes

Bearing Types		Applicable Tolerance Classes					Applicable Tables	Reference Pages
Deep Groove Ball Bearings		Normal	Class 6	Class 5	Class 4	Class 2	Table 7.2	A128 to A131
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 (solid type)		Normal	Class 6	Class 5	Class 4	—		
Spherical Roller Bearings		Normal	Class 6	Class 5	—	—		
Tapered Roller Bearings	Metric Design	Normal Class 6X	—	Class 5	Class 4	—	Table 7.3	A132 to A135
	Inch Design	ANSI/ABMA CLASS 4	ANSI/ABMA CLASS 2	ANSI/ABMA CLASS 3	ANSI/ABMA CLASS 0	ANSI/ABMA CLASS 00	Table 7.4	A136 and A137
Magneto Bearings		Normal	Class 6	Class 5	—	—	Table 7.5	A138 and A139
Thrust Ball Bearings		Normal	Class 6	Class 5	Class 4	—	Table 7.6	A140 to A142
Tapered Roller Thrust Bearings		Normal	—	—	—	—	Table 7.7	A143 and A144
Thrust Spherical Roller Bearings		Normal	—	—	—	—	Table 7.8	A145
Equivalent standards (Reference) JIS(1)		Class 0	Class 6	Class 5	Class 4	Class 2	—	—
Equivalent standards (Reference) DIN(2)		P0	P6	P5	P4	P2	—	—
Equivalent standards (Reference) ANSI/ABMA(3)	Ball Bearings	ABEC 1	ABEC 3	ABEC 5 (CLASS 5P)	ABEC 7 (CLASS 7P)	ABEC 9 (CLASS 9P)	Table 7.2	A128 to A131
	Roller Bearings	RBEC 1	RBEC 3	RBEC 5	—	—	[Table 7.9]	(A146 and A147)
	Tapered Roller Bearings	CLASS 4	CLASS 2	CLASS 3	CLASS 0	CLASS 00	[Table 7.4]	(A136 and A137)

Notes (1) JIS : Japanese Industrial Standards (2) DIN : Deutsch Industrie Norm
(3) ANSI/ABMA : The American Bearing Manufacturers Association

Remark The permissible limit of chamfer dimensions shall conform to Table 7.10 (Pages A148 and A149), and the tolerances and permissible tapered bore diameters shall conform to Table 7.11 (Pages A150 and A151).

(Reference) Rough definitions of the items listed for Running Accuracy and their measuring methods are shown in Fig. 7.1, and they are described in detail in ISO 5593 (Rolling Bearings-Vocabulary) and 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 or housing or central washer is to be rotated.		E

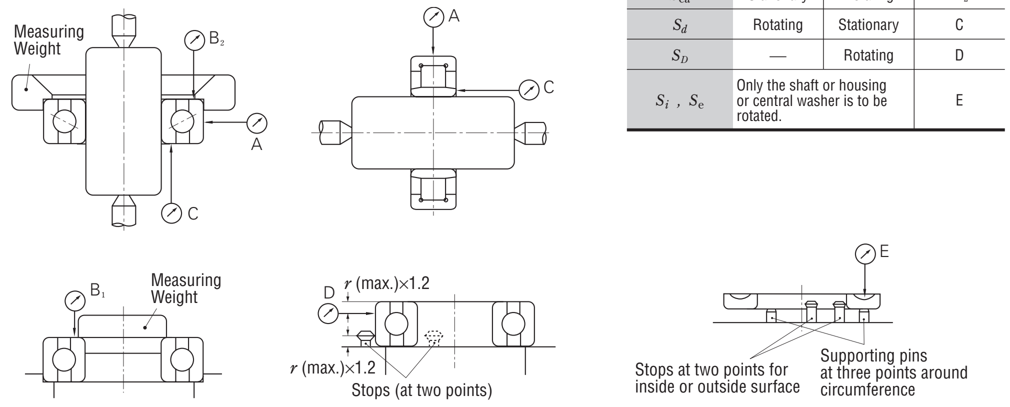


Fig. 7.1 Measuring Methods for Running Accuracy (summarized)

Symbols for Boundary Dimensions and Running Accuracy

d	Brg bore dia., nominal
Δ_{ds}	Deviation of a single bore dia.
Δ_{dmp}	Single plane mean bore dia. deviation
V_{dp}	Bore dia. Variation in a single radial plane
V_{dmp}	Mean bore dia. Variation
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 assembles brg inner ring
S_d	inner ring reference face (backface, where applicable) runout with bore
S_{ia}	Assembled brg inner ring face (back face) runout with raceway
S_i, S_e	Raceway to backface thickness variation of thrust brg
T	Brg width, nominal
Δ_{Ts}	Deviation of the actual brg width
D	Brg outside dia., nominal
Δ_{Ds}	Deviation of a single outside dia.
Δ_{Dmp}	Single plane mean outside dia. Deviation
V_{Dp}	Outside dia. Variation in a single radial plane
V_{Dmp}	Mean outside dia. Variation
C	Outer ring width, nominal
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	Variation of brg outside surface generatrix inclination with outer ring reference face (backface)
S_{ea}	Assembled brg outer ring face (backface) runout with raceway

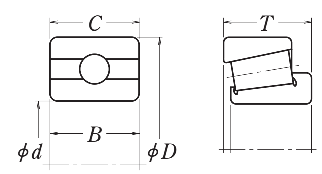


Table 7. 2 Tolerances for Radial Bearings (excluding Tapered Roller Bearings)

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 6		Class 5		Class 4		Class 2		Class 4 Diameter Series 0, 1, 2, 3, 4		Class 2	
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_{dp} (2)											V_{dmp} (2)				
		Normal Diameter Series			Class 6 Diameter Series			Class 5 Diameter Series		Class 4 Diameter Series		Class 2 Diameter Series	Normal	Class 6	Class 5	Class 4	Class 2
over	incl.	9	0, 1	2, 3, 4	9	0, 1	2, 3, 4	9	0,1,2,3,4	9	0,1,2,3,4	0,1,2,3,4					
		max.			max.			max.		max.		max.	max.	max.	max.	max.	max.
0.6(1)	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 (4)						Inner Ring (or Outer Ring) (3)	Inner Ring			
		Normal Class 6		Class 5 Class 4		Class 2		Normal Class 6		Class 5 Class 4		Class 2		Normal	Class 6	Class 5	Class 4	Class 2
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	max.	max.	max.	max.	max.
0.6(1)	2.5	0	− 40	0	− 40	0	− 40	−	−	0	−250	0	−250	12	12	5	2.5	1.5
2.5	10	0	− 120	0	− 40	0	− 40	0	−250	0	−250	0	−250	15	15	5	2.5	1.5
10	18	0	− 120	0	− 80	0	− 80	0	−250	0	−250	0	−250	20	20	5	2.5	1.5
18	30	0	− 120	0	−120	0	−120	0	−250	0	−250	0	−250	20	20	5	2.5	1.5
30	50	0	− 120	0	−120	0	−120	0	−250	0	−250	0	−250	20	20	5	3	1.5
50	80	0	− 150	0	−150	0	−150	0	−380	0	−250	0	−250	25	25	6	4	1.5
80	120	0	− 200	0	−200	0	−200	0	−380	0	−380	0	−380	25	25	7	4	2.5
120	150	0	− 250	0	−250	0	−250	0	−500	0	−380	0	−380	30	30	8	5	2.5
150	180	0	− 250	0	−250	0	−250	0	−500	0	−380	0	−380	30	30	8	5	4
180	250	0	− 300	0	−300	0	−300	0	−500	0	−500	0	−500	30	30	10	6	5
250	315	0	− 350	0	−350	−	−	0	−500	0	−500	−	−	35	35	13	−	−
315	400	0	− 400	0	−400	−	−	0	−630	0	−630	−	−	40	40	15	−	−
400	500	0	− 450	−	−	−	−	−	−	−	−	−	−	50	45	−	−	−
500	630	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	−	−	−	−

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

Notes
(1) 0.6mm is included in the group.
(2) Applicable to bearings with cylindrical bores.
(3) Tolerance for width deviation and tolerance limits for the width variation of the outer ring should be the same bearing. Tolerances for the width variation of the outer ring of Class 5, 4, and 2 are shown in Table 7.2.2.
(4) Applicable to individual rings manufactured for combined bearings.
(5) Applicable to ball bearings such as deep groove ball bearings, angular contact ball bearings, etc.

Remarks
1. The cylindrical bore diameter "no-go side" tolerance limit (high) specified in this table does not necessarily apply within a distance of 1.2 times the chamfer dimension r (max.) from the ring face.
2. 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) over	incl.	Δ_{Dmp} Normal high	Normal low	Class 6 high	Class 6 low	Class 5 high	Class 5 low	Class 4 high	Class 4 low	Class 2 high	Class 2 low	Δ_{Ds} Class 4 (Diameter Series 0, 1, 2, 3, 4) high	Class 4 low	Class 2 high	Class 2 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	−15	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) over	incl.	V_{Dp} (2) Normal Open Type, Diameter Series 9 max.	Normal Open Type 0, 1	Normal Open Type 2, 3, 4	Normal Shielded Sealed 2, 3, 4	Class 6 Open Type 9	Class 6 Open Type 0, 1	Class 6 Open Type 2, 3, 4	Class 6 Shielded Sealed 0,1,2,3,4	Class 5 Open Type 9	Class 5 Open Type 0,1,2,3,4	Class 4 Open Type 9	Class 4 Open Type 0,1,2,3,4	Class 2 Open Type 0,1,2,3,4	V_{Dmp} (2) Normal max.	Class 6 max.	Class 5 max.	Class 4 max.	Class 2 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} Normal max.	Class 6 max.	Class 5 max.	Class 4 max.	Class 2 max.	S_D Class 5 max.	Class 4 max.	Class 2 max.	S_{ea} (3) Class 5 max.	Class 4 max.	Class 2 max.	V_{Cs} (4) Class 5 max.	Class 4 max.	Class 2 max.	Nominal Outside Diameter D (mm) over	incl.
15	8	5	3	1.5	8	4	1.5	8	5	1.5	5	2.5	1.5	2.5 (1)	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) 2.5mm is included in the 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) The tolerances for outer ring width variation of bearings of Classes Normal and 6 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 of 1.2 times the chamfer dimension r (max.) from the ring face.
2. 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 Design 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_{dp}				V_{dmp}				K_{ia}				S_d		S_{ia}
		Normal Class 6X		Class 6 Class 5		Class 4		Class 4		Normal Class 6X	Class 6	Class 5	Class 4	Normal Class 6X	Class 6	Class 5	Class 4	Normal Class 6X	Class 6	Class 5	Class 4	Class 5	Class 4	Class 4
over	incl.	high	low	high	low	high	low	high	low	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.
10	**18**	0	− 8	0	− 7	0	− 5	0	− 5	8	7	5	4	6	5	5	4	15	7	3.5	2.5	7	3	3
18	**30**	0	−10	0	− 8	0	− 6	0	− 6	10	8	6	5	8	6	5	4	18	8	4	3	8	4	4
30	**50**	0	−12	0	−10	0	− 8	0	− 8	12	10	8	6	9	8	5	5	20	10	5	4	8	4	4
50	**80**	0	−15	0	−12	0	− 9	0	− 9	15	12	9	7	11	9	6	5	25	10	5	4	8	5	4
80	**120**	0	−20	0	−15	0	−10	0	−10	20	15	11	8	15	11	8	5	30	13	6	5	9	5	5
120	**180**	0	−25	0	−18	0	−13	0	−13	25	18	14	10	19	14	9	7	35	18	8	6	10	6	7
180	**250**	0	−30	0	−22	0	−15	0	−15	30	22	17	11	23	16	11	8	50	20	10	8	11	7	8
250	**315**	0	−35	0	−25	0	−18	0	−18	35	−	−	−	26	−	−	−	60	25	13	10	13	8	10
315	**400**	0	−40	0	−30	0	−23	0	−23	40	−	−	−	30	−	−	−	70	30	15	12	15	10	14
400	**500**	0	−45	0	−35	0	−27	0	−27	−	−	−	−	−	−	−	−	70	35	18	14	19	13	17
500	**630**	0	−50	0	−40	−	−	−	−	−	−	−	−	−	−	−	−	85	40	20	−	22	−	−
630	**800**	0	−75	0	−60	−	−	−	−	−	−	−	−	−	−	−	−	100	45	22	−	27	−	−

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

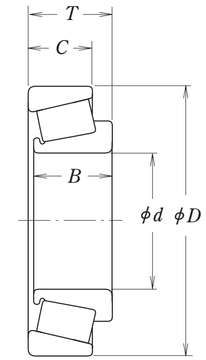
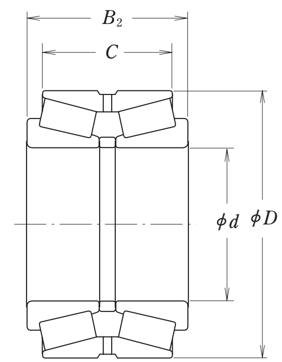


Table 7. 3. 2 Tolerances for Outer Ring Outside Diameter and Running Accuracy

Units : μm

Nominal Outside Diameter D (mm)		Δ_{Dmp}						Δ_{Ds}		V_{Dp}				V_{Dmp}				K_{ea}				S_D		S_{ea}
		Normal Class 6X		Class 6 Class 5		Class 4		Class 4		Normal Class 6X	Class 6	Class 5	Class 4	Normal Class 6X	Class 6	Class 5	Class 4	Normal Class 6X	Class 6	Class 5	Class 4	Class 5	Class 4	Class 4
over	incl.	high	low	high	low	high	low	high	low	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.	max.
18	**30**	0	− 9	0	− 8	0	− 6	0	− 6	9	8	6	5	7	6	5	4	18	9	6	4	8	4	5
30	**50**	0	− 11	0	− 9	0	− 7	0	− 7	11	9	7	5	8	7	5	5	20	10	7	5	8	4	5
50	**80**	0	− 13	0	−11	0	− 9	0	− 9	13	11	8	7	10	8	6	5	25	13	8	5	8	4	5
80	**120**	0	− 15	0	−13	0	−10	0	−10	15	13	10	8	11	10	7	5	35	18	10	6	9	5	6
120	**150**	0	− 18	0	−15	0	−11	0	−11	18	15	11	8	14	11	8	6	40	20	11	7	10	5	7
150	**180**	0	− 25	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	−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	−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	−20	0	−20	40	28	22	15	30	21	14	10	70	35	20	13	13	10	13
400	**500**	0	− 45	0	−33	0	−23	0	−23	45	−	−	−	34	−	−	−	80	40	23	15	15	11	15
500	**630**	0	− 50	0	−38	0	−28	0	−28	50	−	−	−	38	−	−	−	100	50	25	18	18	13	18
630	**800**	0	− 75	0	−45	−	−	−	−	−	−	−	−	−	−	−	−	120	60	30	−	20	−	−
800	**1 000**	0	−100	0	−60	−	−	−	−	−	−	−	−	−	−	−	−	120	75	35	−	23	−	−

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



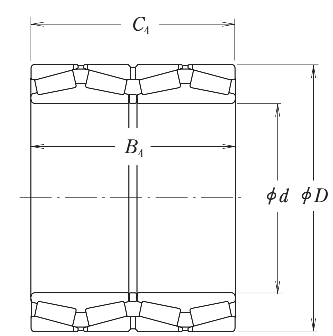


Table 7. 3 Tolerances for Metric Design Tapered Roller Bearings

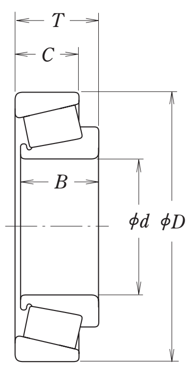
Table 7. 3. 3 Tolerances for Width, Overall Bearing Width, and Combined Bearing Width

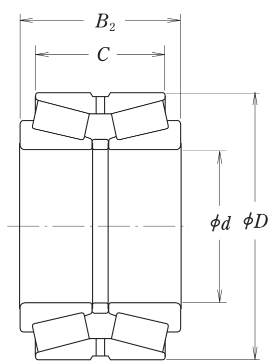
Units : μm

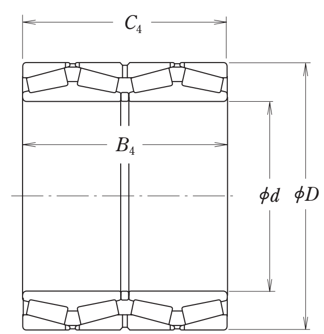
Nominal Bore Diameter d (mm)		Δ_{Bs}						Δ_{Cs}						Δ_{Ts}					
		Normal Class 6		Class 6X		Class 5 Class 4		Normal Class 6		Class 6X		Class 5 Class 4		Normal Class 6		Class 6X		Class 5 Class 4	
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low
10	**18**	0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

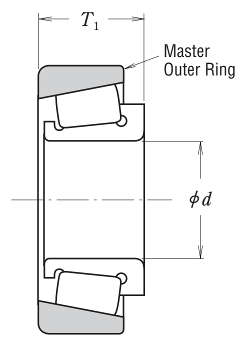
Ring Width with Rollers Δ_{T1s}				Outer Ring Effective Width Deviation Δ_{T2s}				Overall Combined Bearing Width Deviation Δ_{B2s}		Δ_{B4s}, Δ_{C4s}		Nominal Bore Diameter d (mm)	
Normal		Class 6X		Normal		Class 6X		All classes of double-row bearings		All classes of four-row bearings			
high	low	high	low	high	low	high	low	high	low	high	low	over	incl.
+100	0	+ 50	0	+100	0	+ 50	0	+ 200	− 200	−	−	**10**	**18**
+100	0	+ 50	0	+100	0	+ 50	0	+ 200	− 200	−	−	**18**	**30**
+100	0	+ 50	0	+100	0	+ 50	0	+ 200	− 200	−	−	**30**	**50**
+100	0	+ 50	0	+100	0	+ 50	0	+ 300	− 300	+ 300	− 300	**50**	**80**
+100	−100	+ 50	0	+100	−100	+ 50	0	+ 300	− 300	+ 400	− 400	**80**	**120**
+150	−150	+ 50	0	+200	−100	+100	0	+ 400	− 400	+ 500	− 500	**120**	**180**
+150	−150	+ 50	0	+200	−100	+100	0	+ 450	− 450	+ 600	− 600	**180**	**250**
+150	−150	+100	0	+200	−100	+100	0	+ 550	− 550	+ 700	− 700	**250**	**315**
+200	−200	+100	0	+200	−200	+100	0	+ 600	− 600	+ 800	− 800	**315**	**400**
−	−	−	−	−	−	−	−	+ 700	− 700	+ 900	− 900	**400**	**500**
−	−	−	−	−	−	−	−	+ 800	− 800	+1 000	−1 000	**500**	**630**
−	−	−	−	−	−	−	−	+1 200	−1 200	+1 500	−1 500	**630**	**800**

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









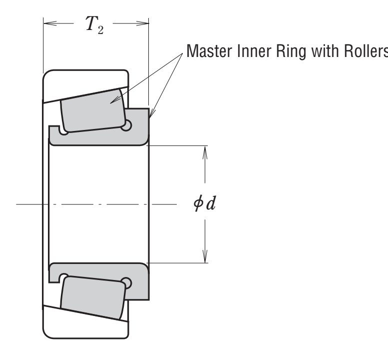
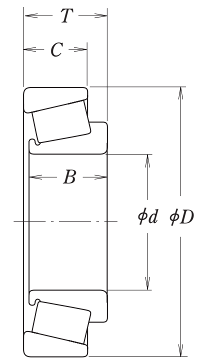
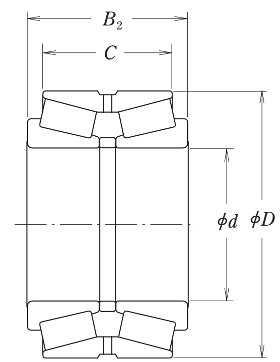


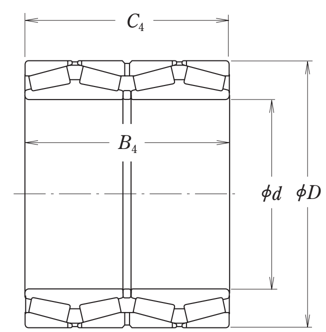
Table 7. 4 Tolerances for Inch Design Tapered Roller Bearings

(Refer to page A126 Table 7. 1 for the tolerance class "CLASS ⁎⁎" that is the tolerance classes of ANSI/ABMA.)





KBE



KV

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	–	–

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)		$\Delta_{d\text{mp}}$						$V_{d\text{p}}$			$V_{d\text{mp}}$			$\Delta_{B\text{s}}$ (or $\Delta_{C\text{s}}$) (1)				$V_{B\text{s}}$ (or $V_{C\text{s}}$) (1)		$\Delta_{T\text{s}}$		K_{ia}			S_d	S_{ia}
		Normal		Class 6		Class 5		Normal	Class 6	Class 5	Normal	Class 6	Class 5	Normal Class 6		Class 5		Normal Class 6	Class 5	Normal Class 6 Class 5		Normal	Class 6	Class 5	Class 5	Class 5
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 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 of 1.2 times the chamfer dimension r (max.) from the ring face.

Table 7. 5. 2 Tolerances for Outer Rings

Units : μm

Nominal Outside Diameter D (mm)		$\Delta_{D\text{mp}}$												$V_{D\text{p}}$			$V_{D\text{mp}}$			K_{ea}			S_{ea}	S_D
		Bearing Series E						Bearing Series EN																
		Normal		Class 6		Class 5		Normal		Class 6		Class 5		Normal	Class 6	Class 5	Normal	Class 6	Class 5	Normal	Class 6	Class 5	Class 5	Class 5
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 of 1.2 times the chamfer dimension r (max.) from the ring face.

Table 7. 6 Tolerances for Thrust Ball Bearings

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

Units : μm

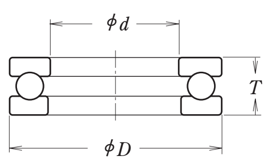
Nominal Bore Diameter d or d_2 (mm)		$\Delta_{d\mathrm{mp}}$ or $\Delta_{d2\mathrm{mp}}$				$V_{d\mathrm{p}}$ or $V_{d2\mathrm{p}}$		S_i or S_e (1)			
		Normal Class 6 Class 5		Class 4		Normal Class 6 Class 5	Class 4	Normal	Class 6	Class 5	Class 4
over	incl.	high	low	high	low	max.	max.	max.	max.	max.	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	–

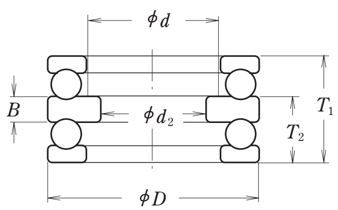
Note (1) For double-direction bearings, the thickness variation doesn't depend on the bore diameter d_2, but on d for single-direction bearings with the same D in the same diameter series.
The thickness variation of housing washers, S_e, applies only to flat-seat thrust bearings.

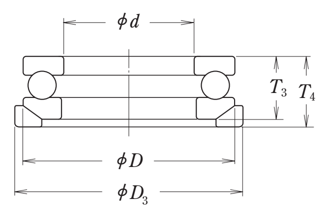
Table 7. 6. 2 Tolerances for Outside Diameter of Housing Washers and Aligning Seat Washers

Units : μm

Nominal Outside Diameter of Bearing or Aligning Seat Washer D or D_3 (mm)		$\Delta_{D\mathrm{mp}}$						$V_{D\mathrm{p}}$		Aligning Seat Washer Outside Diameter Deviation $\Delta_{D3\mathrm{s}}$	
		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	–	–	–	–	–	–	–	–







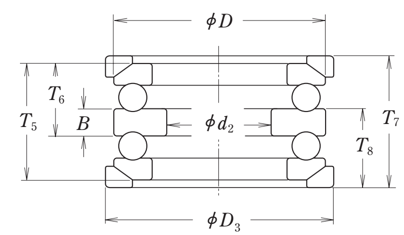


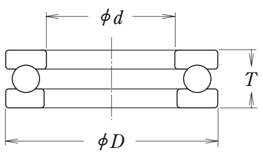
Table 7. 6. 3 Tolerances for Thrust Ball Bearing Height and Central Washer Height

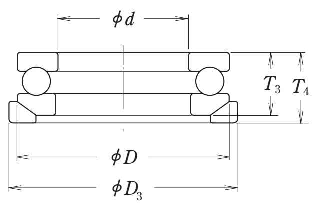
Units : µm

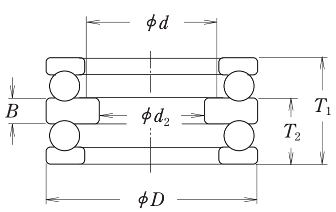
Nominal Bore Diameter d (1) (mm)		Flat Seat Type				Aligning Seat Washer Type				With Aligning Seat Washer				Height Deviation of Central Washer Δ_{Bs}	
		Δ_{Ts} or Δ_{T2s}		Δ_{T1s}		Δ_{T3s} or Δ_{T6s}		Δ_{T5s}		Δ_{T4s} or Δ_{T8s}		Δ_{T7s}			
		Normal, Class 6 Class 5, Class 4		Normal, Class 6 Class 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, its 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 figures below.







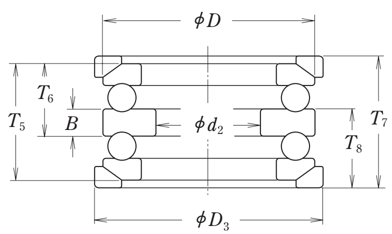


Table 7. 7 Tolerances for Tapered Roller Thrust Bearings

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

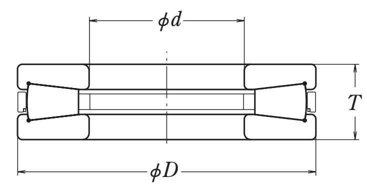
Units : µm

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, Class Normal)

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



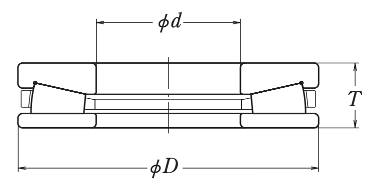


Table 7. 7 Tolerances for Tapered Roller Thrust Bearings

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

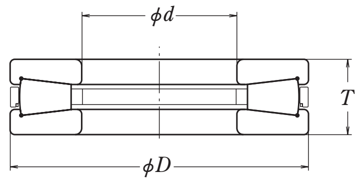
Units : μm

Nominal Bore Diameter d				Δ_{dmp}		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)

Units : μm

Nominal Outside Diameter D				Δ_{Dmp}	
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



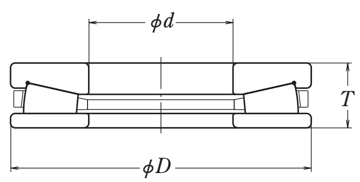


Table 7. 8 Tolerances for Thrust Spherical Roller Bearings

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

Units : μm

Nominal Bore Diameter d (mm)		Δ_{dmp}		V_{dp}	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 of 1.2 times the chamfer dimension r (max.) from the ring face.

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

Units : μm

Nominal Outside Diameter D (mm)		Δ_{Dmp}	
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 of 1.2 times the chamfer dimension r (max.) from the ring face.

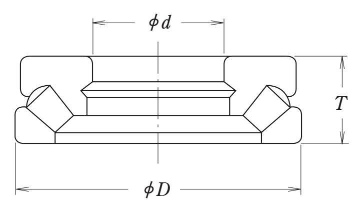


Table 7. 9 Tolerances of Instrument Ball Bearings (Inch design)

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) over	incl.	Δ_{dmp} CLASS 5P CLASS 7P high	low	Δ_{dmp} CLASS 9P high	low	Δ_{ds} CLASS 5P CLASS 7P high	low	Δ_{ds} CLASS 9P high	low	V_{dp} CLASS 5P CLASS 7P max.	V_{dp} CLASS 9P max.	V_{dmp} CLASS 5P CLASS 7P max.	V_{dmp} CLASS 9P max.
–	10	0	−5.1	0	−2.5	0	−5.1	0	−2.5	2.5	1.3	2.5	1.3
10	18	0	−5.1	0	−2.5	0	−5.1	0	−2.5	2.5	1.3	2.5	1.3
18	30	0	−5.1	0	−2.5	0	−5.1	0	−2.5	2.5	1.3	2.5	1.3

Nominal Bore Diameter d (mm) over	incl.	Δ_{Bs} Single Brgs CLASS 5P CLASS 7P CLASS 9P high	low	(or Δ_{Cs}) Combined Brgs (1) CLASS 5P CLASS 7P CLASS 9P high	low	V_{Bs} CLASS 5P max.	V_{Bs} CLASS 7P max.	V_{Bs} CLASS 9P max.	K_{ia} CLASS 5P max.	K_{ia} CLASS 7P max.	K_{ia} CLASS 9P max.	S_{ia} CLASS 5P max.	S_{ia} CLASS 7P max.	S_{ia} CLASS 9P max.	S_d CLASS 5P max.	S_d CLASS 7P max.	S_d CLASS 9P max.
–	10	0	−25.4	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	−25.4	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	−25.4	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 For the CLASS 3P and the tolerances of Metric design Instrument Ball Bearings, it is advisable to consult NSK.

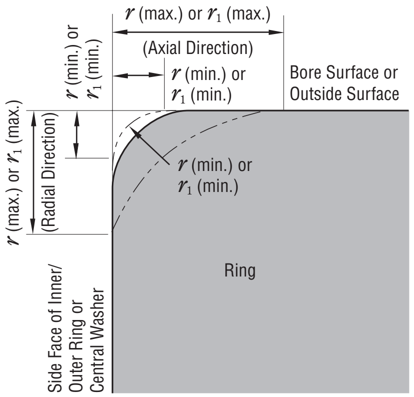
(2) Tolerances for Outer Rings

Units : μm

Nominal Outside Diameter D (mm) over	incl.	Δ_{Dmp} CLASS 5P CLASS 7P high	low	Δ_{Dmp} CLASS 9P high	low	Δ_{Ds} CLASS 5P CLASS 7P Open high	low	Δ_{Ds} CLASS 5P CLASS 7P Shielded Sealed high	low	Δ_{Ds} CLASS 9P Open high	low	V_{Dp} CLASS 5P CLASS 7P Open max.	V_{Dp} CLASS 5P CLASS 7P Shielded Sealed max.	V_{Dp} CLASS 9P Open max.	V_{Dmp} CLASS 5P CLASS 7P Open max.	V_{Dmp} CLASS 5P CLASS 7P Shielded Sealed max.	V_{Dmp} CLASS 9P Open 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) over	incl.	V_{Cs} (1) CLASS 5P max.	V_{Cs} (1) CLASS 7P max.	V_{Cs} (1) CLASS 9P max.	S_D CLASS 5P max.	S_D CLASS 7P max.	S_D CLASS 9P max.	K_{ea} CLASS 5P max.	K_{ea} CLASS 7P max.	K_{ea} CLASS 9P max.	S_{ea} CLASS 5P max.	S_{ea} CLASS 7P max.	S_{ea} CLASS 9P max.	Deviation of Flange Outside Diameter Δ_{D1s} CLASS 5P CLASS 7P high	low	Deviation of Flange Width Δ_{C1s} CLASS 5P CLASS 7P high	low	Flange Backface Runout with Raceway (2) S_{ea1} CLASS 5P CLASS 7P 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 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 shall not intersect an arc of radius r (min.) or r_1 (min.) touching 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 Design 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	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

Remark For bearings with nominal widths less than 2mm, 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	

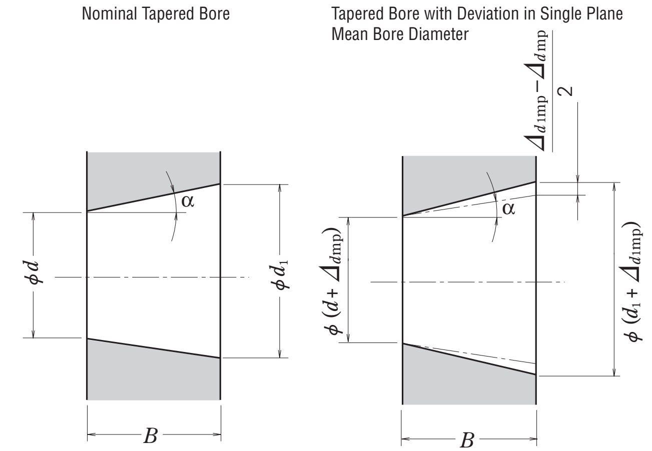
Note ([1]) Inner Rings are classified by d and Outer Rings by D.

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 (Class Normal)



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
α =2°23´9.4
=2.38594°
=0.041643 rad

Taper 1:30
α =57´17.4
=0.95484°
=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

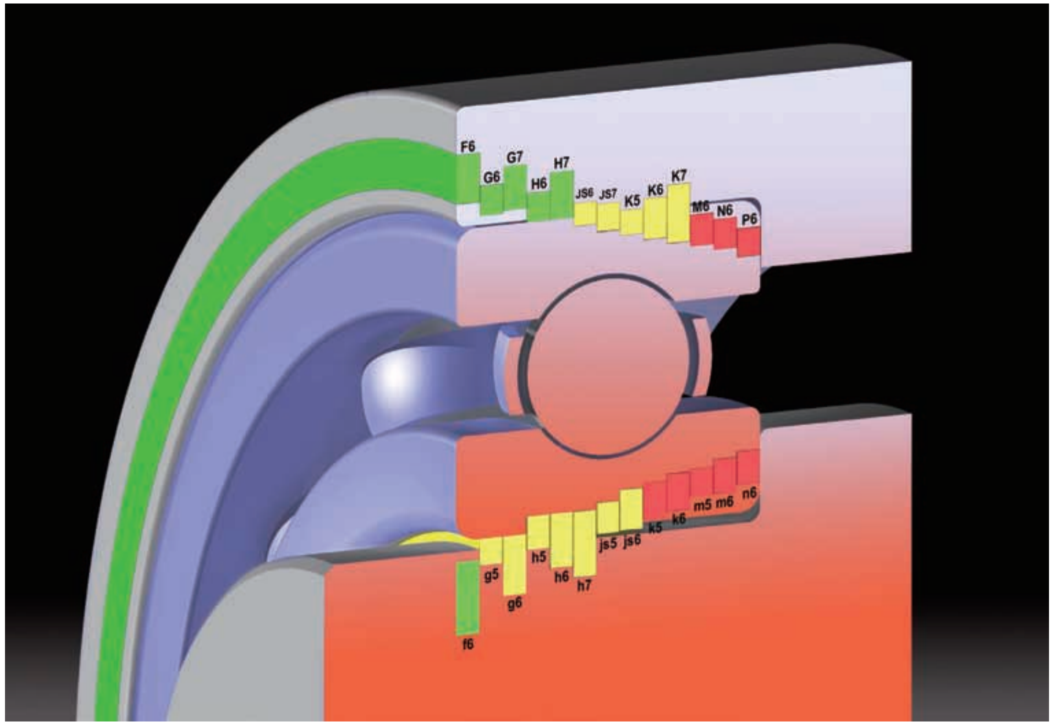
Notes (1) Applicable to all radial planes of tapered bores.
(2) Not applicable to diameter series 7 and 8.
Remark For a value exceeding 630 mm, please contact NSK.

7.2 Selection of Accuracy Classes

For general applications, Class Normal tolerances are adequate in nearly all cases for satisfactory performance, but for the following applications, bearings having an accuracy class of 5,4 or higher are more suitable.
For reference, in Table 7.12, examples of 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 Requirement, Operating Conditions	Examples of Applications	Tolerance Classes
High running accuracy is required	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 is required	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 are required	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 the case of a rolling bearing with the inner ring fitted to the shaft with only slight interference, a harmful circumferential slipping 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 that ring which 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, however, 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 this case, to prevent damage to the fitting surfaces due to creep, lubrication of other applicable methods should be considered.

8.1.2 Selection of Fit

(1) Load Conditions and Fit

The proper fit may 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 the bearing load; therefore, the loss of interference should be estimated using the following equations:

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

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, in the case of heavy loads where the radial load exceeds 20% of the basic static load rating C_{0r}, under the operating condition, interference often becomes shortage. Therefore, interference should be estimated using Equation (8.2):

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

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).

For NU219, with the interference given by Equation (8.1) for loads heavier than 0.25 C_{0r}, the interference 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, sufficient care should be taken to prevent excessive circumferential stress.

Calculation example

For NU219, B=32 (mm) and assume
F_r=98 100 N {10 000 kgf}
C_{0r}=183 000 N {18 600 kgf}

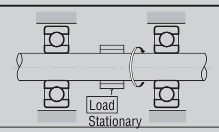
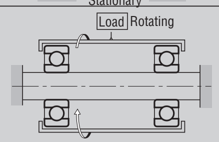
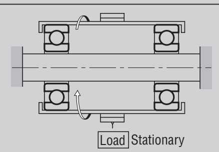
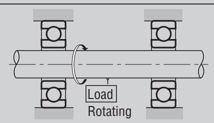
$$\frac{F_r}{C_{0r}}=\frac{98\,100}{183\,000}=0.536>0.2$$

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

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

This result agrees well with Fig. 8.1.

Table 8.1 Loading Conditions and Fits

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

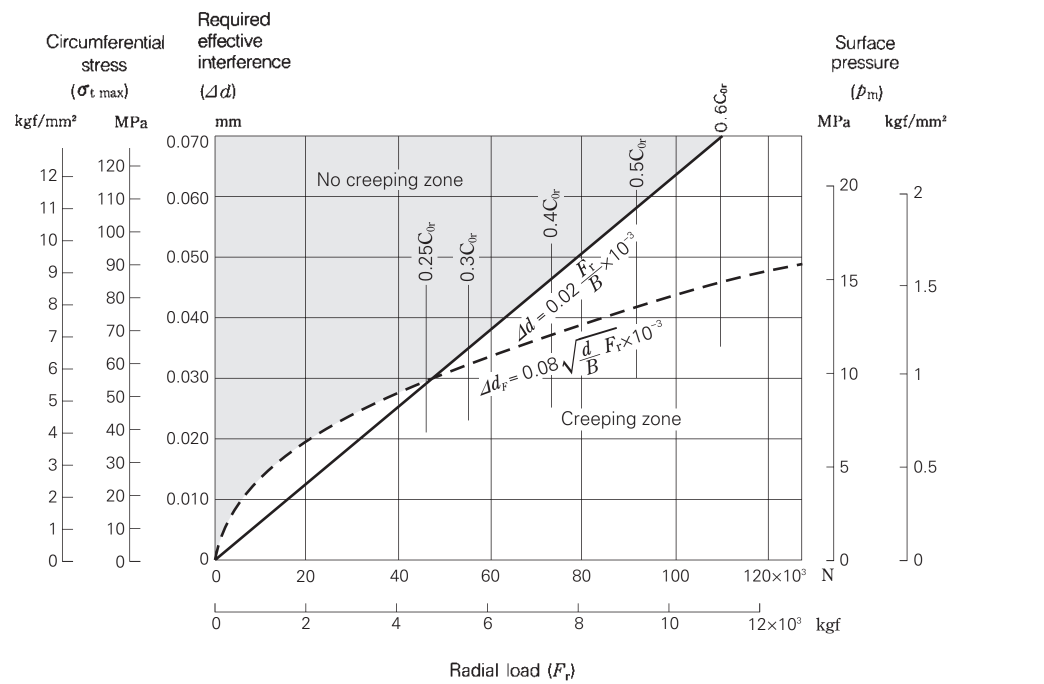


Fig. 8.1 Load and Required Effective Interference for Fit

(3) Interference Variation Caused by Temperature Difference 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 in case that 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):

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

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=12.5×10^{-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, the 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 $\quad \Delta d = \dfrac{d}{d+2} \Delta d_a \quad \cdots\cdots (\mathbf{8.4})$

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

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 guide, the maximum interference should be kept under approximately 7/10 000 of 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 $p_m = \dfrac{\Delta d}{d} \dfrac{1}{\left[\dfrac{m_s-1}{m_s E_s} - \dfrac{m_i-1}{m_i E_i}\right] + 2\left[\dfrac{k_0^2}{E_s(1-k_0^2)} + \dfrac{1}{E_i(1-k^2)}\right]}$ Solid shaft $p_m = \dfrac{\Delta d}{d} \dfrac{1}{\left[\dfrac{m_s-1}{m_s E_s} - \dfrac{m_i-1}{m_i E_i}\right] + \dfrac{2}{E_i(1-k^2)}}$	Housing outside diameter $p_m = \dfrac{\Delta D}{D} \dfrac{1}{\left[\dfrac{m_e-1}{m_e E_e} - \dfrac{m_h-1}{m_h E_h}\right] + 2\left[\dfrac{h^2}{E_e(1-h^2)} + \dfrac{1}{E_h(1-h_0^2)}\right]}$
Expansion of inner ring raceway ΔD_i (mm) Contraction of outer ring raceway ΔD_e (mm)	$\Delta D_i = 2d \dfrac{p_m}{E_i} \dfrac{k}{1-k^2}$ $= \Delta d \cdot k \dfrac{1-k_0^2}{1-k^2 k_0^2}$ (hollow shaft) $= \Delta d \cdot k$ (solid shaft)	$\Delta D_e = 2D \dfrac{p_m}{E_e} \dfrac{h}{1-h^2}$ $= \Delta D \cdot h \dfrac{1-h_0^2}{1-h^2 h_0^2}$
Maximum stress $\sigma_{t\,max}$ (MPa) {kgf/mm²}	Circumferential stress at inner ring bore fitting surface is maximum. $\sigma_{t\,max} = p_m \dfrac{1+k^2}{1-k^2}$	Circumferential stress at outer ring bore surface is maximum. $\sigma_{t\,max} = p_m \dfrac{2}{1-h^2}$
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 for an inner ring, which is the rotating ring, an interference fit is chosen, and for a fixed outer ring, a loose fit is used. To select the fit, the magnitude of the load, the temperature differences among the bearing and shaft and housing, the material characteristics of the shaft and housing, the level of finish, the material thickness, and the bearing mounting/dismounting method must all be considered.

If the interference is insufficient for the operating conditions, ring loosening, creep, fretting, heat generation, etc. may occur. If the interference is excessive, the ring may crack. The magnitude of the interference is usually satisfactory if it is set for the size of the shaft or housing listed in the bearing manufacturer's catalog. To determine the surface pressure and stress on the fitting surfaces, calculations can be made assuming a thick-walled cylinder with uniform internal and external pressures. To do this, the necessary equations are summarized in Table 8.2. For convenience in the fitting of bearing inner rings on solid steel shafts, which are the most common, 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 $\sigma_{t\ max}$ variations with shaft diameter when interference results from the mean values of the tolerance grade shaft and bearing bore tolerances. 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 whether $\sigma_{t\ max}$ exceeds the tolerances. 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.

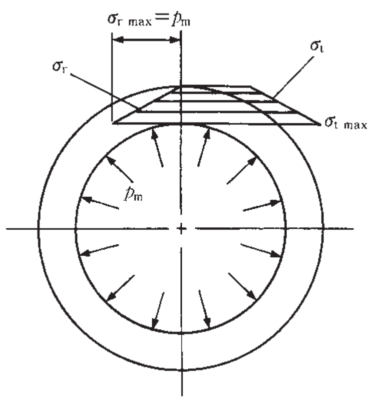


Fig. 8.2 Distribution of Circumferential Stress σ_t and Radial Stress σ_r

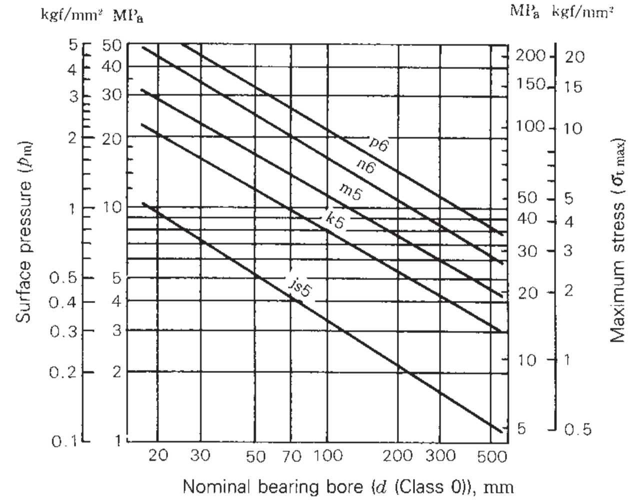


Fig. 8.3 Surface Pressure p_m and Maximum Stress $\sigma_{t\ max}$ for Mean Interference in Various Tolerance Grades

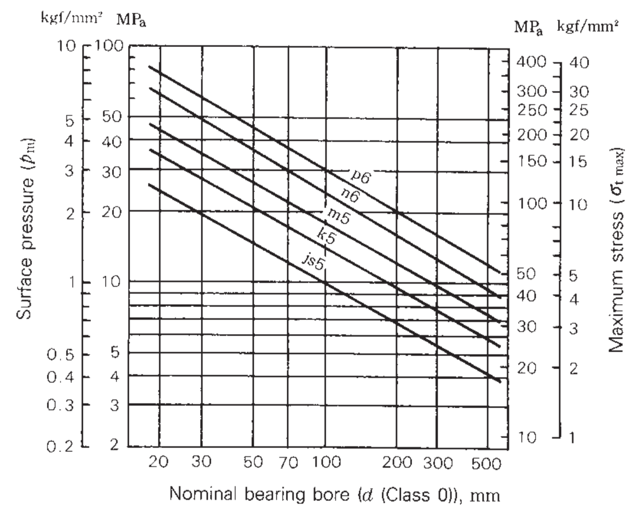


Fig. 8.4 Surface Pressure p_m and Maximum Stress $\sigma_{t\ max}$ for Maximum Interference in Various Tolerance Grades

(7) Mounting and Withdrawal Loads

The push-up load needed to mount bearings on shafts or in a housing hole with interference can be obtained using the thick-walled cylinder theory.

The mounting load (or withdrawal load) depends upon the contact area, surface pressure, and coefficient of friction between the fitting surfaces.

The mounting load (or withdrawal load) K needed to mount inner rings on shafts is given by Equation (8.6).

$$K=\mu\, p_m\, \pi\, d\, B \text{ (N), \{kgf\}} \quad \cdots\cdots \textbf{(8.6)}$$

where μ: Coefficient of friction between fitting surfaces
μ=0.12 (for mounting)
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 the 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) is:

$$\left.\begin{aligned} K &= 118\,000\mu\ \Delta d\ B \text{ (N)} \\ &= 12\,000\mu\ \Delta d\ B \text{ \{kgf\}} \end{aligned}\right\} \quad \cdots\cdots \textbf{(8.7)}$$

Equation (8.7) is shown graphically in Fig. 8.5. The mounting and withdrawal loads for outer rings and housings have been calculated and the results are shown in Fig. 8.6.

The actual mounting and withdrawal loads can become much higher than the calculated values if the bearing ring and shaft (or housing) are slightly misaligned or the load is applied unevenly to the circumference of the bearing ring hole. Consequently, the loads obtained from Figs. 8.5 and 8.6 should be considered only as guides when designing withdrawal tools, their strength should be five to six times higher than that indicated by the figures.

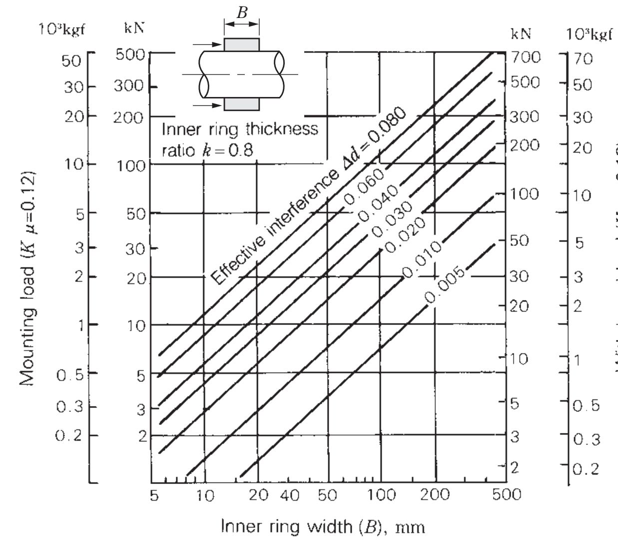


Fig. 8.5 Mounting and Withdrawal Loads for Inner Rings

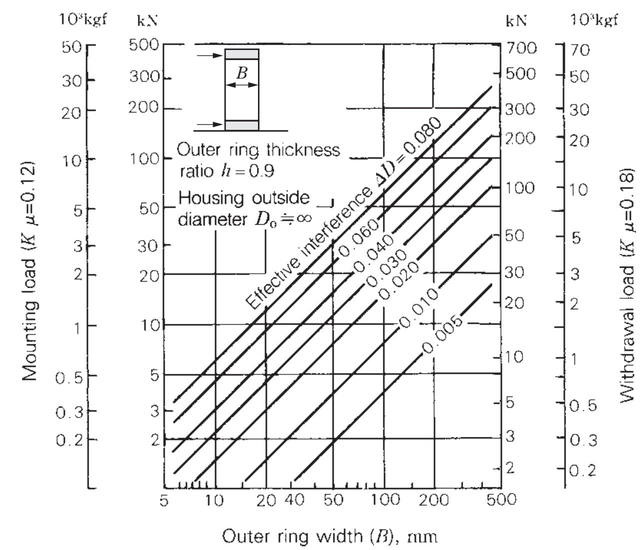


Fig. 8.6 Mounting and Withdrawal Loads for Outer Rings

8.1.3 Recommended Fits

As described previously, many factors, such as the characteristics and magnitude of bearing load, temperature differences, 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 Table 8.3 to 8.8. In the case of unusual operating conditions, it is advisable to consult NSK. For the accuracy and surface finish of shafts and housings, please refer to Section 13.1 (Page A270).

Table 8.3 Fits of Radial Bearings 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 desirable.	Wheels on Stationary Axles	All Shaft Diameters			g6	Use g5 and h5 where accuracy is required. In case of large bearings, f6 can be used to allow easy axial movement.
	Easy axial displacement of inner ring on shaft unnecessary	Tension Pulleys Rope Sheaves				h6	
Rotating Inner Ring Load or Direction of Load Indeterminate	Light Loads or Variable Loads (<0.06C_r(1))	Electrical Home Appliances Pumps, Blowers, Transport Vehicles, Precision Machinery, Machine Tools	<18	—	—	js5	
			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	More than CN bearing internal clearance is necessary.
			—	140 to 200	100 to 140	p6	
			—	over 200	140 to 200	r6	
			—	—	200 to 500	r7	
Axial Loads Only			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)	IT5 and IT7 mean that the deviation of the shaft from its true geometric form, e. g. roundness and cylindricity should be within the tolerances of IT5 and IT7 respectively.
		Transmission Shafts, Woodworking Spindles				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 standard tolerance grades IT.
(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 is applicable 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 Direction of Load Indeterminate	Paper Pulp Refiners, Plastic Extruders	<200	k6	
			200 to 400	m6	
			over 400	n6	

Table 8.5 Fits of Radial Bearings 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		
	Direction of Load Indeterminate	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	Loads of All kinds	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 Desirable under Normal or Light Loads	Grinding Spindle Rear Ball Bearings High Speed Centrifugal Compessor Free Bearings	JS6 (J6)	Possible	—
	Direction of Load Indeterminate		Grinding Spindle Front Ball Bearings High Speed Centrifugal Compressor Fixed Bearings	K6	Generally Impossible	For heavy loads, interference fit tighter than K is used. When high accuracy is required, very strict tolerances should be used for fitting.
	Rotating Inner Ring Load	Accurate Running and High Rigidity Desirable under Variable Loads	Cylindrical Roller Bearings for Machine Tool Main Spindle	M6 or N6	Impossible	
		Minimum 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 those in this table.
2. Refer to the introductory section of the bearing dimension tables (blue pages) for special fits such as 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 Direction of Load Indeterminate		K7	Normal Loads
			M7	Relatively Heavy Radial Loads

Table 8.7 Fits of Inch Design 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≦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. ※ means 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 in the 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 Design 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	

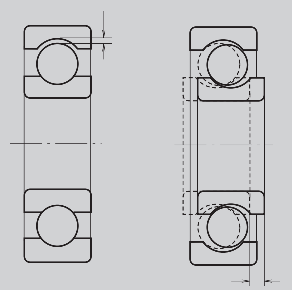
Note (1) For bearings with D greater than 304.8 mm, Class 0 does not exist.

8.2 Bearing Internal Clearances

8.2.1 Internal Clearances and Their Standards

The internal clearance in rolling bearings in operation greatly influences bearing performance including fatigue life, vibration, noise, heat-generation, etc. Consequently, the selection of the 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 is the combined clearances between the inner/outer rings and rolling elements. The radial and axial 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 Clearance Axial 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; therefore, the measured clearance (sometimes called “measured clearance” to make a distinction) is always slightly larger than the theoretical internal clearance (called “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 the one specified as 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 Types

Bearing Types		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 clearances.

Table 8.10 Radial Internal Clearances in Deep Groove Ball Bearings

Units : μm

Nominal Bore Diameter d (mm) over	incl.	C2 min.	C2 max.	CN min.	CN max.	C3 min.	C3 max.	C4 min.	C4 max.	C5 min.	C5 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 for radial clearance increase caused by the measuring load 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) over	incl.	Measuring Load (N)	{kgf}	Radial Clearance Correction Amount 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 min.	MC1 max.	MC2 min.	MC2 max.	MC3 min.	MC3 max.	MC4 min.	MC4 max.	MC5 min.	MC5 max.	MC6 min.	MC6 max.
Clearance	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 in 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 their classification, refer to Table 1 on Page C054.

Table 8.12 Radial Internal Clearances in Magneto Bearings

Units : μm

Nominal Bore Diameter d (mm) over	incl.	Bearing Series	Clearance 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 clearance increase caused by the measuring load is equal to the correction amount for CN clearance 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-Type Needle Roller Bearings

Units : μm

Nominal Bore Dia. d (mm)		Clearances in Bearings with Cylindrical Bores										Clearances in Non-Interchangeable 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 non-Interchangeable cylindrical roller bearings and solid-type needle roller bearings.

Units : μm

Nominal Bore Dia. d (mm)		Clearances in Non-Interchangeable 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 non-Interchangeable cylindrical roller bearings and solid-type 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

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 $\Delta_a = \Delta_r \cot\alpha \doteqdot \frac{1.5}{e}\Delta_r$

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 in Tolerance Classes Normal and 6. For internal axial clearances in bearings in tolerance classes better than 5 and contact angles of 15° and 25°, it is advisable to consult NSK.

Table 8.19 Axial Internal Clearance in Four-Point Contact Ball Bearings (Measured Clearances)

Units : μm

Nominal Bore Dia. d (mm)		Axial Internal Clearance							
		C2		CN		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
10	**18**	15	55	45	85	75	125	115	165
18	**40**	26	66	56	106	96	146	136	186
40	**60**	36	86	76	126	116	166	156	206
60	**80**	46	96	86	136	126	176	166	226
80	**100**	56	106	96	156	136	196	186	246
100	**140**	66	126	116	176	156	216	206	266
140	**180**	76	156	136	196	176	246	226	296
180	**220**	96	176	156	226	206	276	256	326
220	**260**	115	196	175	245	225	305	285	365
260	**300**	135	215	195	275	255	335	315	395
300	**350**	155	235	215	305	275	365	345	425
350	**400**	175	265	245	335	315	405	385	475
400	**500**	205	305	285	385	355	455	435	525

8.2.2 Selection of Bearing Internal Clearances

Among the bearing internal clearances listed in the tables, the CN Clearance is adequate for standard operating conditions. The 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 is tight-fitted on the shaft.

As a measure to reduce bearing noise for electric motors, the radial 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 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 is tight-fitted 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 and 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, Fits (5), 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 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 is greater. 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:

$$\delta_t \fallingdotseq \alpha \Delta_t D_e \quad \text{.................................... (8.8)}$$

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

$$D_e \fallingdotseq \frac{1}{5}(4D+d) \quad \text{..................... (8.9)}$$

For roller bearings

$$D_e \fallingdotseq \frac{1}{4}(3D+d) \quad \text{.................... (8.10)}$$

The clearance after substracting this δ_t from the residual clearance, Δ_f is called the effective clearance, Δ. Theoretically, the longest life of a bearing can be expected 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, it is necessary to provide adequate axial clearance to allow for shaft elongation during operation.

The radial clearances used in some specific applications are given in Table 8.20. Under special operating conditions, it is advisable to consult NSK.

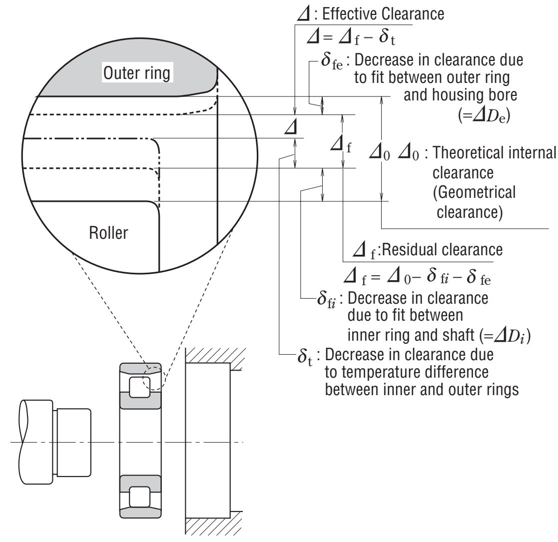


Fig. 8.8 Changes in Radial Internal Clearance of Bearings

Table 8. 20 Examples of 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, it is specified as a rule that measurement of a bearing must be made at 20°C and that the dimensions to be set forth in the standards must be expressed by values at 20°C.

Dimensional change due to temperature change not only affects the dimensional accuracy, but also causes change 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 also possible to achieve shrink fitting with large interference by utilizing dimensional change induced by temperature difference. The dimensional change Δl due to temperature rise can be expressed as in Equation (8.11) below:

$$\Delta l = \Delta T \, \alpha \, l \ \text{(mm)} \quad \cdots\cdots\cdots (\mathbf{8.11})$$

where, Δl: Dimentional change (mm)
ΔT: temperature rise (°C)
α: Coefficient of linear expansion for bearing steel
$\alpha=12.5\times10^{-6}$ (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:

(1) To correct dimensional measurements according to the ambient air temperature
(2) To find the change in bearing internal clearance due to temperature difference between inner and outer rings during operation
(3) To find the relationship between the interference and heating temperature during shrink fitting
(4) To find the change in the interference when a temperature difference exists on the fit surface

Example

To what temperature should the inner ring be heated if an inner ring of 110 mm in bore is to be shrink fitted to a shaft belonging to the n6 tolerance range class?
The maximum interference between the n6 shaft of 110 in diameter and the inner ring is 0.065. To enable insertion of the inner ring with ease on the shaft, there must be a clearance of 0.03 to 0.04. Accordingly, the amount to expand the inner ring must be 0.095 to 0.105.
Intersection of a vertical axis Δl=0.105 and a horizontal axis l=110 is determined on a diagram. ΔT is located in the temperature range between 70°C and 80°C (ΔT≒77°C). Therefore, it is enough to set the inner ring heating temperature to the room temperature +80°C.

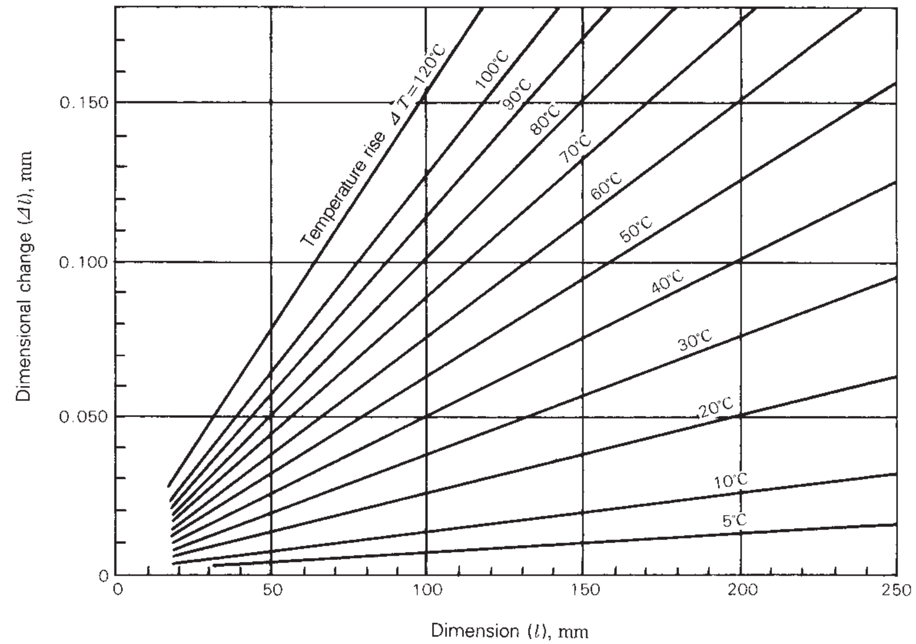


Fig. 8.9 Temperature Rise and Dimensional Change of Bearing Steel

8.3.2 Interference Deviation Due to Temperature Rise (Aluminum Housing, Plastic Housing)

For reducing weight and cost or improving the performance of equipment, bearing housing materials such as aluminum, light alloys, or plastics (polyacetal resin, etc.) are often used.

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 large for plastics which have high coefficients of linear expansion.

The deviation $\varDelta D_T$ of clearance or interference of a fitting surface of a bearing's outer ring due to temperature rise is expressed by the following equation:

$$\varDelta D_T=(\alpha_1\cdot\varDelta T_1-\alpha_2\cdot\varDelta T_2)D\ \text{(mm)} \quad \cdots\cdots\cdots\cdots\cdots(\mathbf{8.12})$$

where $\varDelta D_T$: Change of clearance or interference at fitting surface due to temperature rise

α_1: Coefficient of linear expansion of housing (1/°C)

$\varDelta T_1$: 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)

$\varDelta T_2$: Outer ring temperature rise near fitting surface (°C)

D: Bearing outside diameter (mm)

In general, the housing temperature rise and that of the outer ring are somewhat different, but if we assume they are approximately equal near the fitting surfaces, ($\varDelta T_1 \fallingdotseq \varDelta T_2=\varDelta T$), Equation (8.13) becomes,

$$\varDelta D_T=(\alpha_1-\alpha_2)\ \varDelta T\cdot D\ \text{(mm)} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots(\mathbf{8.13})$$

where $\varDelta T$: Temperature rise of outer ring and housing near fitting surfaces (°C)

In the case of an aluminum housing (α_1=23.7×10^{-6}), Equation (8.13) can be shown graphically as in Fig. 8.10.

Among the various plastics, polyacetal resin is one that is often used for bearing housings. The coefficients of linear expansion of plastics may vary or show directional characteristics. In the case of polyacetal resin, for molded products, it is approximately 9×10^{-5}. Equation (8.13) can be shown as in Fig. 8.11.

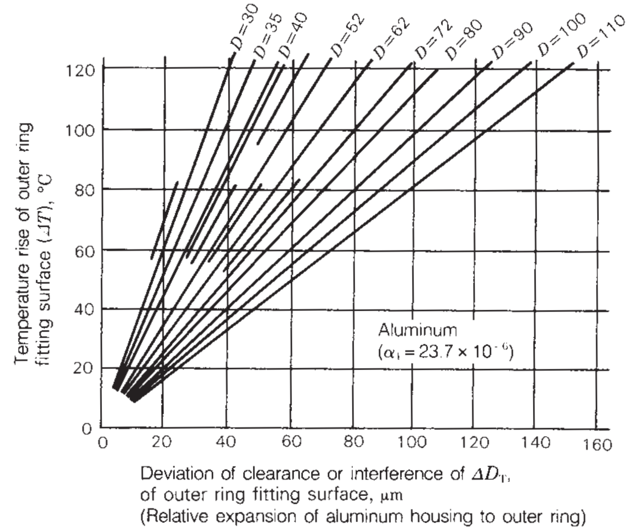


Fig. 8.10 Aluminum Housing

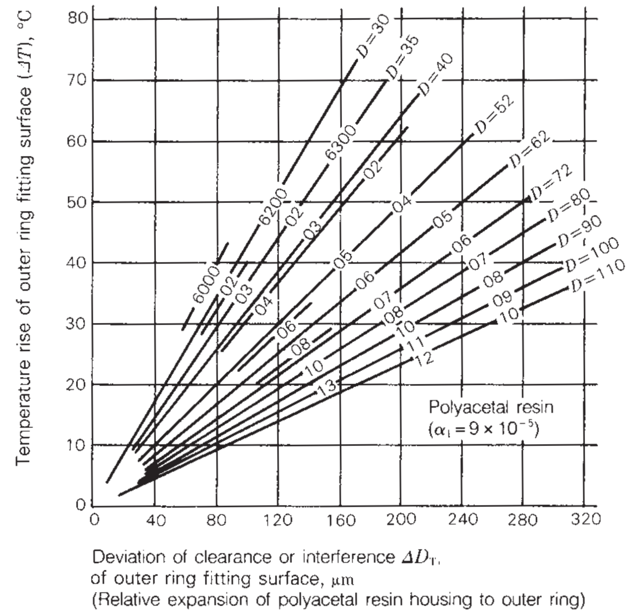


Fig. 8.11 Polyacetal Resin Housing

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. Generally, therefore, 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 at k5, with the housing set at H7. An interference fit is applied only to the inner ring.

Shaft diameter, bore size and radial clearance are the standard bearing measurements. Assuming that 99.7% of the parts are within tolerance, the mean value ($m_{\Delta f}$) and standard deviation ($\sigma_{\Delta f}$) of the internal clearance after mounting (residual clearance) can be calculated. Measurements are given in units of 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, wihtin a 99.7% probability, the variation in internal clearance after mounting ($R_{\Delta f}$), we 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 a 6310 bearing.

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

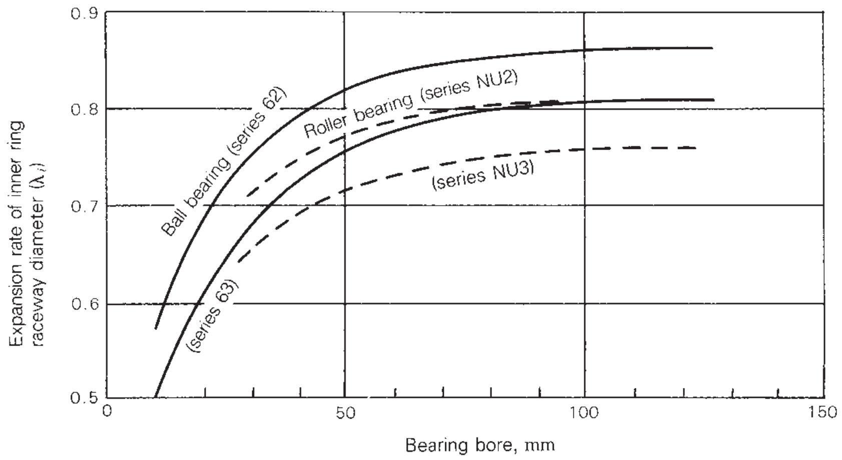


Fig. 8.12 Rate of Inner Ring Raceway Expansion (λ_i) from Apparent Interference

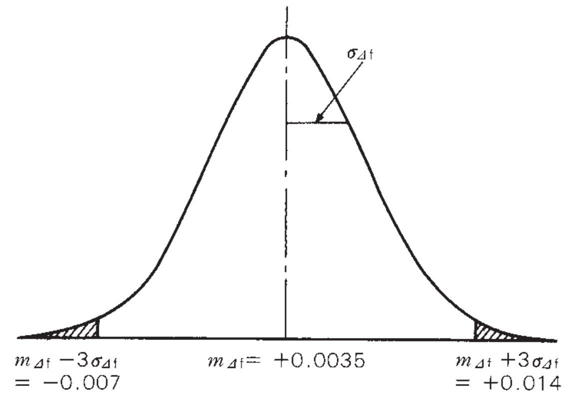


Fig. 8.13 Distribution of Residual Internal Clearance

8.3.4 Effect of Interference Fit on Bearing Raceways (Fit of Inner Ring)

One of the important factors that relates to radial clearance is the reduction in radial clearance resulting from the mounting fit. When 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).

$$\Delta D_i = \Delta d \; k \; \frac{1-k_0^{\,2}}{1-k^2 k_0^{\,2}} \quad \cdots\cdots\cdots\cdots \textbf{(8.14)}$$

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) has been translated into a clearer graphical form in Fig. 8.14.
The vertical axis of Fig. 8.14 represents the inner raceway diameter expansion in relation to the amount of interference. The horizontal axis is the ratio of inside and outside diameter of the hollow shaft (k_0) and uses as its parameter the ratio of bore diameter and raceway diameter of the inner ring (k).
Generally, the decrease in radial clearance is calculated to be approximately 80% of the interference. However, this is for solid shaft mountings only. For hollow shaft mountings the decrease in radial clearance varies with the ratio of inside to outside diameter of the shaft. 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.
Let's take as an example 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 and determine the decrease in radial clearance.
The ratio between bore diameter and raceway diameter, k is 0.87 as shown in Fig. 8.15. The ratio of inside to outside diameter for shaft, 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.

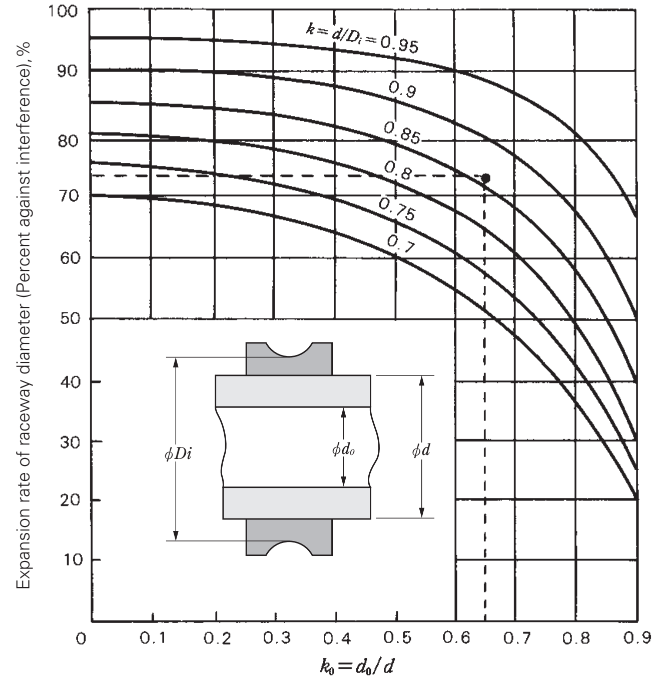


Fig. 8.14 Raceway Expansion in Relation to Bearing Fit (Inner Ring Fit upon Steel Shaft)

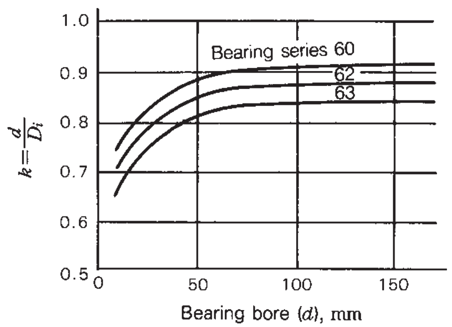


Fig. 8.15 Ratio of Bore Size to Raceway Diameter for Single Row Deep Groove Ball Bearings

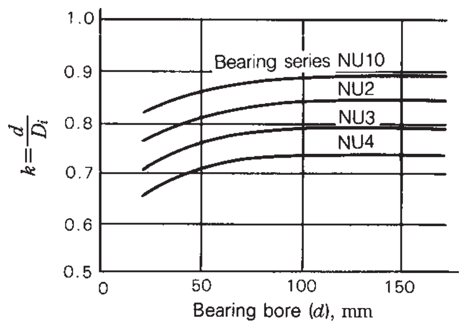


Fig. 8.16 Ratio of Bore Size to Raceway Diameter for Cylindrical Roller Bearings

8.3.5 Effect of Interference Fit on Bearing Raceways (Fit of Outer Ring)

We continue with the calculation of the raceway contraction of the outer ring after fitting.
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 the bearing 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.
Actually, because the amount of interference that can be applied to the outer ring is limited by stress, and because the constraints of most bearing applications make it difficult to apply a large amount of interference to the outer ring, and instances where there is an indeterminate load are quite rare compared to those where a rotating inner ring carries the load, 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).

$$\Delta D_e = \Delta D \cdot h \, \frac{1-h_0^{\,2}}{1-h^2 h_0^{\,2}} \cdots\cdots\cdots\cdots\cdots\cdots \textbf{(8.15)}$$

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 in graphic form.
The vertical axis show the outer-ring raceway contraction as a percentage of interference, and the horizontal axis is the housing thickness ratio h_0. The data are plotted for constant values of the outer-ring thickness ratio 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 for 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, we estimate the amount of decrease in radial clearance assuming a 6207 ball bearing is mounted in a steel housing with an N7 fit. The outside diameter of the housing is assumed to be D_0=95, and the bearing outside diameter is D=72. From Fig. 8.18, the outer-ring thickness ratio, h, is 0.9. Because $h_0=D/D_0$=0.76, from Fig. 8.17, the amount of raceway contraction is 71%. Taking the mean value for N7 interference as 18 μm, the amount of contraction of the outer raceway, or the amount of decrease in radial clearance is 0.71×18=13 μm.

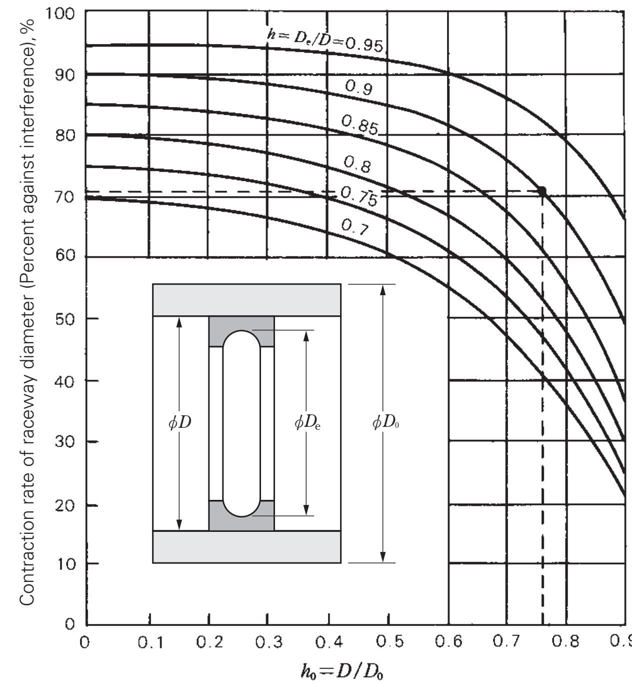


Fig. 8.17 Raceway Contraction in Relation to Bearing Fit (Outer Ring Fit in Steel Housing)

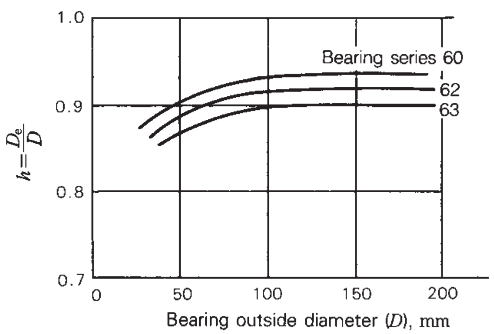


Fig. 8.18 Ratio of Outside Diameter to Raceway Diameter for Single Row Deep Groove Ball Bearings

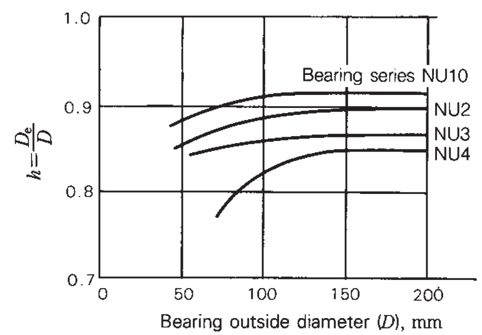


Fig. 8.19 Ratio of Outside Diameter to Raceway Diameter for Cylindrical Roller Bearings

8.3.6 Measuring Method of Internal Clearance of Combined Tapered Roller Bearings (Offset Measuring Method)

Combined tapered roller bearings are available in two types: a back-to-back combination (DB type) and a face-to-face combination (DF type) (see Fig. 8.20 and Fig. 8.21). The advantages of these combinations can be obtained by assembly as one set or combined with other bearings to be a fixed- or free-side bearing.
For the DB type of combined tapered roller bearing, as its cage protrudes from the back side of the outer ring, the outer ring spacer (K spacer in Fig. 8.20) is mounted to prevent mutual contact of cages. For the inner ring, the inner ring spacer (L spacer in Fig. 8.20), having an appropriate width, is provided to secure the clearance. For the DF type, as shown in Fig. 8.21, bearings are used with a K spacer.

In general, to use such a bearing arrangement either an appropriate clearance is required that takes into account the heat generated during operation or an applied preload is required 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.
Hereunder, we introduce you to a clearance measurement method for a DB arrangement.
(1) As shown in Fig. 8.22, put the bearing A on the surface plate and after stabilization of rollers by rotating the outer ring (more than 10 turns), measure the offset $f_A = T_A - B_A$.
(2) Next, as shown in Fig. 8.23, use the same procedure to measure the other bearing B for its offset $f_B = T_B - B_B$.
(3) Next, 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 combined tapered roller bearing can be obtained, with the use of symbols shown in Figs. 8.22 through 8.24 by Equation (8.16):

$$\Delta_a = (L - K) - (f_A + f_B) \quad \cdots\cdots (\mathbf{8.16})$$

As an example, for the combined tapered roller bearing HR32232JDB+KLR10AC3, confirm the clearance of the actual product conforms to the specifications.
First, refer to Table 8.17 and notice that the C3 clearance range is Δ_r=110 to 140 μm.
To compare this specification with the offset measurement results, convert it into an axial clearance Δ_a by using Equation (8.17):

$$\Delta_a = \Delta_r \cot\alpha \fallingdotseq \Delta_r \frac{1.5}{e} \quad \cdots\cdots (\mathbf{8.17})$$

where, e: Constant determined for each bearing No. (Listed in the Bearing Tables of NSK Rolling Bearings Catalog)

refering to the said catalog (Page C205), with use of e=0.44, the following is obtained:

$$\Delta_a = (110 \text{ to } 140) \times \frac{1.5}{e}$$
$$\fallingdotseq 380 \text{ to } 480\ \mu\text{m}$$

It is possible to 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.

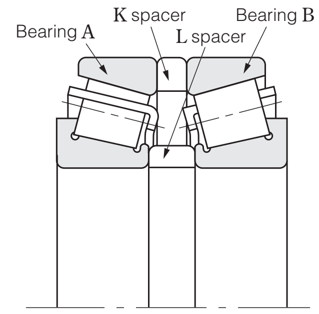


Fig. 8.20 DB Arrangement

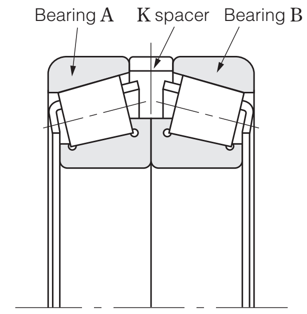


Fig. 8.21 DF Arrangement

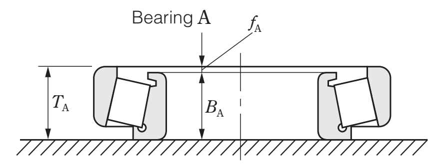


Fig. 8.22

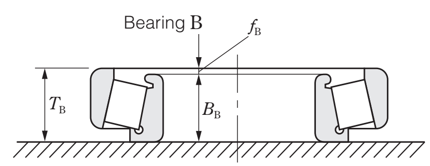


Fig. 8.23

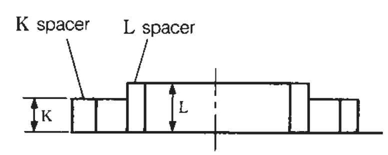


Fig. 8.24

8.3.7 Internal Clearance Adjustment Method when Mounting a Tapered Roller Bearing

The two single row tapered roller bearings are usually arranged in a configuration opposite each other and the clearance is adjusted in the axial direction. There are two types of opposite placement methods: back-to-back arrangement (DB arrangement) and face-to-face arrangement (DF arrangement).
The clearance adjustment of the 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, it is necessary that the fit of the tightening side inner ring with the shaft be a loose fit to allow displacement of the inner ring in the axial direction.
For the face-to-face arrangement, 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 (Fig. 8.26). In this case, it is necessary to use a loose fit between the tightening side of the outer ring and the housing in order to allow appropriate displacement of the outer ring in the axial direction. When the structure is designed to install the outer ring into the retaining cover (Fig. 8.27), the above measure becomes unnecessary and both mounting and dismounting become easy.
Theoretically when the bearing clearance is slightly negative during operation, the fatigue life becomes the longest, but if the negative clearance becomes much bigger, then the fatigue life becomes very short and heat generation quickly increases. Thus, it is generally arranged that the clearance be slightly positive (a little bigger than zero) while operating. In consideration of the clearance reduction caused by temperature difference of inner and outer rings during operation and difference of thermal expansion of the shaft and housing in the axial direction, the bearing clearance after mounting should be decided.
In practice, the clearance C1 or C2 is frequently adopted which is listed in Table 8.17.
In addition, the relationship between the 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 NSK Rolling Bearing 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. Such a method is called a preload method. There are two different modes of preloading: position preload and constant pressure preload. The position preload is used most often.
For the position preload, there are two methods: one method is to use an already adjusted arrangement of bearings and the other method is to apply the specified preload by tightening an adjustment nut or using an adjustment shim.
The constant pressure preload is a method to apply an appropriate preload to the bearing by means of spring or hydraulic pressure, etc. Next we introduce several examples that use these methods:
Fig. 8.28 shows the automotive final reduction gear. For pinion gears, the preload is adjusted by use of an inner ring spacer and shim. For large gears on the other hand, the preload is controlled by tightening the torque of the outer ring retaining screw.
Fig. 8.29 shows the rear wheel of a truck. This is an example of a preload application by tightening the inner ring in the axial direction with a shaft nut. In this case, the preload is controlled by measuring the starting friction moment of the bearing.
Fig. 8.30 shows an example of the head spindle of the lathe, the preload is adjusted by tightening the shaft nut.
Fig. 8.31 shows an example of a constant pressure preload for which the preload is adjusted by the displacement of the spring. In this case, first find a relationship between the spring's preload and displacement, then use this information to establish a constant pressure preload.

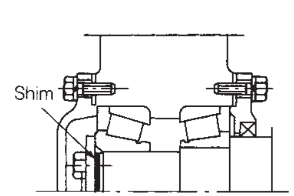


Fig. 8.25 DB Arrangement whose Clearance is Adjusted by Inner Rings.

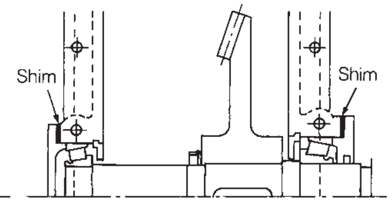


Fig. 8.26 DF Arrangement whose Clearance is Adjusted by Outer Rings.

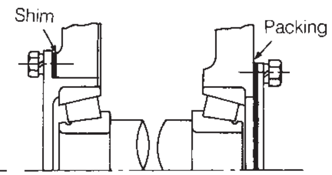


Fig. 8.27 Examples of Clearance Adjusted by Shim Thickness of Outer Ring Cover

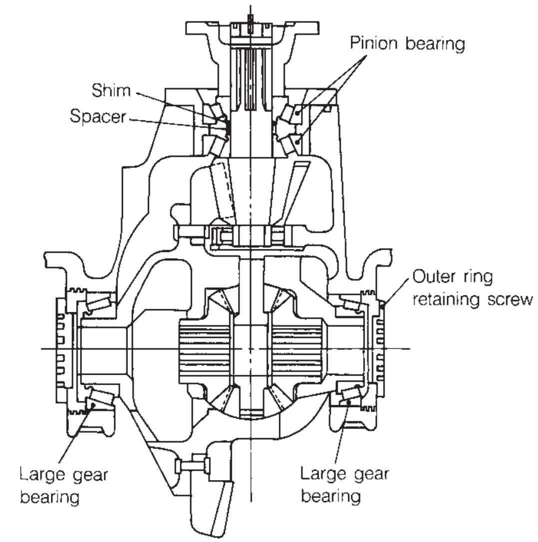


Fig. 8.28 Automotive Final Reduction Gear

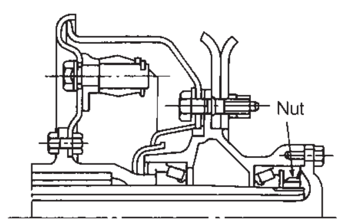


Fig. 8.29 Rear Wheel of Truck

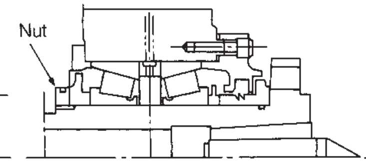


Fig. 8.30 Head Spindle of Lathe

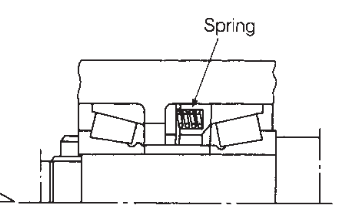
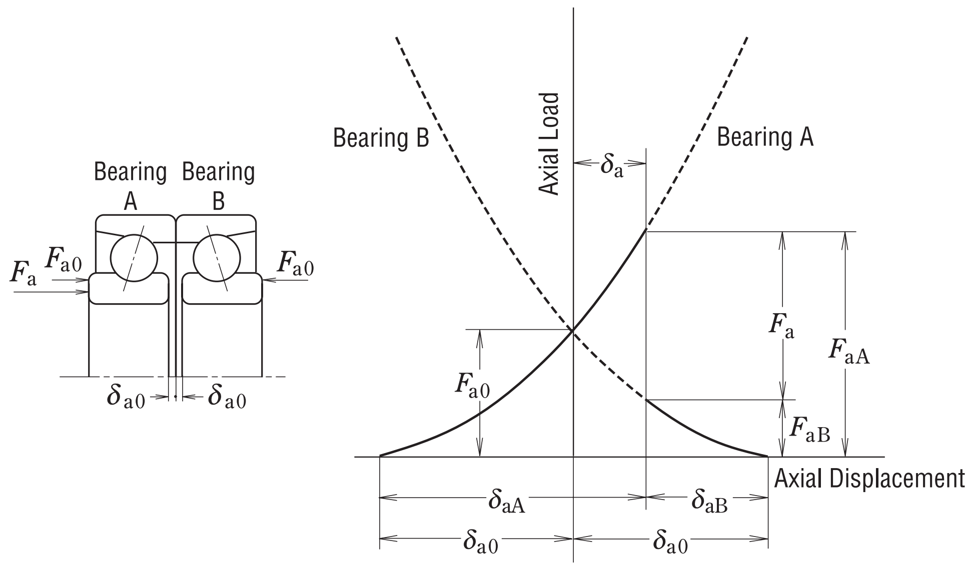


Fig. 8.31 Constant Pressure Preload Applied by Spring


Bearing A
Bearing B
F_a
F_{a0}
F_{a0}
δ_{a0}
δ_{a0}
Bearing B
Bearing A
Axial Load
δ_a
F_a
F_{aA}
F_{a0}
F_{aB}
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, it is desirable to provide a negative clearance to keep them internally stressed. This is called "preloading". A preload is usually applied to bearings in which the 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 duplex set 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 exact position both radially and axially and to maintain the running accuracy of the shaft.
…Main shafts of machine tools, precision instruments, etc.
(2) To increase bearing rigidity
…Main shafts of machine tools, pinion shafts of final drive gears of automobiles, etc.
(3) To minimize noise due to axial vibration and resonance
…Small electric motors, etc.
(4) To prevent sliding between the rolling elements and raceways due to gyroscopic moments
…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
…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, remain unchanged while in operation.
In practice, the following three methods are generally used to obtain a position preload.

(1) By installing a duplex bearing set with previously adjusted stand-out dimensions (see Page A007, Fig. 1.1) and axial clearance.
(2) By using a spacer or shim of proper size to obtain the required spacing and preload. (Refer to Fig. 9.1)
(3) By 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 using a coil or leaf spring to impose a constant preload. Even if the relative position of the bearings changes during operation, the magnitude of the preload remains relatively constant (refer to Fig. 9.2)

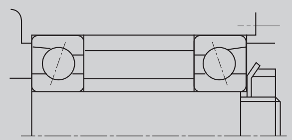


Fig. 9.1 Position Preload

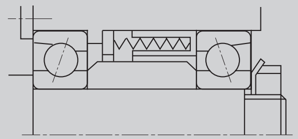


Fig. 9.2 Constant-Pressure Preload

9.3 Preload and Rigidity

9.3.1 Position Preload and Rigidity

When the inner rings of the duplex bearings shown in Fig.9.3 are fixed axially, bearings A and B are displaced δ_{a0} and axial space $2\delta_{a0}$ between the inner rings is eliminated. With this condition, a preload F_{a0} is imposed on each bearing. A preload diagram showing bearing rigidity, that is the relation between load and displacement with a given axial load F_a imposed on a duplex set, is shown in Fig. 9.4.

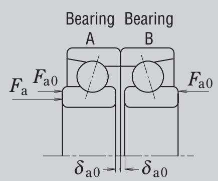


Fig. 9.3 Back-to-Back Duplex Bearing Preload

9.3.2 Constant-Pressure Preload and Rigidity

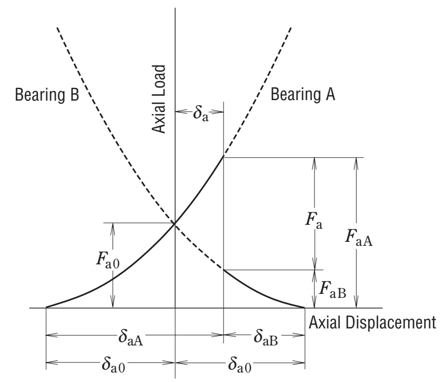
A preload diagram for duplex bearings under a constant-pressure preload is shown in Fig. 9.5. The deflection curve of the spring is nearly parallel to the horizontal axis because the rigidity of springs is lower than that of the bearing. As a result, the rigidity under a constant-pressure preload is approximately equal to that for a single bearing with a preload F_{a0} applied to it. Fig. 9.6 presents a comparison of the rigidity of 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 using both preloading methods is shown in Fig. 9.6. The position preload and constant-pressure preload may be compared as follows:

(1) When both of the preloads are equal, the position preload provides greater bearing rigidity, in other words, the deflection due to external loads is less for bearings with a position preload.
(2) In the case of a position preload, the preload varies depending on such factors as a difference in axial expansion due to a temperature difference between the shaft and housing, a difference in radial expansion due to a temperature difference between the inner and outer rings, deflection due to load, etc.



F_a : Axial load applied from outside
F_{aA}: Axial load imposed on Bearing A
F_{aB}: Axial load imposed on Bearing B
δ_a : Displacement of duplex bearing set
δ_{aA} : Displacement of Bearing A
δ_{aB} : Displacement of Bearing B

Fig. 9.4 Axial Displacement with Position Preload

In the case of a constant-pressure preload, it is possible to minimize any change in the preload because the variation of the spring load with shaft expansion and contraction is negligible. From the foregoing explanation, it is seen that position preloads are generally preferred for increasing rigidity and constant-pressure preloads are more suitable for high speed applications, for prevention of axial vibration, for use with thrust bearings on horizontal shafts, etc.

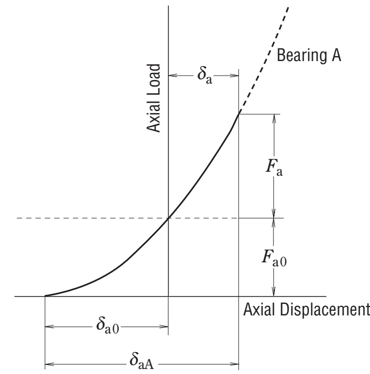


Fig. 9.5 Axial Displacement with Constant-Pressure Preload

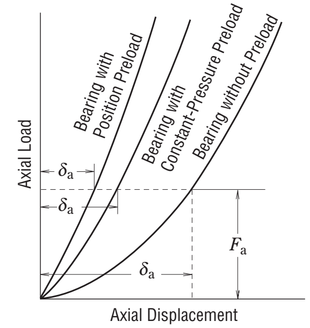


Fig. 9.6 Comparison of Rigidities and Preloading Methods

9.5 Amount of Preload

If the preload is larger than necessary, abnormal heart generation, increased frictional torque, reduced fatigue life, etc. may occur. The amount of the 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, etc. At NSK, preloads are divided into four graduated classifications — Extra light (EL), Light (L), Medium (M), and Heavy (H) — to allow the customer to freely choose the appropriate preload for the specific application. These four preload classes are expressed in symbols, EL, L, M, and H, respectively, when applied to DB and DF bearing sets.

The average preload and axial clearance (measured) for duplex angular contact ball bearing sets with contact angles 15° and 30° (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 the 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. The housing fits should be selected in the lower part of the standard clearance for bearings in fixed-end applications and the higher part of the standard clearance for bearings in free-end applications.

As general rules when selecting preloads, grinding machine spindles or machining center spindles require extra light to light preloads, whereas lathe spindles, which need rigidity, require medium preloads.
The bearing preloads, if the bearing set is mounted with tight fit, are larger than those shown in Tables 9.3 to 9.5. Since excessive preloads cause bearing temperature rise and seizure, etc., it is necessary to pay attention to fitting.
When speeds result in a value of $D_{pw}\times n$ ($d_m n$ value) higher than 500000, the preload should be very carefully studied and selected. In such a case, please consult with NSK beforehand.

Table 9.1 Measuring Load of Axial Clearance

Nominal bearing outside diameter D (mm) over	incl	Measuring load (N)
10*	50	24.5
50	120	49
120	200	98
200	—	196

*10 mm is included in this range.

Table 9.2 Target of Fitting

Units : μm

Bore or outside diameter d or D (mm) over	incl	Shaft and inner ring: Target interference	Housing and outer ring: 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 Preloads and Axial Clearance for Bearing Series 79C

Bearing No.	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)
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 In the axial clearance column, the measured value is given.

Table 9.4 Average Preloads and Axial Clearance for Bearing Series 70C

Bearing No.	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 In the axial clearance column, the measured value is given.

Table 9.5 Average Preloads and Axial Clearance for Bearing Series 72C

Bearing No.	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 In the axial clearance column, the measured value is given.

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

$$F_{a\,min}=\frac{C_{0a}}{100}\left(\frac{n}{N_{max}}\right)^2 \quad \textbf{(9.1)}$$

$$F_{a\,min}=\frac{C_{0a}}{1000} \quad \textbf{(9.2)}$$

where $F_{a\,min}$: Minimum axial load (N), {kgf}
n : Speed (min^{-1})
C_{0a} : Basic static load rating (N), {kgf}
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:

$$F_{a\,min}=\frac{C_{0a}}{1000} \quad \textbf{(9.3)}$$

9.6 Technical Data

9.6.1 Load and Displacement of Position-Preloaded Bearings

Two (or more) ball or tapered roller bearings mounted side by side as a set are termed duplex (or multiple) bearing sets. The bearings most often used in multiple arrangements are single-row angular contact ball bearings for machine tool spindles, since there is a requirement to reduce the bearing displacement under load as much as possible.
There are various ways of assembling sets depending on the effect desired. Duplex angular contact bearings fall into three types of arrangements, Back-to-Back, with lines of force convergent on the bearing back faces, Face-to-Face, with lines of force convergent on the bearing front faces, and Tandem, with lines of force being parallel. The symbols for these are DB, DF, and DT arrangements respectively (Fig. 9.7).

DB and DF arrangement sets can take axial loads in either direction. Since the distance of the load centers of DB bearing set is longer than that of DF bearing set, they are widely used in applications where there is a moment. DT type sets can only take axial loads in one direction. However, because the two bearings share some load equally between them, a set can be used where the load in one direction is large.
By selecting the DB or DF bearing sets with the proper preloads which have already been adjusted to an appropriate range by the bearing manufacturer, the radial and axial displacements of the bearing inner and outer ring can be reduced as much as allowed by certain limits. However, the DT bearing set cannot be preloaded.
The amount of preload can be adjusted by changing clearance between bearings, δ_{a0}, as shown in Figs. 9.9 to 9.11. Preloads are divided into four graduated classification — Extra light (EL), Light (L), Medium (M), and Heavy (H). Therefore, DB and DF bearing sets are often used for applications where shaft misalignments and displacements due to loads must be minimized.
Triplex sets are also available in three types (symbols: DBD, DFD, and DTD) of arrangements as shown in Fig. 9.8. Sets of four or five bearings can also be used depending on the application requirements.
Duplex bearings are often used with a preload applied. Since the preload affects the rise in bearing temperature during operation, torque, bearing noise, and especially bearing life, it is extremely important to avoid applying an excessive preload.
Generally, the axial displacement δ_a under an axial load F_a for single-row angular contact ball bearings is calculated as follows,

$$\delta_a = c\,F_a^{2/3} \quad \cdots\cdots (9.4)$$

where, c: Constant depending on the bearing type and dimensions.

Fig. 9.9 shows the preload curves of duplex DB arrangement, and Figs. 9.10 and 9.11 show those for triplex DBD arrangement.
If the inner rings of the duplex bearing set in Fig. 9.9 are pressed axially, A-side and B-side bearings are deformed δ_{a0A} and δ_{a0B} respectively and the clearance (between the inner rings), δ_{a0}, becomes zero. This condition means that the preload F_{a0} is applied on the bearing set. If an external axial load F_a is applied on the preloaded bearing set from the A-side, then the A-side bearing will be deformed δ_{a1} additionally 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. That is, the load on A-side bearing including the preload is ($F_{a0}+F_a-F_a'$) and the B-side bearing is ($F_{a0}-F_a'$).

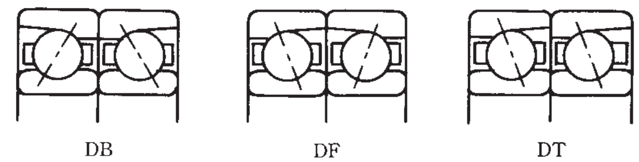


Fig. 9.7 Duplex Bearing Arrangements

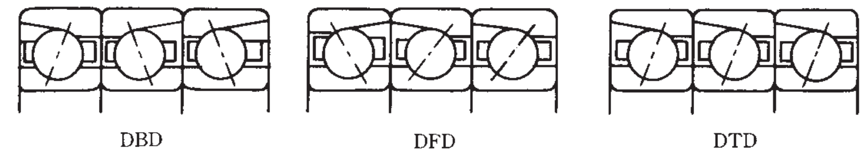


Fig. 9.8 Triplex Bearing Arrangements

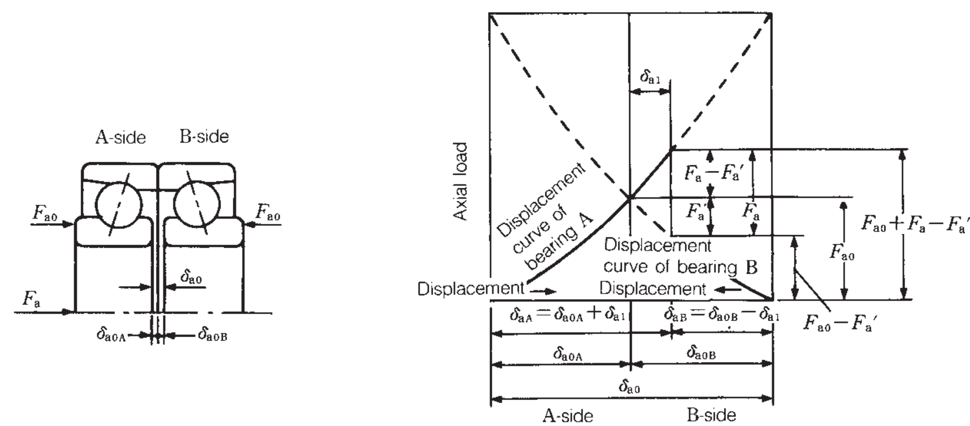


Fig. 9.9 Preload Graph of DB Arrangement Duplex Bearings

If the bearing set has an applied preload, the A-side bearing should have a sufficient life and load capacity for an axial load ($F_{a0}+F_a-F_a'$) under the speed condition. The axial clearance δ_{a0} is shown in Tables 9.3 to 9.5 of Section 9.5.1 (Pages A195 to A197).

In Fig. 9.10, with an external axial load F_a applied on the AA-side bearings, the axial loads and displacements of AA- and B-side bearings are summarized in Table 9.6.

In Fig. 9.11, with an external axial load F_a applied on the A-side bearing, the axial loads and displacements of A- and BB-side bearings are summarized in Table 9.7.

The examples, Figs. 9.12 to 9.17, show the relation of the axial loads and axial displacements using duplex DB and triplex 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	$\delta_{a0B}-\delta_{a1}$	$F_{a0}-F_a'$

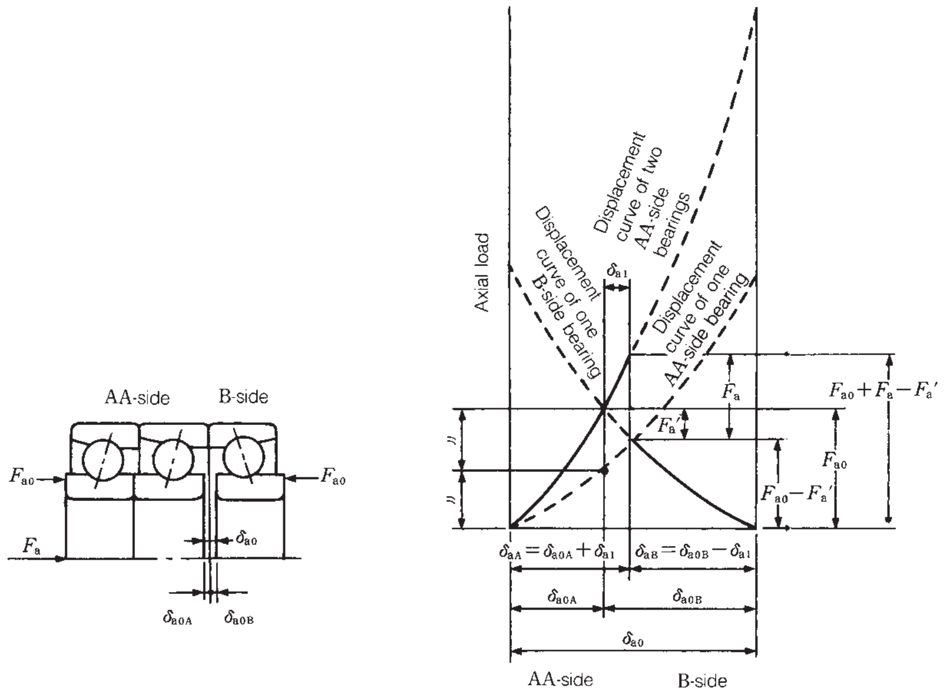


Fig. 9.10 Preload Graph of Triplex DBD Bearing Set (Axial load is applied from AA-side)

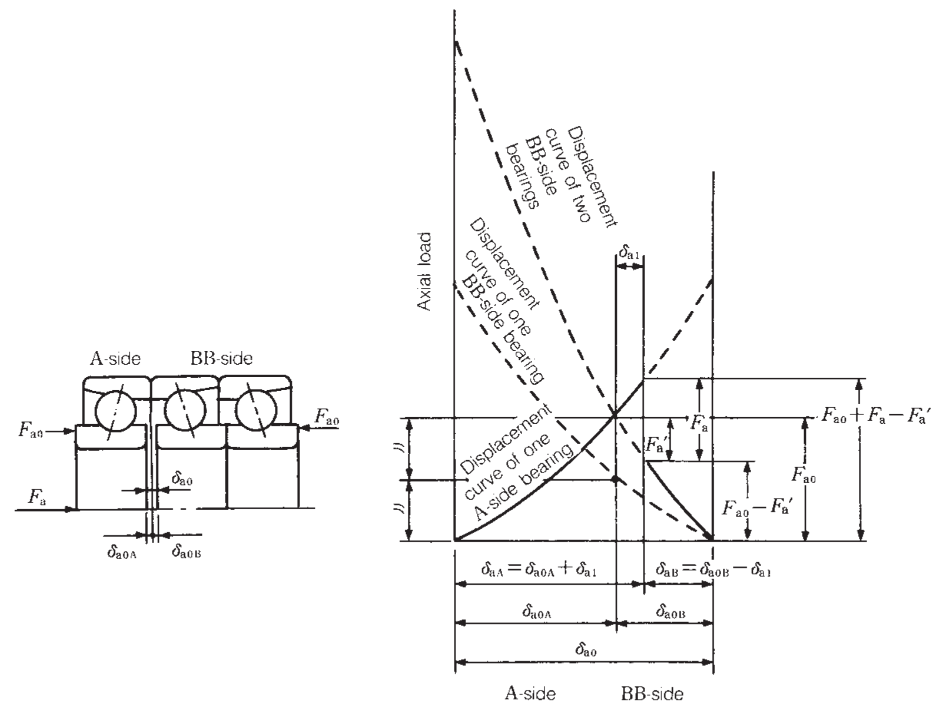


Fig. 9.11 Preload Graph of Triplex DBD Bearing set (Axial load is applied from A-side)

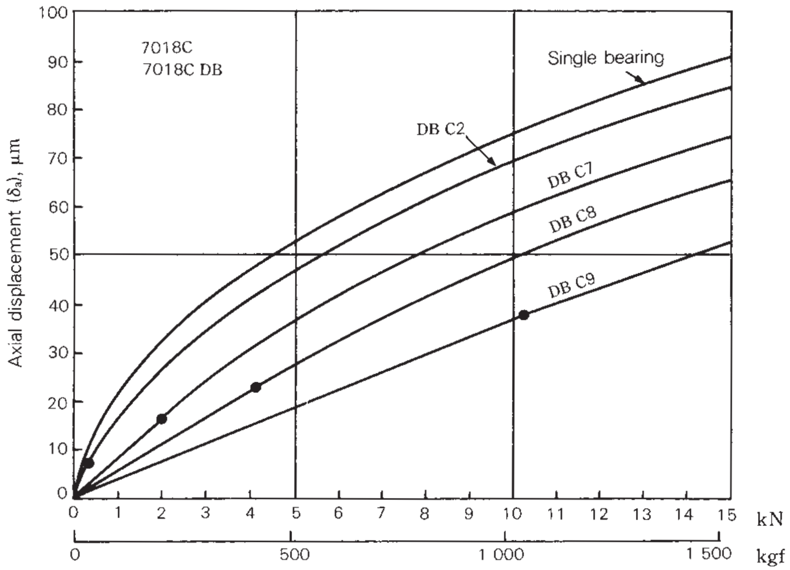


Fig. 9.12

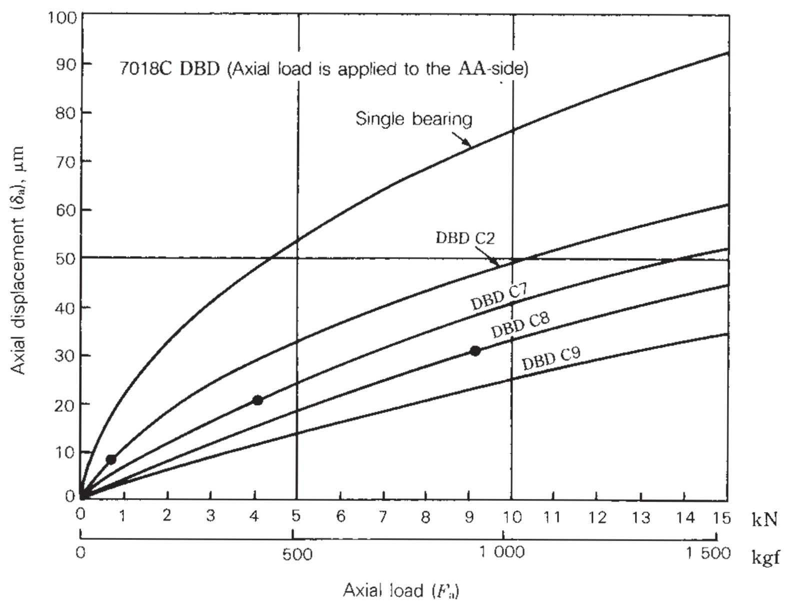


Fig. 9.13

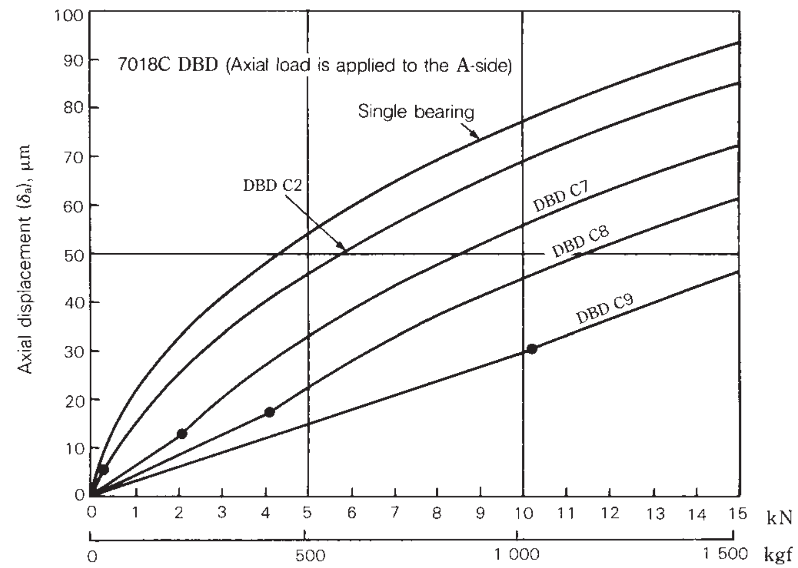


Fig. 9.14

Remark A (•) 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.

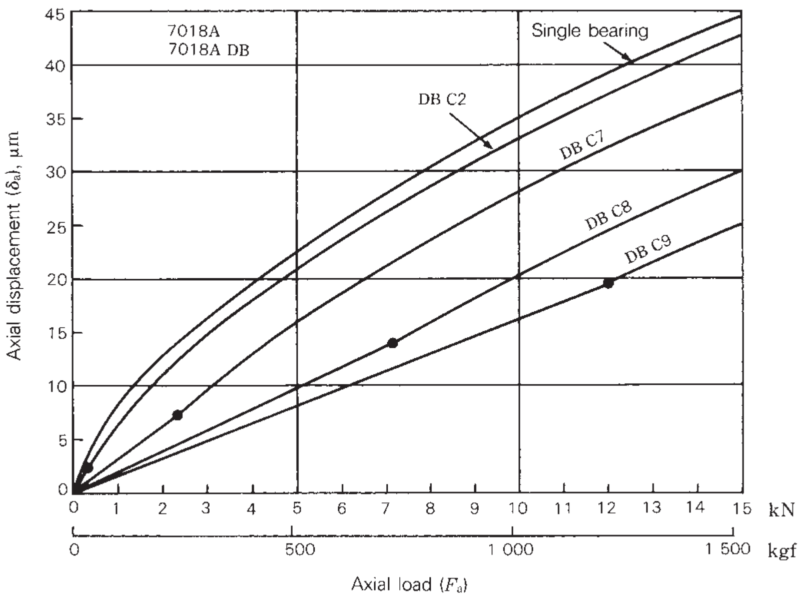


Fig. 9.15

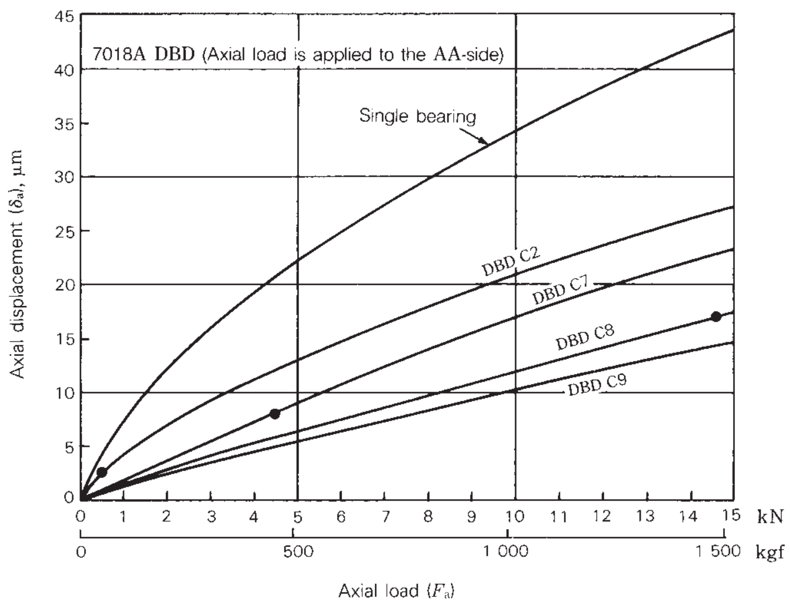


Fig. 9.16

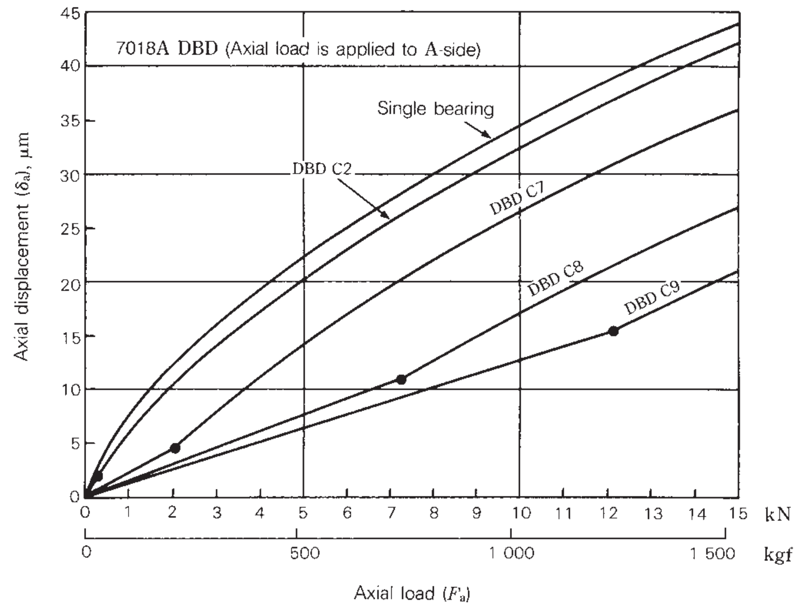


Fig. 9.17

Remark A (•) 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 a contact angle α_0 and the inner ring is displaced δ_a, the center Oi of the inner ring raceway radius is also moved to Oi' resulting in the 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.

$$(m_0+\delta_N)^2=(m_0\cdot\sin\alpha_0+\delta_a)^2+(m_0\cdot\cos\alpha_0)^2$$

$$\therefore\ \delta_N=m_0\left\{\sqrt{\left(\sin\alpha_0+\frac{\delta_a}{m_0}\right)^2+\cos^2\alpha_0}-1\right\}\ \cdots\cdots\cdots\cdots\ (\mathbf{9.5})$$

Also there is the following relationship between the rolling element load Q and elastic deformation δ_N.

$$Q=K_N\cdot\delta_N^{3/2}\ \cdots\cdots\cdots\cdots\ (\mathbf{9.6})$$

where, K_N: Constant depending on bearing material, type, and dimension

∴ 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 $K_N=K\cdot\dfrac{\sqrt{D_w}}{B^{3/2}}$

$$Q=K\cdot D_w^2\left\{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}-1\right\}^{3/2}\cdots\cdots\cdots\cdots\ (\mathbf{9.7})$$

On the other hand, the relation between the bearing axial load and rolling element load is shown in Equation (9.8) using Fig. 9.19:

$$F_a=Z\cdot Q\cdot\sin\alpha\ \cdots\cdots\cdots\cdots\ (\mathbf{9.8})$$

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),

$$\sin\alpha=\frac{\sin\alpha_0+h}{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}}\ \cdots\cdots\cdots\cdots\ (\mathbf{9.9})$$

That is, 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).

$$F_a=K\cdot Z\cdot D_w^2\cdot\frac{\left\{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}-1\right\}^{3/2}\times(\sin\alpha_0+h)}{\sqrt{(\sin\alpha_0+h)^2+\cos^2\alpha_0}}\ \cdots\cdots\cdots\cdots\ (\mathbf{9.10})$$

where, K: Constant depending on the bearing material and design
D_w: Ball diameter
Z: Number of balls
α_0: Initial contact angle
In case of single-row deep groove ball bearings, the initial contact angle can be obtained using Equation (5) of Page C012

Actual axial deformation varies depending on the bearing mounting conditions, such as the material and thickness of the shaft and housing, and bearing fitting. For details, consult with NSK regarding the axial deformation after mounting.

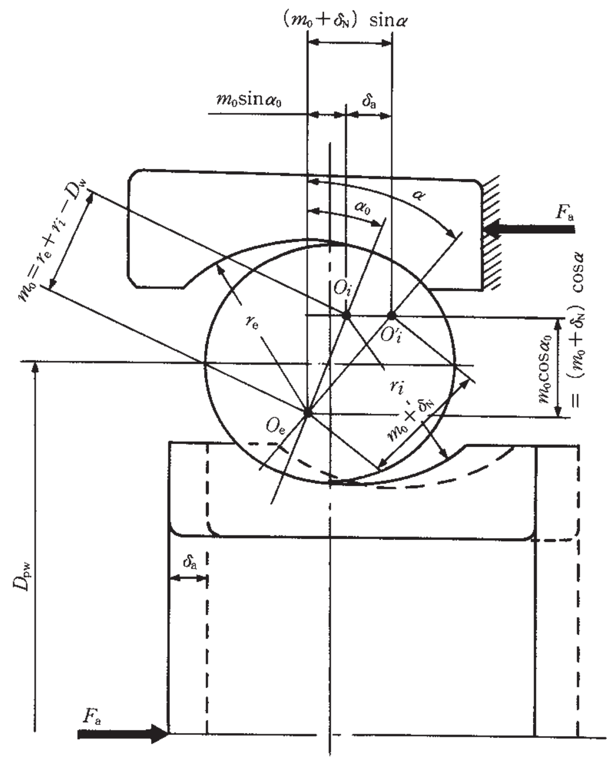


Fig. 9.18

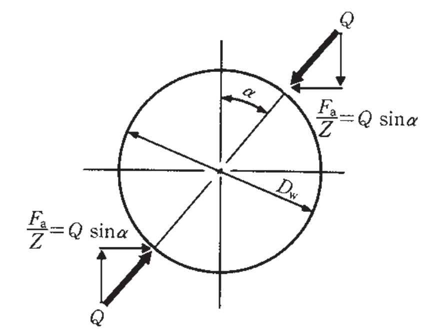


Fig. 9.19

Fig. 9.20 gives the relation between axial load and axial displacement for 6210 and 6310 single-row deep groove ball bearings with initial contact angles of α_0=0°, 10°, 15°. The larger the initial contact angle α_0, the more rigid the bearing will be in the axial direction and also the smaller the difference between the axial displacements of 6210 and 6310 under the same axial load. The angle α_0 depends upon the groove radius and the radial clearance.

Fig. 9.21 gives the relation between axial load and axial displacement for 72 series angular contact ball bearings with initial contact angles of 15° (C), 30° (A), and 40° (B). Because 70 and 73 series bearings with identical contact angles and bore diameters can be considered to have almost the same values as 72 series bearings.
Angular contact ball bearings that sustain loads in the axial direction must maintain their running accuracy and reduce the bearing elastic deformation from applied loads when used as multiple bearing sets with a preload applied.

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 $\delta_a \propto F_a^{2/3}$. That is, the axial displacement δ_a is proportional to the axial load F_a to the 2/3 power. When this axial load index is less than one, it indicates the relative axial displacement will be small with only a small increase in the axial load. (Fig. 9.21) The underlying reason for applying a preload is to reduce the amount of displacement.

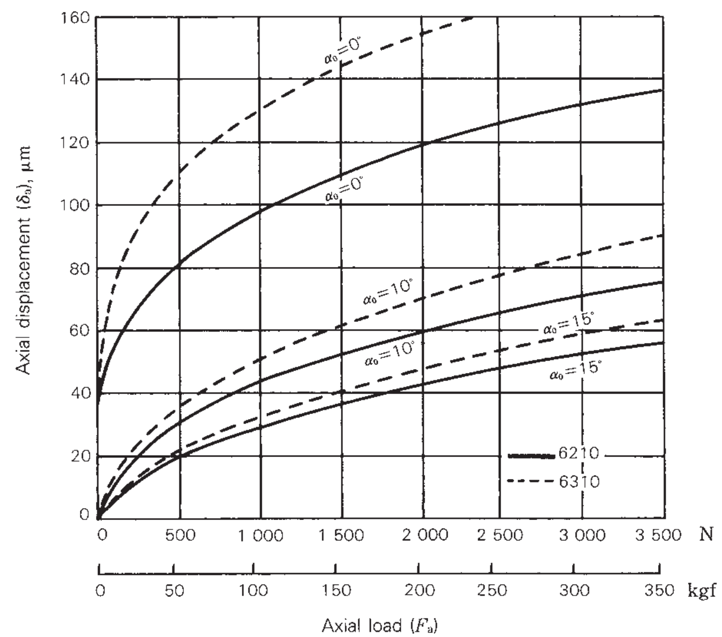


Fig. 9.20 Axial Load and Axial Displacement of Deep Groove Ball Bearings

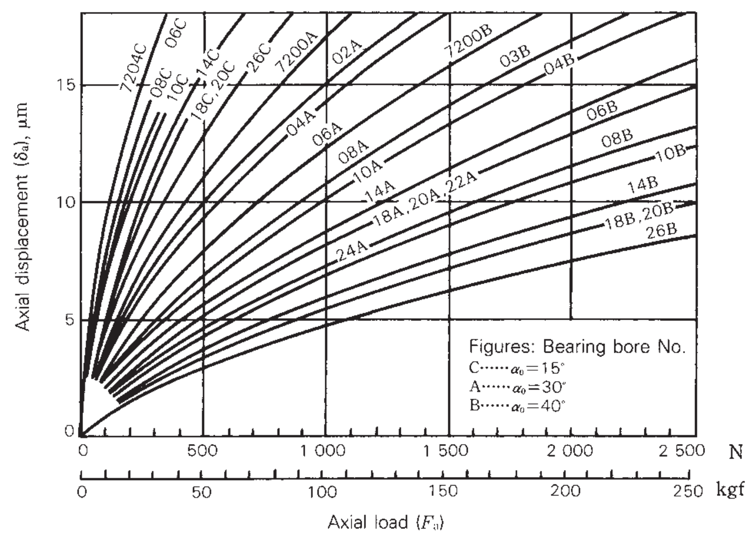


Fig. 9.21 Axial Load and Axial Displacement of Angular Contact Ball Bearings

9.6.3 Axial Displacement of Tapered Roller Bearings

Tapered roller bearings are widely used in pairs like angular contact ball bearings. Care should be taken to select 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 in an application, 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, 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 bearing fitting. For details, consult with NSK regarding the axial deformation after mounting.

$$\left.\begin{aligned}\delta_a &= \frac{0.000077}{\sin\alpha}\cdot\frac{Q^{0.9}}{L_{we}^{\,0.8}} && \text{(N)}\\ &= \frac{0.0006}{\sin\alpha}\cdot\frac{Q^{0.9}}{L_{we}^{\,0.8}} && \{\text{kgf}\}\end{aligned}\right\} \quad \cdots\cdots \ (\mathbf{9.11})$$

where, δ_a: Axial displacement of inner, outer ring (mm)
α: Contact angle...1/2 the cup 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: Coefficient determined by the 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.
Caution should be taken 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.

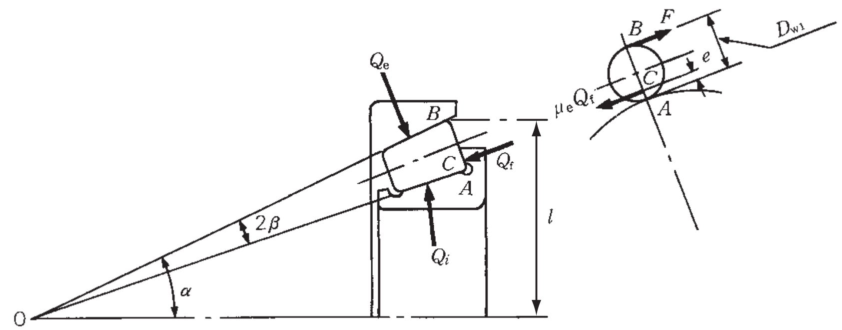


Fig. 9.22

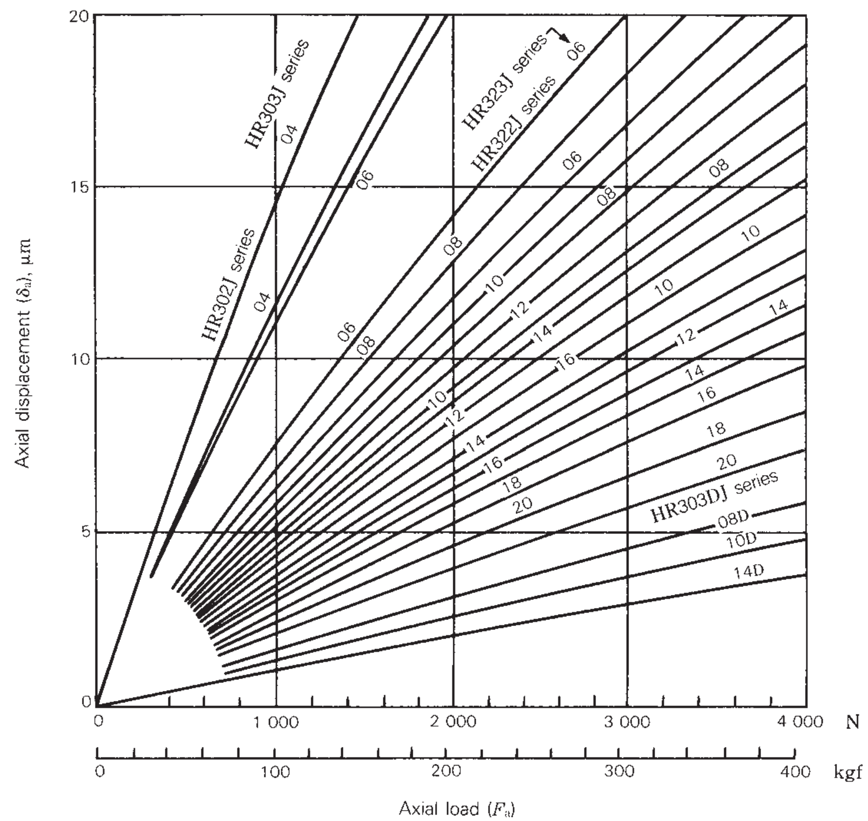
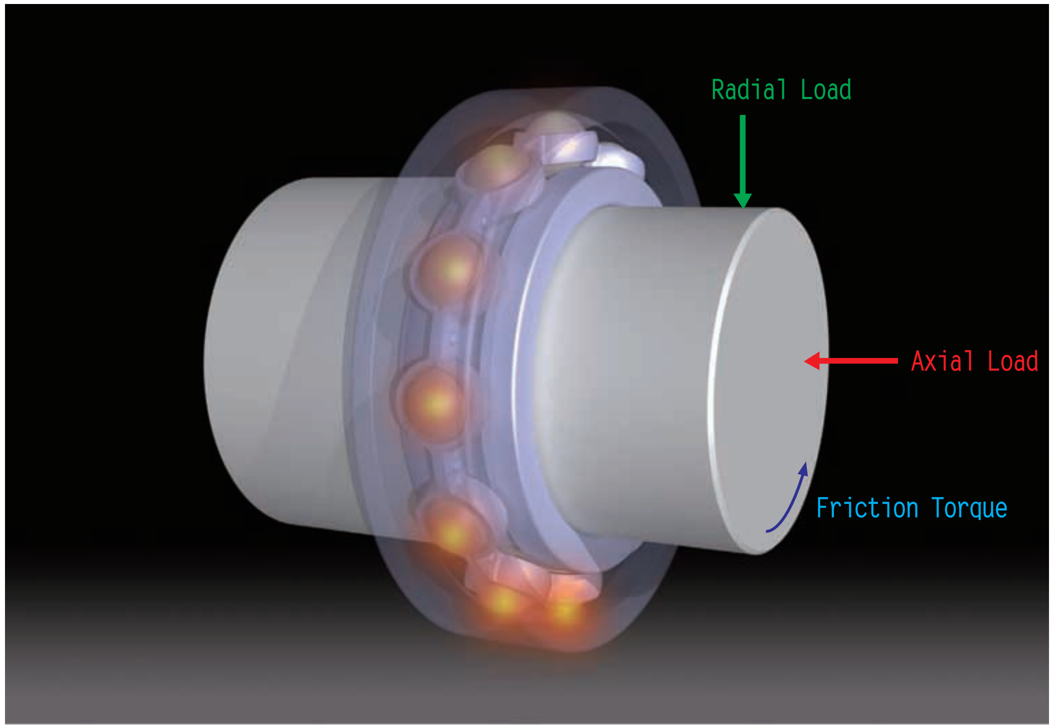


Fig. 9.23 Axial Load and Axial Displacement for Tapered Roller Bearings



10. FRICTION

10. FRICTION

10.1 Coefficients of Dynamic Friction

10.1.1 Bearing Types and Their Coefficients of Dynamic Friction μ

$$\mu=\frac{M}{P\cdot\frac{d}{2}} \quad \cdots\cdots\cdots\cdots \quad \textbf{(10.1)}$$

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 torque of bearing (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 for the bearing will increase in direct proportion to the preload.

The starting torque of angular contact ball bearings is mainly the torque 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,

$$M=M_s\cdot Z\sin\alpha \text{ (mN·m), {kgf·mm}} \quad \cdots\cdots\cdots \quad \textbf{(10.2)}$$

where, M_s: Spin friction for contact angle α centered on the shaft,

$$M_s=\frac{3}{8}\mu_s\cdot Q\cdot a\cdot E(k)$$

(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 70C and 72C series bearings.

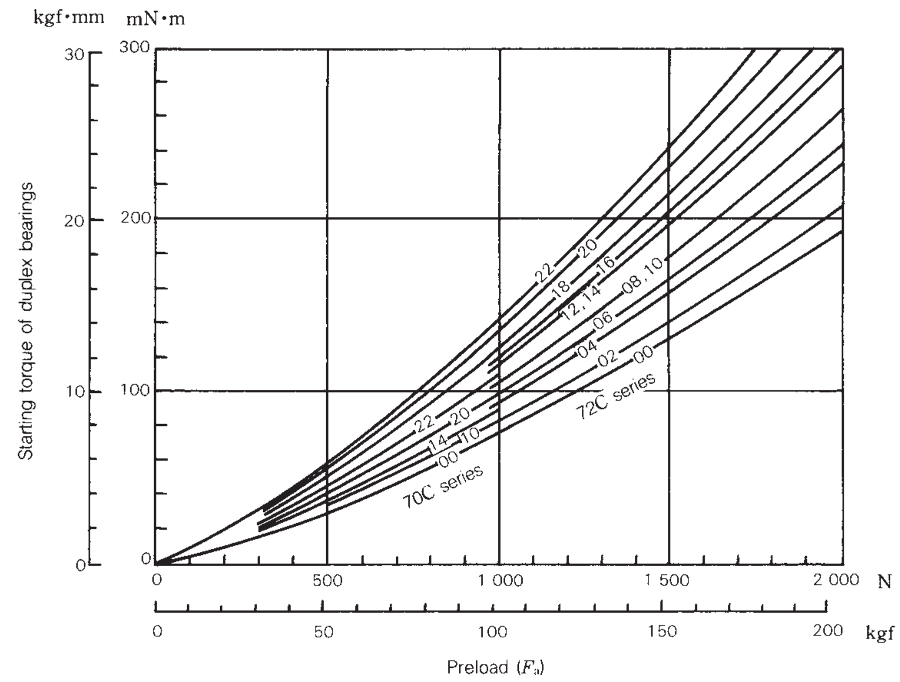


Fig. 10.1 Preload and Starting Torque for Angular Contact Ball Bearings (α=15°) of DF and DB Duplex Sets

10.3.2 Empirical Equation of Running Torque of High-Speed Ball Bearings

We present here empirical equations 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 bigger bearings.

The running torque M can be obtained as the sum of a load term M_l and speed term M_v as follows:

$$M = M_l + M_v \ (\mathrm{mN \cdot m}),\ \{\mathrm{kgf \cdot mm}\} \qquad \cdots\cdots\cdots\cdots\ \mathbf{(10.3)}$$

The load term M_l is the term for friction, which has no relation with speed or fluid friction, and is expressed by Equation (10.4) which is based on experiments.

$$\left.\begin{aligned} M_l &= 0.672 \times 10^{-3} D_{pw}^{0.7} F_a^{1.2} \ (\mathrm{mN \cdot m}) \\ &= 1.06 \times 10^{-3} D_{pw}^{0.7} F_a^{1.2} \ \{\mathrm{kgf \cdot mm}\} \end{aligned}\right\} \cdots\cdots\cdots\ \mathbf{(10.4)}$$

where, D_{pw}: Pitch diameter of rolling elements (mm)
F_a: Axial load (N), {kgf}

The speed term M_v is that for fluid friction, which depends on angular speed, and is expressed by Equation (10.5).

$$\begin{aligned} M_v &= 3.47 \times 10^{-10} D_{pw}^{3} n_i^{1.4} Z_B^{a} Q^{b} \ (\mathrm{mN \cdot m}) \\ &= 3.54 \times 10^{-11} D_{pw}^{3} n_i^{1.4} Z_B^{a} Q^{b} \ \{\mathrm{kgf \cdot mm}\} \end{aligned} \qquad \cdots\cdots\cdots\ \mathbf{(10.5)}$$

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)

The exponents a and b, that affect the oil viscosity and flow rate factors, depend only on the angular speed and are given by Equations (10.6) and (10.7) as follows:

$$a = 24 n_i^{-0.37} \qquad \cdots\cdots\cdots\ \mathbf{(10.6)}$$

$$b = 4 \times 10^{-9} n_i^{1.6} + 0.03 \qquad \cdots\cdots\cdots\ \mathbf{(10.7)}$$

An example of the estimation of 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 big, 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 ($\phi 20 \times \phi 47 \times 14$) under the following conditions:

n_i=70 000 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),

$$\begin{aligned} M_l &= 0.672 \times 10^{-3} D_{pw}^{0.7} F_a^{1.2} \\ &= 0.672 \times 10^{-3} \times 33.5^{0.7} \times 590^{1.2} \\ &= 16.6 \ (\mathrm{mN \cdot m}) \end{aligned}$$

$$\begin{aligned} M_l &= 1.06 \times 10^{-3} \times 33.5^{0.7} \times 60^{1.2} \\ &= 1.7 \ \{\mathrm{kgf \cdot mm}\} \end{aligned}$$

From Equations (10.6) and (10.7),

$$\begin{aligned} a &= 24 n_i^{-0.37} \\ &= 24 \times 70\,000^{-0.37} = 0.39 \\ b &= 4 \times 10^{-9} n_i^{1.6} + 0.03 \\ &= 4 \times 10^{-9} \times 70\,000^{1.6} + 0.03 = 0.26 \end{aligned}$$

From Equation (10.5),

$$\begin{aligned} M_v &= 3.47 \times 10^{-10} D_{pw}^{3} n_i^{1.4} Z_B^{a} Q^{b} \\ &= 3.47 \times 10^{-10} \times 33.5^{3} \times 70\,000^{1.4} \times 10^{0.39} \times 1.5^{0.26} \\ &= 216 \ (\mathrm{mN \cdot m}) \end{aligned}$$

$$\begin{aligned} M_v &= 3.54 \times 10^{-11} \times 33.5^{3} \times 70\,000^{1.4} \times 10^{0.39} \times 1.5^{0.26} \\ &= 22.0 \ \{\mathrm{kgf \cdot mm}\} \end{aligned}$$

$$M = M_l + M_v = 16.6 + 216 = 232.6 \ (\mathrm{mN \cdot m})$$

$$M = M_l + M_v = 1.7 + 22 = 23.7 \ \{\mathrm{kgf \cdot mm}\}$$

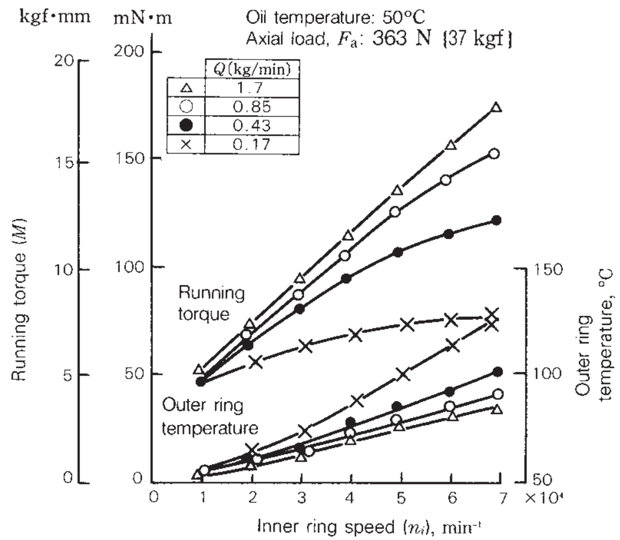


Fig. 10.2 Typical Test Example

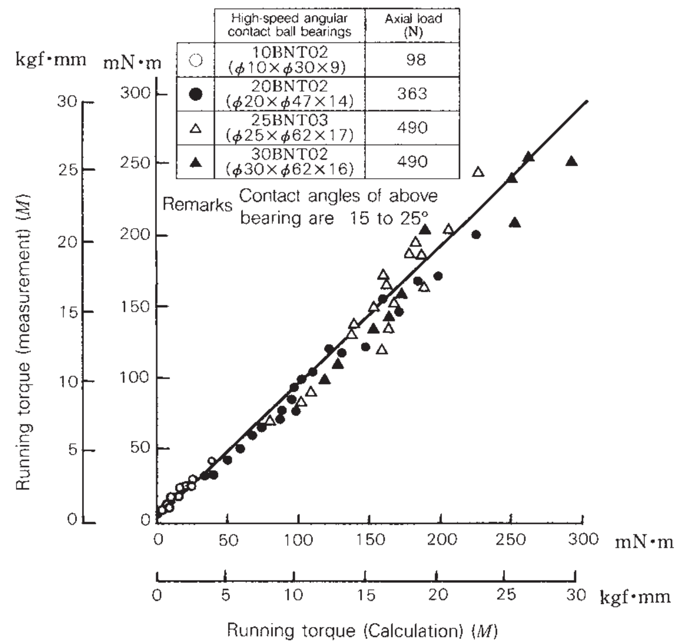


Fig. 10.3 Comparison of Actual Measurements and Calculated Values

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 (10.8), (10.9), and (10.10):

$$Q_e = \frac{F_a}{Z \sin\alpha} \quad \cdots\cdots \quad \textbf{(10.8)}$$

$$Q_i = Q_e \cos 2\beta = \frac{\cos 2\beta}{Z \sin\alpha} F_a \quad \cdots\cdots \quad \textbf{(10.9)}$$

$$Q_f = Q_e \sin 2\beta = \frac{\sin 2\beta}{Z \sin\alpha} F_a \quad \cdots\cdots \quad \textbf{(10.10)}$$

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...(1/2) of the cup angle (°)
β : (1/2) of 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,

$$D_{w1} F = e\,\mu_e Q_f \ (\text{mN}\cdot\text{m}),\ \{\text{kgf}\cdot\text{mm}\} \quad \cdots\cdots \quad \textbf{(10.11)}$$

where, μ_e: Friction coefficient between inner ring large rib and roller endface

The starting torque M for one bearing is given by,

$$M = F Z l$$

$$= \frac{e\,\mu_e\, l \sin 2\beta}{D_{w1} \sin\alpha} F_a \quad (\text{mN}\cdot\text{m}),\ \{\text{kgf}\cdot\text{mm}\} \quad \cdots\cdots \quad \textbf{(10.12)}$$

because, $D_{w1} = 2\,\overline{OB}\sin\beta$, and $l = \overline{OB}\sin\alpha$.

If we substitute these into Equation (10.12) we obtain,

$$M = e\,\mu_e \cos\beta\, F_a \ (\text{mN}\cdot\text{m}),\ \{\text{kgf}\cdot\text{mm}\} \quad \cdots\cdots \quad \textbf{(10.13)}$$

The 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. The torque caused by other factors can be ignored. Values for e and β in Equation (10.12) are determined by the bearing design. Consequently, assuming a value for μ_e, the starting torque can be calculated.

The values for μ_e and for e have to be thought of as a dispersion, thus, even for bearings with the same number, the individual starting torques can be quite diverse. When using a value for e determined by the bearing design, the average value for the bearing starting torque can be estimated using μ_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.

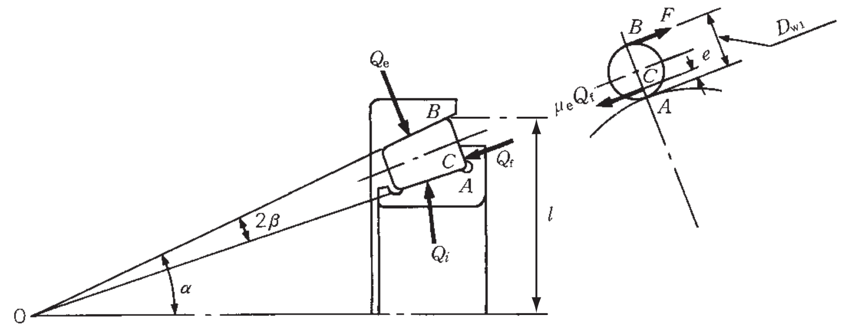


Fig. 10.4

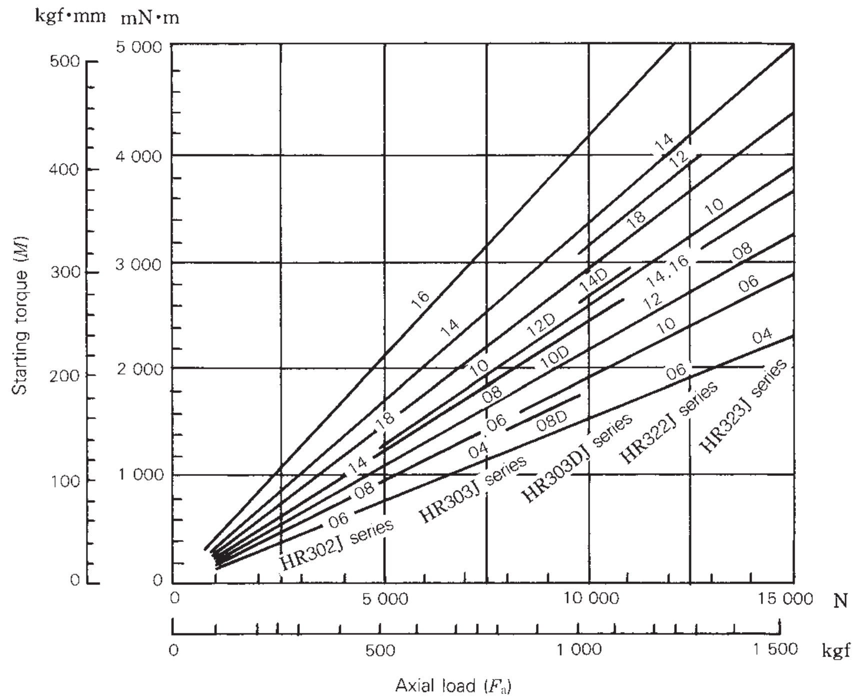


Fig. 10.5 Axial Load and Starting Torque for Tapered Roller Bearings

10.3.4 Empirical Equations for Running Torque of Tapered Roller Bearings

When tapered roller bearings operate under axial load, we reanalyzed the torque of tapered roller bearings 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.

$$Q_e \doteqdot Q_i = \frac{F_a}{Z \sin\alpha} \quad \cdots\cdots \quad (10.14)$$

$$Q_f = \frac{F_a \sin 2\beta}{Z \sin\alpha} \quad \cdots\cdots \quad (10.15)$$

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…(1/2) of the cup angle
β: (1/2) of tapered roller angle

For simplification, a model using the average diameter D_{we} as shows in Fig. 10.7 can be used.

Where, 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

In Fig. 10.7, when the balance of sliding friction and moments on the rollers are considered, the following equations are obtained:

$$F_{se} - F_{si} = F_{sf} \quad \cdots\cdots \quad (10.16)$$

$$M_i + M_e = \frac{D_w}{2} F_{se} + \frac{D_w}{2} F_{si} + \left(\frac{D_w}{2} - e\right) F_{sf} \quad \cdots\cdots \quad (10.17)$$

When the running torque M applied on the outer (inner) ring is calculated using Equations (10.16) and (10.17) and multiplying by Z, which is the number of rollers:

$$\begin{aligned} M &= Z\,(R_e F_{se} - M_e) \\ &= \frac{Z}{D_w}(R_e M_i + R_i M_e) + \frac{Z}{D_w} R_e\, e\, F_{sf} \\ &= M_R + M_S \end{aligned}$$

Therefore, the friction on the raceway surface M_R and that on the ribs M_S are separately obtained. Additionally, M_R and M_S are rolling friction and sliding friction respectively.

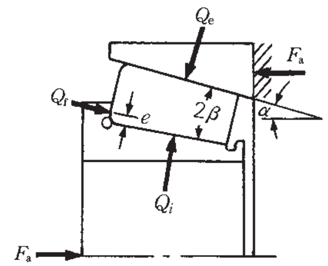


Fig. 10.6 Loads Applied on Roller

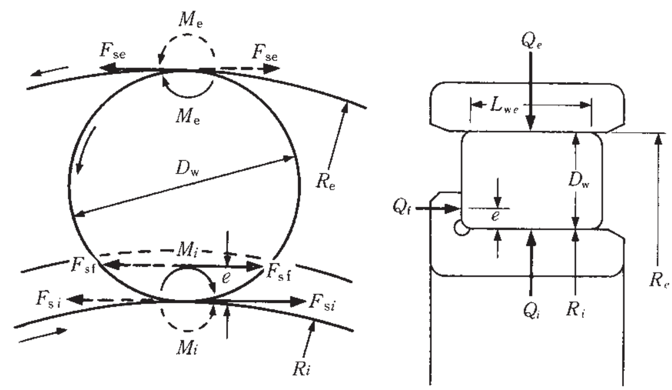


Fig. 10.7 Model of Parts where Friction is Generated

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.

$$M=M_R+M_S=\frac{Z}{D_w}(R_eM_i+R_iM_e)+\frac{Z}{D_w}R_e\ e\ F_{sf} \quad \cdots\cdots \quad (10.18)$$

Sliding Friction on Rib M_S

As a part of M_S, F_{sf} is the tangential load caused by sliding, so we can write $F_{sf}=\mu Q_f$ using the coefficient of dynamic friction μ. Further, by substitution of the axial load F_a,
the following equation is obtained:

$$M_S=e\,\mu\cos\beta\,F_a \quad \cdots\cdots \quad (10.19)$$

This is the same as the equation for starting torque, but μ is not constant and it decreases depending on the conditions or running in. For this reason, Equation (10.19) can be rewritten as follows:

$$M_S=e\,\mu_0\cos\beta\,F_a f'(\Lambda, t, \sigma) \quad \cdots\cdots \quad (10.20)$$

Where μ_0 is approximately 0.2 and $f'(\Lambda, t, \sigma)$ is a function which decreases with running in and oil film formation, but it is set equal to one when starting.

Rolling Friction on Raceway Surface M_R

Most of the rolling friction on the raceway is viscous oil resistance (EHL rolling resistance). M_i and M_e in Equation (10.18) correspond to it. A theoretical equation exists, but it should be corrected as a result of experiments. We obtained the following equation that includes corrective terms:

$$M_{i,e}=\left[f(w)\left(\frac{1}{1+0.29L^{0.78}}\right)\frac{4.318}{\alpha_0}(G\cdot U)^{0.658}W^{0.0126}R^2L_{we}\right]_{i,e} \quad \cdots\cdots \quad (10.21)$$

$$f(w)=\left(\frac{kF_a}{E'\,D_wL_{we}Z\sin\alpha}\right)^{0.3} \quad \cdots\cdots \quad (10.22)$$

Therefore, M_R can be obtained using Equations (10.21) and (10.22) together with the following equation:

$$M_R=\frac{Z}{D_w}(R_eM_i+R_iM_e)$$

Running Torque of Bearings M

From these, the running torque of tapered roller bearings M is given by Equation (10.23)

$$M=\frac{Z}{D_w}(R_eM_i+R_iM_e)+e\,\mu_0\cos\beta\,F_af'(\Lambda, t, \sigma) \quad \cdots\cdots \quad (10.23)$$

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. When needed, please consult NSK.

[Explanation of Symbols]

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 (Half of cup angle)
R_i, R_e:	Inner and outer ring raceway radii (center)
β:	Half angle of roller
i, e:	Indicate inner ring or outer ring respectively
L_{we}:	Effective roller length

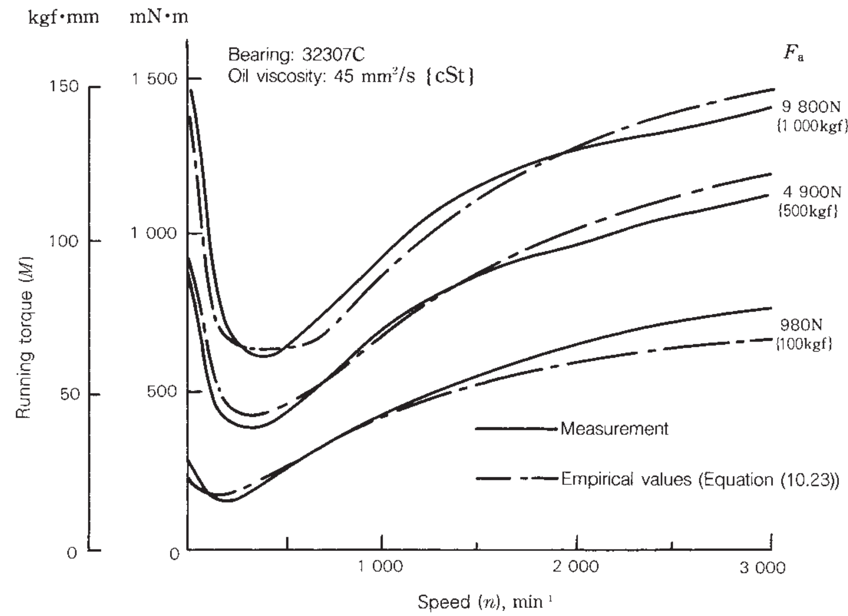


Fig. 10.8 Comparison of Empirical Values with Actual Measurements

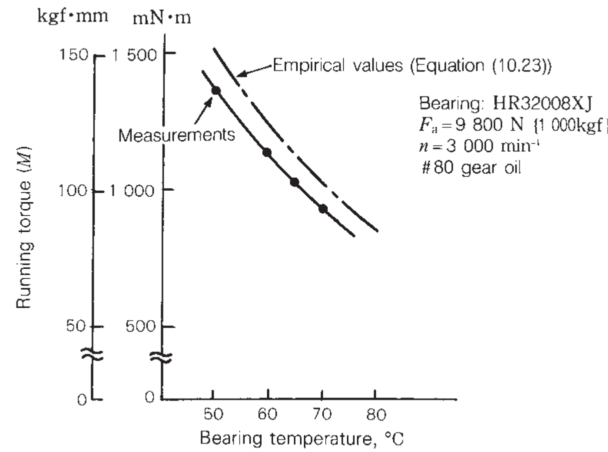


Fig. 10.9 Viscosity Variation and Running Torque

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 bearing rings, rolling elements and cage, which are the basic components of a bearing, is prevented by an oil film which reduces the friction and wear in the 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. A heavy film thickness prolongs the fatigue life, but it is shortened if the viscosity of the oil is too low so the film thickness is insufficient.

(3) Dissipation of Frictional Heat and Cooling
Circulation lubrication may be used to carry away frictional heat or heat transferred from the outside to prevent the bearing from overheating and the oil from deteriorating.

(4) Others
Adequate lubrication also helps to prevent foreign material from entering the bearings and guards against corrosion or rusting.

11.2 Lubricating Methods

The various lubricating methods are first divided into either grease or oil lubrication. Satisfactory bearing performance can be achieved by adopting the lubricating method which is most suitable for the particular application and operating condition.
In general, oil offers superior lubrication; however, grease lubrication allows 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 and Sealing Method	Simple	May be complex, Careful maintenance required.
Speed	Limiting speed is 65% to 80% of that with oil lubrication.	Higher limiting speed.
Cooling Effect	Poor	Heat transter is possible using forced oil circulation.
Fluidity	Poor	Good
Full Lubricant Replacement	Sometimes difficult	Easy
Removal of Foreign Matter	Removal of particles from grese is impossible.	Easy
External Contamination due to Leakage	Surroundings seldom contaminated by leakage.	Often leaks without proper countermeasures. Not suitable if external 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, the bearings for the main shafts of machine tools, where the accuracy may be impaired by a small temperature rise, require only a small amount of grease. The quantity of grease for ordinary bearings is determined as follows.
Sufficient grease must be packed inside the bearing including the cage guide face. The available space inside the housing to be packed with grease depends on the speed as follows:
1/2 to 2/3 of the space … When the speed is less than 50% of the limiting speed.
1/3 to 1/2 of the space … When the speed is more than 50% of the limiting speed.

(2) Replacement of Grease
Grease, once packed, usually need not be replenished for a long time; however, for 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 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 is made larger than the partitioned side so it can retain the old grease, which is removed periodically by removing the cover.

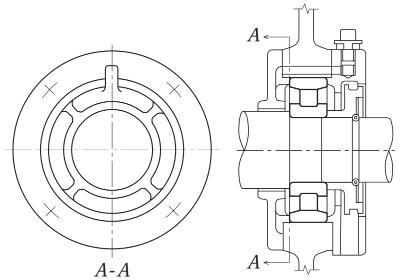
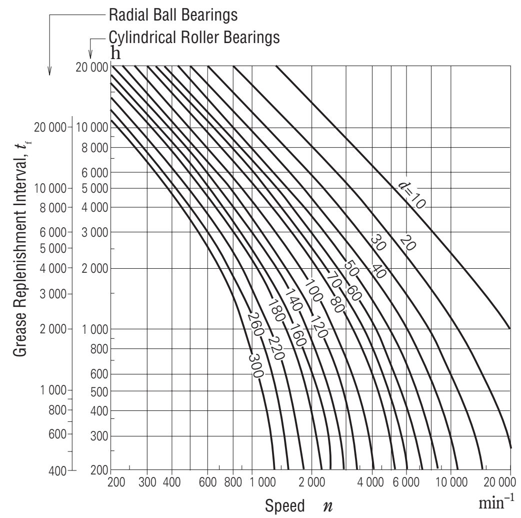
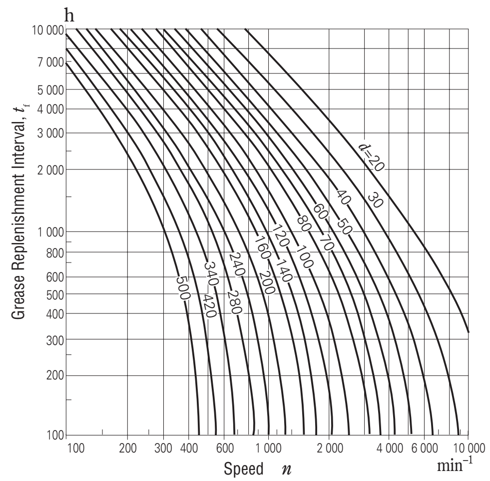


Fig. 11.1 Combination of Partitioned Grease Reservoir and Grease Valve

(3) Replenishing Interval
Even if high-quality grease is used, there is deterioration of its properties with time; therefore, periodic replenishment is required. Figs 11.2 (1) and (2) show the replenishment time intervals for various bearing types running at different speeds. Figs.11.2 (1) and (2) apply for the condition of high-quality lithium soap-mineral oil grease, bearing temperature of 70°C, and normal load (P/C=0.1).
· Temperature
If the bearing temperature exceeds 70°C, the replenishment time interval must be reduced by half for every 15°C temperature rise of the bearings.
· Grease
In case of ball bearings especially, the replenishing time interval can be extended depending on used grease type. (For example, high-quality lithium soap-synthetic oil grease may extend about two times of replenishing time 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.)
It is advisable to consult NSK.
· Load
The replenishing time interval depends on the magnitude of the bearing load.
Please refer to Fig.11.2 (3).
If P/C exceeds 0.16, it is advisable to 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) or (11.2) or Fig. 11.3:

(General purpose grease ([1]))

$$\log t = 6.54 - 2.6\,\frac{n}{N_{max}} - \left(0.025 - 0.012\,\frac{n}{N_{max}}\right)T \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\textbf{(11.1)}$$

(Wide-range grease ([2]))

$$\log t = 6.12 - 1.4\,\frac{n}{N_{max}} - \left(0.018 - 0.006\,\frac{n}{N_{max}}\right)T \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\textbf{(11.2)}$$

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 listed in the bearing tables)
T : Operating temperature °C

Equations (11.1) and (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 $\frac{n}{N_{max}} = 0.25$

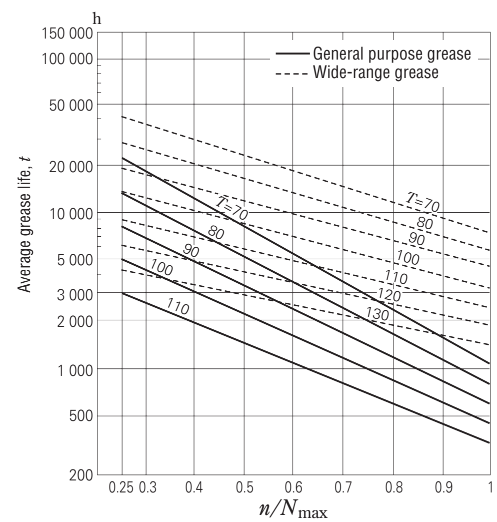


Fig. 11.3 Grease Life of Sealed Ball Bearings

(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 of the basic load rating C_r.

Notes ([1]) Mineral-oil base greases (e.g. lithium soap base grease) which are often used over a temperature range of around – 10 to 110 °C.
([2]) Synthetic-oil base greases are usable over a wide temperature range of around – 40 to 130 °C.

11.2.2 Oil Lubrication

(1) Oil Bath Lubrication

Oil bath lubrication is a widely used with low or medium speeds. The oil level should be at the center of the lowest rolling element. It is desirable to 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 the screw in the top.

(3) Splash Lubrication

With this lubricating method, oil is splashed onto the bearings by gears or a simple rotating disc installed near bearings without submerging the bearings in oil.
It is commonly used in automobile transmissions and final drive gears. Fig. 11.6 shows this lubricating method used on a reduction gear.

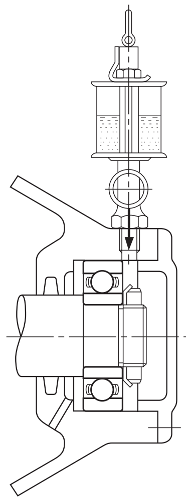


Fig. 11.5 Drip Feed Lubrication

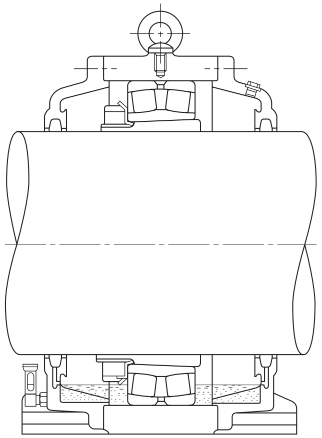


Fig. 11.4 Oil Bath Lubrication

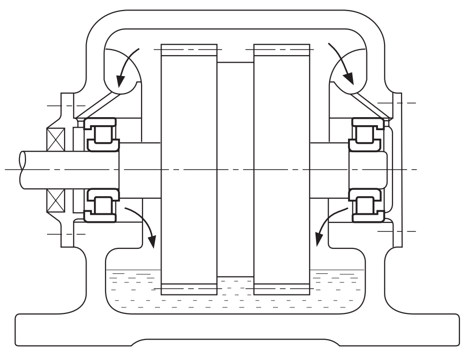
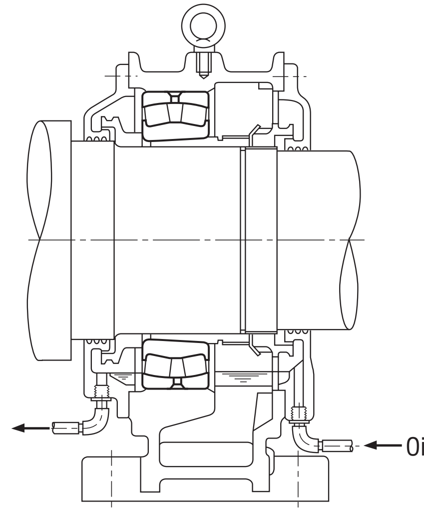


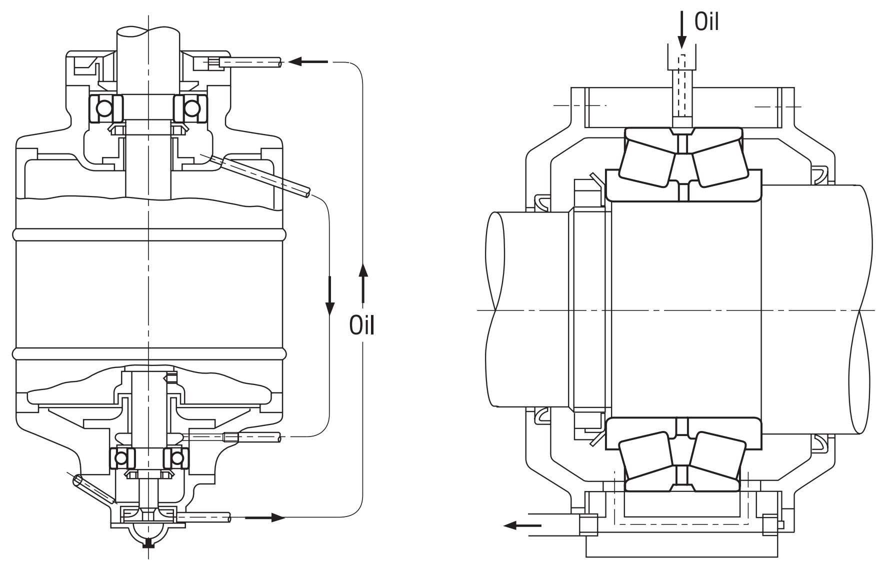
Fig. 11.6 Splash Lubrication

(4) Circulating Lubrication

Circulating lubrication is commonly used for high speed operation requiring bearing cooling and for bearings used at high temperatures. As shown in Fig. 11.7 (a), oil is supplied by the pipe on the right side, it travels through the bearing, and drains out through the pipe on the left. After being cooled in a reservoir, it returns to the bearing through a pump and filter.
The oil discharge pipe should be larger than the supply pipe so an excessive amount of oil will not back up in the housing.



(a)



(b) (c)

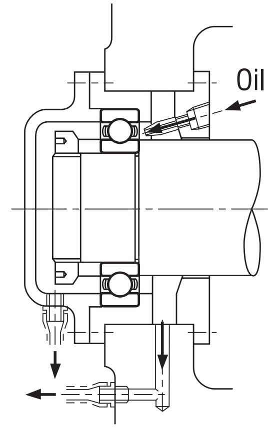
Fig. 11.7 Circulating Lubrication

(5) Jet Lubrication

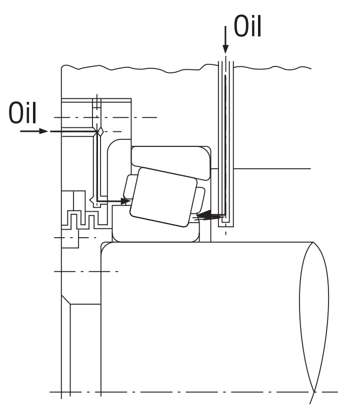
Jet lubrication is often used for ultra high speed bearings, such as the bearings in jet engines with a $d_m n$ valve (d_m: pitch diameter of rolling element set in mm; n: rotational speed in min^{-1}) exceeding one million. Lubricating oil is sprayed under pressure from one or more nozzles directly 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 the case of 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 (which is also the guide face for the cage).

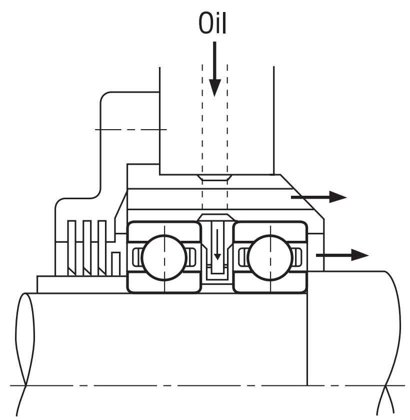
More uniform cooling and a better temperature distribution is achieved using more nozzles for a given amount of oil. It is desirable for the oil to be forcibly discharged so the agitating resistance of the lubricant can be reduced and the oil can effectively carry away the heat.



(a)



(b)



(c)

Fig. 11.8 Jet Lubrication

(6) Oil Mist Lubrication

Oil mist lubrication, also called 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 slight because the oil leakage is small.

(c) It is relatively easy to continuously supply fresh oil; therefore, the bearing life is extended.

This lubricating method is used in bearings for the high speed spindles of machine tools, high speed pumps, roll necks of rolling mills, etc (Fig. 11.9).

For oil mist lubrication of large bearings, it is advisable to consult NSK.

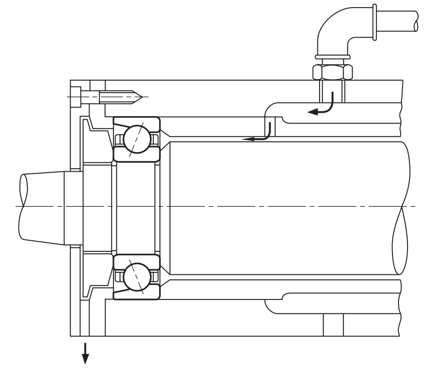


Fig. 11.9 Oil Mist Lubrication

(7) Oil/Air Lubricating Method

Using 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:

(a) Since the minimum necessary amount of oil is supplied, this method is suitable for high speeds because less heat is generated.

(b) Since the minimum amount of oil is fed continuously, bearing temperature remains stable. Also, because of the small amount of oil, there is almost no atmospheric pollution.

(c) Since only fresh oil is fed to the bearings, oil deterioration need not be considered.

(d) Since compressed air is always fed to the bearings, the internal pressure is high, so dust, cutting fluid, etc. cannot enter.

For these reasons, this method is used in the main spindles of machine tools and other high speed applications (Fig. 11.10).

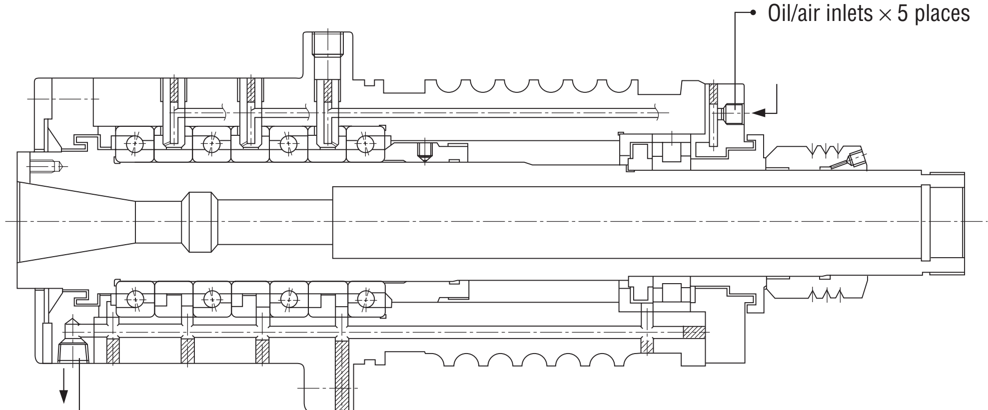


Fig. 11.10 Oil/Air Lubrication

11.3 Lubricants

11.3.1 Lubricating Grease

Grease is a semi-solid lubricant consisting of base oil, a thickener and additives. The main types and general properties of grease are shown in Table 11.2. It should be remembered 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 mainly 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 greases made with high viscosity base oils are 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

As thickeners for lubricating grease, there are 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, please be aware 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 Based Oil)
Dropping Point, °C	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, °C	−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. Pay attention to rust caused by insulation varnish.	Mainly for high temperature applications. Unsuitable for bearings for high and low speeds or heavy loads or those having numerous sliding-contact areas (roller bearings, etc.)	Long and short fiber types are available. Long fiber grease is unsuitable for high speeds. Attention to water and high temperature is requred.	Extreme pressure grease containing high viscosity mineral oil and extreme pressure additive (Pb soap, etc.) has high pressure resistance.	Often used for roller bearings and large ball bearing.	Suitable for extreme pressures mechanically stable	Mineral oil base grease is middle and high temperature purpose lubricant. Synthetic oil base grease is recommended for low or high temperature. Some silicone and fluoric oil based grease have poor rust prevention and noise.	

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. It is recommended that extreme pressure additives be used in 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 working 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 1/10mm. 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 the 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 oil or synthetic oil that have a high oil film strength and superior oxidation and corrosion resistance. When selecting a lubricating oil, the viscosity at the operating conditions is important. If the viscosity is too low, a proper oil film is not formed and abnormal wear and seizure may occur. On the other hand, if the viscosity is too high, excessive viscous resistance may cause heating or large power 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.

For use when selecting the proper lubricating oil, Fig. 11.11 shows the relationship between oil temperature and viscosity, and examples of selection 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)

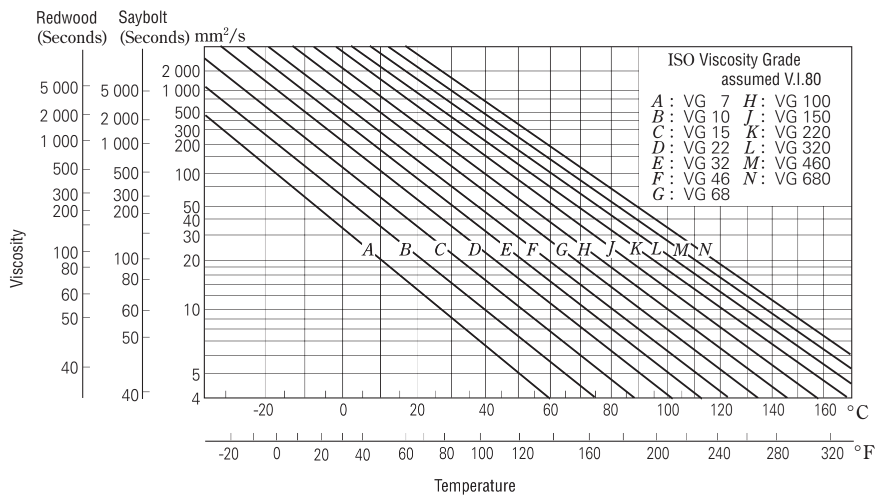


Fig. 11.11 Temperature-Viscosity Chart

Oil Replacement Intervals

Oil replacement intervals depend on the operating conditions and oil quantity.

In those cases where the operating temperature is less than 50°C, and the environmental conditions are good with little dust, the oil should be replaced approximately once a year. However, in cases where the oil temperature is about 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.

Mixing different brands of oil must be prevented for the same reason given previously for grease.

Table 11. 5 Examples of Selection Lubricating Oils

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. For the limiting speed, use the values listed in the bearing tables.
2. Refer to Refrigerating Machine Oils (JIS K 2211), Bearing Oils (JIS K 2239), Turbine Oils (JIS K 2213), Gear Oils (JIS K 2219).
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 , it is advisable to consult NSK.

11.4 Technical Data

11. 4. 1 Brands and Properties of Lubricating Greases

Table 11. 6 Brands of Lubricating Greases

Brands	Thickeners	Base Oils	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 Brack	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 sufficient of the temperature range or in a special environment such as vacuum, it is advisable to consult NSK.
(2) For short-term operation or when cooling is 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

The bearing rings and rolling elements of rolling bearings are subjected to repetitive high pressure with a small amount of sliding. The cages are subjected to tension and compression and sliding contact with the rolling elements and either or both of the bearing rings.

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 easy 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 among the JIS steel types listed in Table 12.1, while the larger bearings generally use SUJ3. The chemical composition of SUJ2 is approximately the same as AISI 52100 specified in the USA, DIN 100 Cr6 in Germany, and BS 535A99 in England.

For bearings that are subjected to very severe shock loads, carburized low-carbon alloy steels such as chrome steel, chrome molybdenum steel, nickel chrome molybdenum steel, etc. are often used. 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 the softer energy-absorbing core. The chemical composition of common carburized bearing steels is listed in Table 12.2.

Table 12.1 Chemical Composition of High-Carbon Chromium Bearing Steel (Major Elements)

Standard	Symbols	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	Symbols	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	Symbols	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

NSK uses highly pure vacuum-degassed bearing steel containing a minimum of 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 having 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 low carbon steels shown in Table 12.5 are the main ones for the pressed cages for bearings. Depending on the purpose, brass or stainless steel may be used. For machined cages, high strength brass (Table 12.6) or carbon steel (Table 12.5) is used. Sometimes synthetic resin is also used.

Table 12.4 Chemical Composition of Stainless Steel for Rolling Bearing (Major Elements)

Standard	Symbols	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	Symbols	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 is Japanese Bearing Association Standard.

Table 12.6 Chemical Composition of High Strength Brass for Machined Cages

Standard	Symbols	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 and Shaft/Housing Materials

Rolling bearings must be able to resist high load, run at high speed, and endure long-time operation. It is also important to know the characteristics of the shaft and housing materials if the bearing performance is to be fully exploited. Physical and mechanical properties or typical materials of a bearing and shaft/housing are shown for reference in Table 12.7.

Table 12.7 Physical and Mechanical Properties of Bearing and Shaft/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°C) ×10⁻⁶/°C	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 structure
	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°C tempering	7.83	0.48	47	18	12.8	207 000 {21 100}	440 {45}	735 {75}	25	217	Carbon steel for machine structure
	SCr430	Quenching, 520 to 620°C 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°C fast cooling			45	23			*784 {80} Min.	*930 {95} Min.	*13 Min.	*269 to 331	
	SCM420	Quenching, 150 to 200°C air cooling		0.47	48	21	12.8		—	*930 {95} Min.	*14 Min.	*262 to 352	Chrome molybdenum steel
	SNCM439	Quenching, 650°C 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°C oil cooling, 400°C 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 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 as reference.

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 treatment, alloy, and free cutting steels / Part 17 ball and roller bearing steels). However, materials are also standardized according to standards of individual countries and, in some cases, makers are even making their own modifications.

As internationalization of products incorporating bearings and references to the standards of these kinds of steels are increasing nowadays, applicable standards are compared and their features described for some representative bearing steels.

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		
SUJ1	—	—	0.95 to 1.10	0.15 to 0.35	≦0.50	0.90 to 1.20	—	*1	Not used generally	Equivalent to each other though there are slight differences in the ranges.
—	51100	—	0.98 to 1.10	0.15 to 0.35	0.25 to 0.45	0.90 to 1.15	≦0.10	*1		
SUJ2	—	—	0.95 to 1.10	0.15 to 0.35	≦0.50	1.30 to 1.60	—	*1	Typical steel type 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	—	*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	Scarcely 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 ultralarge size 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)

It is well known that 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. The life test executed for long periods showed that oxide non-metallic inclusions exert a particularly adverse effect on the rolling fatigue life.

Fig. 12.1 shows the parameter (oxygen content) indicating the amount of oxide non-metallic inclusions vs. life.
The oxygen amount in steel was minimized as much as possible by reducing impurities (Ti, S) substantially, thereby achieving a decrease in the oxide non-metallic inclusions. The resulting long-life steel is the Z steel.
The Z steel is an achievement of improved steelmaking facility and operating conditions made possible by cooperation with a steel maker on the basis of numerous life test data. A graph of the oxygen content in steel over the last 25 years is shown in Fig. 12.2.

The result of the life test with sample material in Fig. 12.2 is shown in Fig. 12.3. The life tends to become longer with decreasing oxygen content in steel. The high-quality Z steel has a life span which is about 1.8 times longer than that of conventional degassed steel.

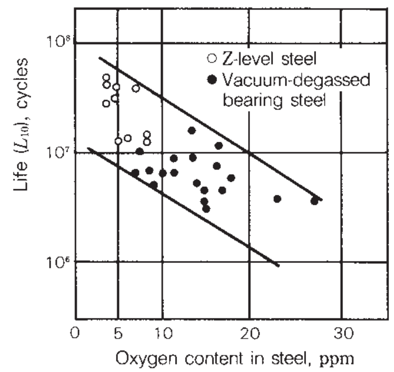


Fig. 12.1 Oxygen Content in Steel and Life of Bearing Steel

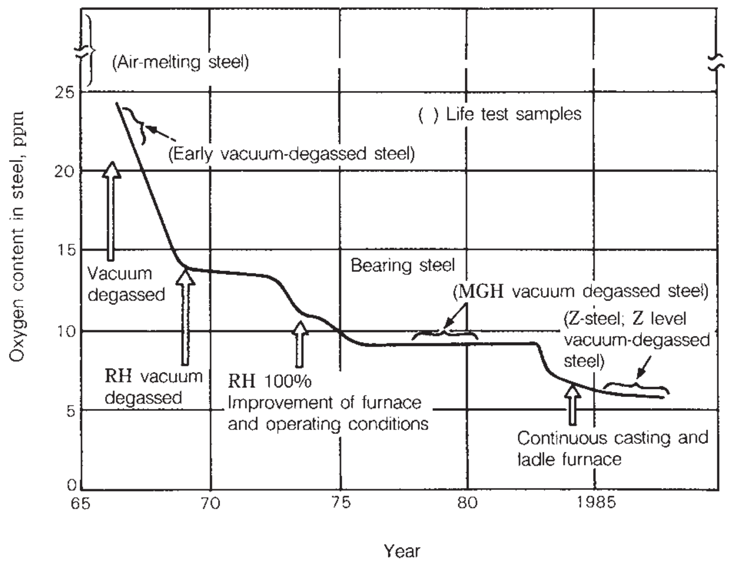
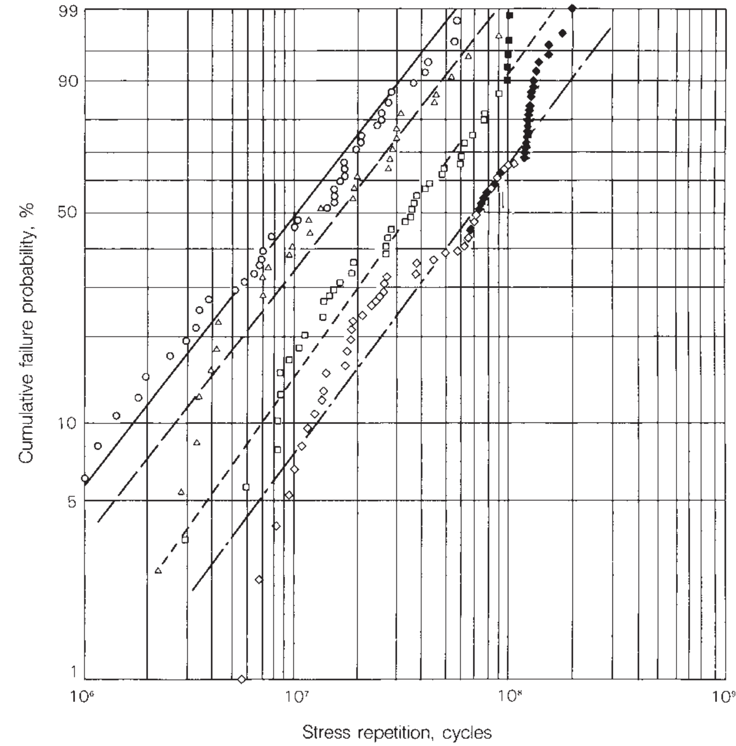


Fig. 12.2 Transition of Oxygen Content in NSK Bearing Steels



Classification	Test quantity	Failured quantity	Weibull slope	L_{10}	L_{50}
○ Air-melting 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 dark ■ and ◆ has not been finished testing yet.

Fig. 12.3 Result of Thrust Life Test of Bearing Steel

12.4.3 Dimensional Stability of Bearing Steel

Sectional changes or changes in the dimensions of rolling bearings as time passes during operation is called aging deformation. When the inner ring develops expansion due to such deformation, the result is a decrease in the interference between the shaft and inner ring. This becomes one of the causes of inner ring creep. Creep phenomenon, by which the shaft and inner ring slip mutually, causes the bearing to proceed from heat generation to 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 the tempering during heat treatment and the bearing's operating temperature. The bearing dimensional stability increases with rising tempering temperature while the retained austenite decomposition gradually expands as the bearing's operating temperature rises.

Fig. 12.4 shows how temperature influences the bearing's dimensional stability. In the right-hand portion of the figure, the interference between the inner ring and shaft in various shaft tolerance classes is shown as percentages for the shaft diameter. As is evident from Fig. 12.4, the bearing 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 needs to be prevented by using a bearing which has received dimension stabilization treatment.

When the bearing temperature is high, there is a possibility of inner ring creep. Since due attention is necessary for selection of an appropriate bearing, it is essential to consult NSK beforehand.

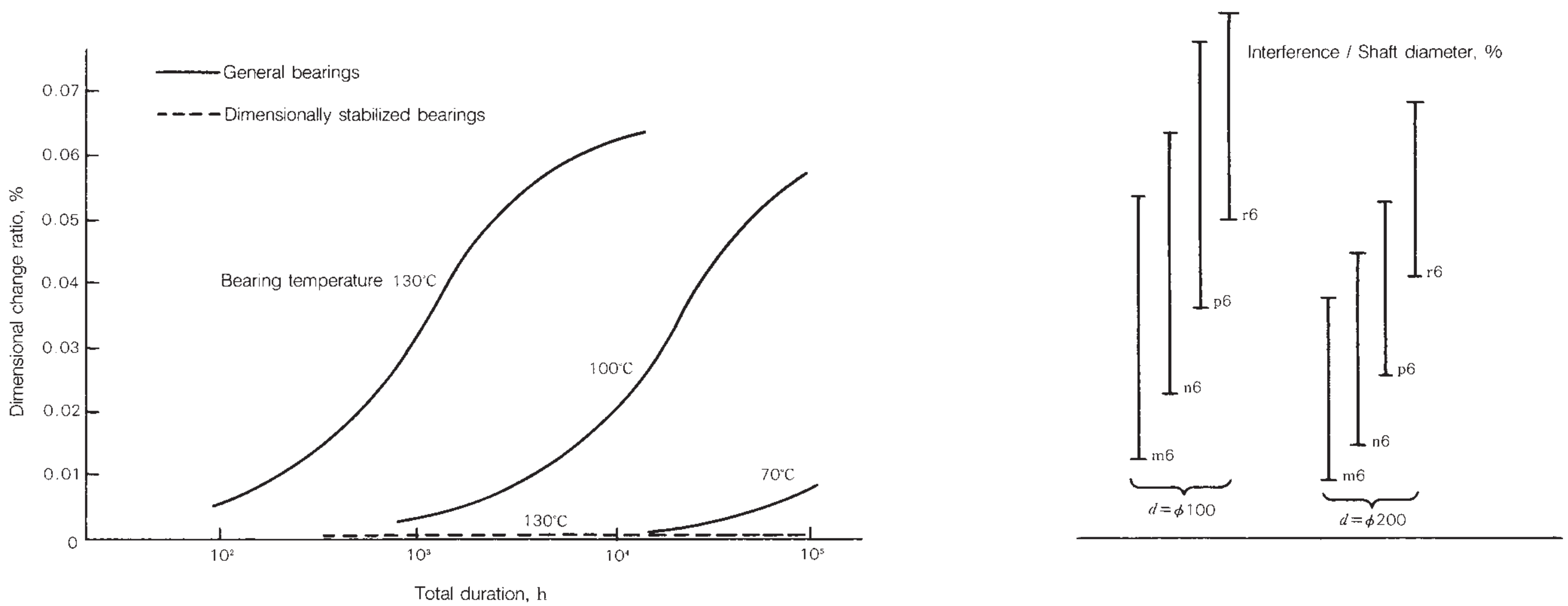


Fig. 12.4 Bearing Temperature and Dimensional Change Ratio

12.4.4 Fatigue Analysis

It is necessary for prediction of the fatigue life of rolling bearings and estimation of the residual life to know all fatigue break-down phenomena of bearings. But, it will take some time before we reach a stage enabling prediction and estimation. Rolling fatigue, however, is fatigue proceeding under compressive stress at the contact point and known to develop extremely great material change until breakdown occurs. In many cases, it is possible to estimate the degree of fatigue of bearings by detecting material change. However, this estimation method is not effective in the cases where the defects in the raceway surface cause premature cracking or chemical corrosion occurs on the raceway. In these two cases, flaking grows in advance of the 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 the residual stress, diffraction half-value width, and retained austenite amount.

These values change as the fatigue progresses as shown in Fig. 12.5. Residual stress, which grows early and approaches the saturation value, can be used to detect extremely small fatigue. For large fatigue, change of the diffraction half-value width and retained austenite amount may be correlated to the progress of fatigue. These measurements with X-ray are put together into one parameter (fatigue index) to determine the relationship with the endurance test period of a bearing.

Measured values were collected by carrying out endurance test with many ball, tapered roller, and cylindrical roller bearings under various load and lubrication conditions. Simultaneously, measurements were made on bearings used in actual machines.

Fig. 12.6 summarizes the data. Variance is considerable because data reflects the complexity of the fatigue phenomenon. But, there exists correlation between the fatigue index and the endurance test period or operating hours. If some uncertainty is allowed, the fatigue degree can be handled quantitatively.

Description of "sub-surface fatigue" in Fig. 12.6 applies to the case when fatigue is governed by internal shearing stress. "Surface fatigue" shows correlation when the surface fatigue occurs earlier and more severely than sub-surface fatigue due to contamination or oil film breakdown of lubricating oil.

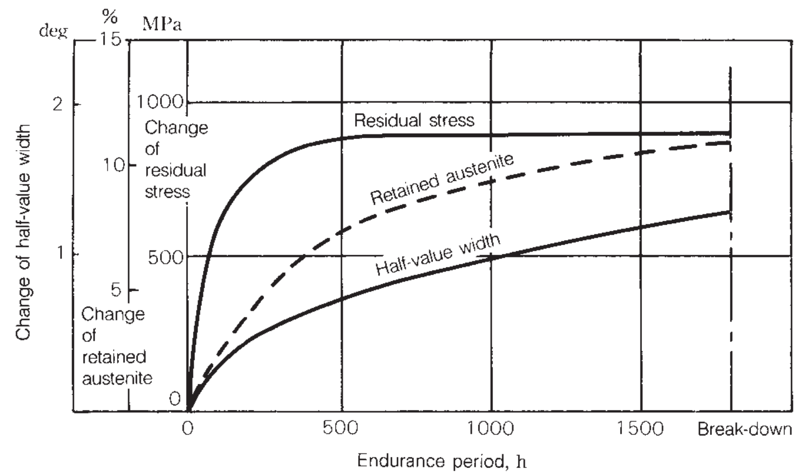


Fig. 12.5 Change in X-ray Measurements

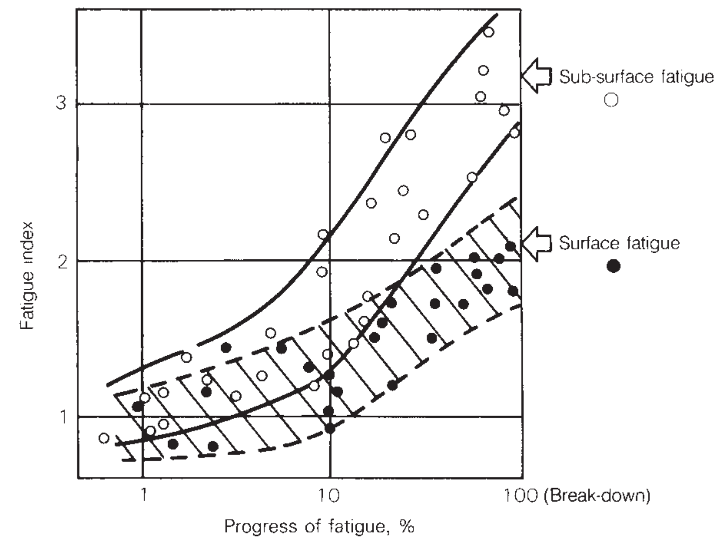


Fig. 12.6 Fatigue Progress and Fatigue Index

(2) Surface and Sub-Surface Fatigues

Rolling bearings have an extremely smooth finish surface and enjoy relatively satisfactory lubrication conditions. It has been considered that internal shearing stress below the rolling surface governs the failure of a bearing.

Shearing stress caused by rolling contact becomes maximum at a certain depth below the surface, with a crack (which is an origin of break-down) occurring initially under the surface. When the raceway is broken due to such sub-surface fatigue, the fatigue index as measured in the depth direction is known to increase according to the theoretical calculation of shearing stress, as is evident from an example of the ball bearing shown in Fig. 12.7.

The fatigue pattern shown in Fig. 12.7 occurs mostly when lubrication conditions are satisfactory and oil film of sufficient thickness is formed in rolling contact points. The basic dynamic load rating described in the bearing catalog is determined using data of bearing failures according to the above internal fatigue pattern.

Fig. 12.8 shows an example of a cylindrical roller bearing subject to endurance test under lubrication conditions causing unsatisfactory oil film. It is evident that the surface fatigue degree rises much earlier than the calculated life.

In this test, all bearings failed before sub-surface fatigue became apparent. In this way, 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 matters or moisture into lubricant.

Needless to say, bearing failure induced by surface fatigue occurs in advance of that by sub-surface fatigue. Bearings in many machines are exposed frequently to danger of initiating such surface fatigue and, in most of the cases, failure by surface fatigue prior to failure due to sub-surface fatigue (which is the original life limit of bearings).

Fatigue analysis of bearings used in actual machines shows not the sub-surface fatigue pattern, but the surface fatigue pattern as shown in the figure in overwhelmingly high percentage.

In this manner, knowing the distribution of the fatigue index in actually used bearings leads to an understanding of effective information not only on residual life of bearings, but also on lubrication and load conditions.

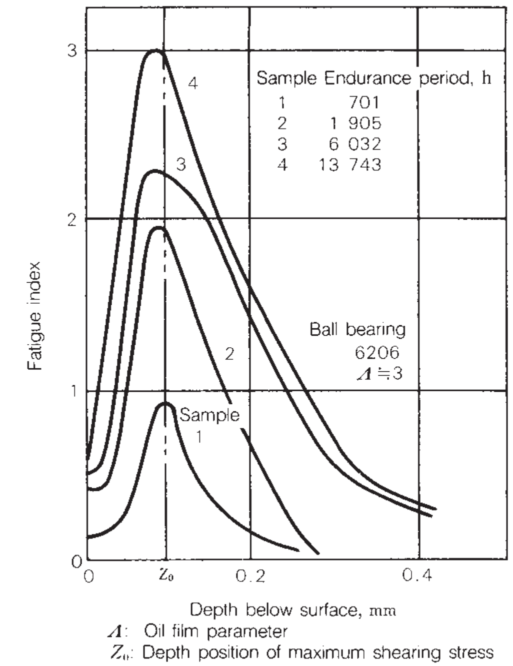


Fig. 12.7 Progress of Sub-Surface Fatigue

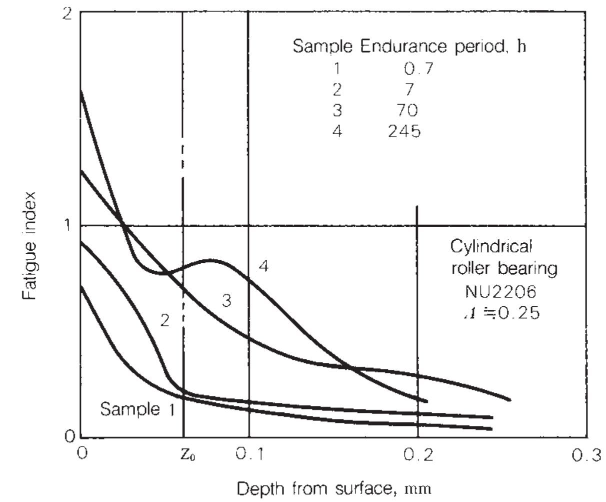


Fig. 12.8 Progress of Surface Fatigue

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 and researching and developing materials, heat treatment processes and operating conditions. The range of approaches to achieving longer service life taken by our research team are shown in Fig. 12.9. The 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-originating flaking.

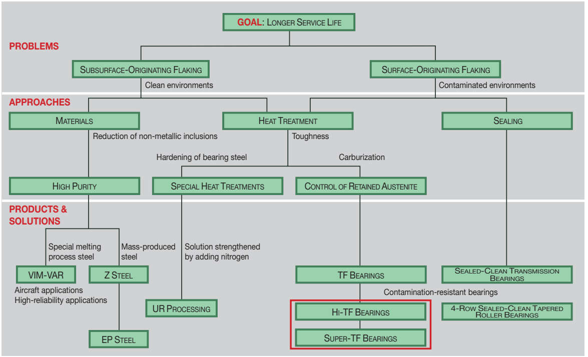


Fig. 12.9 Approaches to Achieving Longer Service Life in Bearings

(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-originating 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 material properties to 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.

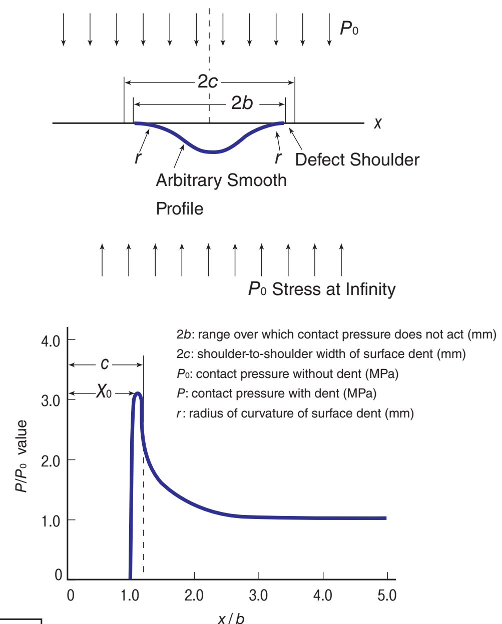


Fig. 12.10 Concentration of Stress around a Surface Dent

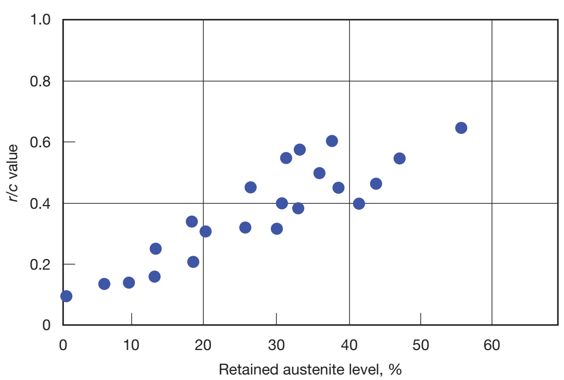


Fig. 12.11 Relationship of r/c Value to Retained Austenite Level

(3) Material Properties of Hi-TF Bearings and Super-TF Bearings

NSK has developed the Hi-TF Bearings and Super-TF Bearings as two series of bearings that offer longer service life exceeding that of TF Bearings. As we have seen, the approach to achieving long service life taken in the Super-TF Bearings is to minimize the concentration of stress around the shoulders of surface dents. A high level of retained austenite helps to maximize the value of r/c and reduce the concentration of stress around the dents. 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, we decided to adopt a technique that would both promote the uniform distribution and reduce the diameter of carbide and carbonitride particles in the bearing material.

To this end, our researchers have developed a new type of steel that has added the proper quantity of element used in the formation of carbides, and have developed the carbonitriding heat treatment to extract minute carbide and nitride compulsorily for the first time in the world. Hi-TF Bearings adopt a new type of steel, which has a specific amount of chrome added to it. Super-TF Bearings adopt a new type of steel, which has a specific amount of chrome and molybdenum added to it. Figures 12.12 and 12.13 illustrate the image analysis results of carbide distribution in the structures of Super-TF Bearings and an ordinary carburized steel bearing. It is clear that the Super-TF Bearings has a greater amount of fine-size carbide and carbonitride particles. Fig. 12.14 shows that the formations of finer carbide and carbonitride particle 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)

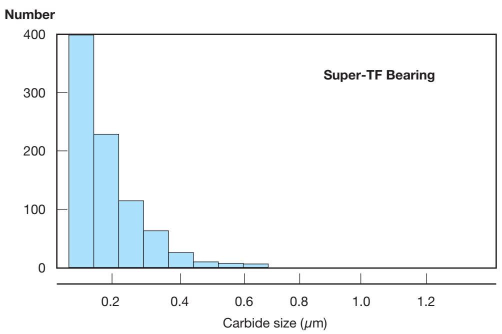


Fig. 12.12 Average Diameter of Carbide and Carbonitride Particles in a Super-TF Bearing

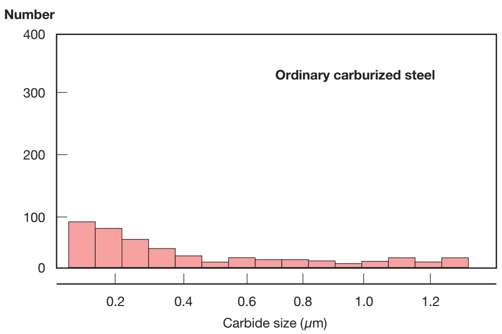


Fig. 12.13 Average Diameter of Carbide Particles in an Ordinary Carburized Steel Bearing

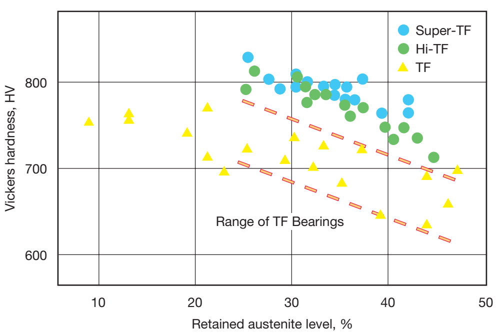


Fig. 12.14 Relationship of Material Hardness and Retained Austenite Level

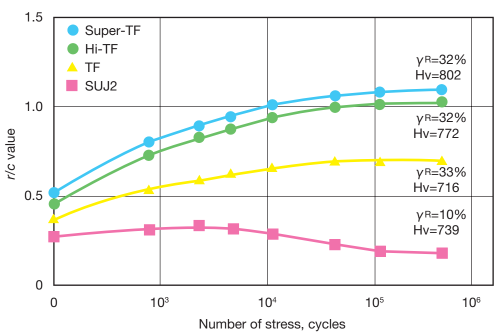


Fig. 12.15 Change of r/c Value under Repeated Stress

(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 NSK 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 time and 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 lubricated conditions, service life may fall to as little as 1/5 of the catalog life.

As a result of attempting longer service life under contaminated lubrication, Hi-TF Bearings and Super-TF Bearings can achieve service life that exceeds the catalog life of existing products under contaminated lubrication for the first time.

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

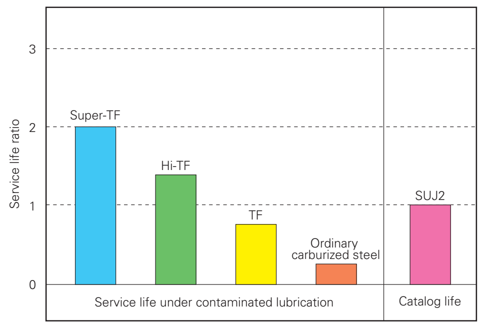


Fig. 12.16 Comparison of Service Life under Contaminated Lubrication

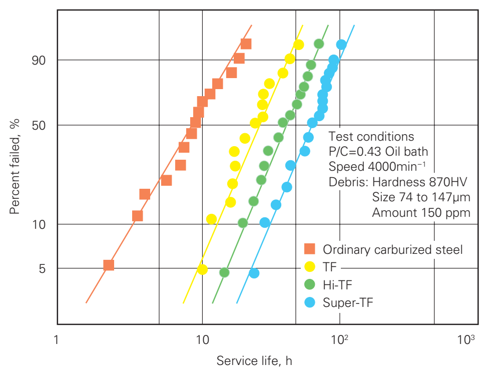


Fig. 12.17 Service Life of L44649/10 Bearings under Contaminated Lubrication

(5) Service Life under Clean Lubrication Conditions

Fig. 12.18 shows the result of service life tests under clean lubrication conditions using 6206 deep groove ball bearings. Under clean lubrication, Hi-TF Bearings and Super-TF Bearings show a slightly longer service life than those made of SUJ2. 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.

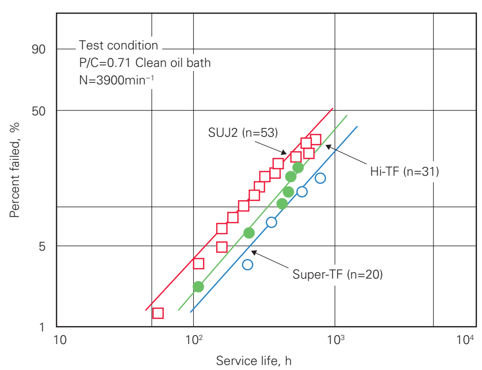


Fig. 12.18 Service Life Tests of 6206 Bearings under Clean Lubrication

(6) Service Life under Boundary Lubrication Conditions

Under boundary lubrication conditions where there is an insufficient amount of 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). When Λ is very small, 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.

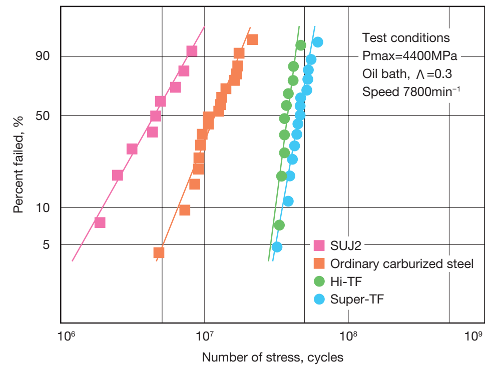


Fig. 12.19 Service Life Tests under Boundary Lubrication Conditions

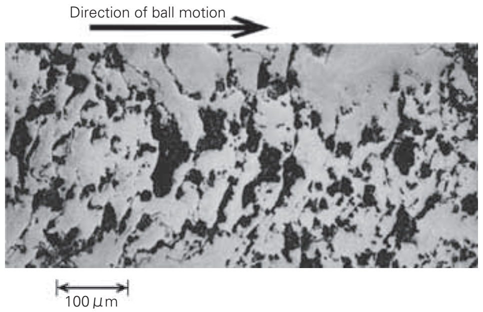


Fig. 12.20 Peeling Damage

(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-type wear test, showing 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% and 40% more resistant to seizure than both SUJ2 steel and TF Bearings.

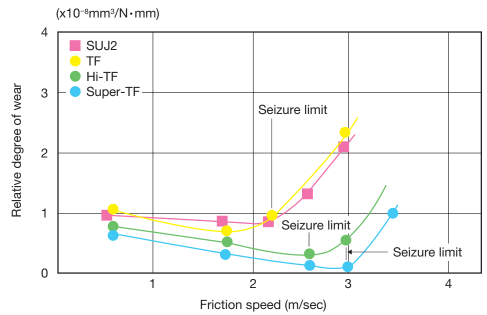


Fig. 12.21 Comparison of Wear Resistance

(8) Heat Resistance

Fig. 12.22 shows the results of service life tests conducted with 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.

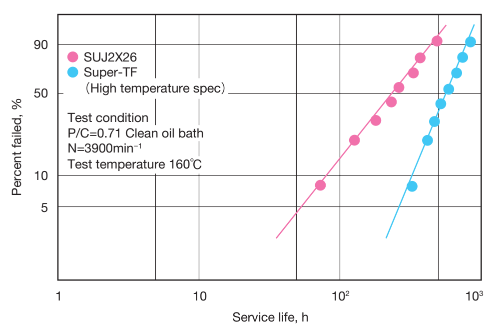


Fig. 12.22 Service Life Test of 6206 under High Temperature Clean Lubrication

12.4.6 Physical Properties of Representative Polymers Used as Bearing Material

Because of lightweight, easy formability, and high corrosion resistance, polymer materials are used widely as a material for cages. 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. In this way composites can be designed to be bearing materials. For example, fillers can be used to improve such properties as low friction, low wear, non-stick slip characteristic, high limit *PV* value, non-scrubbing of counterpart material, mechanical properties, and heat resistance, etc.

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 °C	Glass transition temp °C	Thermal deformation temperature °C (2)	Continuous operating temperature °C	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 and adhesion, inert. Stable at 290°C
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°C 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°C
Polyphenylene sulfide PPS (GF 40%)	4.2	0.14	1.64	256.1	8.5	275	94	>260	220	Hot cured at 360°C
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 resistance 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°C
	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°C for bearing. 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°C as adhesive or enamel Improved polyimide of melting forming
Polyether imide (Aromatic polyimide) PI	3.6	0.107	1.27	240.9	—	—	215	210/200	170	Improved polyimide of melting forming
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, then the value to the left of the "/" applies to 451 kPa, If there, the value relates to 1.82 MPa.
(3) Reference value

12.4.7 Characteristics of Nylon Material for Cages

In various bearings these days, plastic cages have come to replace metal cages increasingly. Advantages of using plastic cages may be summarized as follows:

(1) Lightweight and favorable for use with high-speed rotation
(2) Self-lubricating and low wear. Worn powders are usually not produced when plastic cages are used. As a result, a highly clean internal state is maintained.
(3) Low noise appropriate atm silent environments
(4) Highly corrosion resistant, without rusting
(5) Highly shock resistant, proving durable under high moment loading
(6) Easy molding of complicated shapes, ensures high freedom for selection of cage shape. Thus, better cage performance can be obtained.

As to disadvantages when compared with metal cages, plastic cages have low heat resistance and limited operating temperature range (normally 120°C). Due attention 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 used in large quantity because of its high heat resistance and mechanical properties.
Polyamide resin contains the amide coupling (-NHCO-) with hydrogen bonding capability in the molecular chain and is characterized by its regulation of mechanical properties and water absorption according to the 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 or deterioration of rigidity. On the other hand, however, 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 causes improvement is toughness which is effective for shock absorption during use. In this way, a 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. In view of maintaining deformation of the cage during assembly of bearings, it is common to use a relatively small amount of glass fiber to reinforce the cage. (Table 12.12)
Nylon 66 demonstrates vastly superior performance under mild 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, due attention should be paid to 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. It is known that 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. Besides, materials reinforced with glass fiber can suppress deterioration of the strength through material deterioration by means of the reinforcement effect of glass fibers, thereby, helping to extend the durability period.

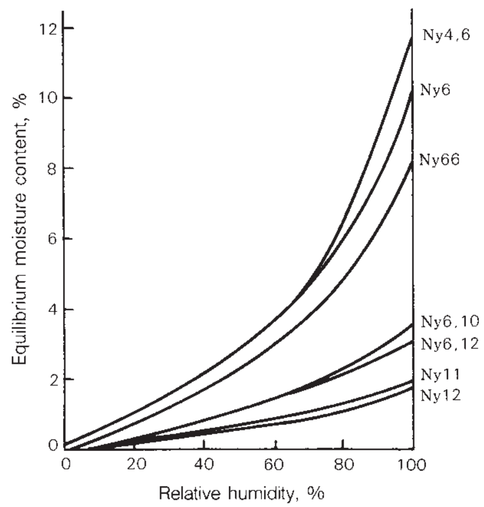


Fig. 12.23 Equilibrium Moisture Content and Relative Humidity of Various Nylons

Table 12.12 Examples of Applications with Fiber Reinforced Nylon Cages

	Bearing type	Main aplication	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		Temper-ature, ℃	Glass content	Hours for the physical property value to drop 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		

Class content: A<B<C<D

12.4.8 Heat-Resistant Resin Materials for Cages

Currently, polyamide resin shows superior performance under medium operating environmental conditions. This feature plus its relative inexpensiveness lead to its use in increasing quantities. But, the material suffers from secular material deterioration or aging which creates a practical problem during continuous use at 120°C or more or under constant or intermittent contact with either oils (containing an extreme pressure agent) or acids.

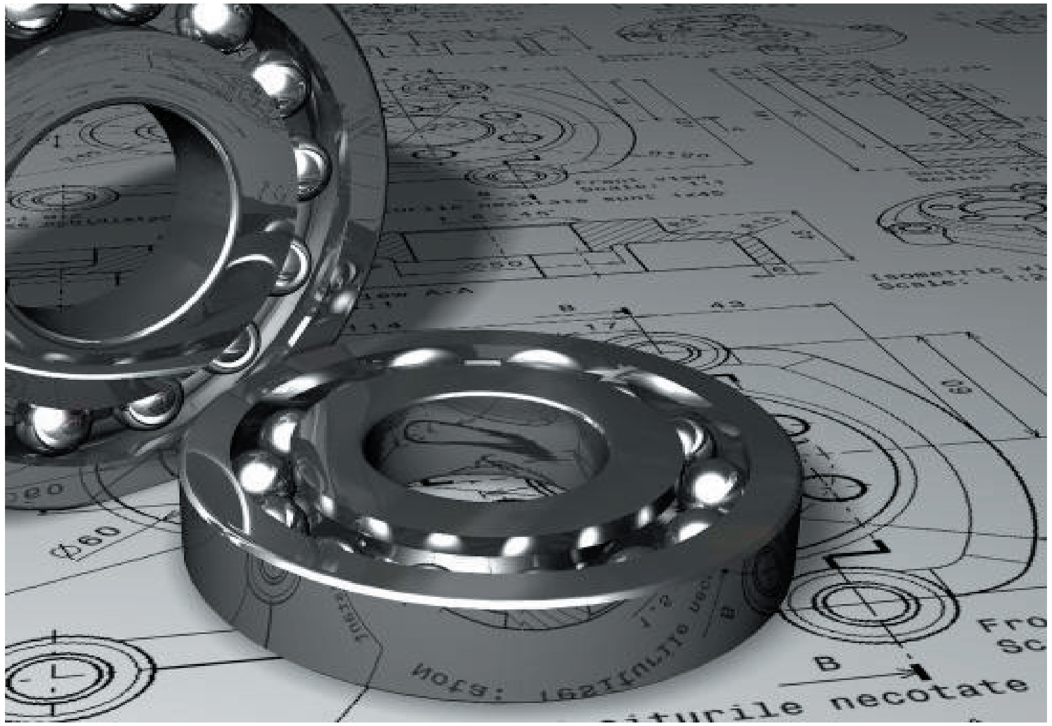
Super-engineering plastics should be used for the cage materials of bearings running in severe environments such as high temperature over 150°C or corrosive chemicals. Though super-engineering plastics have good material properties like heat resistance, chemical resistance, rigidity at high temperature, mechanical strength, they have problems with characteristics required for the cage materials like toughness when molding or bearing assembling, weld strength, fatigue resistance. Also, the material cost is expensive. Table 12.14 shows the evaluation results of typical super-engineering plastics, which can be injection molded into cage shapes.

Among the materials in Table 12.14, though the branch type polyphenylene sulfide (PPS) is popularly used, the cage design is restricted since forced-removal from the die is difficult due to poor toughness and brittleness. Moreover, PPS is not always good as a cage material, since the claw, stay, ring, or flange of the cage is easily broken on the bearing assembling 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 branch or crosslinking so they have high toughness compared to the former material (branch type PPS). Linear PPS is not only superior in heat resistance, oil resistance, and chemical resistance, but also has good mechanical characteristics such as snap fitting (an important characteristic for cages), and high temperature rigidity.

NSK has reduced the disadvantages associated with linear PPS: difficulty of removing from the die and slow crystallization speed, thereby establishing it as a material suitable for cages. Thus, linear PPS is thought to satisfy the required capabilities for a heat resistant cage material considering the relation between the 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 type polyphenylene sulfide (PPS)	Linear type 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 (Pay attention to cage shape) •Low weld strength •Small fatigue resistance	•Poor toughness •Small weld strength •Small 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 environment resistance	•Good environment resistance	•Good environment resistance	•Good environment 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 its performance	•Reasonable cost for its 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 provide 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 same reason. Housings should be rigid in order to provide firm bearing support. High rigidity housings are advantageous also from he standpoint of 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 one single-piece housing, the fitting surfaces of the housing bore should be designed so both bearing seats may be finished together with one operation such as in -line boring. In the case of split housings, care must be taken in the fabrication of the housing so the outer ring will not become deformed during installation. The accuracy and surface finish of shafts and housings are listed in Table 13.1 for normal operating conditions.

Table 13. 1 Accuracy 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 is for general recommendation using radius measuring method, the basic tolerance (IT) class should be selected in accordance with the bearing precision class. Regarding the figures of IT, please refer to the Appendix Table 11 (page E016).
In cases that the outer ring is mounted in the housing bore with interference or that a thin cross-section bearing is mounted on a shaft and housing, the accuracy of the shaft and housing should be higher 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 .

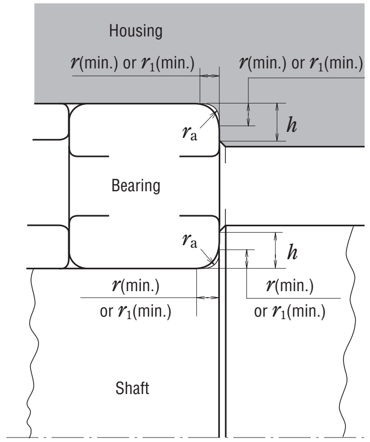


Fig. 13.1 Chamfer Dimensions, Fillet Radius of Shaft and Housing, and Shoulder Height

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 are listed in the bearing tables including the proper shoulder diameters. Sufficient shoulder height is particularly important for supporting the side ribs of tapered roller bearings and 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, Self-Aligning Ball Bearings, Cylindrical Roller Bearings, Solid Needle Roller Bearings	Angular Contact Ball Bearings, Tapered Roller Bearings, 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

Remarks
1. When heavy axial loads are applied, the shoulder height must be sufficiently higher than the values listed.
2. The fillet radius of the corner is also applicable to thrust bearings.
3. The shoulder diameter is listed instead of shoulder height in the bearing tables.

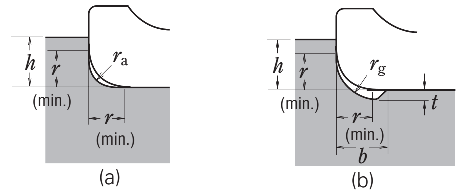


Fig. 13. 2 Chamfer Dimensions, Fillet Radius, and Shoulder Height

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, it is advisable for the full contact length between rollers and rings to be supported by the shaft and housing shoulder (Fig. 13.4).
These diameters d_a and D_a are listed in the bearing tables.

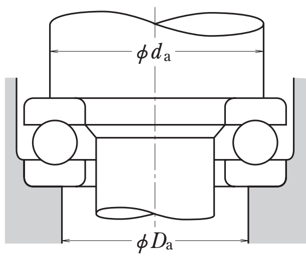


Fig. 13.3 Face Supporting Diameters for Thrust Ball Bearings

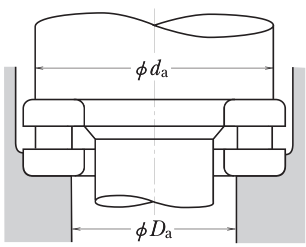


Fig. 13.4 Face Supporting Diameters for Thrust Roller Bearings

13.3 Bearing Seals

To insure the longest possible life of a bearing, it may be necessary to provide seals to prevent leakage of lubricant and entry of dust, water and other harmful material like metallic particles. The seals must be free from excessive running friction and the probability of seizure. They should also be easy to assemble and disassemble. It is necessary to select a suitable seal for each application considering the lubricating method.

13.3.1 Non-Contact Type Seals

Various sealing devices that do not contact the shaft, such as oil grooves, flingers, and labyrinths, are available. 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

The effectiveness of oil groove seals is obtained by means of the small gap between the shaft and housing bore and by multiple grooves on either or both of the housing bore and 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 type seal is often combined with an oil groove seal (Fig. 13.5 (c)). The entry of dust is impeded by packing grease with a consistency of about 200 into the grooves.
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 5mm, with a depth of about 4 to 5mm. In the case of sealing methods using grooves only, there should be three or more grooves.

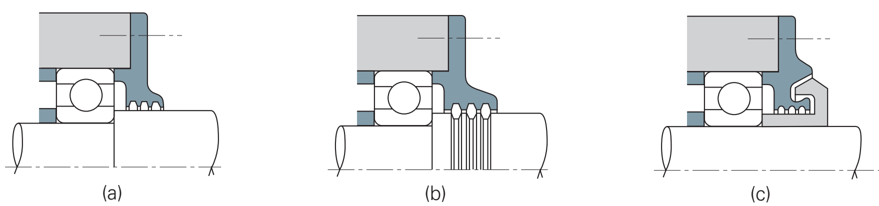


Fig. 13.5 Examples of Oil Grooves

(2) Flinger (Slinger) Type Seals

A flinger is designed to force water and dust away by means of the centrifugal force acting on any contaminants on the shaft. Sealing mechanisms with flingers inside the housing as shown in Fig. 13.6 (a), (b) are mainly intended to prevent oil leakage, and are used in environments with relatively little dust. Dust and moisture are prevented from entering by the centrifugal force of flingers shown in Figs 13.6 (c), (d).

(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), (c) have better seal effectiveness.

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

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

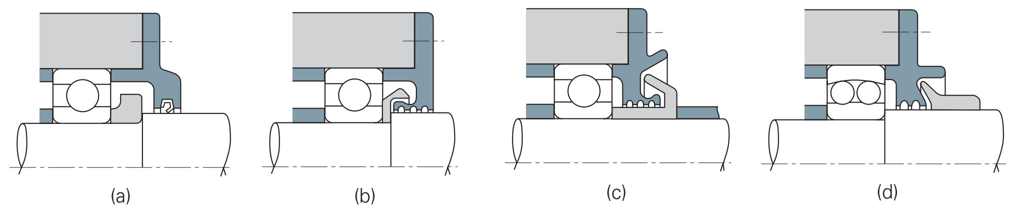


Fig. 13.6 Examples of Flinger Configurations

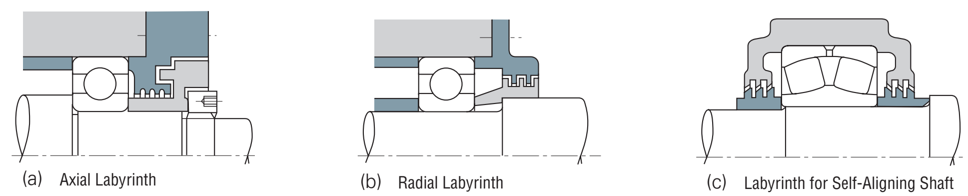


Fig. 13.7 Examples of Labyrinth Designs

13.3.2 Contact Type Seals

The effectiveness of contact seals is achieved by the 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 out as well as to prevent dust, water, and other foreign matter from entering (Figs. 13.8 and 13.9)
In Japan, such oil seals are standardized (Refer to JIS B 2402) on the basis of type and size. 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.
Seal lip materials are usually synthetic rubber including nitrile, acrylate, silicone, and fluorine. Tetrafluoride ethylene is also used. The maximum allowable operating temperature for each material increases in this same order.
Synthetic rubber oil seals may cause trouble 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. It is also desirable for the lubricant inside the housing to spread a little between the sliding surfaces. However, please be aware that ester-based grease will cause acrylic rubber material to swell. Also, 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 the type, the 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 temperature permitted 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 the shaft eccentricity should be less than 0.02 to 0.05 mm.
The hardness of the shaft's contact surface should be made higher than HRC40 by means of heat treatment or hard chrome plating in order to gain abrasion resistance. If possible, a hardness of more than HRC 55 is recommended.
The approximate level of contact surface finish required for several shaft circumferential surface speeds is given in Table 13.7.

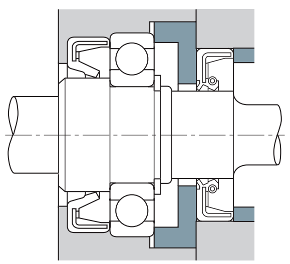


Fig. 13.8 Example of Application of Oil Seal (1)

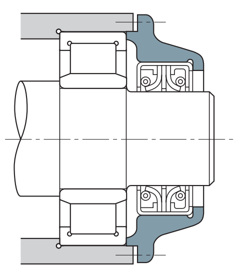


Fig. 13.9 Example of Application of Oil Seal (2)

Table 13. 6 Permissible Circumferential Surface Speeds and Temperature Range for Oil Seals

Seal Materials		Permissible Circumferential Speeds (m/sec)	Operating Temperature Range (°C) (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 °C for operation for 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 being used for transmission shafts, etc.
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 entering. Felt seals are not suitable for circumferential surface speeds exceeding 4m/sec; therefore, it is preferable to replace them with synthetic rubber seals depending on the application.