NTN

Needle Roller Bearings

TECHNICAL DATA CONTENTS

DIMENSIONAL DATA CONTENTS

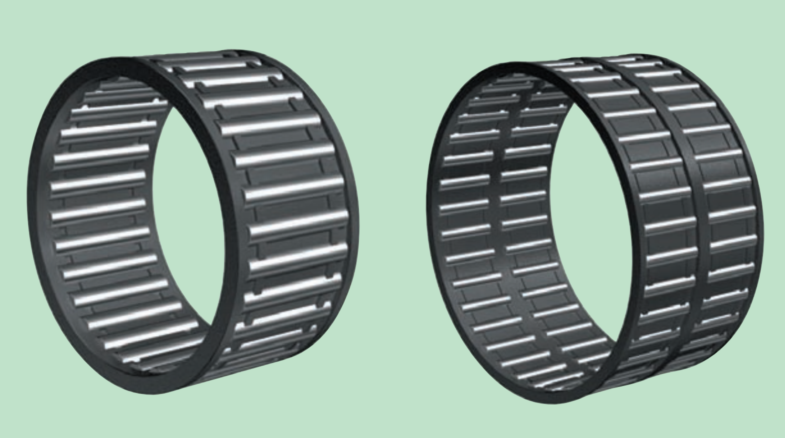
1. Classification and Characteristics of Needle Bearings

Needle roller bearings have relatively small diameter cylindrical rolling elements whose length is much larger than their diameter.

Compared with other types of rolling bearings, needle roller bearings have a small cross-sectional height and significant load-bearing capacity and rigidity relative to their volume. Also, because the inertial forces acting on them is limited, they are an ideal choice for applications with oscillating motion. Needle roller bearings also work well in compact and lightweight machine designs and they serve as a ready replacement for sliding bearings.

NTN offers several different types of needle roller bearings.

Needle roller and cage assembly



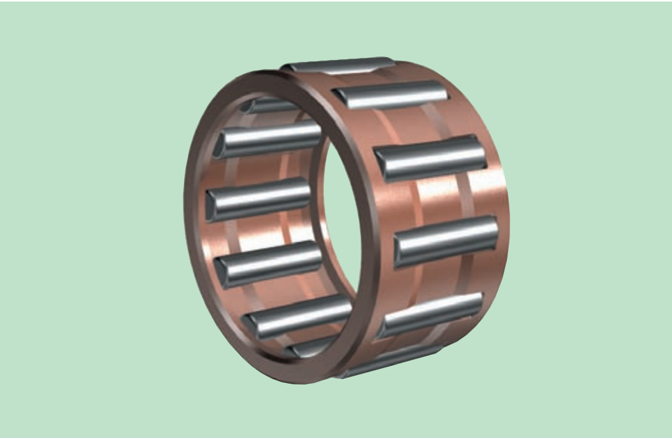
A needle roller and cage assembly includes needle rollers and a cage that guides and retains the rollers.

- These assemblies use both the shaft and housing as raceway surfaces. Consequently, the cross-sectional thickness of the assembly is small, roughly equivalent to the diameter of the needle rollers.
- Because this bearing type has no inner or outer rings, the installation is much easier.
- These assemblies are available in both single-row and double-row configurations.
- As long as the tolerance limits of the shaft and housing are satisfied, the bearing radial internal clearance can be adjusted.

Needle roller and cage assembly for connecting rods

A needle roller and cage assembly for connecting rods includes needle rollers and a cage that guides and retains the rollers. This bearing type is used for connecting rods in compact and mid-sized internal combustion engines (e.g. outboard engines and multipurpose engines), as well as reciprocating compressors.

Needle roller and cage assembly for large end



- This product boasts a unique light-weight high-strength design to cope with crank motion involving the simultaneous rotation and revolution on the large-end side of connecting rod. At the same time, the outer diameter of the cage surface is precision-finished so that the assembly maintains the appropriate cage-riding clearance.
- The cage is made of high-tensile special steel with a surface hardened treatment.
- The assembly uses an outer diameter-guided system.
- If an application has poor lubrication, the cage can be protected with a surface treatment using a non-ferrous metal.
- For applications with a one-piece crank shaft, split-type cage design is also available.

1. Classification and Characteristics of Needle Bearings

Needle roller and cage assembly for small end connecting rods

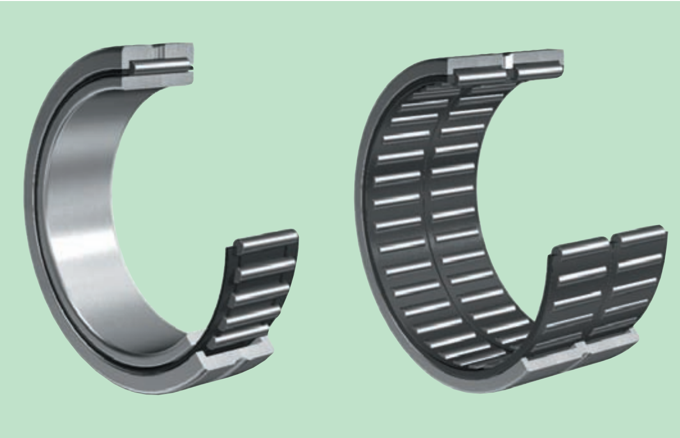
- The small end of connecting rods are subjected to high impact loads and high-speed oscillation. To address this condition, these bearings boast a unique light-weight high-strength design. In addition the cage bore surface is precision-finished so that the assembly maintains an appropriate cage-riding clearance.
- The cage is made from high tensile special steel and the cage surface is hardened.
- The cage is bore-guided and the guide surface is designed to be as long as possible to minimize surface pressure.
- Rollers with the longest possible length are used, At the same time, the maximum number of smaller diameter rollers are incorporated in order to reduce the contract pressure on the rollers.

Drawn-cup needle roller bearing

This bearing type includes an outer ring and needle rollers, which are both drawn from special thin steel plate by precision deep drawing, and a cage which guides the needle rollers precisely.

- This bearing product comprises an outer ring formed through precision deep-drawing process from a thin special steel blank; needle rollers; and cage that guides the rollers.
- A hardened and ground shaft or inner ring (IR Series) is used as the raceway.
- This bearing needs no axial clamping due to easy installation and a press-fit in the housing.
- Both a closed end type to close around the end of the shaft and an open end type are available.
- Furthermore, a type with a seal installed on a single side or on both sides is also available.
- The standard type includes a needle roller and cage assembly. In addition to this type, a special type with full complement rollers is available as an option.

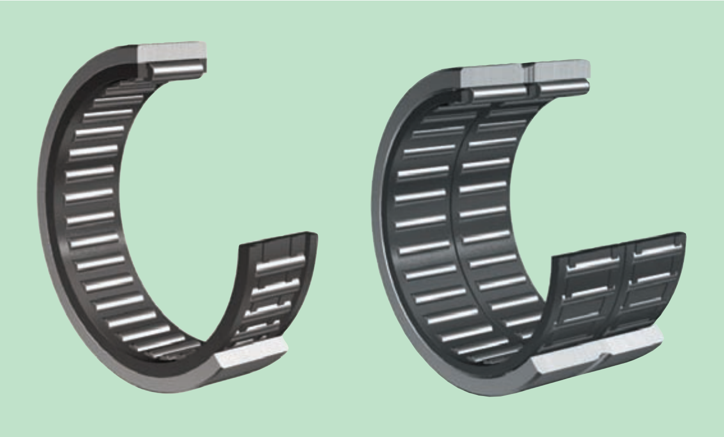
Machined-ring needle roller bearings



This product mainly includes machined components — an outer ring and inner ring, needle rollers and a cage that guides the rollers. In this bearing, the cage or needle rollers are guided by the rib or side plate of the outer ring. Consequently, the roller and cage assembly cannot be separated from the outer ring. When the user wants to use the shaft as the raceway surface, **NTN** can offer a variant without an inner ring.

- Available in both metric dimensions and inch dimensions.
- This product is best-suited to a space-saving design due to its low section height,and large load capacity.
- Another advantage is high rigidity and high bearing accuracy due to the machined outer ring
- This bearing can be used with a housing made of light metal, because of its highly rigid outer ring. (Other than NKS small size products)
- The outer ring has a lubrication hole and lubrication groove.
- Both single-row and double-row types are available.
- A type with seal a installed on a single side or on both sides is also available.

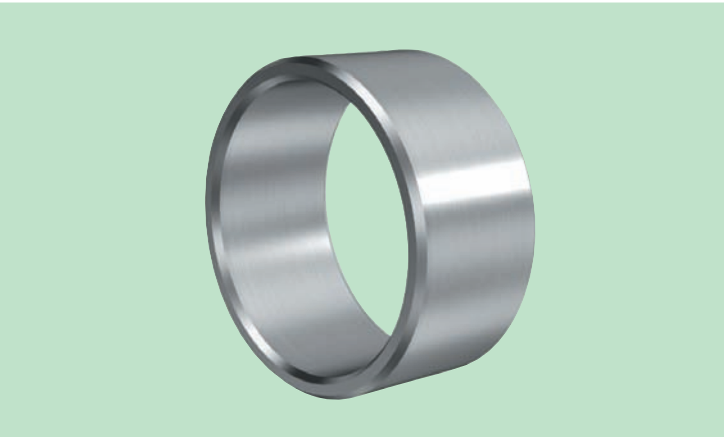
Machined-ring needle roller bearing separable type



This product is essentially comprised of a machined outer ring, inner ring, and, needle rollers with a cage to guide the rollers. With this bearing, the roller and cage assembly can be separated from the outer ring. If the user wants to use the shaft directly as a raceway surface, **NTN** offers a variant that lacks inner ring.

- Easy to install: The following components can be mounted independently: cage and roller assembly, and the inner and outer rings.
- Radial Internal Clearance: Radial internal clearance is selected by combining individual independent components with the desired clearance.
- Space Saving Design: Best-suited to save space because of its low section height and large load capacity.
- High Rigidity: The machined (precut) outer ring allows the bearing to have high rigidity and high bearing.
- Housing Material: This bearing can be mounted in light alloy metal housings because of the outer ring high rigidity.
- Single and double row types bearings available. The outer ring of the double row bearing has a lubrication hole and groove.

Inner ring



Most needle roller bearings lack an inner ring and use the shaft as raceway surface. However, there may be cases where the shaft surface cannot be changed on the machine to the required hardness and/or roughness so in this case an inner ring may be used. **NTN** inner rings are made of high carbon chromium bearing steel blank that is heat-treated, and then fInish-ground to higher precision.

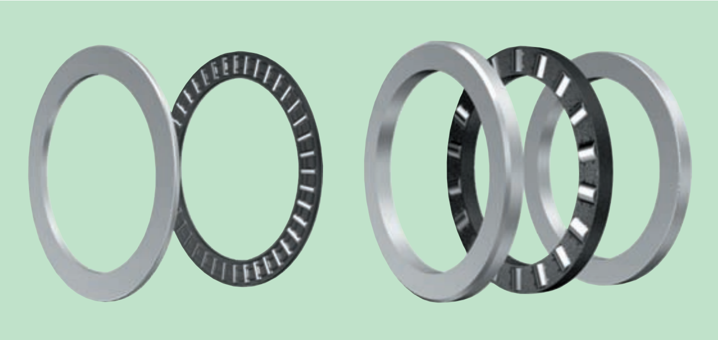
- Can also be used as a bushing.
- Available in both metric and inch series.
- Lubrication hole type at the raceway center is also available.

Clearance-adjustable needle roller bearing

This product is essentially comprised of a machined outer ring, inner ring, and, needle rollers with a cage to guide the rollers. This product features an outer ring with a unique cross-sections shape machined from a solid blank material. With this bearing, the roller and cage assembly cannot be separated from the outer ring. If the user wants to use the shaft directly as a raceway surface, **NTN** offers a variant that lacks inner ring.

- Clearance Reduction: The outer ring raceway diameter is reduced by clamping the outer ring axially,which then reduces the roller assembly bore diameter.
- Clearance Adjustment: Axial clamping force on the bearing can be adjusted to alter the reduction on outer ring raceway diameter.
- Application: This bearing is used on machine tools main spindle and other similar applications which require high speed rotational accuracy of JIS Grade-4.

Thrust roller bearing



The product is comprised of needle or cylindrical rollers, a cage that guides and retains the rollers, and a disk shaped bearing ring, and is capable of holding an axial load in one direction. The mounting surface can be used as raceway surface when the mounting surface are beat-treated and finished. As a result, the bearing can be supplied without bearing ring raceways.

- Space Saving Design: Best-suited to save space because of its small section height and large load capacity.
- Bearing Types: Current available bearing ring types are AS, WS, GS, and ZS. The AS type consists of a thin steel disk having undergone surface-hardening, while the WS, GS and ZS types are machined.

Complex needle roller bearings — Needle roller bearing with thrust bearing —

This complex bearing is comprised of a needle roller bearing for supporting radial load and a thrust bearing for supporting axial load which are assembled integrally. Both thrust ball bearing and thrust roller bearing type are available to support axial load.

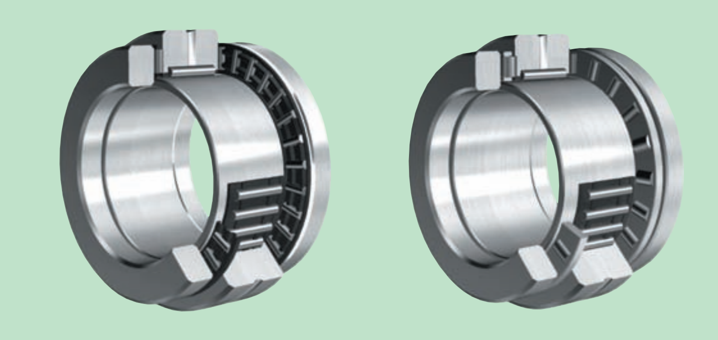
- A variant of thrust bearing are equipped with a dust cover that positively prevents outward release of oil splash and protects the bearing against ingress of dust.

Complex needle roller bearings
— Needle roller bearing with angular contact ball bearing, needle roller bearing with three-point contact ball bearing —

This complex bearing is comprised of a needle roller bearing for supporting radial load, a ball bearing for supporting comparatively small axial load and machined inner and outer rings which are all assembled integrally. Both angular contact ball bearing and three-point contact ball bearing are available to support the axial load.

- The complex needle roller bearings (NKIA Series) use an angular contact ball bearing as the thrust bearing to support a one-directional axial load.
- The complex needle roller bearings (NKIB Series) use a three-point contact ball bearing as the thrust bearing to support a double-directional axial load in addition the position in axial direction can be fixed.

Needle roller bearing with double thrust roller bearing



This is a complex bearing wherein a thrust needle roller bearing or a thrust cylindrical roller bearing intends to support an axial load is configured at the double sides of a radial needle roller bearing for supporting radial load.

- Bi-Direction Axial Loading: This bearing can support large axial loads from both sides.
- Application: This complex bearing is designed to support a machine tool precision ball screw.

The track roller bearing is a needle roller bearing with thick outer ring, which is applied to cam roller, guide roller, eccentric roller or rocker arm.

The track roller bearings are mainly categorized into a stud type track roller bearing (cam follower) and a yoke type track roller bearing (roller follower). Various types of the roller follower and the cam follower are available.

Cam follower — Needle roller type —

This is a bearing designed for rotation of the outer ring. A needle roller and cage assembly and a stud instead of inner ring are fitted in the thick-walled outer ring. The stud is threaded to be mounted easily. This cam follower (bearing) uses needle rollers as its rolling element and it is available with cage or full complement roller bearing type without cage.

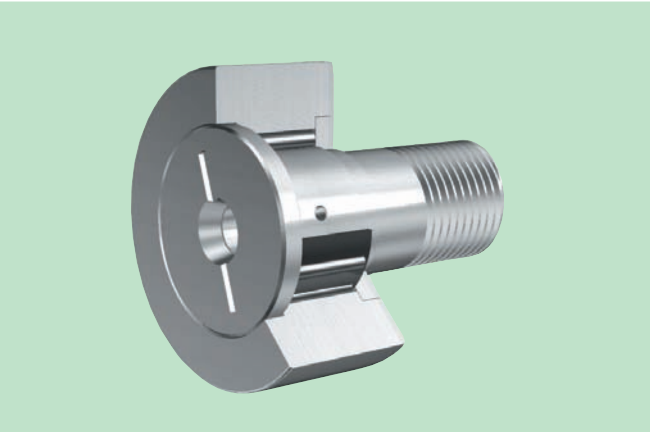
- The bearing type with cage is suitable to comparatively high speed running because its rollers are guided by the cage.
- Having more rollers relative to a given size, a full complement roller type boasts greater load capacity.
- The outer surface is available in both spherical (crowning) profile and cylindrical profile.
- This cam follower (bearing) is selectively available is both metric and inch sizes.
- A seal built-in type is also available.
- The stud is either a recessed head type allowing use of a screwdriver or hexagon socket head type so as to be mounted and adjusted easily.

Cam follower — Cylindrical roller type —

This is a full complement roller bearing designed for rotation of the outer ring. Double-row cylindrical rollers and a stud instead of inner ring are fitted in the thick-walled outer ring. The stud is threaded be mounted easily.

- Compared with needle roller type of a given size, cylindrical roller type of a similar size boasts greater load capacity.
- A steel plate is press-fitted in the outer ring and a labyrinth seal is formed between the face ring and the outer ring.
- The outer surface is available in both spherical (crowning) profile and cylindrical profile.
- The stud is either a recessed head type allowing use of a screwdriver or hexagon socket head type so as to be mounted and adjusted easily.

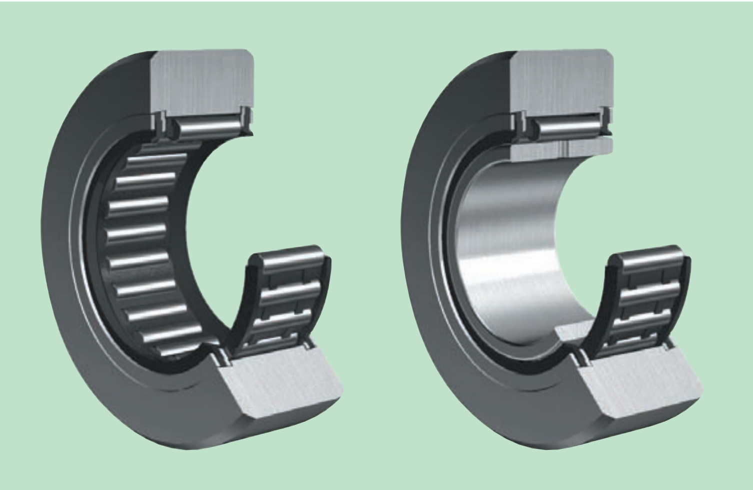
Cam follower — Eccentric type —



This is a cam follower (bearing) where the studs of the needle roller type and cylindrical roller type a prescribed are made eccentric. It can then be adjusted by making eccentric the outer ring relative position against the raceway.

- Load distribution is easily adjustable in configuring two or more cam followers in linear form.
- Preload can be applied by adjustment of load distribution.
- Alignment is possible even when the mounting hole is not processed in high accuracy.
- The outer surface is selectively available in both spherical (crowning) profile and cylindrical profile.
- The stud is either a recessed head type allowing use of a screwdriver or hexagon socket head type so as to be mounted and adjusted easily.

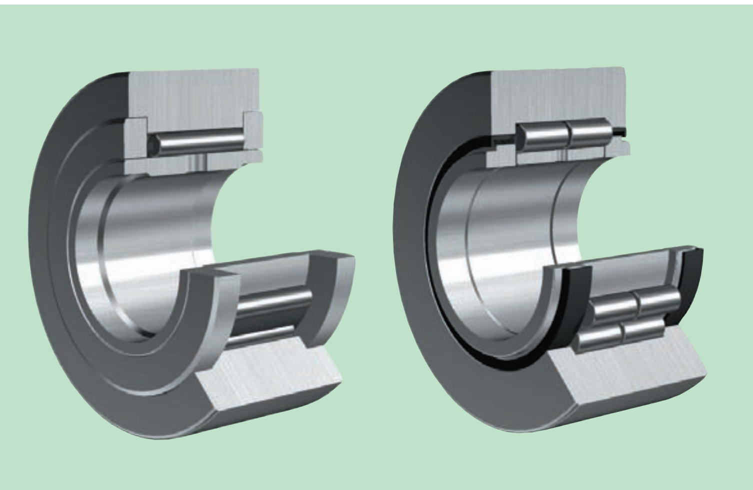
Roller follower — Without axial guide —



This roller follower is a bearing designed for rotation of the outer ring. A needle roller and cage assembly and a synthetic rubber seal reinforced with steel plate are assembled in a thick-walled outer ring.

- The outer ring, the needle roller and cage assembly, and the rubber seal are non-separable from each other.
- The outer ring is thick-walled type so that it is resistible to high load and impact load.
- **A shaft must be provided with a thrust washer and a flange, because the outer ring has no ribs (or face ring) and no axial guide function.**
- The outer surface is available in both spherical (crowning) profile and cylindrical profile.
- The spherical outer ring is effective in damping offset load which is caused by deviation in installing.
- The bearing with cylindrical outer ring is suitable for cases of large load and low-hardness track surface,due to its large area of contact with the mating track surface.

Roller follower — With axial guide —



This roller follower is a bearing designed for rotation of the outer ring. A needle roller and cage assembly, an inner ring, and a face ring are assembled in a thick-walled outer ring.

This bearing uses needle rollers as its rolling element. It is available with a cage or full complement roller bearing without cage. The outer ring is guided axially by a face ring which is press-fitted in the inner ring.

- The outer ring is thick-walled type so that it is resistible to high load and impact load.
- The outer surface is available in both spherical (crowning) profile and cylindrical profile.
- The spherical outer ring is effective in damping offset load which is caused by deviation in installing.
- The bearing with cylindrical outer ring is suitable for cases of large load and low-hardness track surface,due to its large area of contact with the mating track surface.
- This bearing is easier to handle because it needs no mounting of a guide (thrust washer, etc.) on the shaft unlike other types without axial guide (RNA22, NA22).

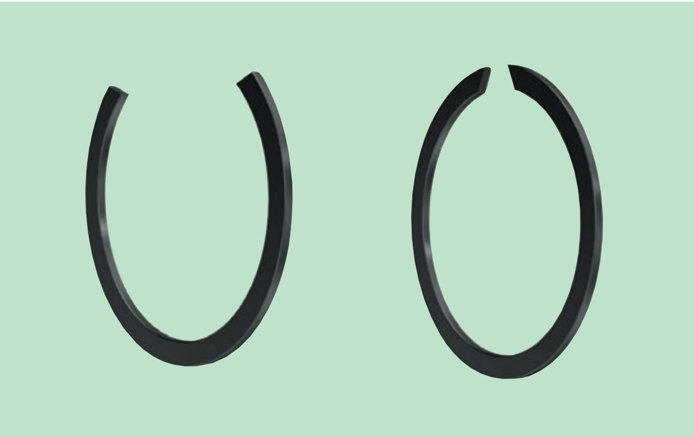
The components described below are for needle roller bearing.

Needle rollers

The needle rollers with flat end round end faces are standard. These rollers are made of high-carbon chrome bearing steel, surface-finished by grinding and buffing after heat-treatment.

- A-Inter-diameter tolerance of the needle rollers is 2mm maximum.
- Rollers with crowned rolling surfaces are also available, which can reduce edge load.
- These needle rollers are supplied individually for applications (pin, shaft).

Snap rings



These are special-purposed rings used for axially positioning, guiding the inner and outer rings, or the needle roller and cage assembly in needle roller bearing.

- Two types are available, for either shaft and/or housing use.
- The snap ring product range cover smaller cross-sectional height products for use in needle roller bearings. The product range also covers snap rings of smaller dimensional range.
- For the axial guide it is recommended to provide a spacer between the cage and the snap ring.

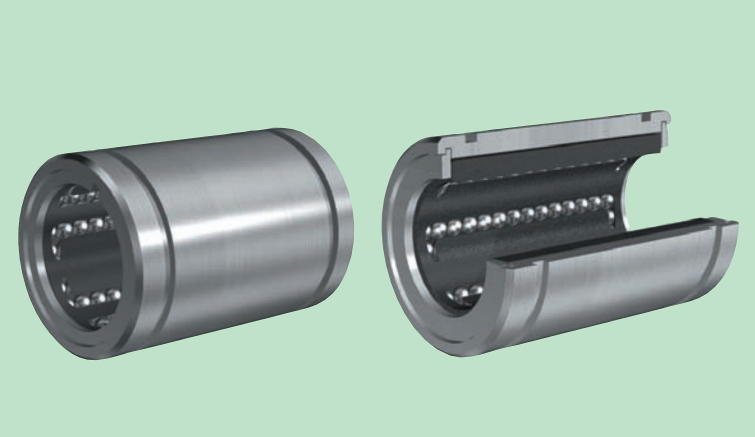
Seals

This product line covers special seals that have been designed for use with low profile needle roller bearings. The product prevents ingress of contamination and help retain grease.

- G-type seal with one lip and GD-type seal with two lips are selectively available on application.
- These seals consist of a ring section formed from steel sheet as well as synthetic rubber material. Their operating temperature ranges from -25 to 120˚C. They are capable of continuous range at a maximum temperature of 100˚C.
- These seals act to prevent the ingress of contamination and over-consumption of lubrication grease.
- The radial section height of each seal is designed to match the drawn-cup needle roller bearings. Hence, these seals require no additional finishing of the housing. This facilitates handling.

This catalogue describes the following ones of linear motion bearings.

Linear ball bearing — Machined ring type —



The product assembly includes a machined outer ring, side plate, steel balls, and a synthetic resin cage that retains the steel balls. This high-precision linear motion bearing develops infinite linear motion on the shaft.

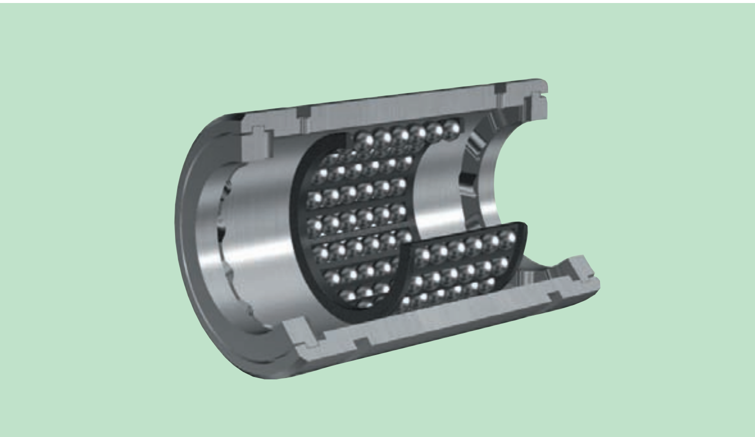
- Standard type, clearance-adjustable type and open type are selectively available on application.
- Some bearings of these types are provided with a synthetic rubber seal at single side or double sides to prevent invasion of foreign matter.
- The steel balls in this product are reliably guided by the cage. Consequently, this product develops stable linear motion on the shaft with minimum frictional resistance.
- **No rotational motion is available.**

Linear ball bearing — Drawn cup type —

The product assembly includes an outer ring formed through precision deep-drawing of thin sheet steel material, steel balls, and a synthetic resin cage that retains the steel balls. This high-precision linear motion bearing develops infinite linear motion on the shaft.

- The outer ring made of thin steel plate creates a smaller section height and allows for a move compact linear motion system.
- Easy to install — This bearing is press-fitted in the housing so that it requires no axial fixing.
- **No rotational motion available.**
- Some bearings of this type are provided with a synthetic rubber seal at double sides to prevent invasion of foreign matter therein.

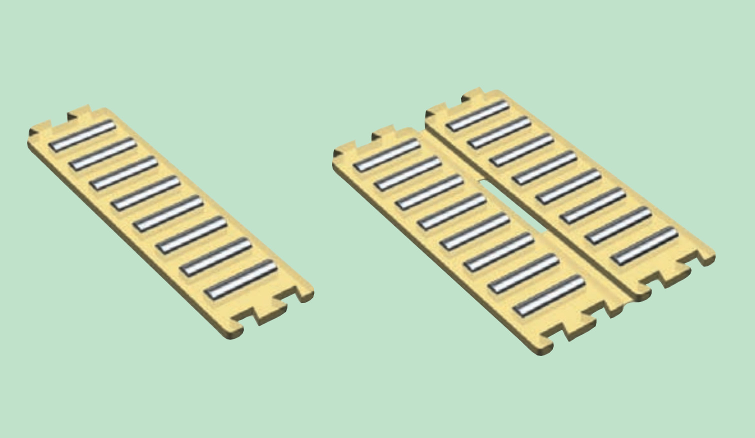
Linear ball bearing — Stroking type —



The product assembly includes a machined outer ring, side plate, steel balls, and a synthetic resin cage that retains the steel balls. This high-precision bearing rotates and develops finite linear motion on the shaft. The outer ring is provided with a snap ring on both sides and a wavy spring washer is provided between the snap ring and the cage to damp on the impact acting on the cage and to prevent wear of the cage.

- Some bearings of this type are provided with a synthetic rubber seal on each side to prevent invasion of foreign matter.
- The outer ring is grooved so that the snap ring can be fitted and fixed easily.

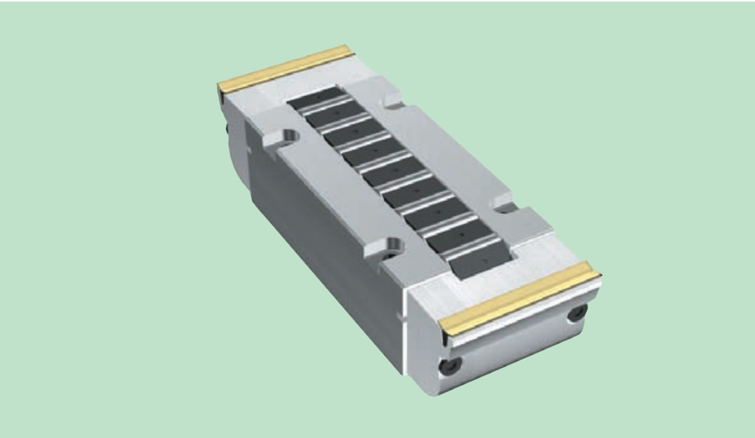
Linear flat roller



This flat roller bearing,comprised of a flat cage and needle rollers, reciprocates on a flat raceway by motion of linear movable components.

- Two material types are available for the cage—synthetic resin and pressed sheet steel.
- FF type molded resin cage – Multiple cages may be joined together in a serial configuration.
- Press-formed steel plate cage – Cage to cage jointing is unavailable, but it can be supplied at any specified length.
- double-row synthetic resin cage has an elastic seam along its center line. When immersed in a hot oil bath heated to 70 to 90˚C the cage can be "folded" to any desired cross-sectional angle so that it can be fitted to a V-sectioned face.

Linear roller bearing

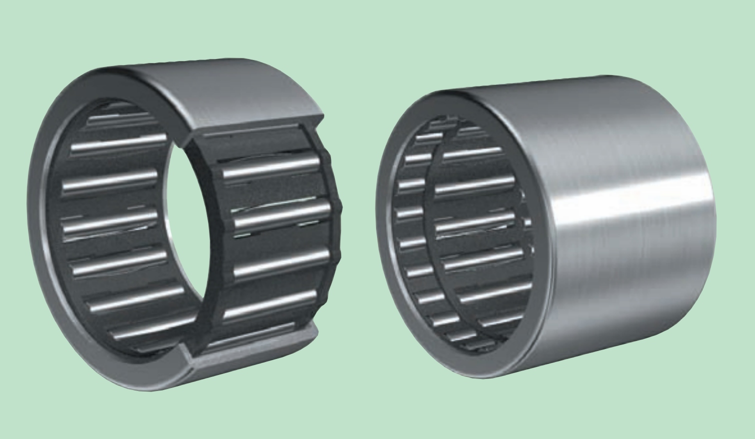


This type has the function of enabling cylindrical rollers to circulate within a track frame and ensures infinite linear motion on a plane.

- Low friction factor due to the cage assembly preventing neighboring rollers from touching each other.
- High load rating due to use of cylindrical rollers

This catalogue describes the following products, too.

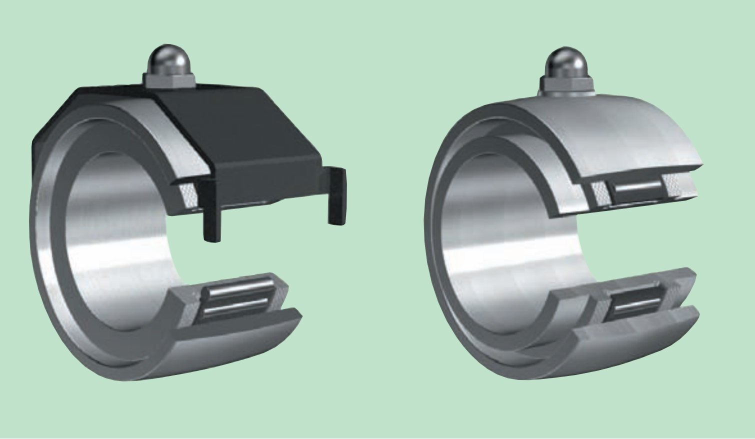
One-way clutch



Comprised of an outer ring drawn from thin special steel plate by precision deep drawing, a spring, needle rollers and a cage, the one-way clutch can transmit torque in only one direction.

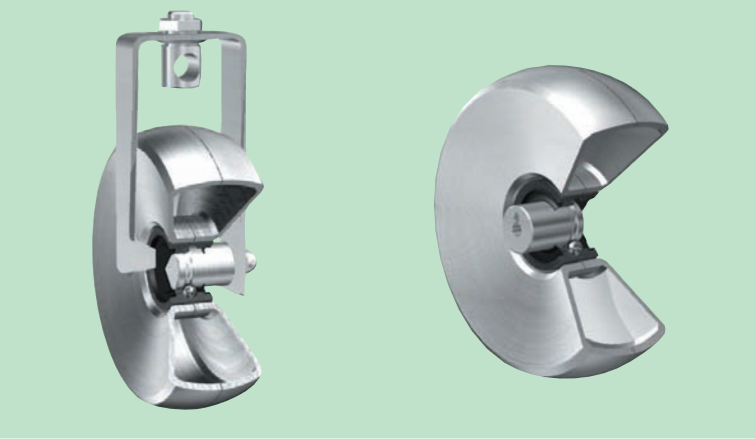
- Boasting low frictional torque during over-running, this one-way clutch also features high transmittable torque despite its small cross-sectional height.
- A certain one-way clutch variant has a built-in bearing that supports radial loading. Another variant has a plated outer ring for improved corrosion resistance.
- HF HFL types can be retained axially by merely press-fitting into a housing.
- These one-way clutches use the outer ring drawn by precision deep drawing, which requires a housing with wall thickness of a specified value or more.
- The HF type unit alone is not capable of bearing radial loads, and both ends must be supported with external radial bearings. (On the other hand, HFL type includes integrated radial bearings on each side.)

Bottom roller bearing — For textile machinery —



This product has a built-in needle roller bearing pre-filled with grease and is used to support bottom rollers. The spherical outer surface of the outer ring can allow a degree of bottom roller installation error. In order to prevent fiber entry into the bearing, tight clearances are maintained between the outer ring and double-ribbed inner ring, and the rib outer diameter surfaces are knurled.

Tension Pulley — For Textile Machinery —



These pulleys are used to guide and tension the tapes and belts driving the spindles of a fine spinning machine, a roving frame, a false twister, etc. The structure is comprised of a precision deep-drawn plate steel pulley which is press-fitted to the outer ring of a bearing.

2. Load Rating and Life

2.1 Bearing life

Even in bearings operating under normal conditions, the surfaces of the raceways and rolling elements are constantly subjected to repeated compressive stresses which causes **flaking** of these surfaces to occur. This flaking is due to material fatigue and will eventually cause the bearing to fail. The effective life of a bearing is usually defined in terms of **the total number of revolutions a bearing can undergo before flaking of either the raceway surface or the rolling element surfaces occur**.

Other causes of bearing failure are often attributed to problems such as seizing, abrasions, cracking, chipping, scuffing, rust, etc. However these so called "causes" of bearing failure are usually themselves caused by improper lubrication, faulty sealing or inaccurate bearing selection. Since the above mentioned "causes" of bearing failure can be avoided by taking the proper precautions, and are not simply caused by material fatigue, they are considered separately from fatigue or flaking.

2.2 Basic rated life and basic dynamic load rating

A group of seemingly identical bearings, when subjected to identical operating conditions will exhibit a wide diversity in their durability. This disparity in lives can be accounted for by differences in the fatigue of the bearing material itself. This disparity is considered statistically when calculating bearing life.

The basic rated life is based on a 90% statistical model. In this model 90% of an identical group of bearings subjected to identical operating conditions will attain or surpass the stated number of revolutions without any flaking due to rolling fatigue. For bearings operating at fixed constant speeds, the basic operating life (90% reliability) is expressed in the total number of hours of operation.

Basic dynamic load rating expressed a rolling bearing's capacity to support a dynamic load. The basic dynamic load rating is the load under which the basic rating life of the bearing is 1 million revolutions. This is expressed as pure radial load for radial bearings and pure axial load load for thrust bearings. These are referred to as **basic dynamic radial load rating** (C_r), and **Basic dynamic axial load rating** (C_a).

The basic dynamic load ratings given in the tables of this catalog are for bearings constructed of **NTN** standard bearing materials using standard manufacturing technologies. For information about the basic dynamic load rating for a bearing using non-standard material and/or manufacturing techniques, contact **NTN** Engineering.

The relationship between the basic rated life, the basic dynamic load rating and the bearing load can be expressed in formula (2.1).

Basic Rated Life specified in ISO 281.

$$L_{10} = \left(\frac{C}{P}\right)^p \quad \cdots\cdots \textbf{(2.1)}$$

where,

p= 10/3 ……………For roller bearing

p= 3. ………………For ball bearings

L_{10} : Basic rated life (10^6 revolutions)

C : Basic dynamic rated load, (N) (kgf)
(radial bearings: C_r, thrust bearings: C_a)

P : Bearing load, (N) (kgf)
(radial bearings: P_r, thrust bearings: P_a)

Furthermore, the basic rated life can be expressed in hours using **formula (2.2)**

$$L_{10h} = 500 f_h^{\,p} \quad \cdots\cdots \textbf{(2.2)}$$

$$f_h = f_n \frac{C}{P} \quad \cdots\cdots \textbf{(2.3)}$$

$$f_n = \left(\frac{33.3}{n}\right)^{1/p} \quad \cdots\cdots \textbf{(2.4)}$$

where,

L_{10h} : Basic rated life, h

f_h : Life factor

f_n : Speed factor

n : Rotational speed, r/ min

Formula (2.2) can also be expressed as **formula (2.5).**

$$L_{10h} = \frac{10^6}{60\,n}\left(\frac{C}{P}\right)^p \quad \cdots\cdots \textbf{(2.5)}$$

When several bearings are incorporated into a piece of equipment it is possible to calculate the bearing life of the whole system by way of **formula (2.6)**.

$$L = \frac{1}{\left(\frac{1}{L_1^{\,e}} + \frac{1}{L_2^{\,e}} + \cdots \frac{1}{L_n^{\,e}}\right)^{1/e}} \quad \cdots\cdots \textbf{(2.6)}$$

where,

e = 9/8 ………For roller bearings

e = 10/9 ………For ball bearings

L : Total basic rated life of bearing as a whole, h

L_1 , L_2 $\cdots L_n$: Individual basic rated life of bearings, 1, 2,⋯n, h

2.3 Required bearing life for a give application

When selecting a bearing, it is essential to determine the required life of the bearing under the intended operating conditions. The life requirement is usually determined by the durability and reliability required for the particular application. General guidelines for required life are shown in **Table 2.1**.

While the fatigue life of bearing is an important factor to consider when sizing the bearing it is also important to consider the strengths and rigidities of shaft and housing.

Table 2.1 Operating conditions and required life (reference information)

Operation profile	Life Requirement L_{10h} ×10³ hrs. ~4	4~12	12~30	30~60	60~
Machine to be run for a short time or only occasionally.	Home electric appliances Power tools	Agricultural machinery Office equipment			
Machine to be run for a short time or only occasionally; however, the machine needs to perform reliably.	Medical equipment Measuring instruments	Home air-conditioner Construction machinery Elevator Cranes	Cranes (sheave)		
Machine to be run for a prolonged time (but not continuous).	Passenger cars Motor cycles	Compact electric motors Buses and trucks General gearing equipment Woodworking machinery	Spindle on machine tool Multi-purpose electric motor for production plant Crusher Vibration screen	Critical gearing equipment Calender rolls for rubber or plastic materials Offset printing press	
Machine to be always run at least 8 hours a day.		Roll neck on steel rolling machinery Escalator Conveyor Centrifugal separator	Axles on rolling stocks Air-conditioning equipment Large electric motor Compressor and pump	Axles on locomotives Traction motors Hoist for mines Press flywheels	Pulp or paper making machinery Propulsion system for ships
Machine to be run 24 hours a day, and must continue operating even in the event of accident.					City water facility Drain and ventilation system for mines Electric power station equipment

2.4 Adjusted rating life

While the basic rating life (90% reliability) for a given bearing can be calculated with the formulas in Subsection 2.2 a number of factors may be present which adjust that life. In some applications it may be necessary to calculate bearing life at greater than 90% reliability. Special materials or manufacturing processes may be applied to the bearing in an effort to increase life. Furthermore, bearing life may be affected by the operating conditions (lubrication, temperature, running speed, etc.).

The basic rating life can be adjusted to consider these factors. The resultant basic rating life is called the **adjusted rating life**, and can be determined by **formula (2.7)**:

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \ (C/P)^p \quad \cdots\cdots\cdots\cdots\mathbf{(2.7)}$$

where,

L_{na} : Adjusted life rating 10^6 revolutions
a_1 : Reliability adjustment factor
a_2 : Bearing material adjustment factor
a_3 : Operating condition adjustment factor

2.4.1 Reliability adjustment factor a_1

The reliability adjustment factor, a_1, is used when a reliability higher than 90% is required. Values are shown in **Table 2.2**.

2.4.2 Bearing material adjustment factor for a_2

When non-standard bearing materials or manufacturing processes are used , the life-related bearing characteristics are inevitably changed. In this case, the bearing life is adjusted using the life adjustment factor, a_2.

The basic dynamic load ratings found in the "Bearing Dimensions Table" of the catalog assume the use of standard **NTN** materials and manufacturing processes / techniques. In this case a_2=1.

When special materials or manufacturing techniques are used in the manufacture of the bearing an a_2≠1 will need to be applied. In such a case, feel free to contact **NTN** for further information.

When bearings made of high carbon chrome bearing steel are used at temperatures greater than 120˚C for a significant period of time significant dimensional changes will occur in the bearing. To limit these changes and their effect on bearing life a special dimension-stabilizing heat-treatment **(TS treatment)** is used. The specific treatment is determined according to the maximum operating temperature. However, this dimension-stabilizing treatment results in lower bearing hardness which reduces bearing life. To account for this, the bearing life is adjusted using the a_2 factor shown in **Table 2.3.**

Table 2.2 Values of reliability adjustment factor a_1

Reliability %	L_n	Reliability adjustment factor a_1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

Table 2.3 Life adjustment values (a_2) for dimension-stabilizing heat-treated (TS-treated) bearings

Code	Maximum operating temperature	Life adjustment factor for bearing material a_2
TS2-	160˚C	1.00
TS3-	200˚C	0.73
TS4-	250˚C	0.48

2.4.3 Life adjustment factor for operating conditions a_3

The life adjustment factor for operating conditions (a_3) is used to adjust the bearing life when operating under non-ideal conditions such as deteriorated lubricated, the ingress of foreign matter (contamination) or excessively high the rotational speeds.

Generally the life adjustment factor in the case of optimum lubrication and no contamination is a_3=1. When the bearing operates under particularly good conditions it is possible to have a_3>1. However, a_3<1 is applied in the following cases.

- **Low dynamic viscosity of grease or oil at bearing operating temperature**
  Radial needle roller bearing 13mm²/s and less
  Thrust needle roller bearing 20mm²/s and less
- **Particularly low rotational speed**
  (The product of rotational speed n min⁻¹ by pitch circle diameter (D_{pw} mm) of rolling element is $D_{pw} \cdot n$<10000.)
- **High operating temperature of bearing**
  When standard bearings operate at high temperatures hardness of the raceway hardness is reduced, impacting bearing life, In such cases the bearing life is adjusted by multiplying the value shown in **Fig.2.1**. **However, this does not apply to bearings having undergone dimension-stabilizing (TS) treatment.**
- **Ingress of foreign matter (contamination) and/or moisture into lubricant**
  When using a bearing operating under suboptimal conditions please feel free to **NTN** for assistance in applying the adjustment factors.
  If the lubricating conditions are not favorable a factor of $a_2 \times a_3$<1 is usually applied. This is true even if special materials and manufacturing techniques are used that would result in a life adjustment factor a_2>1.

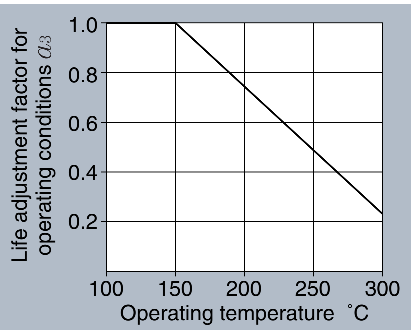


Fig. 2.1 Life adjustment factor for operating conditions depending on operating temperature

2.5 Effect of surface hardness on basic dynamic load rating

It is possible to use the shaft or housing surface as the raceway surface. Under these conditions the surface layer of the shaft/housing must be hardened to HRC58 to 64 and a proper hardening depth must be achieved.

Methods such as ordinary quenching, carburizing or induction quenching can be used to harden the shaft/housing. If it is not possible to sufficiently harden the surface the load rating of the bearing will need to be reduced. The basic load rating must be adjusted by multiplying the hardness factor shown in **Fig.2.2**.

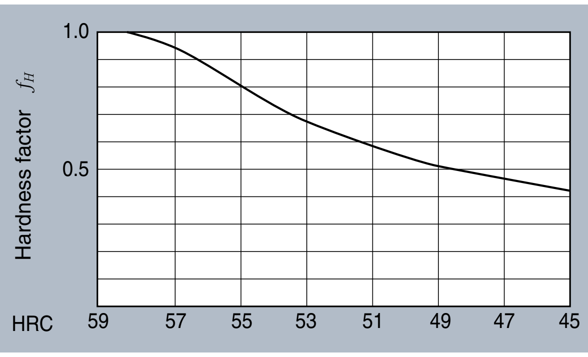


Fig. 2.2 Hardness factor

2.6 Bearing life under oscillating motion

The life of a bearing under oscillating motion can be determined by **formula (2.8)**.

$$L_{osc} = \Omega L_{Rot} \qquad \textbf{(2.8)}$$

where,

L_{osc} : Life of bearing with oscillating motion
L_{Rot} : Life of bearing subject to rotational speed min⁻¹ identical to oscillation frequency cpm
Ex.) Rating life determined from 90 min⁻¹ that is equivalent to cyclic rate of 90 cpm.
Ω : Oscillation factor (showing the relation with half angle β of oscillation angle per **Fig.2.3**).

Generally, **Fig.2.3** applies to cases where the critical oscillation angle 2 β is greater than the critical oscillation angle 2 β_C. Critical oscillation angle is principally governed by the internal design of the bearing; in particular, the number of rolling elements included in one row.

There may be a case where the bearing needs to be used at an angle smaller than its critical oscillation angle: however, the bearing life will be shorter than the

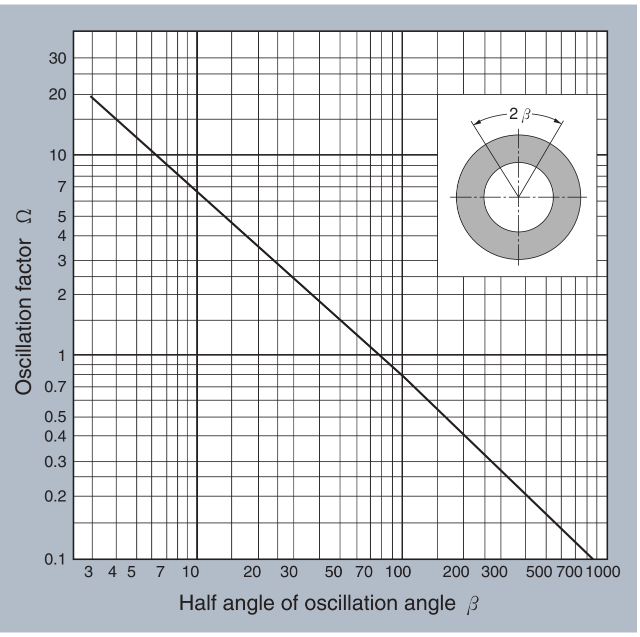


Fig. 2.3 Relationship of oscillation angle β to factor Ω

calculated life determined using the data in **Fig.2.3**. If the oscillation angle of the bearing is unknown, determine Ω, assuming that $\beta = \beta_C$. For the data about an intended bearing, contact **NTN** Engineering.

When the oscillation angle 2β is very small, difficulty in forming an oil film on the contact surface of rolling ring to rolling element could result in **fretting** corrosion.

In the case of inner ring oscillation, the critical oscillation angle is expressed in **formula (2.9)**.

$$\text{Critical oscillation angle } 2\beta_c \geqq \frac{360^\circ}{Z} \cdot \frac{D_{pw}}{D_{pw} - D_w \cos\alpha} \quad \cdots\textbf{(2.9)}$$

Where,

Z : Number of rolling elements (per row)
d_p : Pitch circle diameter (PCD) of rolling element
D_p: Rolling element diameter
e : Contact angle

(In the case of outer ring oscillation, the right side denominator is $D_{pw} + D_w \cos\alpha$.)

2.7 Life of bearing with linear motion

In the case of bearings with linear motion such as linear ball bearing, linear flat roller bearing, etc., the relationship among axial travel distance, bearing load and load rating can be expressed in **formulas (2.10)**, **(2.11)**.

When the rolling elements are balls;

$$L = 50 \times \left(\frac{C_r}{P_r}\right)^3 \quad \cdots\textbf{(2.10)}$$

When the rolling elements are rollers;

$$L = 100 \times \left(\frac{C_r}{P_r}\right)^{10/3} \quad \cdots\textbf{(2.11)}$$

where,

L : Load rating km
C_r : Basic dynamic load rating N (kgf)
P_r : Bearing load N (kgf)

Fig.2.4 shows the relationship of C_r/P_r to L.

Furthermore, when the travel motion frequency and travel distance remain unchanged, the lifetime of bearing can be determined by **formulas (2.12)**, **(2.13)**.

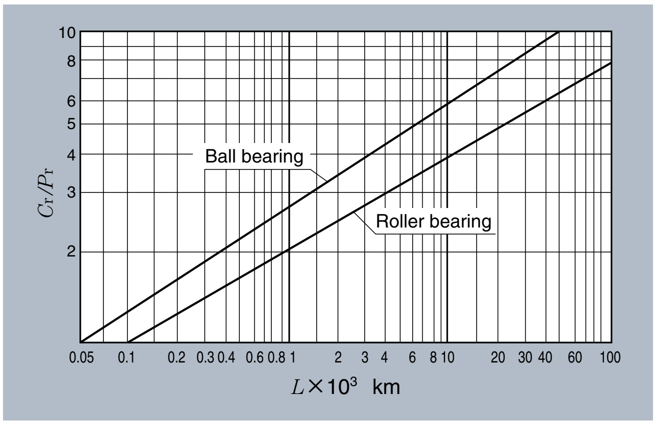


Fig. 2.4 Life of bearing with axial motion

When the rolling elements are balls;

$$L_h = \frac{50 \times 10^3}{60 \cdot S}\left(\frac{C_r}{P_r}\right)^3 \quad \cdots\textbf{(2.12)}$$

When the rolling elements are rollers;

$$L_h = \frac{100 \times 10^3}{60 \cdot S}\left(\frac{C_r}{P_r}\right)^{10/3} \quad \cdots\textbf{(2.13)}$$

where,

L_h: Travel life h
S : Travel distance per minute m/min
$S = 2 \cdot L \cdot n$
L : Stroke length m
n : Stroke cycle cpm

2.8 Fitting misalignment and crowning

Generally it is well known that stress concentrations at the edge portion of the roller (so called, edge load) arising from fitting misalignment could result in rapid reduction of bearing lifetime. "Crowning" is adopted as a countermeasure against such rapid reduction of bearing lifetime. In that case, however, unless it is designed properly this crowning would cause the effective contact length of the roller to be reduced, which could then lead to shorter bearing life. It is therefore necessary to calculate a proper crowning based on the extent of fitting misalignment and load condition.

For Reference purposes, **Figs. 2.5** to **2.7** show computer generated examples of contact surface pressure profiles for various scenarios. These profiles demonstrate how crowning can reduce edge surface contact pressure in conditions of misalignment.

Fig. 2.8 shows an example of a computer generated relationship between allowable fitting misalignment and bearing life. It is possible to see from this Figure how the bearing lifetime is influenced by fitting misalignment.

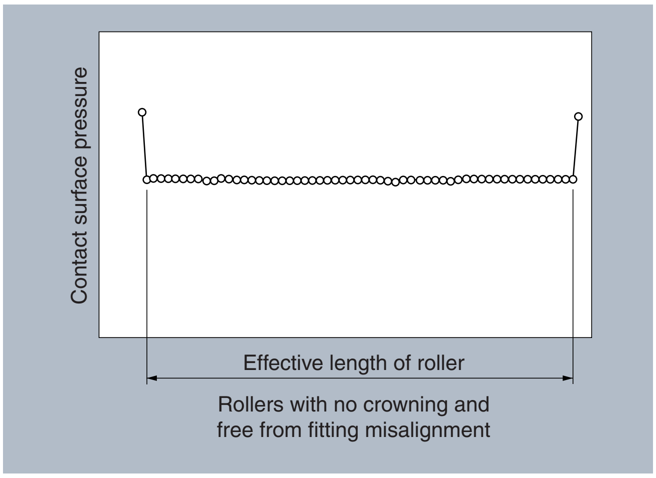


Fig. 2.5

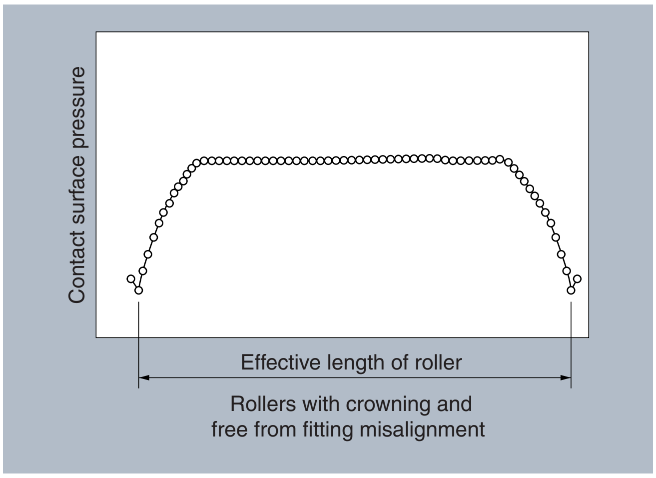


Fig. 2.7

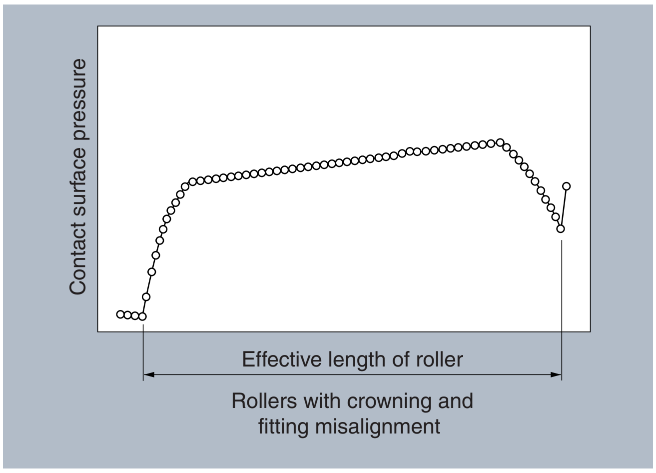


Fig. 2.6

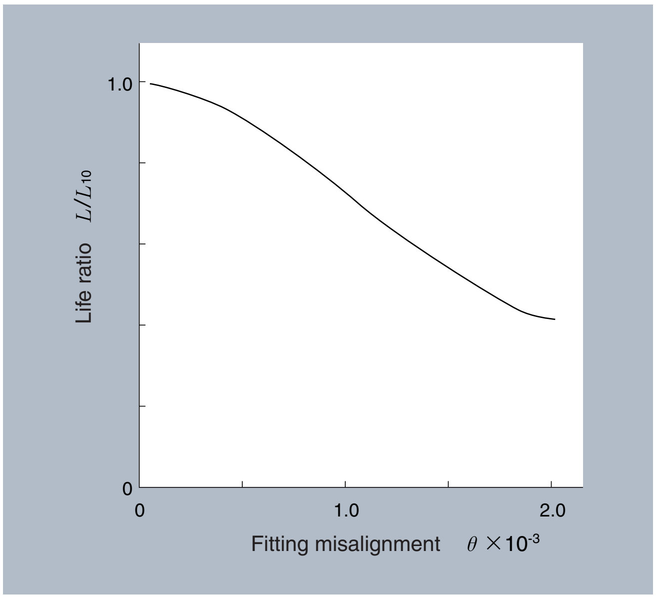


Fig. 2.8 Relationship of fitting misalignment to bearing lifetime

2.9 Basic static load rating

"Basic static load rating" is defined as the minimum static load acting on the center of a rolling element which results in a calculated contact stress value of:

4,000 MPa (408kgf/mm^2) for Roller bearings.
4,200 MPa (428kgf/mm^2) for Ball Bearings.

It has been empirically shown that the resulting permanent deformation on the rolling element and raceway caused by these magnitudes of contact stress is approximately 0.0001 time as great as the diameter of rolling element, and that this deformation level is maximum allowable deformation for smooth running of the bearing.

Basic static load rating for radial bearings is known as **"basic static radial load rating"**, and that for axial thrust bearing as **"basic static axial load rating"**. The bearing dimension tables in this catalog provide data for these load rating types under the parameter names C_{or}, and C_{oa}.

2.10 Allowable static bearing load

The basic static load rating described in **Subsection 2.9** is generally deemed as an allowable static bearing limit load, but in some cases this allowable limit load is set up larger than the basic static load rating and in some other cases it is set up smaller, according to the requirements for revolving smoothness and friction.

Generally this allowable limit load is decided considering the safety factor So in the following **formula (2.14)** and **Table 2.4**.

$$S_o = C_o / P_o \quad \cdots\cdots (2.14)$$

where,

S_o : Safety factor
C_o : Basic static rated load, N (kgf)
(For radial bearings: C_{or},
For thrust bearings: C_{oa})
$P_{o\,max}$: Maximum static bearing load, N (kgf)
(For radial bearings: P_{or} max,
For thrust bearings: C_{oa} max)

Table 2.4 Lower limit value of safety factor S_o

Operating conditions	Roller bearings	Ball bearings
Requirement for high revolving accuracy	3	2
Requirement for ordinal revolving accuracy (ordinary-purposed)	1.5	1
Where minor deterioration of revolving accuracy is allowed (Ex. Low speed revolution, duty load application, etc.)	1	0.5

Remarks: 1. The lower limit of S_o for drawn cup needle roller bearings is set at 3; for Premium Shell Product, the limit is set at 2.
2. The lower limit of S_o is set at 3 for an application where the AS type raceway is used in an axial thrust bearing.
3. Where vibration and shock load act on bearing, P_o max shall be determined considering the shock load factor.

3. Calculation of Bearing Loads

To compute bearing roads, the forces which act on the shaft being supported by the bearing must be determined. These forces include the inherent dead weight of the rotating body (the weight of the shafts and components themselves), loads generated by the working forces of the machine, and loads arising from transmitted power.

It is possible to calculate theoretical values for these loads; however, there are many instances where the load acting on the bearing is usually determined by the nature of the load acting on the main power transmission shaft.

3.1 Load acting on shafts

3.1.1 Load factors

The actual shaft loads on a machine that uses a bearing are usually greater than the theoretically determined values owing to vibration and impact occurring on the machine. For this reason, loads actually acting on a shaft system are often determined through multiplication by an appropriate load factors listed in **Table 3.1**.and **Table 3.2**.

$$K = f_W \cdot f_Z \cdot K_C \quad \cdots\cdots (3.1)$$

where

K : Actual load acting on shaft N (kgf)
K_c : Theoretically calculated value N (kgf)
f_w : Load factor **(Table 3.1)**
f_z : Gear factor **(Table 3.2)**

Table 3.1 Load factor f_w

Extent of shock	f_w	Application
Nearly no shock	1.0–1.2	Electrical machines, machine tools, measuring instruments
Light shock	1.2–1.5	Railway vehicles, automobiles, rolling mills, metal working machines, paper making machines, rubber mixing machines, printing machines, aircraft, textile machines, electrical units, office equipment
Heavy shock	1.5–3.0	Crushers, agricultural machines, construction machines, cranes

Table 3.2 Gear factor f_z

Types of gear	f_z
Precision ground gears (Pitch and profile errors of less than 0.02mm)	1.05–1.1
Ordinary machined gears (Pitch and profile errors of less than 0.1mm)	1.1–1.3

3.1.2 Load acting on gears

The loads acting on gears can be divided into tangential load (K_t), radial load (K_s) and axial load (K_a). The magnitude and acting direction of each load differ depending on the types of gear. This paragraph describes how to calculate the loads acting on parallel shaft gears and cross shaft gears for general use.

(1) Load acting on parallel shaft gear

Figs. 3.1 to **3.3** illustrate the loads acting on spur gear and helical gear which are used with a parallel shaft. The magnitude of each load can be determined using the **formulas (3.2)** to **(3.5)**.

$$\left.\begin{array}{ll} K_t = \dfrac{19.1\times10^6 \cdot H}{D_p \cdot n} & \text{N} \\[2ex] K_t = \dfrac{1.95\times10^6 \cdot H}{D_p \cdot n} & (\text{kgf}) \end{array}\right\} \quad \cdots\cdots (3.2)$$

$$K_s = K_t \cdot \tan\alpha \text{ (Spur gear)} \quad \cdots\cdots (3.3a)$$

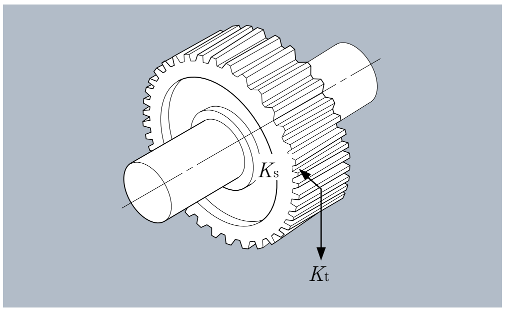


Fig. 3.1 Load acting on spur gear

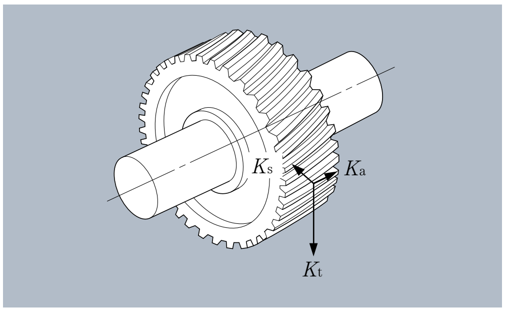


Fig. 3.2 Load acting on helical gear

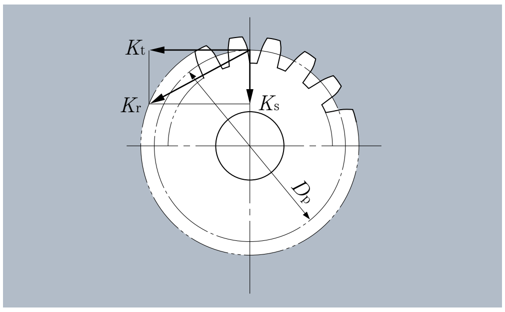


Fig. 3.3 Composite radial force acting on gear

$$=K_t \cdot \frac{\tan\alpha}{\cos\beta} \text{ (for helical gear)} \quad \cdots\cdots \textbf{(3.3b)}$$

$$K_r = \sqrt{K_t^2 + K_s^2} \quad \cdots\cdots \textbf{(3.4a)}$$

$$K_a = K_t \cdot \tan\beta \text{ (for helical gear)} \quad \cdots\cdots \textbf{(3.5)}$$

where,

K_t : Tangential load acting on gear (Tangential force) N (kgf)
K_s : Radial load acting on gear (separating force) N (kgf)
K_r : Load acting perpendicularly on gear shaft (composite force of tangential force and separating force) N (kgf)
K_a : Parallel load acting on gear shaft N (kgf)
H : Transmission power kw
n : Rotational speed min^{-1}
D_p : Pitch circle diameter of gear mm
α : Gear pressure angle deg
β : Gear helix angle deg

(2) Loads acting on cross shaft gears

Figs. 3.4 and **3.5** illustrate the loads acting on straight-tooth bevel gears and spiral bevel gears which are used with cross shafts.

The calculation methods for these gear loads are shown in **Table 3.3**. Herein, to calculate gear loads for straight bevel gears, the helix angle (β) is 0.

K_t : Tangential load acting on gear (Tangential force) N (kgf)
K_s : Radial load acting on gear (separating force) N (kgf)
K_a : Parallel load acting on gear shaft (axial load) N (kgf)
H : Transmission power kw
n : Rotational speed min^{-1}
D_{pm} : Mean pitch circle diameter mm
α : Gear pressure angle deg
β : Gear helix angle deg
δ : Pitch cone angle of gear deg

In general, the relationship between the loads acting on pinion and pinion gear can be expressed as follows, due to the perpendicular intersection of two shafts.

$$K_{sp} = K_{ag} \quad \cdots\cdots \textbf{(3.6)}$$

$$K_{ap} = K_{sg} \quad \cdots\cdots \textbf{(3.7)}$$

where,

K_{sp}, K_{sg} : Pinion and pinion gear separating force N (kgf)
K_{ap}, K_{ag} : Axial load acting on pinion and pinion gear N (kgf)

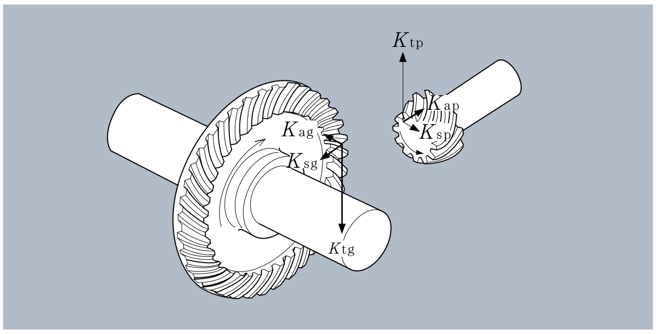


Fig. 3.4 Load acting on bevel gears

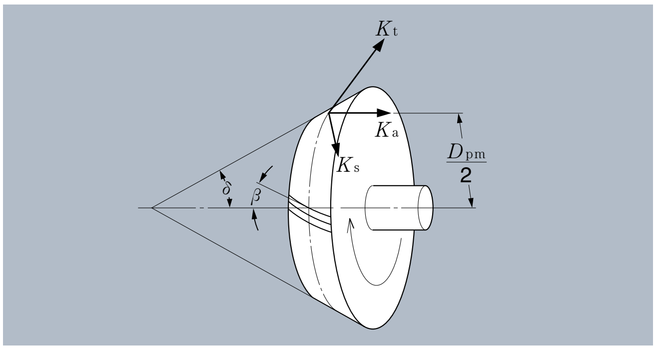


Fig.3.5 Bevel gear diagram

Table 3.3 Calculation formulas for determining loads acting on bevel gears

Unit N

Type of load	Rotational direction	Clockwise	Counter clockwise	Clockwise	Counter clockwise
	Helix angle	To right	To left	To left	To right
Tangential load (tangential force) K_t		$K_t = \frac{19.1\times10^6 \cdot H}{D_{pm}\cdot n}$, $\left\{\frac{1.95\times10^6 \cdot H}{D_{pm}\cdot n}\right\}$			
Radial load (separating force) K_s	Drive side	$K_s = K_t\left[\tan\alpha\frac{\cos\delta}{\cos\beta} + \tan\beta\sin\delta\right]$		$K_s = K_t\left[\tan\alpha\frac{\cos\delta}{\cos\beta} - \tan\beta\sin\delta\right]$	
	Driven side	$K_s = K_t\left[\tan\alpha\frac{\cos\delta}{\cos\beta} - \tan\beta\sin\delta\right]$		$K_s = K_t\left[\tan\alpha\frac{\cos\delta}{\cos\beta} + \tan\beta\sin\delta\right]$	
Load parallel to gear train (Axial load) K_a	Drive side	$K_a = K_t\left[\tan\alpha\frac{\sin\delta}{\cos\beta} - \tan\beta\cos\delta\right]$		$K_a = K_t\left[\tan\alpha\frac{\sin\delta}{\cos\beta} + \tan\beta\cos\delta\right]$	
	Driven side	$K_a = K_t\left[\tan\alpha\frac{\sin\delta}{\cos\beta} + \tan\beta\cos\delta\right]$		$K_a = K_t\left[\tan\alpha\frac{\sin\delta}{\cos\beta} - \tan\beta\cos\delta\right]$	

The orientation of loading on a spiral bevel gear will vary depending on the direction of the helix angle, the direction of rotation and whether the gear is a driving or driven gear.

The separating force (K_s) and the axial load (K_a) are shown in the positive direction in **Fig. 3.5**. The direction of rotation and the helix direction are defined as viewed from the large end of the gear. For the gear illustrated in **Fig. 3.5** these directions are clockwise and to the right.

3.1.3 Loads acting on chain and belt shafts

When power is transmitted by means of a chain or belt as illustrated in **Fig. 3.6**, the loads acting on the sprocket or pulley can be determined by **formula (3.8).**

$$K_t = \frac{19.1 \times 10^6 \cdot H}{D_p \cdot n} \left(\frac{1.95 \times 10^6 \cdot H}{D_p \cdot n} \right) \quad \cdots\cdots\cdots (3.8)$$

where,

K_t : Load acting on sprocket or pulley N (kgf)

HP : Transmission power kW

D_p : Pitch circle diameter of sprocket or pulley mm

For belt drives, an initial tension is applied to ensure sufficient normal force between the belt and pulley during operation.

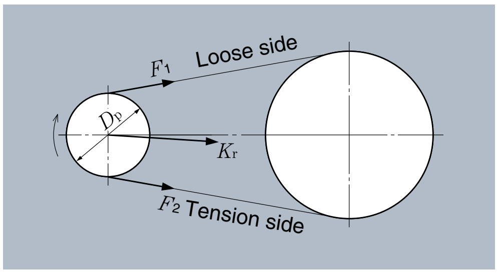


Fig. 3.6 Loads acting on chain/ belt

Taking into account the initial tension, the radial load acting on the pulley can be determined by **formula (3.9)**. For chain drives, the radial load can be expressed using the same formula, if vibration and shock are taken into consideration.

$$K_r = f_b \cdot K_t \quad \cdots\cdots\cdots (3.9)$$

where,

K_r : Radial load acting on sprocket or pulley N (kgf)

f_b : Chain/belt factor **(Table 3.4)**

Table 3.4 Chain/belt factor f_b

Type of chain / belt	f_b
Chain (single row type)	1.2–1.5
Vee-belt	1.5–2.0
Timing belt	1.1–1.3
Flat belt (with tension pulley)	2.5–3.0
Flat belt	3.0–4.0

3.2 Bearing load distribution

Any loads acting on shafts are distributed to the bearings. The bearing load distribution is determined by considering the shaft to be a static beam supported by the bearings.

For example, the loads acting on the bearings supporting the gear shaft illustrated in **Fig. 3.7** can be expressed using **formulas (3.10)** and **(3.11).**

$$F_{rA} = K_{r\,\mathrm{I}} \frac{b}{l} - K_{r\,\mathrm{II}} \frac{c}{l} - K_a \frac{D_p}{2l} \quad \cdots\cdots\cdots (3.10)$$

$$F_{rB} = K_{r\,\mathrm{I}} \frac{a}{l} + K_{r\,\mathrm{II}} \frac{a+b+c}{l} + K_a \frac{D_p}{2l} \quad \cdots\cdots\cdots (3.11)$$

where,

F_{rA} : Radial load acting on bearing-A N (kgf)

F_{rB} : Radial load acting on bearing-B N (kgf)

$K_{r\,\mathrm{I}}$: Radial load acting on gear-Ⅰ N (kgf)

K_a : Axial load acting on gear-Ⅰ N (kgf)

$K_{r\,\mathrm{II}}$: Axial load acting on gear-Ⅱ N (kgf)

D_p : Pitch circle diameter of gear-Ⅰ mm

l : Bearing to bearing distance mm

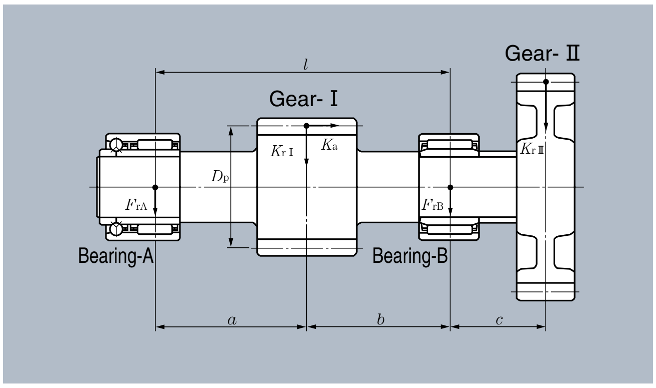


Fig. 3.7 Gear shaft

3.3 Mean load

The load on bearings used in machines will often fluctuate according to a fixed duty cycle. The load on bearings operating under such conditions can be converted to a mean load (F_m) . The mean load is a load which gives the bearings the same life they would have under constant operating conditions.

(1) Stepped fluctuating load

The mean bearing load, F_m, for stepped loads is calculated using **formula (3.12)**, where F_1 , F_2 .... F_n are the bearing loads, and the rotational speed and running time are n_1, n_2.......n_n and t_1, t_2....... t_n respectively.

$$F_m = \left[\frac{\Sigma \left(F_i^{\,p} n_i t_i \right)}{\Sigma \left(n_i t_i \right)} \right]^{1/p} \cdots\cdots\cdots \textbf{(3.12)}$$

where:

p=10/3 for roller bearing
p=3 for ball bearing

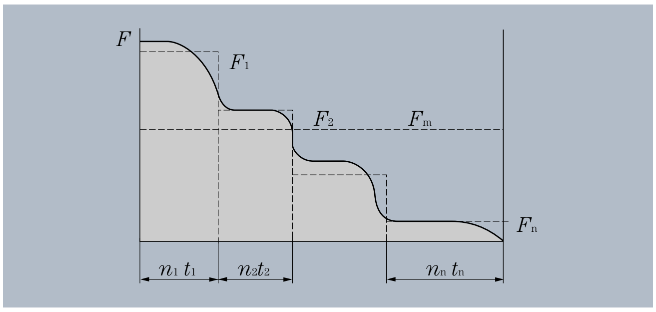


Fig. 3.8 Stepped fluctuating load

(2) Cyclical load

Where the bearing load can be expressed as a function of time $F(t)$, repeating with cycle time (t),1 the mean load can be expressed **formula (3.13)**.

$$F_m = \left[\frac{1}{t_0} \int_0^{t_0} F(t)^p d_t \right]^{1/p} \cdots\cdots\cdots \textbf{(3.13)}$$

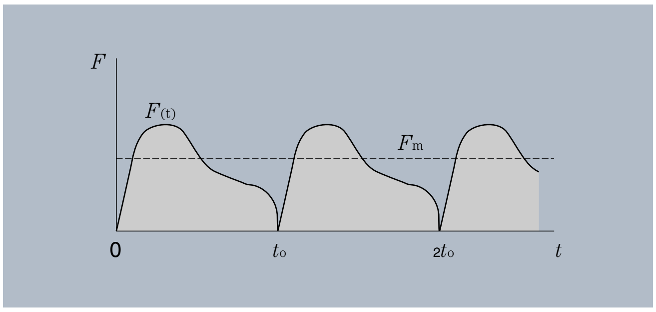


Fig. 3.9 Load fluctuating as cyclical function of time

(3) Linearly fluctuating load

The mean load F_m can be approximated by **formula (3.14)**.

$$F_m = \frac{F_{min} + 2F_{max}}{3} \cdots\cdots\cdots \textbf{(3.14)}$$

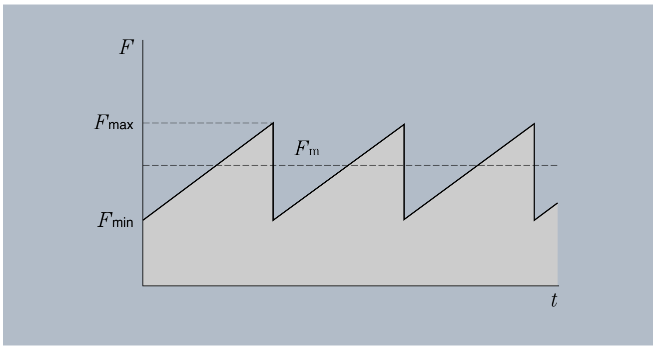


Fig. 3.10 Linearly fluctuating load

(4) Sinusoidal load

The mean load F_m, can be approximated by **formulas (3.15)**, and **(3.16)**.

case of (a) $F_m = 0.75 F_{max}$ $\cdots\cdots\cdots$ **(3.15)**
case of (b) $F_m = 0.65 F_{max}$ $\cdots\cdots\cdots$ **(3.16)**

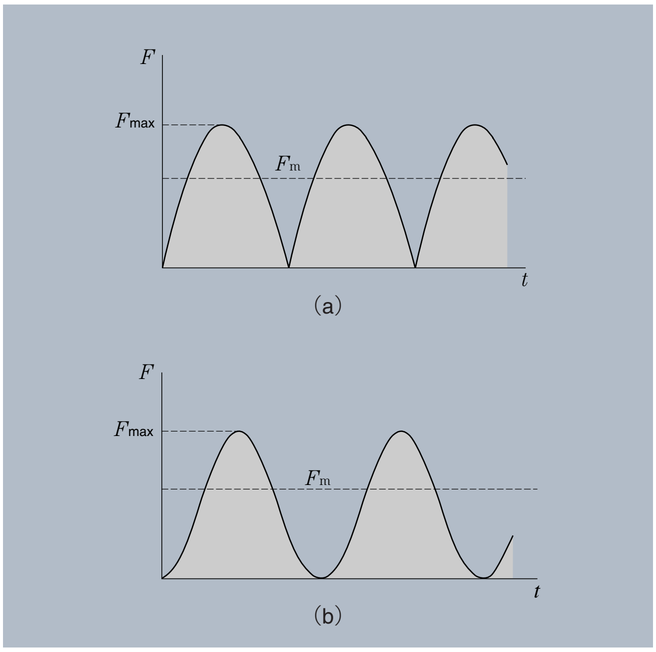


Fig. 3.11 Sinusoidal load

4. Bearing Accuracy

The dimensional, profile and running accuracies of rolling bearings are specified in ISO Standard as applicable and JIS B 1514 (Accuracy of Rolling Bearings).

"**Dimensional accuracy**" and "**Profile accuracy**" are the items indispensable in installing the rolling bearings on a shaft and in a bearing housing, and allowable bearing run-out in running is specified as the running accuracy.

Dimensional accuracy:

Dimensional accuracy means the respective allowable values for bore diameter, outer diameter, width or height (limited to thrust bearing) and chamfering dimension.

Profile accuracy:

Profile accuracy relates to tolerances for inside diameter variation, mean inside diameter variation, outside diameter variation, mean outside diameter variation, and ring width variation.

Running accuracy:

Running accuracy relates to tolerances for radial runout and axial runout with inner ring and outer ring, perpendicularity of ring face, perpendicularity of outside surface, and raceway thickness variation (thrust bearing).

Regarding the accuracy class of the machined ring needle roller bearings, class-0 is equivalent to bearings of the normal precision class, and precision becomes progressively higher as the class number becomes smaller; i.e. Class 6 is less precise than Class 5, which is less precise than Class 4, and so on.

Bearings of Class-0 are mostly used for general applications while bearings of Class-5 or Class-4 are used where the required running accuracies and revolutions are high or less friction and less fluctuation are required for bearings.

Various bearing types are available for **NTN** needle roller bearings and the representative types and the accuracy classes applicable to them are as shown in **Table 4.1**.

Dimensional item symbols used in the accuracy standard are given in **Table 4.2**, the radial bearing accuracy specified every accuracy class given in **Table 4.3**, the thrust bearing accuracy specified every accuracy class given in **Table 4.4**, and the allowable values for chamfering dimension given in **Table 4.5**.

Table 4.2 Dimensional item symbols used in applicable standards

●Radial bearings

Classification	Symbols	Symbol representation	Symbols under JIS B 0021 (Reference)
Dimensional accuracy	Δd_{mp}	Dimensional tolerance for in-plane mean bore diameter	―
	Δd_s	Dimensional tolerance for bore diameter	―
	ΔD_{mp}	Dimensional tolerance for in-plane mean outer diameter	―
	ΔD_s	Dimensional tolerance for outer diameter	―
	ΔB_s	Dimensional tolerance for inner ring width	―
	ΔC_s	Dimensional tolerance for outer ring width	―
Profile accuracy	V_{dp}	Variation of in-plane bore diameter	Roundness ○ 1)
	V_{dmp}	Variation of in-plane mean bore diameter	Cylindricity ⌭ 2)
	V_{Dp}	Variation of in-plane outer diameter	Roundness ○ 1)
	V_{Dmp}	Variation of in-plane mean outer diameter	Cylindricity ⌭ 2)
	V_{Bs}	Variation of inner ring width	Parallelism //
	V_{Cs}	Variation of outer ring width	Parallelism //
Running accuracy	K_{ia}	Radial run-out of inner ring	Run-out ↗
	K_{ea}	Radial run-out of outer ring	Run-out ↗
	S_{ia}	Axial run-out of inner ring	―
	S_{ea}	Axial run-out of outer ring	―
	S_d	Perpendicularity of face (inner ring)	Perpendicularity ⊥
	S_D	Perpendicularity of outside surface (outer ring)	Perpendicularity ⊥

●Thrust bearings

Classification	Symbols	Symbol representation	Symbols under JIS B 0021 (Reference)
Dimensional accuracy	Δd_{mp}	Single plane mean bore diameter deviation on single-direction bearing	―
	Δd_{2mp}	Single plane mean bore diameter deviation on central washer	―
	ΔD_{mp}	Dimensional tolerance for in-plane mean outer diameter	―
Profile accuracy	V_{dp}	Bore diameter variation in a single radial plane on single-direction bearing	Roundness ○ 1)
	V_{d2p}	Bore diameter variation in a single radial plane on central washer	Roundness ○ 1)
	V_{Dp}	Variation of in-plane outer diameter	Roundness ○ 1)
Running accuracy	S_i	Raceway thickness variation on shaft washer	Run-out ↗
	S_e	Raceway thickness variation on housing washer	Run-out ↗

1) The roundness specified in JIS B 0021 is applicable to the tolerance V_{dp} for variation of radial in-plane bore diameter or nearly half of V_{Dp}.
2) The cylindricity specified in JIS B 0021 is applicable to the tolerance V_{dmp} for in-uniformity of radial in-plane mean diameter or nearly half of V_{Dmp}.

Table 4.1 Bearing types and corresponding accuracy classes

Bearing type		Applicable accuracy class				Applicable table
Needle roller bearing, Clearance-adjustable needle roller bearing		JIS Class-0 —	JIS Class-6 —	JIS Class-5 —	JIS class-4 JIS class-4	**Table 4.3** **Table 4.3**
Complex bearing	Radial bearing Thrust bearing	JIS Class-0 NTN Class 0	JIS Class-6 NTN Class 6	JIS Class-5 NTN Class 5	— NTN Class 4	**Table 4.3** **Table 4.4**
Needle roller bearing with double-direction thrust roller bearing	Radial bearing Thrust bearing	— —	— —	JIS Class-5 NTN Class 5	JIS Class-4 NTN Class 4	**Table 4.3** **Table 4.4**
Thrust roller bearing Roller follower/cam follower		NTN Class 0 JIS Class-0	NTN Class 6 —	NTN Class 5 —	NTN Class 4 —	**Table 4.4** **Table 4.3**

Table 4.3 Tolerances for radial bearings
Table 4.3(1) Inner rings

Nominal bore diameter d mm		Dimensional tolerance for mean bore diameter Δ_{dmp}								Variation of mean bore diameter V_{dp}				Allowable variation of bore diameter V_{dmp}			
		Class 0		Class 6		Class 5		Class 4		Class 0	Class 6	Class 5	Class 4	Class 0	Class 6	Class 5	Class 4
over	incl.	high	low	high	low	high	low	high	low	max				max			
2.5❶	10	0	−8	0	−7	0	−5	0	−4	10	9	5	4	6	5	3	2
10	18	0	−8	0	−7	0	−5	0	−4	10	9	5	4	6	5	3	2
18	30	0	−10	0	−8	0	−6	0	−5	13	10	6	5	8	6	3	2.5
30	50	0	−12	0	−10	0	−8	0	−6	15	13	8	6	9	8	4	3
50	80	0	−15	0	−12	0	−9	0	−7	19	15	9	7	11	9	5	3.5
80	120	0	−20	0	−15	0	−10	0	−8	25	19	10	8	15	11	5	4
120	150	0	−25	0	−18	0	−13	0	−10	31	23	13	10	19	14	7	5
150	180	0	−25	0	−18	0	−13	0	−10	31	23	13	10	19	14	7	5
180	250	0	−30	0	−22	0	−15	0	−12	38	28	15	12	23	17	8	6
250	315	0	−35	0	−25	0	−18	—	—	44	31	18	—	26	19	9	—
315	400	0	−40	0	−30	0	−23	—	—	50	38	23	—	30	23	12	—
400	500	0	−45	0	−35	—	—	—	—	56	44	—	—	34	26	—	—

❶ 2.5mm is included in this dimensional category.
❷ This table is applied to the ball bearings.

Table 4.3 (2) Outer rings

Nominal outer diameter D mm		Dimensional tolerance for mean outer diameter Δ_{Dmp}								Allowable variation of outer diameter V_{Dp}				Allowable variation of mean outer diameter V_{Dmp}			
		Class 0		Class 6		Class 5		Class 4		Class 0	Class 6	Class 5	Class 4	Class 0	Class 6	Class 5	Class 4
over	incl.	high	low	high	low	high	low	high	low	max.				max.			
6❶	18	0	−8	0	−7	0	−5	0	−4	10	9	5	4	6	5	3	2
18	30	0	−9	0	−8	0	−6	0	−5	12	10	6	5	7	6	3	2.5
30	50	0	−11	0	−9	0	−7	0	−6	14	11	7	6	8	7	4	3
50	80	0	−13	0	−11	0	−9	0	−7	16	14	9	7	10	8	5	3.5
80	120	0	−15	0	−13	0	−10	0	−8	19	16	10	8	11	10	5	4
120	150	0	−18	0	−15	0	−11	0	−9	23	19	11	9	14	11	6	5
150	180	0	−25	0	−18	0	−13	0	−10	31	23	13	10	19	14	7	5
180	250	0	−30	0	−20	0	−15	0	−11	38	25	15	11	23	15	8	6
250	315	0	−35	0	−25	0	−18	0	−13	44	31	18	13	26	19	9	7
315	400	0	−40	0	−28	0	−20	0	−15	50	35	20	15	30	21	10	8
400	500	0	−45	0	−33	0	−23	—	—	56	41	23	—	34	25	12	—
500	630	0	−50	0	−38	0	−28	—	—	63	48	28	—	38	29	14	—

❶ 6mm is included in this dimensional category.
❷ This table is applied to the ball bearings.

Unit μm

Radial run-out K_{ia}				Perpendicularity of face S_d		Axial run-out S_{ia} [2]		Allowable width deviation Δ_{Bs}				Allowable width variation V_{Bs}				Nominal bore diameter d mm	
Class 0	Class 6	Class 5	Class 4	Class 5	Class 4	Class 5	Class 4	Class 0,6		Class 5,4		Class 0	Class 6	Class 5	Class 4		
max				max		max		high	low	high	low	max				over	incl.
10	6	4	2.5	7	3	7	3	0	−120	0	−40	15	15	5	2.5	2.5 [1]	10
10	7	4	2.5	7	3	7	3	0	−120	0	−80	20	20	5	2.5	10	18
13	8	4	3	8	4	8	4	0	−120	0	−120	20	20	5	2.5	18	30
15	10	5	4	8	4	8	4	0	−120	0	−120	20	20	5	3	30	50
20	10	5	4	8	5	8	5	0	−150	0	−150	25	25	6	4	50	80
25	13	6	5	9	5	9	5	0	−200	0	−200	25	25	7	4	80	120
30	18	8	6	10	6	10	7	0	−250	0	−250	30	30	8	5	120	150
30	18	8	6	10	6	10	7	0	−250	0	−250	30	30	8	5	150	180
40	20	10	8	11	7	13	8	0	−300	0	−300	30	30	10	6	180	250
50	25	13	—	13	—	15	—	0	−350	0	−350	35	35	13	—	250	315
60	30	15	—	15	—	20	—	0	−400	0	−400	40	40	15	—	315	400
65	35	—	—	—	—	—	—	0	−450	—	—	50	45	—	—	400	500

Unit μm

Radial run-out K_{ea}				Perpendicularity of outside surface S_D		Axial run-out S_{ea} [2]		Allowable width deviation Δ_{Cs}	Allowable width variation V_{Cs}				Nominal outer diameter D mm	
Class 0	Class 6	Class 5	Class 4	Class 5	Class 4	Class 5	Class 4	Class 0,6,5,4	Class 0	Class 6	Class 5	Class 4		
max				max		max			max				over	incl.
15	8	5	3	8	4	8	5	Depending on the tolerance of Δ_{Bs} for d of same bearing.	Depending on the applicable allowable value of V_{Bs} for d of same bearing.		5	2.5	6 [1]	18
15	9	6	4	8	4	8	5				5	2.5	18	30
20	10	7	5	8	4	8	5				5	2.5	30	50
25	13	8	5	8	4	10	5				6	3	50	80
35	18	10	6	9	5	11	6				8	4	80	120
40	20	11	7	10	5	13	7				8	5	120	150
45	23	13	8	10	5	14	8				8	5	150	180
50	25	15	10	11	7	15	10				10	7	180	250
60	30	18	11	13	8	18	10				11	7	250	315
70	35	20	13	13	10	20	13				13	8	315	400
80	40	23	—	15	—	23	—				15	—	400	500
100	50	25	—	18	—	25	—				18	—	500	630

Table 4.4 Tolerances of thrust roller bearings
Table 4.4 (1) Inner rings and center rings

Unit μm

Nominal bore diameter d or d_2 mm		Allowable deviation of mean diameter Δ_{dmp} or Δ_{d2mp}				Allowable variation of bore diameter V_{dp} or V_{d2p}		Allowable variation of raceway thickness ❶ S_i			
		Class 0, 6, 5		Class 4		Class 0, 6, 5	Class 4	Class 0	Class 6	Class 5	Class 4
over	incl.	high	low	high	low	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

❶ The complex bearings are applicable to the category of single plane bearing d which corresponds to the same nominal outer diameter of same diameter series, without being applicable to d_2 category.

Table 4.4 (2) Outer rings

Unit μm

Nominal outer diameter D mm		Allowable deviation of mean outer diameter Δ_{Dmp}				Allowable variation of outer diameter V_{Dp}		Allowable variation of raceway thickness S_e
		Class 0,6,5		Class 4		Class 0,6,5	Class 4	Class 0,6,5,4
over	incl.	high	low	high	low	max		max
10	18	0	−11	0	−7	8	5	Depending on the applicable allowable value of S_1 for d or d_2 of same bearing.
18	30	0	−13	0	−8	10	6	
30	50	0	−16	0	−9	12	7	
50	80	0	−19	0	−11	14	8	
80	120	0	−22	0	−13	17	10	
120	180	0	−25	0	−15	19	11	
180	250	0	−30	0	−20	23	15	
250	315	0	−35	0	−25	26	19	
315	400	0	−40	0	−28	30	21	
400	500	0	−45	0	−33	34	25	
500	630	0	−50	0	−38	38	29	
630	800	0	−75	0	−45	55	34	

Table 4.5 Allowable critical value for chamfering dimension
Table 4.5 (1) Radial bearings

Unit mm

$r_{s\,min}$ ❶	Nominal bore diameter d		Radial direction	Axial direction
	over	incl.	$r_{s\,max}$	
0.15	—	—	0.3	0.6
0.2	—	—	0.5	0.8
0.3	—	40	0.6	1
	40	—	0.8	1
0.6	—	40	1	2
	40	—	1.3	2
1	—	50	1.5	3
	50	—	1.9	3
1.1	—	120	2	3.5
	120	—	2.5	4
1.5	—	120	2.3	4
	120	—	3	5
2	—	80	3	4.5
	80	220	3.5	5
	220	—	3.8	6
2.1	—	280	4	6.5
	280	—	4.5	7
2.5	—	100	3.8	6
	100	280	4.5	6
	280	—	5	7
3	—	280	5	8
	280	—	5.5	8
4	—	—	6.5	9

❶ Allowable minimum values for the chamfering dimension "r" .

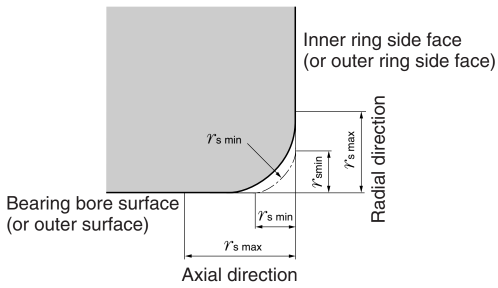


Table 4.5 (2) Thrust bearings

Unit mm

$r_{s\,min}$ ❶	Radial and axial directions $r_{s\,max}$
0.3	0.8
0.6	1.5
1	2.2
1.1	2.7
1.5	3.5
2	4
2.1	4.5
3	5.5

❶ Allowable minimum values for the chamfering dimension "r" .

Table 4.6 Basic tolerances

Unit μm

Basic dimension (mm)		IT basic tolerance classes									
over	incl.	IT1	IT2	IT3	IT4	IT5	IT6	IT7	IT8	IT9	IT10
	3	0.8	1.2	2	3	4	6	10	14	25	40
3	6	1	1.5	2.5	4	5	8	12	18	30	48
6	10	1	1.5	2.5	4	6	9	15	22	36	58
10	18	1.2	2	3	5	8	11	18	27	43	70
18	30	1.5	2.5	4	6	9	13	21	33	52	84
30	50	1.5	2.5	4	7	11	16	25	39	62	100
50	80	2	3	5	8	13	19	30	46	74	120
80	120	2.5	4	6	10	15	22	35	54	87	140
120	180	3.5	5	8	12	18	25	40	63	100	160
180	250	4.5	7	10	14	20	29	46	72	115	185
250	315	6	8	12	16	23	32	52	81	130	210
315	400	7	9	13	18	25	36	57	89	140	230
400	500	8	10	15	20	27	40	63	97	155	250
500	630	9	11	16	22	30	44	70	110	175	280
630	800	10	13	18	25	35	50	80	125	200	320
800	1 000	11	15	21	29	40	56	90	140	230	360
1 000	1 250	13	18	24	34	46	66	105	165	260	420
1 250	1 600	15	21	29	40	54	78	125	195	310	500
1 600	2 000	18	25	35	48	65	92	150	230	370	600
2 000	2 500	22	30	41	57	77	110	175	280	440	700
2 500	3 150	26	36	50	69	93	135	210	330	540	860

5. Bearing Internal Clearance

5.1 Bearing internal clearance

Bearing radial internal clearance (free clearance) is the amount of internal clearance a bearing has before being installed on a shaft or into a housing. When either the inner ring or the outer ring is fixed and the other ring is free to move, displacement takes place in the radial direction. This amount of displacement is called the radial internal clearance.

The radial internal clearance values of **NTN** machined ring needle roller bearings are listed in **Table 5.1**. **Table 5.1 (1)** shows the interchangeable clearances, which remain unchanged even if inner or outer ring are switched with those from different bearings. **Table 5.1 (2)** shows non-interchangeable clearances, which are supplied as matched sets due to the tighter clearance ranges. Bearing clearances are represented by the symbols C2, normal, C3, and C4 in increasing order from smallest to largest. Non-interchangeable clearances symbols are followed by "NA" for identification.

For radial clearance values for bearings other than machined ring needle roller bearings, refer to "Commentary" provided with the appropriate dimension tables.

5.2 Running clearance

5.2.1 Running clearance selection

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

Theoretically, regarding bearing life, the optimum operating internal clearance of any bearing would be a slight negative clearance after the bearing has reached normal operating temperature.

Unfortunately, under actual operating conditions, maintaining such optimum tolerances is often difficult at best. Due to various fluctuating operating conditions this slight minus clearance can quickly become a large minus, greatly lowering the life of the bearing and causing excessive heat to be generated. Therefore, an initial internal clearance that will result in a slightly greater than minus internal operating clearance should be selected.

Under normal operating conditions (e.g. normal load, fit, speed, temperature, etc.), a standard internal clearance will give a very satisfactory operating clearance.

Table 5.1 Radial internal clearance in machined ring needle roller bearing

Table 5.1 (1) Interchangeable bearings

Unit μm

Nominal bore diameter d (mm)		Radial internal clearance							
		C2		Normal ❶		C3		C4	
over	incl.	min	max	min	max	min	max	min	max
—	10	0	30	10	40	25	55	35	65
10	18	0	30	10	40	25	55	35	65
18	24	0	30	10	40	25	55	35	65
24	30	0	30	10	45	30	65	40	70
30	40	0	35	15	50	35	70	45	80
40	50	5	40	20	55	40	75	55	90
50	65	5	45	20	65	45	90	65	105
65	80	5	55	25	75	55	105	75	125
80	100	10	60	30	80	65	115	90	140
100	120	10	65	35	90	80	135	105	160
120	140	10	75	40	105	90	155	115	180
140	160	15	80	50	115	100	165	130	195
160	180	20	85	60	125	110	175	150	215
180	200	25	95	65	135	125	195	165	235
200	225	30	105	75	150	140	215	180	255
225	250	40	115	90	165	155	230	205	280
250	280	45	125	100	180	175	255	230	310
280	315	50	135	110	195	195	280	255	340
315	355	55	145	125	215	215	305	280	370
355	400	65	160	140	235	245	340	320	415
400	450	70	190	155	275	270	390	355	465

❶ Supplementary suffix codes of clearance is not added to bearing numbers.

Table 5.1 (2) Non-interchangeable bearings

Unit μm

Nominal bore diameter d (mm)		Radial internal clearance							
		C2NA		NA ❷		C3NA		C4NA	
over	incl.	min	max	min	max	min	max	min	max
—	10	10	20	20	30	35	45	45	55
10	18	10	20	20	30	35	45	45	55
18	24	10	20	20	30	35	45	45	55
24	30	10	25	25	35	40	50	50	60
30	40	12	25	25	40	45	55	55	70
40	50	15	30	30	45	50	65	65	80
50	65	15	35	35	50	55	75	75	90
65	80	20	40	40	60	70	90	90	110
80	100	25	45	45	70	80	105	105	125
100	120	25	50	50	80	95	120	120	145
120	140	30	60	60	90	105	135	135	160
140	60	35	65	65	100	115	150	150	180
160	180	35	75	75	110	125	165	165	200
180	200	40	80	80	120	140	180	180	220
200	225	45	90	90	135	155	200	200	240
225	250	50	100	100	150	170	215	215	265
250	280	55	110	110	165	185	240	240	295
280	315	60	120	120	180	205	265	265	325
315	355	65	135	135	200	225	295	295	360
355	400	75	150	150	225	255	330	330	405
400	450	85	170	170	255	285	370	370	455

❷ For bearing with normal clearance, only NA is added to bearing numbers.
EX. NA4920NA

5.2.2 Calculation of running clearance

The internal clearance differential between the free clearance and the operating (running) clearance (the amount of clearance reduction caused by interference fits, or clearance variation due to the temperature difference between the inner and outer rings) can be calculated by the following formula:

$$\delta_{eff} = \delta_o - (\delta_f + \delta_t) \quad \cdots\cdots\cdots\cdots\cdots\cdots \textbf{(5.1)}$$

where,

δ_{eff} : Running clearance mm

δ_o : Free clearance mm

δ_f : Reduction in internal clearance by interference mm

δ_t : Reduction in internal clearance due to inner/outer ring temperature difference mm

(1) Reduction in radial clearance by interference

When bearings are installed with interference fits on shafts and in housings, the inner ring will expand and the outer ring will contract; thus reducing the bearing' s internal clearance. The amount of expansion or contraction varies depending on the shape of the bearing, the shape of the shaft or housing, dimensions of the respective parts, and the type of materials used. The differential is approximately 85% of the effective interference. For details, refer to **Table 6.4** on page A-35.

$$\delta_f \approx 0.85 \cdot \Delta_{deff} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \textbf{(5.2)}$$

where,

δ_f : Reduction in internal clearance by interference mm

Δ_{deff} : Effective interference mm

(2) Reduction in radial clearance due to inner/outer ring temperature difference

During operation, normally the outer ring will be from 5 to l0˚C cooler than the inner ring or rolling elements. However, if the cooling effect of the housing is large, the shaft is connected to a heat source, or a heat substance is conducted through a hollow shaft; the temperature difference between the two rings can be even greater. The amount of internal clearance is thus further reduced by the differential expansion of the two rings.

$$\delta_t = \alpha \cdot \Delta_T \cdot D_o \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \textbf{(5.3)}$$

where,

δ_t : Reduction in internal clearance due to inner/outer ringtemperature difference mm

α : Linear expansion coefficient of bearing steel 12.5×10^{-6}/˚C

Δ_T : Inner ring – outer ring temperature difference ℃

D_o : Outer ring raceway diameter mm

When a shaft or housing is directly used as a raceway, the temperature difference (Δ_T) can be determined, treating the shaft as an inner ring and the housing as an outer ring.

5.3 Fits and bearing radial internal clearance

Once the dimensional tolerances for the shaft outside diameter and the housing bore diameter have been determined, a simple nomogram such as a one in **Fig. 5.1** may be used as a guide to determine the initial radial internal clearance for the bearing that will later lead to an appropriate internal clearance of the installed bearing. The nomogram in **Fig. 5.1** is used as the guideline as stated above. For details feel free to contact **NTN**.

For example, where the fit condition of a needle roller bearing with an inner ring is already given as J7 m6, **Fig.5.1** shows that clearance C3 must be used to get the standard running clearance after installation.

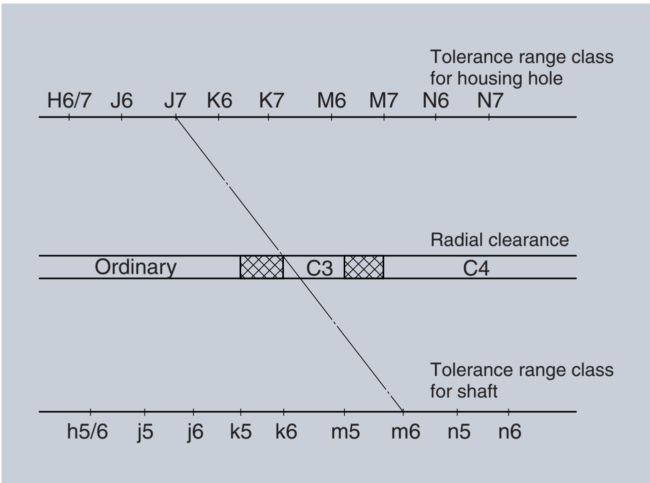


Fig. 5.1 Relationship between bearing fits and radial clearance

6. Bearing Fits

6.1 About bearing fits

For rolling bearings, the inner ring and outer ring are fixed on the shaft or in the housing so that relative movement does not occur between the fitted surfaces of the bearing ring and the shaft or housing in radial, axial and rotational directions when a load acts on the bearing. Depending on presence/absence of interference, fit modes can be categorized into **“interference fit”**, **“transition fit”**, and **“loose fit”**.

The most effective practice to position a bearing is to provide an interference on the fit surfaces between the bearing ring and shaft or housing. Furthermore, as its advantage this tight fit method supports the thin-walled bearing ring with uniform load throughout its entire circumference without any loss of load carrying capacity.

The needle roller bearing is a bearing type which allows separation of the inner ring and the outer ring from one another and, therefore, it can be installed on a shaft or in a housing with an interference applied to both. In the case of "tight fit", the ease of bearing installation and removal. The bearing ring subjected to stationary load can be "loose-fitted". In contrast, tight fitting may not apply to all bearing applications because ease of mounting or removal of the bearing will be jeopardized.

6.2 Necessity of proper fit

Improper fit could lead to damage and shorter life of the bearing. Therefore, advance careful analysis is needed for selection of proper fit. Representative examples of bearing defects caused by improper fit are as described below.

- Fracture of bearing ring, and displacement of bearing ring
- Wear of bearing ring, shaft and housing caused by creep and fretting corrosion
- Seizure caused by insufficient internal clearance
- Insufficient running accuracy and abnormal noise caused by deformed raceway surface

6.3 Fit selection

Fit selection is generally done in accordance with the rule specified hereunder.

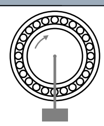
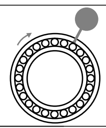
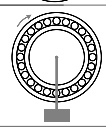
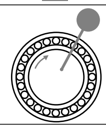
The loads acting on each bearing ring are divided into running load, stationary load and directionally unstable load according to the direction and characteristic of loads acting on the bearing.

A bearing ring that carries both running load and indeterminate direction load is provided with tight fit while a bearing ring that carries static load may be provided with either transition fit or loose fit (refer to **Table 6.1**).

Where load of high magnitude or vibration and shock loads act on a bearing or if a light alloy/plastic housing is used, it is necessary to secure a large interference. **However, if this type of practice is applied, it is necessary to consider the rigidity of housing in order to avoid problems including deformation or fracture of the housing, deformation of the bearing, galling on fit surfaces, as well as resultant poor fit accuracy.**

For an application subjected to high running accuracy, bearings of high accuracy must be used with a shaft and a housing of higher dimensional accuracy so as not to require a large interference Applying a large interference would cause the shaft or housing profile to be transferred to the bearing track, which could then interfere with the bearing running accuracy.

Table 6.1 Radial load and bearing fit

Bearing running conditions	Sketch	Load characteristic	Bearing fit	
			Inner ring	Outer ring
Inner ring : Rotation Outer ring: static Load direction: constant		Rotating inner ring load	Needs to be tight fit	May be loose fit
Inner ring: static Outer ring: rotation Load direction: rotating with outer ring		Static outer ring load		
Inner ring: static Outer ring: rotation Load direction: constant		Rotating inner ring load	May be loose fit	Needs to be tight fit
Inner ring: rotation Outer ring: static Load direction : rotating with inner ring		Static outer ring load		
Inner ring: rotation or static Outer ring: rotation or static Load direction: The direction can not be fixed.	Load direction is non-constant due to directional fluctuation, unbalanced load, etc.	Directionally unstable load	Needs to be tight fit	Needs to be tight fit

6.4 Recommended fits

The dimensional tolerances for the diameter of a shaft and the bore diameter of a bearing housing, on/in which a bearing is installed, are standardized under the metric system in ISO 286 and JIS B 0401 (Bases of tolerances, deviations and fits). Hence, bearing fits are determined by selection of the dimensional tolerances for shaft diameter and housing bore diameter as applicable.

Table 6.2 shows the recommended fits for the machined ring needle roller bearings (with inner ring) that are generally selected based on the dimensional and load conditions. **Table 6.3** shows the numerical fit values.

For the recommended fits for others than the machined ring needle roller bearings, refer to "Commentary" described in the respective Dimension Tables.

Table 6.2 General standards for fits of machined ring needle roller bearing (JIS Class 0, Class 6)

Table 6.2 (1) Tolerance range classes for shaft (recommended)

Conditions			Tolerance range class
Load characteristic	Load magnitude	Shaft diameter d mm	
Rotating inner ring load or directionally unstable load	Light load	— 50	j5
	Ordinary load	— 50	k5
		50 — 150	m5
		150 —	m6
	Heavy load and shock load	— 150	m6
		150 —	n6
Inner ring static load	Medium- and low-speed rotation, light load	All dimensions	g6
	General application		h6
	When high rotational accuracy is required		h5

Table 6.2 (2) Tolerance range classes for housing bore (recommended)

Conditions		Tolerance range class
Outer ring static load	Ordinary and heavy load	J7
	Two-split housing, ordinary load	H7
Rotating outer ring load	Light load	M7
	Ordinary load	N7
	Heavy load and shock load	P7
Directionally unstable load	Light load	J7
	Ordinary load	K7
	Heavy load and shock load	M7
When high rotational accuracy under light load is required		K6

Remarks: Light load, ordinary load and heavy load are classified per the following criteria.
Light load : $P_r \leqq 0.06C_r$
Ordinary load : $0.06C_r < P_r \leqq 0.12C_r$
Heavy load : $P_r > 0.12C_r$

6.5 Lower limit and upper limit of interference

When an intended bearing application requires an interference on the bearing, determine the appropriate interference taking into account the following considerations:

- Determine the lower limit taking into account the following factors:
  (1) Reduction in interference due to radial load
  (2) Reduction in interference due to temperature difference
  (3) Reduction in interference due to poor roughness on fit surfaces
- Recommended upper limit is 1/1000 as large as the shaft diameter or smaller.

The formulas for calculating the required interference are presented below:

(1) Radial load and required interference

When a radial load acts on a bearing, the interference between the inner ring and shaft will decrease. The interference required to maintain an effective interference can be determined by **formulas (6.1)** and **(6.2)** below:

For $F_r \leqq 0.3\, C_{0r}$,

$$\Delta d_F = 0.08\sqrt{\frac{d \cdot F_r}{B}} \ , \left(0.25\sqrt{\frac{d \cdot F_r}{B}}\right) \quad \cdots\cdots\cdots (6.1)$$

For $F_r > 0.3\, C_{0r}$,

$$\Delta d_F = 0.02\frac{F_r}{B} \ , \left(0.2\frac{F_r}{B}\right) \quad \cdots\cdots\cdots (6.2)$$

Where,
Δd_F : Required effective interference mm
d : Bearing bore diameter mm
B : Inner ring width mm
F_r : Radial load N (kgf)
C_{0r} : Basic static load rating N (kgf)

Table 6.3 Numerical fit values for radial bearing (JIS Class-0)
Table 6.3(1) Bearing fits on shaft

Unit μm

Nominal bore diameter d mm		Allowable deviation of mean bore diameter Δ_{dmp}		g6	h5	h6	j5	k5	m5	m6	n6
over	incl.	high	low	Bearing / Shaft	Bearing / Shaft	Bearing / Shaft	Bearing / Shaft	Bearing / Shaft	Bearing / Shaft	Bearing / Shaft	Bearing / Shaft
3	6	0	−8	4T〜12L	8T〜 5L	8T〜 8L	11T〜 2L	14T〜1T	17T〜 4T	20T〜 4T	24T〜 8T
6	10	0	−8	3T〜14L	8T〜 6L	8T〜 9L	12T〜 2L	15T〜1T	20T〜 6T	23T〜 6T	27T〜10T
10	18	0	−8	2T〜17L	8T〜 8L	8T〜11L	13T〜 3L	17T〜1T	23T〜 7T	26T〜 7T	31T〜12T
18	30	0	−10	3T〜20L	10T〜 9L	10T〜13L	15T〜 4L	21T〜2T	27T〜 8T	31T〜 8T	38T〜15T
30	50	0	−12	3T〜25L	12T〜11L	12T〜16L	18T〜 5L	25T〜2T	32T〜 9T	37T〜 9T	45T〜17T
50	80	0	−15	5T〜29L	15T〜13L	15T〜19L	21T〜 7L	30T〜2T	39T〜11T	45T〜11T	54T〜20T
80	120	0	−20	8T〜34L	20T〜15L	20T〜22L	26T〜 9L	38T〜3T	48T〜13T	55T〜13T	65T〜23T
120 140 160	140 160 180	0	−25	11T〜39L	25T〜18L	25T〜25L	32T〜11L	46T〜3T	58T〜15T	65T〜15T	77T〜27T
180 200 225	200 225 250	0	−30	15T〜44L	30T〜20L	30T〜29L	37T〜13L	54T〜4T	67T〜17T	76T〜17T	90T〜31T
250 280	280 315	0	−35	18T〜49L	35T〜23L	35T〜32L	42T〜16L	62T〜4T	78T〜20T	87T〜20T	101T〜34T
315 355	355 400	0	−40	22T〜54L	40T〜25L	40T〜36L	47T〜18L	69T〜4T	86T〜21T	97T〜21T	113T〜37T
400 450	450 500	0	−45	25T〜60L	45T〜27L	45T〜40L	52T〜20L	77T〜5T	95T〜23T	108T〜23T	125T〜40T

Table 6.3 (2) Bearing fits in housing hole

Unit μm

Nominal outer diameter D mm		Allowable deviation of mean outer diameter Δ_{Dmp}		H7	J7	K6	K7	M7	N7	P7
over	incl.	high	low	Housing / Bearing	Housing / Bearing	Housing / Bearing	Housing / Bearing	Housing / Bearing	Housing / Bearing	Housing / Bearing
6	10	0	−8	0〜 23L	7T〜16L	7T〜10L	10T〜13L	15T〜 8L	19T〜 4L	24T〜 1T
10	18	0	−8	0〜 26L	8T〜18L	9T〜10L	12T〜14L	18T〜 8L	23T〜 3L	29T〜 3T
18	30	0	−9	0〜 30L	9T〜21L	11T〜11L	15T〜15L	21T〜 9L	28T〜 2L	35T〜 5T
30	50	0	−11	0〜 36L	11T〜25L	13T〜14L	18T〜18L	25T〜11L	33T〜 3L	42T〜 6T
50	80	0	−13	0〜 43L	12T〜31L	15T〜17L	21T〜22L	30T〜13L	39T〜 4L	52T〜 8T
80	120	0	−15	0〜 50L	13T〜37L	18T〜19L	25T〜25L	35T〜15L	45T〜 5L	59T〜 9T
120	150	0	−18	0〜 58L	14T〜44L	21T〜22L	28T〜30L	40T〜18L	52T〜 6L	68T〜10T
150	180	0	−25	0〜 65L	14T〜51L	21T〜29L	28T〜37L	40T〜25L	52T〜13L	68T〜 3T
180	250	0	−30	0〜 76L	16T〜60L	24T〜35L	33T〜43L	46T〜30L	60T〜16L	79T〜 3T
250	315	0	−35	0〜 87L	16T〜71L	27T〜40L	36T〜51L	52T〜35L	66T〜21L	88T〜 1T
315	400	0	−40	0〜 97L	18T〜79L	29T〜47L	40T〜57L	57T〜40L	73T〜24L	98T〜 1T
400	500	0	−45	0〜108L	20T〜88L	32T〜53L	45T〜63L	63T〜45L	80T〜28L	108T〜 0

Remarks: Fit symbols “L” and “T” represent bearing clearance and interference respectively.

(2) Temperature difference and required interference

Heat is generated in a running bearing, and temperature difference occurs across the inner ring and outer ring: as a result, the interference between the inner ring and shaft will decrease. When the difference between bearing temperature and ambient temperature is taken as ΔT, the interference needed for maintaining an effective interference can be determined by **formula (6.3)**:

$$\Delta d_F = 0.0015 \cdot d \cdot \Delta T \quad \cdots\cdots (6.3)$$

Where,

Δd_T : Required effective interference for temperature difference μm

ΔT : Difference between bearing temperature and ambient temperature ˚C

d : Bearing bore diameter mm

(3) Fitting surface roughness and required interference

The fitting surface is smoothed (surface roughness is made less) by bearing fits so that the interference reduces correspondingly. The interference reduction differs depending on the fitting surface roughness and generally the following reduction values must be used.

For ground shafts : 1.0 to 2.5mm

For lathe-turned shafts : 5.0 to 7.0 mm

(4) Maximum interference

Tensile stress or compressive stress occurs on a bearing ring that has been installed to a shaft or housing with possible interference between these members. Excessively large interference can cause the bearing ring to fracture or shorten the fatigue life of the bearing. Therefore, usually **set the maximum allowable interference at 1/1000 as large as the shaft diameter or smaller; or such that the maximum circumferential tensile stress occurring on the fitting surfaces is not greater than 130 MPa** (refer to **Table 6.4**).

(5) Stress and deformation caused by interference

When bearing ring (solid) is fitted with interference, it deforms elastically and this elastic deformation results in stress.(See **Fig.6.1**) The fitting surface pressure of bearing ring, circumferential tensile stress (inner ring), compressive stress (outer ring) and radial expansion of raceway (inner ring), and shrinkage(outer ring) can be calculated from **Table 6.4**.

Table 6.4 Deformation and stress caused by bearing fit

Item	Inner ring	Outer ring
Surface pressure p MPa	$p_i = \dfrac{E}{2} \dfrac{\Delta d_{eff}}{d} \dfrac{(1-k^2)(1-k_0^2)}{1-k^2k_0^2}$	$p_e = \dfrac{E}{2} \dfrac{\Delta D_{eff}}{D} \dfrac{(1-h^2)(1-h_0^2)}{1-h^2h_0^2}$
Circumferential maximum stress σ MPa	$\sigma_i = p_i \dfrac{1+k^2}{1-k^2}$ (Tensile stress)	$\sigma_e = p_e \dfrac{2}{1-h^2}$ (Compressive stress)
Radial elastic deformation of raceway Δ	$\Delta_i = \Delta d_{eff} \cdot k \dfrac{1-k_0^2}{1-k^2k_0^2}$ (Expansion)	$\Delta_e = \Delta D_{eff} \cdot h \dfrac{1-h_0^2}{1-h^2h_0^2}$ (Shrinkage)

Where,

$$k = \frac{d}{d_i},\quad k_0 = \frac{d_0}{d},\quad h = \frac{D_e}{D},\quad h_0 = \frac{D}{D_0}$$

Remarks (Symbol representation)

d : Inner ring bore diameter (shaft diameter) mm

d_0 : Hollow shaft bore diameter (For solid shaft, $d_0 = 0$) mm

d_i : Inner ring raceway diameter mm

Δd_{eff} : Effective interference for inner ring mm

D : Outer ring outer diameter (housing hole diameter) mm

D_0 : Housing outer diameter (For sufficient housing size, $D_0 = \infty$) mm

D_e : Outer ring raceway diameter mm

ΔD_{eff} : Effective interference for outer ring mm

E : Modulus of elasticity (Young factor) 2.07×10^6 (21200) MPa (kgf/mm^2)

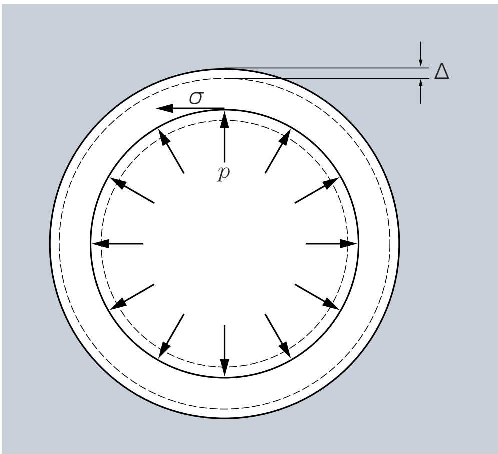


Fig.6.1

7. Limiting Speeds

At a higher bearing running speed, the bearing temperature will be higher due to frictional heat generated inside the bearing, possibly leading to failures such as seizure. As a result, the bearing will fail to continue stable operation. A maximum running speed that allows a bearing to run without developing such a problem heat buildup is known as a **limiting speed** (min^{-1}) and can vary depending on the bearing type, dimensions, cage type, load, acceleration/deceleration conditions, lubrication conditions and cooling conditions.

As a guideline, each bearing dimension table contains data about limiting bearing speeds obtained from grease lubrication and oil lubrication. However, it should be noted that these values are based on the following assumptions:

- Bearing that has been manufactured per **NTN** standard design specification and is provided with an appropriate internal clearance has been correctly mounted.
- Bearing is lubricated with a good quality lubricant, which is resupplied and replaced at correct intervals.
- Bearing is operated under ordinary loading conditions ($P \leqq 0.09\, C_r$) and at an ordinary operating temperature.

If the user is thinking of a bearing application whose running speed exceeds the limiting speed in the relevant dimension table, the user has to adopt a bearing that satisfies stricter requirements for cage specification, internal clearance, bearing accuracy, etc. and make special considerations which typically include adoption of forced circulating lubrication system.

8. Shaft and Housing Design

Even if the bearing to be used is selected correctly, it can not fulfill its specific function unless the shaft/housing on/in which it is installed is designed correctly. For needle roller bearings, special attention must be paid to shaft and housing designs, since the bearing ring thickness is thinner compared to other rolling bearing types.

8.1 Design of bearing installing portions

For needle roller and cage assemblies, attention must be paid to the axial guidance surface, such as a shaft shoulder. This guiding surface should be smooth and free from burrs. Under challenging load and/or speed conditions, a hardened and ground surface is required.

In cases where a snap ring is used as a locating shoulder (**Fig. 8.1**), a thrust ring should be used between the snap ring and bearing cage to prevent the cut section of the snap ring from contacting the cage directly. **NTN** offers WR type snap rings that are customdesigned for axial retention of needle roller & cage assemblies. (Refer to the Dimensions Table on page B-267.)

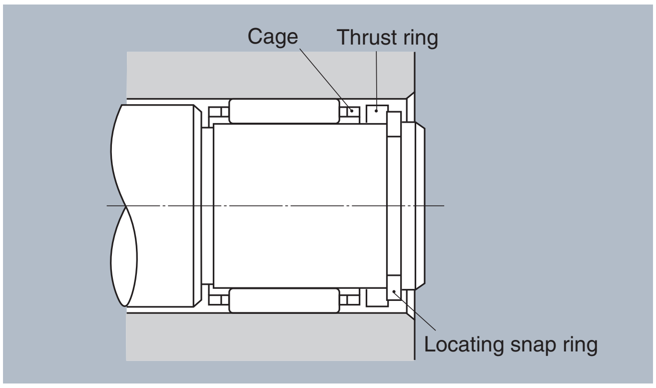


Fig. 8.1 Bearing fixing by thrust ring

Since a radial needle roller bearing can move freely on the shaft along the axial direction, a ball bearing or thrust bearing is used on the side opposite to the radial needle roller bearing in order to locate the shaft in the axial direction. With an application where the axial load is low and the running speed is not high (for example, an idle gear in gearbox), a thrust ring may be installed to a shaft as shown in **Fig. 8.2** to form a sliding bearing between the thrust ring and the housing end face in order to axially position the bearing. **Fig.8.3** illustrates an example of the above thrust ring with oil groove on its guide surface. The boundary between this oil groove and the plane area must be chamfered for deburring.

In general, for proper installation of needle roller bearing the inner ring and outer ring are both positioned in axial direction so that the bearing displaces in axial direction while running.

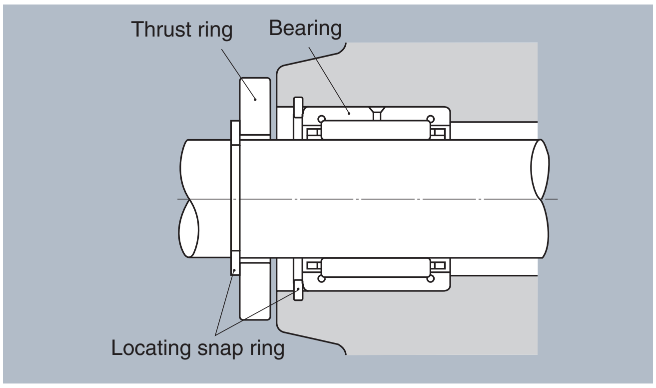


Fig. 8.2 Bearing fixing in axial direction

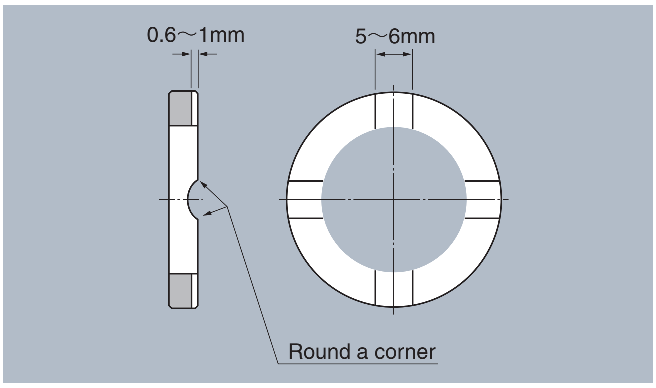


Fig. 8.3 Design of thrust ring guide surface

(1) Inner ring

For fixing inner ring correctly on a shaft, the shaft shoulder face is finished at the right angle against the shaft axial center and, in addition, the shaft corner radius must be smaller than the inner ring chamfer dimension.

To simplify inner ring extraction work, cutout grooves for engagement with jaws of an extraction jig are formed at the shoulder of the shaft as shown in **Fig. 8.4**, and the inner ring is extracted with the extraction jig according to a method illustrated in **Fig. 8.5**. Furthermore, for facilitating inner ring pull-out work the shaft shoulder is provided with a notched groove,as illustrated in **Fig. 8.5**, to accept an inner ring pull-out jig (puller).

NTN snap ring WR type for shaft use (Refer to Dimensions Table on page B-267) can be used for simply fixing inner ring in axial direction. (**Fig. 8.6**) Moreover,

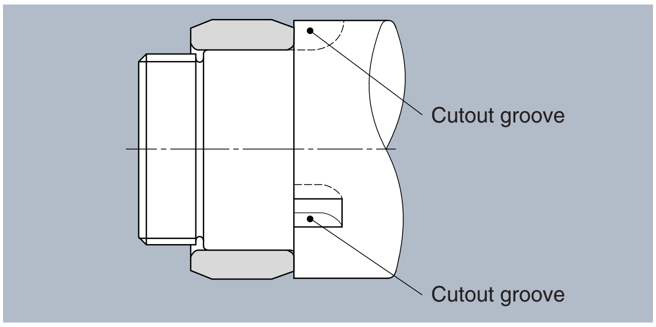


Fig. 8.4 Cutout groove for inner ring extraction

inner ring can be fixed in axial direction using an end plate or a side ring as illustrated in **Figs. 8.7** and **8.8**.

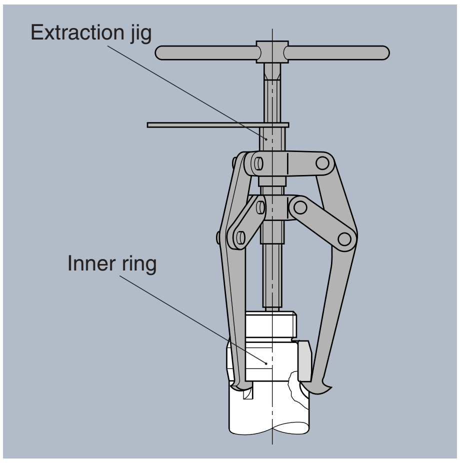


Fig. 8.5 Inner ring extraction with extraction jig

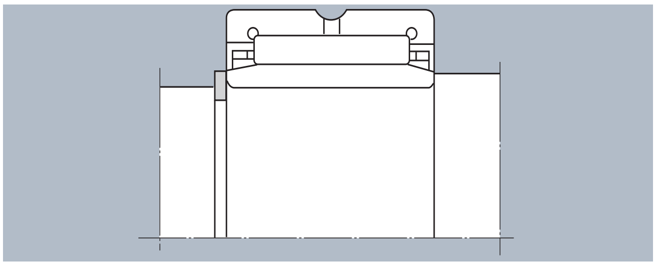


Fig. 8.6 Inner ring fixing method with snap ring

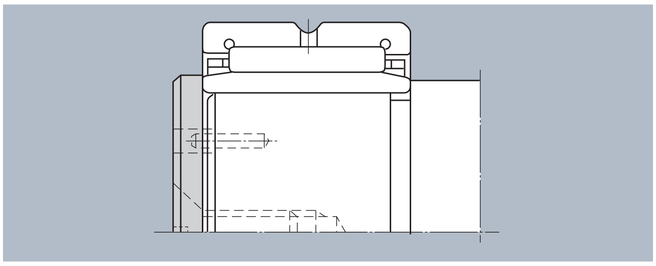


Fig. 8.7 Inner ring fixing method with end plate

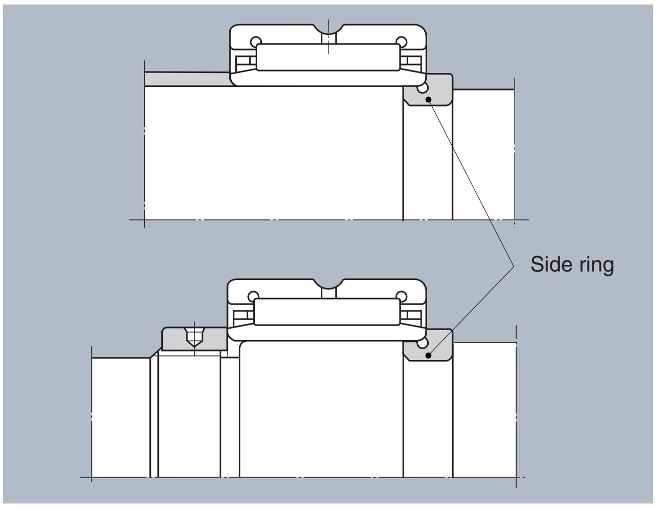


Fig.8.8 Inner ring fixing method with side ring

(2) Outer ring

Similarly to Para. 8.1(1) "Inner Ring", good care must be exercised of the shoulder profile of bearing housing for fixing outer ring in axial direction.

Figs. 8.9 and **8.10** illustrate the methods of fixing outer ring in axial direction.

For the outer ring also, the **NTN** snap ring type BR for housing (refer to the dimension table in page B-269) can be used. **NTN** BR type snap rings are designed to the dimensions adaptable to the needle roller bearings with low section height. However, commercially available snap rings conforming to JIS standard as applicable can also be used for the same bearings with adequately high section height.

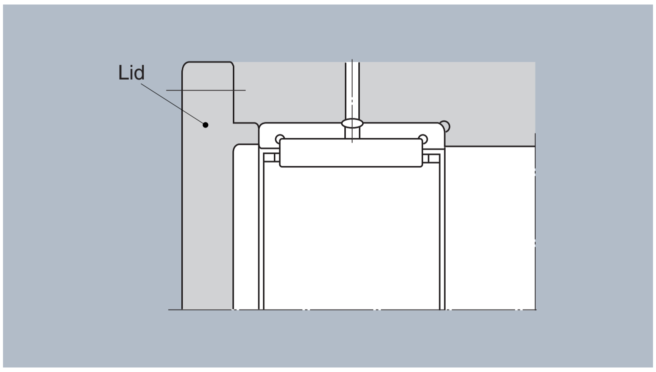


Fig. 8.9 Outer ring fixing by lid

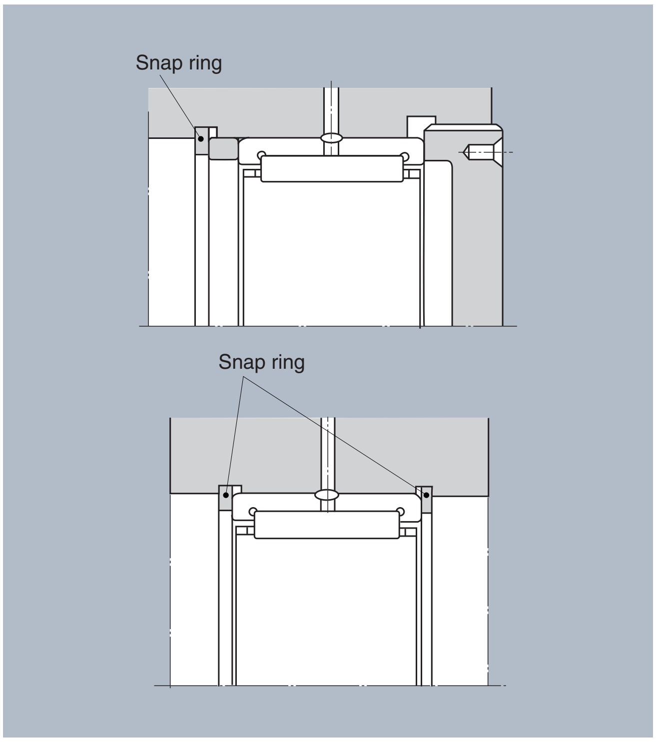


Fig. 8.10 Outer ring fixing method with snap ring

8.2 Bearing fitting dimensions

8.2.1 Shoulder height and corner roundness

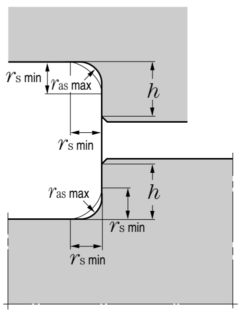
The respective shoulder heights "h", of the shaft and housing are designed to be larger than the maximum bearing chamfer dimension $r_{s\ max}$, so the bearing end face comes in contact with the flat zone. The corner roundness "r_{as}" is designed to be smaller than the minimum bearing chamfer dimension "r_s" so as not to interfere with the bearing. Generally the radius of the shaft and housing corner roundness shown in **Table 8.1** is used as the shoulder heights of the shaft and housing.

The dimensions of the shafts and housings related to bearing installation are as described in the dimensions table for each bearing type. The shoulder diameter shown in this table means the effective shoulder diameter which comes in contact with the side face of bearing excluding the chamfered portion of the shoulder.

Table 8.1 Radius of shaft /housing corner roundness and shoulder height

Unit mm

$r_{s\ min}$	$r_{as\ max}$	h (min)
0.15	0.15	0.6
0.2	0.2	0.8
0.3	0.3	1
0.6	0.6	2
1	1	2.5
1.1	1	3.25
1.5	1.5	4
2	2	4.5
2.1	2	5.5
2.5	2	5.5
3	2.5	6.5
4	3	8



8.2.2 Applications of spacer and relief grinding

There may be cases where corner roundness r_{as} max needs to be greater than the chamfering dimension on the bearing in order to mitigate stress concentration and enhance shaft strength (**Fig. 8.11a**), or where sufficiently large contact area is not available because of a low shaft shoulder (**Fig. 8.11b**). Then, a spacer can be inserted between the shaft shoulder and bearing.

Dimensions of ground-finished fit surfaces on shafts and housings are listed in **Table 8.2**.

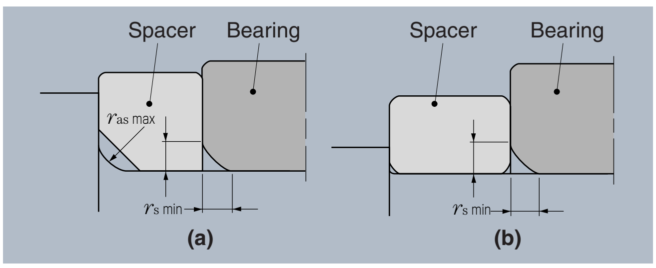
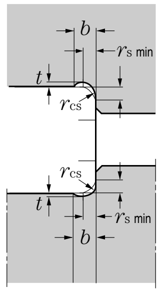


Fig. 8.11 Spacer applications

Table 8.2 Relief grinding dimension for shaft and housing corners

Unit mm

$r_{s\ min}$	b	t	r_{cs}
1	2	0.2	1.3
1.1	2.4	0.3	1.5
1.5	3.2	0.4	2
2	4	0.5	2.5
2.1	4	0.5	2.5
3	4.7	0.5	3
4	5.9	0.5	4



8.2.3 Mounting dimensions for thrust bearings

To be able to satisfy requirements for load capacity and rigidity, the surface of bearing ring on any thrust bearing needs to be sufficiently large. Therefore, the mounting dimensions in the dimension table for the intended bearing needs to be satisfied (**Fig. 8.12**).

Because of this, shoulder heights for thrust bearings have to be greater compared to radial bearings. (For the mounting dimensions of a particular thrust bearing, refer to the dimension table for that bearing.)

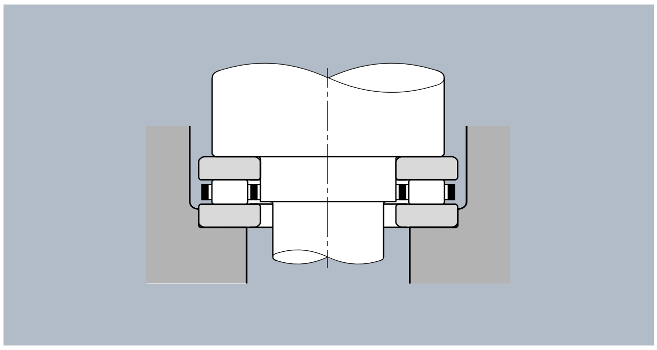


Fig. 8.12

8.3 Shaft and housing accuracy

The bearing ring of a needle roller bearings is thin-walled. Consequently, degree of physical accuracy of the raceway surface of the bearing ring is governed by physical accuracy of the fit surface of the shaft or housing to which the intended bearing is installed.

Table 8.3 summarizes recommended physical accuracy (tolerances) on the fit surfaces of the shaft and housing under ordinary bearing operating conditions: the characteristics in question are dimensional accuracy, shape accuracy, surface roughness, and runout of the shaft shoulder relative to fit surface.

For an application that adopts a double-split housing, the bore side of each housing half may be relieved: consequently, when the housing halves are joined together and the mating surfaces are forced together, resultant deformation on the outer ring is minimal.

Table 8.3 Shaft and housing accuracy (recommended)

Characteristic item	Shaft	Housing
Dimensional accuracy	IT6（IT5）	IT7（IT6）
Roundness cylindricality (max)	IT3	IT4
Shoulder perpendicularity (max)	IT3	IT3
Fitting surface roughness	0.8a	1.6a

Remarks: The parenthesized values are applied to the bearings of accuracy class 5 and higher.

8.4 Raceway surface accuracy

For needle roller bearings, the shaft and housing are used as the raceway surface on applications. The raceway dimensional accuracy, profile accuracy and surface roughness of the shaft/housing must be equivalent to the raceway accuracy of the bearing itself. **Table 8.4** shows the specified surface accuracy and surface roughness of the shaft/housing raceway.

Table 8.4 Raceway surface accuracy (recommendation)

Characteristic item	Shaft	Housing
Dimensional accuracy	IT5（IT4）	IT6（IT5）
Roundness cylindricality (max)	IT3（IT2）	IT4（IT3）
Shoulder perpendicularity (max)	IT3	IT3
Surface roughness	For shaft diameter of ϕ 80 and less :0.2a For shaft diameter of over ϕ 80 to 120 :0.3a For shaft diameter of over ϕ 120 :0.4a	

Note) The parenthesized values are applied where high rotational accuracy is required.

8.5 Material and hardness of raceway surface

When the outer surface or bore surface of the shaft(hollowed) or housing is used as raceway, it must be hardened to HRC58 to 64 in order to obtain sufficient load capacity. For that, the materials shown in **Table 8.5** are used after heat-treated properly.

Table 8.5 Materials used for raceway

Kinds of steel	Representative example	Related standards
High carbon chrome bearing steel	SUJ2	JIS G 4805
Carbon tool steel	SK85 (previously: SK5)	JIS G 4401
Nickel chrome molybdenum steel	SNCM420	JIS G 4053 (previously: JIS G 4103)
Chrome steel	SCr420	JIS G 4053 (previously: JIS G 4104)
Chrome molybdenum steel	SCM420	JIS G 4053 (previously: JIS G 4105)
Stainless steel	SUSU440C	JIS G 4303

When steel is surface-hardened by carburizing or carbonitriding, JIS Standard defines the depth from surface up to HV550 as an effective hardened layer. The minimum value of effective hardened layer depth is approximately expressed in **formula (8.1)**.

$$E_{ht\ \min} \geqq 0.8 D_w (0.1 + 0.002 D_w) \quad \cdots\cdots\cdots \textbf{(8.1)}$$

Where,

$E_{ht\ \min}$: Minimum effective hardened layer depth mm

D_w : Roller diameter mm

8.6 Allowable bearing inclination

The inner ring and outer ring of the bearing incline a little eventually against one another depending on shaft deflection, shaft /housing machining accuracy, fitting deviation, etc. Although this allowable inclination differs depending on bearing type, bearing load, internal clearance, etc., the inclination degree shown in **Table 8.6** must be used as a guideline in the case of general applications because even minor inclination of the inner and outer ring could cause a reduction of bearing life and damage the cage.

Table 8.6

Bearing type	Allowable inclination
Radial needle roller bearing	1/2 000
Thrust bearing	1/10 000

9. Lubrication

9.1 Purpose of lubrication

When a bearing is lubricated, its rolling and sliding surfaces are covered with a thin oil film that prevents the occurrence of metal-to-metal contact. Lubrication of rolling bearings offers the following benefits:

(1) Reduction of friction and wear
(2) Discharge of friction heat
(3) Further extension of bearing life
(4) Rust prevention
(5) Prevention of foreign matter invasion

To fully realize these benefits developed, the bearing user has to adopt a lubrication system that best suits the projected bearing operating conditions, select quality lubricant, and adopt a relevant sealing design that helps regulate the amount of lubricant retained, prevent the ingress of foreign materials and leakage of the lubricant.

9.2 Lubrication systems and characteristics

In general, bearing lubrication systems usually available as grease lubrication and oil lubrication, each featuring unique advantages and disadvantages. The user needs to select an appropriate lubrication system that best suits the user's bearing performance requirements.

Table 9.1 summarizes the different features of grease and oil lubrication.

Table 9.1 Characteristic comparison of grease and oil lubrication

Lubrication method / Comparative items	Grease lubrication	Oil lubrication
Handling	◎	△
Reliability	○	◎
Cooling effect	×	○ (Recirculation needed)
Seal structure	○	△
Power loss	○	○
Environmental pollution	○	△
High speed operation of bearing	×	○

◎ : Extraordinarily advantageous ○ : Advantageous
△ : Fairly advantageous × : Disadvantageous

9.3 Grease lubrication

Grease lubrication is the simplest lubrication method. This method enables a simplified design of the seal structure, and is broadly used.

Important points for this lubrication method are to select an optimum grease and to fill it securely in the bearing. Particularly where the cage is guided by the inner ring or outer ring of bearing, care must be exercised so the guide surface is fully greased throughout its entire area.

If requiring refilling of grease, the bearing should be provided with grease sectors as a refilling means and a grease valve or an equivalent as a means of discharge.

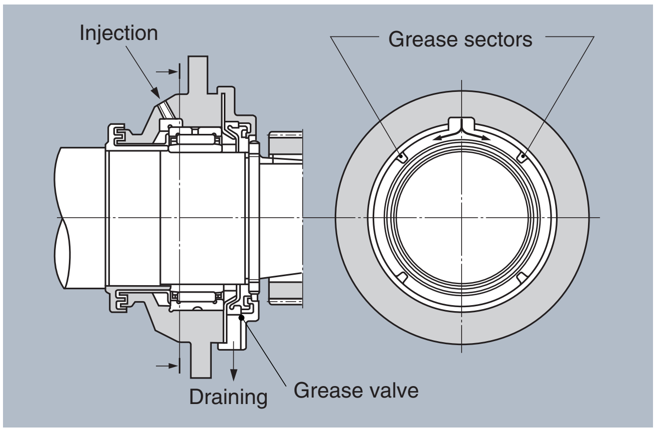


Fig. 9.1 An example of bearing unit with grease sector and grease valve

Fig. 9.1 illustrates an example of bearing unit with grease sector and a grease valve. An amount of grease injected through a port, such as a grease nipple, is blocked by the grease sectors, then fills the space and the excess fluid flows into the bearing. Grease is circulated through the interior of bearing, and excess amount of grease pushed out of the bearing is allowed to drain through the grease valve.

9.3.1 About grease

Grease lubrication is **composed of a lubrication base oil (ex. mineral oil base or a synthetic oil base) held with a thickener, and various additives added thereto.** The properties of grease are determined by the kinds and combination of base oil, thickener, and additives.

Commons grease types and their characteristics are summarized in **Table 9.2**. Characteristics of greases of a similar type can vary greatly depending on the brands. Therefore, **to be able to select an optimal grease brand, it is necessary to check grease characteristic data, available from grease manufacturers.**

(1) Base oil

Base oils used in grease are **mineral oil**, or synthetic oils such as **ester oil** and **ether oil**.

Lubricating performance of a given lubricant is mainly governed by lubricating performance of the base oil. Generally, greases comprising a low-viscosity base oil excel in low-temperature characteristics and high-speed performance, while greases with a high-viscosity base oil boast excellent high-temperature, high-load characteristics.

(2) Thickener

Thickeners are blended and diffused in base oil to hold grease in a semi-solid form. Commonly used thickeners include: metal soaps derived from **lithium**, **sodium** and **calcium**; non-metal soap thickeners made from inorganic materials such as **silica gel** and **bentonite**, and organic materials such as **urea** and **fluoro carbon**. The grease characteristics such as **critical operating temperature, mechanical stability, durability,** etc. are mainly

Table 9.2 Grease varieties and characteristics

Grease name	Lithium grease			Sodium grease (Fiber grease)	Calcium compound base grease	Aluminum grease	Non-soap grease	
Thickener	Li soap			Na soap	Ca+Na soap Ca+Li soap	Al soap	Bentone, silica gel, urea, carbon black, etc.	
Base oil	Mineral oil	Diester oil	Silicone oil	Mineral oil	Mineral oil	Mineral oil	Mineral oil	Synthetic oil
Dropping point ˚C	170～190	170～190	200～250	150～180	150～180	70～90	250 or more	250 or more
Operating temperature range ˚C	−30～+130	−50～+130	−50～+160	−20～+130	−20～+120	−10～+80	−10～+130	−50～+200
Mechanical stability	Excellent	Good	Good	Excellent to good	Excellent to good	Good to poor	Good	Good
Pressure resistance	Good	Good	Poor	Good	Excellent to good	Good	Good	Good
Water resistance	Good	Good	Good	Good to poor	Good to poor	Good	Good	Good
Applications	Broadest application. Grease for universal type rolling bearings.	Excellent in low temperature characteristic and anti-friction characteristic.	suited to high temperature and low temperature. Low oil film strength and unsuitable for high load application.	emulsified by inclusion of water content. Comparatively excellent in high temperature characteristic.	Excellent in water resistance and mechanical stability. Suitable for bearing subjected to shock load.	Excellent in viscosity characteristic. Suitable for bearing subjected to vibration.	Available for use in wide temperature range from low to high temperature. Some of non-soap base greases are excellent in heat resistance, cold resistance, chemical resistance, etc. subject to proper combination of base oil and thickener. Grease for universal type rolling bearings.	

Remarks: The operating temperature range in this table is the general characteristic value, not the guaranteed value.

determined by the kind of thickener used. Generally, water resistance of sodium soap grease is poor. Non-soap thickeners made from bentone and urea feature excellent high-temperature characteristics.

(3) Additives

Any greases contain various additives to improve the performance. For example: **oxidation inhibitors, extreme pressure additives** (EP additives), **rust inhibitors, corrosion inhibitors,** etc.

A grease containing extreme pressure additives is used for bearings subjected to high load or shock load. A grease containing oxidation stabilizer is used for bearing applications wherein the operating temperature is comparatively high and the grease is not replenished for a long time.

(4) Consistency

"Consistency" is an index showing the hardness or fluidity of grease. **The higher the numerical value, the harder the consistency**. Lubricants commonly used for lubrication of rolling bearings are those having NLGI consistency number 1, 2 or 3.

Table 9.3 shows the general relationship of grease consistency to application.

(5) Grease mixing

Mixing dissimilar greases will alter the characteristics of grease: for example, consistency will vary (usually, the grease mixture will be softer compared with original greases) and the permissible operating temperature will be lower. **Therefore, in principle, do not mix greases other than mixing of portions of same grease brand.**

Where mixing of different greases is inevitable, greases composed of the same thickener and similar base oil must be

Table 9.3 Grease consistency

NLGI consistency No.		JIS (ASTM) 60-cycle mixed grease consistency	Application
0	Soft	355–385	For centralized greasing
1	↕	310–340	For centralized greasing
2		265–295	For general application, for tight-sealed bearing
3		220–250	For general application, for high temperature
4	Hard	175–205	Special application

selected. Even when greases of the same kind are mixed together, the properties of the mixed grease could vary depending on difference in additives, etc. It is therefore necessary to check the property variation in advance.

9.3.2 Grease fill amount

Grease fill amount differs depending on housing design, available volume, rotational speed, kind of grease, etc.

As a guideline, approximately 50% to 80% of the static volume within a bearing and housing is filled with grease. When intending higher running speed, or wanting to limit temperature rise, fill grease sparingly. **Too much grease fill would cause the grease temperature to rise, which would then lead to reduction of the specific lubrication performance due to leak of the softened grease, or quality change such as oxidation, etc.**

9.3.3 Grease replenishment

A bearings grease must be replenished at proper intervals because its lubrication performance deteriorates with running time. This replenishing interval differs depending on bearing type, dimension, rotational speed, bearing temperature, kind of grease used, etc.

Fig.9.2 gives the replenishing interval chart as a guideline. This chart is subject to use of a grease for ordinary rolling bearings under usual operating conditions.

Needless to say, the grease replenishing interval must be shortened as the bearing temperature gets higher. As an approximate guideline, when the bearing temperature is 80˚C or more, the replenishing interval shall be shortened by 1/1.5 whenever the bearing temperature rises by 10˚C.

[Ex.] Let us determine grease replenishment intervals for NA4910R that is run at a speed of n = 1600 min^{-1}. From the dimension table for **NA4910R**, the shaft diameter (bearing bore diameter) d = 50 mm, limiting speed no = 4700 mn^{-1}:

Accordingly,

$$\frac{n_o}{n} = \frac{4700}{1600} \fallingdotseq 2.9$$

Plot a line horizontally from d =50 point in **Fig. 9.2** and deem the intersection point with the vertical line Ⅰ as A. Thereafter, connect n_o/n=3 point B on the vertical line Ⅱ and said A point together, with a straight line, and determine the intersection point C with the vertical line Ⅲ.

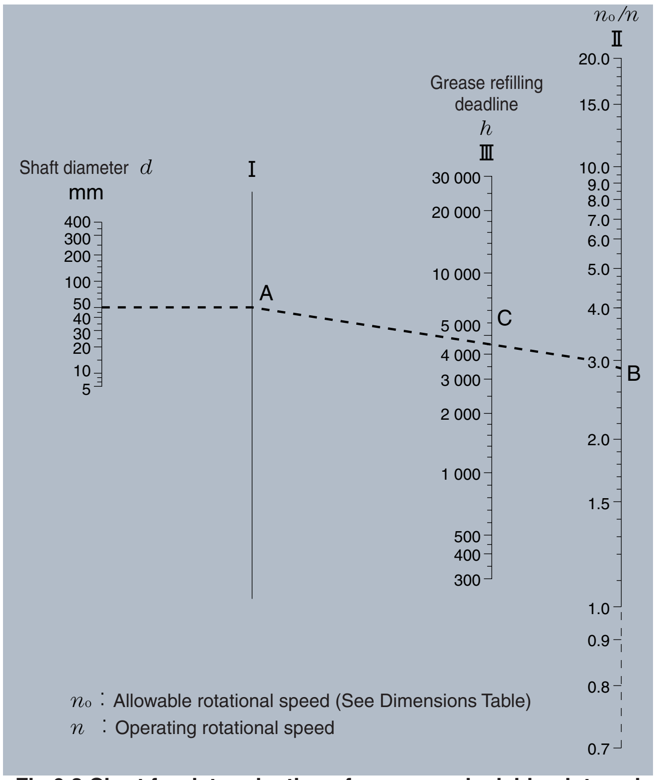


Fig.9.2 Chart for determination of grease replenishing interval

The grease replenishing interval of approximately 4600 hours can be read from the intersection point C.

9.3.4 Solid grease
(lubricant for special “Polylube” bearings)

This unique solid grease consists of lubricating grease and ultra heavy molecular weight polyethylene (UHMW-PE) as main components. For more detailed information, refer to page A-54 in this document or the special catalog (Japanese only) “Polylube Needle Bearings” (**NTN** CAT. NO. 3605).

9.4 Oil lubrication

In general, oil lubrication is more suitable for high speed or high temperature applications than grease lubrication. Oil lubrication is suitable for the cases where heat generated in a bearing or heat transferred to a bearing must be discharged outside the bearing.

9.4.1 Lubrication method

(1) Oil bath lubrication

Oil bath lubrication is the most common lubrication scheme among various oil lubrication systems. It is used for low-speed and medium-speed bearing applications. An important point in this method is control of oil level in an oil bath.

For that, when bearings are installed on a horizontal shaft, it is common that a point close to the center of the rolling element in the lowest position should be deemed as the oil level to be secured during shutdown. In this case, the housing must be designed with such a profile as to minimize variation in oil level therein. Furthermore, it is desirable to provide the housing with an oil gauge to facilitate level check during running as well as shutdown.

When bearings are installed on a vertical shaft, it is okay if 50 to 80% of the rolling elements are dipped in an oil bath under low speed running, but in the cases of high speed running and bearings used in multiple rows it is desirable to adopt the drip lubrication and circulating lubrication methods, or others described hereunder.

(2) Spray lubrication

This method sprays lubrication oil by an impeller of simple structure, which is mounted on the shaft, without directly dipping a bearing in an oil batch. This can be applied to bearings running at considerably high speed.

(3) Drip lubrication

This lubrication method is used where bearing runs at comparatively high speed with medium and less loads act thereon. In this method, oil drips from an oiler on the top of a bearing unit, striking the rolling elements for atomizing lubrication (**Fig. 9.3**) and a small amount of oil passes through the bearing. In many cases the bearing is lubricated with several drips per minute though the number of oil drips per specific unit differs depending on bearing type and dimension.

(4) Circulating lubrication

This circulating lubrication method is adopted to cool down bearings or to lubricate by a centralized lubrication system. As added features with this method the oil feed line is equipped with a cooler to cool down the lubrication oil and an oil filter to purify the lubrication oil.

Under this circulating lubrication system, the lubrication oil must be discharged from each bearing after having passed through it. For that, it is important to provide an oil inlet and an oil outlet on each bearing in opposite position and to make the oil discharge port size as large as possible or otherwise to discharge the oil compulsorily. (**Fig.9.4**)

(5) Others

Jet lubrication, oil mist lubrication, air-operated oil lubrication, etc. are available as other lubrication methods.

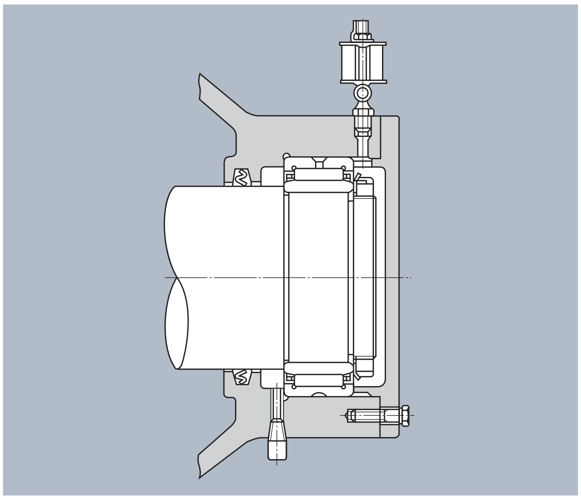


Fig. 9.3 Drip lubrication

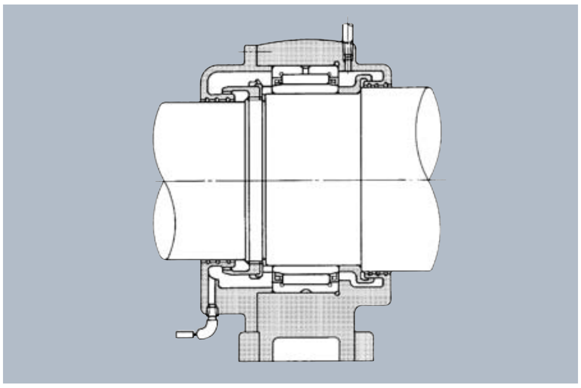


Fig.9.4 Circulating lubrication

9.4.2 Lubrication oil

To lubricate rolling bearings, mineral oil lubricants are often used, the examples of which include **spindle oil, machine oil** and **turbine oil**. When a rolling bearing is used in a demanding operating environment where the ambient temperature can be **not lower than 150˚C** or **not higher than -30˚C**, a rolling bearing should be lubricated with **synthetic oils** such as **diester oil, silicone oil** and **fluoro carbon oil.**

With lubrication oil, its viscosity is one of the important characteristics that determine the lubrication performance. Too low viscosity of lubrication oil causes imperfect forming of an oil film reading to damage of bearing surface, while too high viscosity of lubrication oil causes great viscosity resistance, which then leads to temperature rise and increase of friction loss.

Generally lubrication oil of lower viscosity is used for the faster rotational speed of bearing, while lubrication oil of higher viscosity is used for the heavier bearing loads.

A lubricant for a rolling bearing has to satisfy viscosity listed in **Table 9.4** at the operating temperature of that rolling bearing. **Fig. 9.5** shows the lubrication oil viscosity - temperature characteristic chart, which should be referred to when selecting a lubrication oil of optimal viscosity under actual operating temperature.

Furthermore, **Table 9.5** shows the criterion for selection of the lubrication oil viscosity according to the actual bearing operating conditions.

Table 9.4 Oil viscosity required for each bearing type

Bearing type	Required viscosity mm²/s
Radial needle roller bearing	13
Thrust needle roller bearing	20

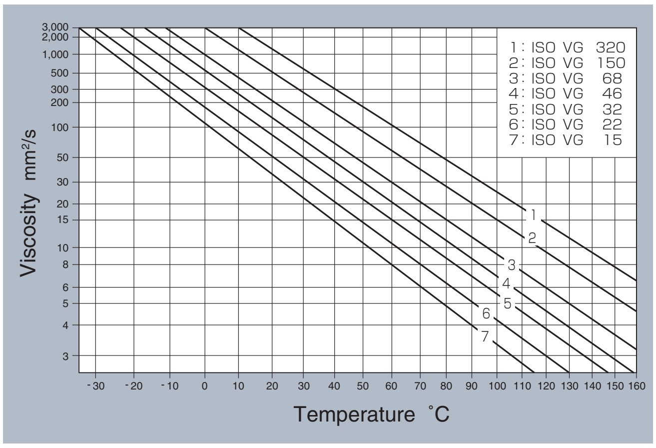


Fig. 9.5 Lubrication oil viscosity - temperature characteristic chart

Table 9.5 Criteria for selection of lubrication oil (Reference)

Bearing operating temperature ˚C	dn value	ISO viscosity grades for lubrication oil (VG)	
		Ordinary load	Heavy load or shock load
−30～0	up to allowable rotational speed	22 32	46
0～60	up to 15000	46 68	100
	15 000～80 000	32 46	68
	80 000～150 000	22 32	32
60～100	up to 15000	150	220
	15 000～80 000	100	150
	80 000～150 000	68	100 150
100～150	up to allowable rotational speed	320	

Remarks:
1. Subject to oil bath lubrication or circulating lubrication.
2. Apply to **NTN** for other operating conditions other than those specified in this Table.

9.4.3 Oil supply rate

When lubricating oil is force-fed into a bearing, the amount of heat generated in the bearing is equal to a sum of amount of heat diffused from the housing and amount of heat removed by lubricating oil.

A standard oil supply rate to be used as a guideline when using an ordinary housing can be determined by **formula (9.1)**. The amount of heat diffused can vary depending on the shape of housing. Therefore, for bearing operation on an actual machine, begin with an oil supply rate approximately 1.5 to 2 times as much as the value determined by **formula (9.1)**, and determine an optimal supply rate through a series of adjustment efforts. It may be convenient to perform calculations with an assumption that there is no heat radiation from the housing and all the heat generated is removed with the lubricating oil. In such a case, take the shaft diameter d = 0 and then determine the oil supply rate q.

$$Q = K \cdot q \quad \text{..........} \quad \textbf{(9.2)}$$

Where,

Q: Oil supply rate per bearing assembly cm³/min

K: Coefficient governed by temperature rise with lubricating oil in operating mode (**Table 9.6**)

q : Oil supply rate determined from the chart cm³/min (**Fig. 9.6**)

9.4.4 Guideline for lubricating oil change

How often the lubricating oil needs to be changed varies depending on the factors including bearing operating conditions, amount of oil in the lubrication system, and lubricating oil type. As a guideline, perform oil change for an oil bath lubrication system approximately once a year if the oil temperature in the bath is regulated at 50˚C or lower, or at least every three months if the oil temperature in the bath reaches a range from 80 to 100˚C.

For a critical machine involving needle roller bearings, the user is recommended to monitor current lubrication performance of the lubricating oil and deterioration in oil cleanness at regular intervals to establish the user's unique oil change schedule.

Table 9.6 K value

Temperature of discharged oil – Temperature of supply oil ˚C	K
10	1.5
15	1
20	0.75
25	0.6

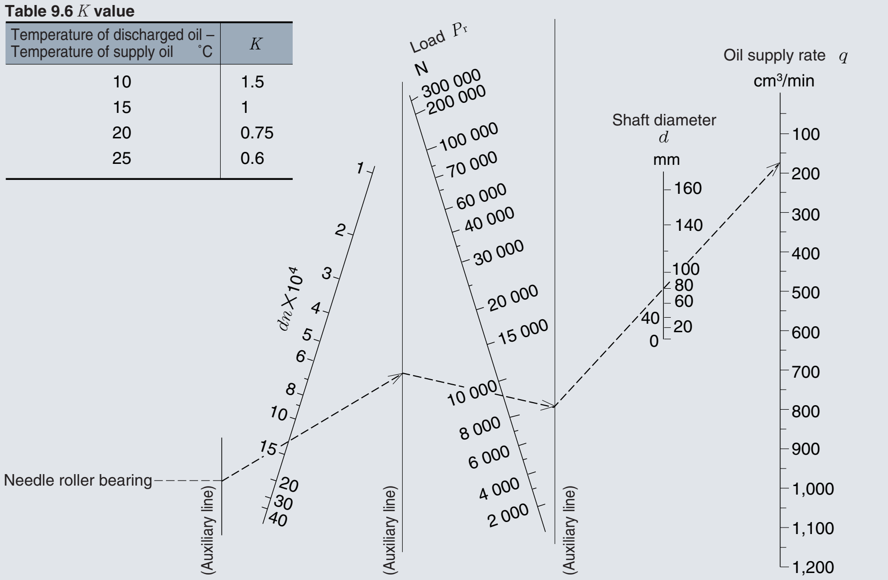


Fig. 9.6 Chart for determining oil supply rate

10. Sealing Devices

10.1 Non-contact seal and contact seal

The purpose of using a seal is to prevent a lubricant held in a bearing from leaking outside the bearing and to prevent powder, water content, etc. from invading into the bearing from outside.

It is very important to design a sealing device with full consideration of the operating conditions, lubricating condition, environmental condition, economical merit, etc., so that the bearing is not adversely affected by the sealing device during operation.

The bearing seals are mainly classified into non-contact seal, contact seal types. as shown in **Tables 10.1** and **10.2**, which must then be selected correctly according to each application,under full consideration of the characteristics of each sealing type.

Table 10.1 Seals (Non-contact seals)

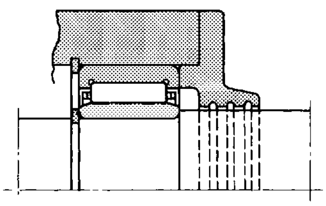
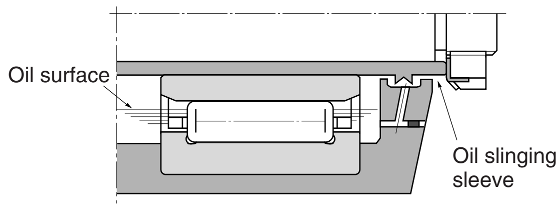
<table>
<tr><th></th><th colspan="3">Non-contact seals</th></tr>
<tr><td>Seal name</td><td>Oil groove seal
Labyrinth seal (axial, radial)</td><td>Slinger seal</td><td>Air seal</td></tr>
<tr><td>Features</td><td>**<Oil groove seal>**
This seal is fitted at either one side of a housing or a shaft, or fitted at the both sides for sealing In this case, this seal has an effect in preventing invasion of foreign matter from outside by retaining grease in the oil grooves.
< Labyrinth seal >
This seal having a high sealing effect due to its multiple labyrinths and long passage is mainly used for grease lubrication. Generally it is suited to a high speed bearing, but it has a dust-proofing effect even under low speed running if the seal grooves are filled up with grease. It is convenient if this seal is provided with a grease nipple.</td><td>In oil lubrication, this seal has an effect in slinging and returning the oil thrown out along its sleeve by centrifugal force if its sleeve is provided with projections.
A seal example illustrated in **Fig. 10.6** prevents invasion of foreign matter from outside.</td><td>—</td></tr>
<tr><td rowspan="3">Application examples</td><td>
Fig. 10.1 Oil groove seal</td><td colspan="2">

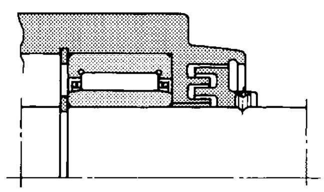
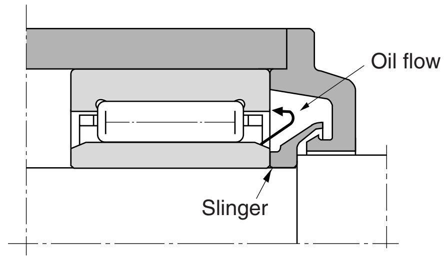
Fig. 10.4 Slinger with projections</td></tr>
<tr><td>
Fig. 10.2 Axial labyrinth seal</td><td colspan="2">

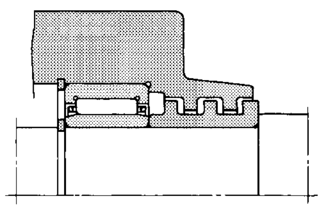
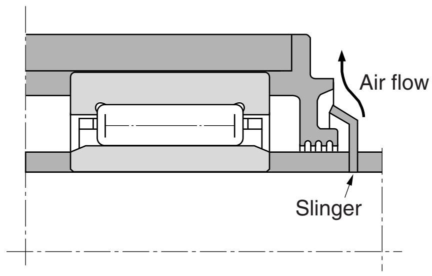
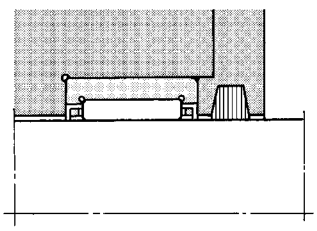
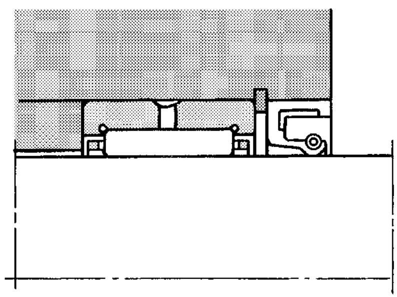
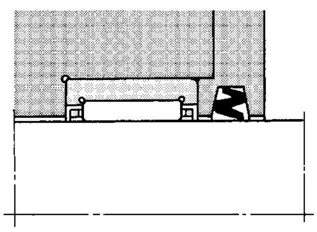
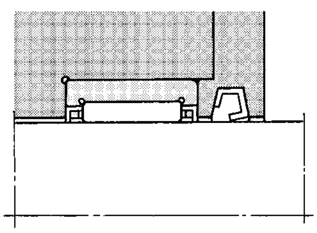
Fig. 10.5 Slinger intended for back flow of flown-out oil by centrifuge</td></tr>
<tr><td>
Fig. 10.3 Radial labyrinth seal</td><td colspan="2">

Fig. 10.6 Slinger provided at outer side</td></tr>
</table>

Table 10.2 Seals (Contact seals)

	Contact seals	
Seal name	Seal ring (felt seal, etc.) O-ring, piston ring	Oil seal, V-shaped ring seal, mechanical seal
Features	**<O-ring seal>** This seal type seals a fluid by pressing its elastic body onto the sliding surface with a constant contact pressure. Generally the contact seals are better in sealing performance than the non-contact seals, but the friction torque and temperature rise are greater than those of the non-contact seals. **<Felt seal>** This is the simplest of the contact seals, which is mainly used for grease lubrication and suited to prevention of fine dust, but oil penetration and purging are occasionally unavoidable to some extent.	**< Oil seal >** This seal type intended to seal lubricant at the sliding portion between its lip and a shaft. The oil seal ia an effective seal and is the most frequently used. The lip must be oriented outward to prevent invasion of water content and foreign matter from outside and oriented inward to prevent lubricant from leaking out of the housing. Furthermore, another seal type with two or more lips is also available for preventing lubricant purge and dust contamination.
Application examples	 **Fig. 10. 7 Felt seal**	 **Fig. 10.10 Oil seal**
	 **Fig.10. 8 Z type grease seal**	
	 **Fig. 10. 9 GS type grease seal**	

10.2 Combined seals

Several seal types are used in combination for an application in an environment where dust, water components, etc. exist as well as for mechanical portions which cannot to be contaminated by lubricant leak.

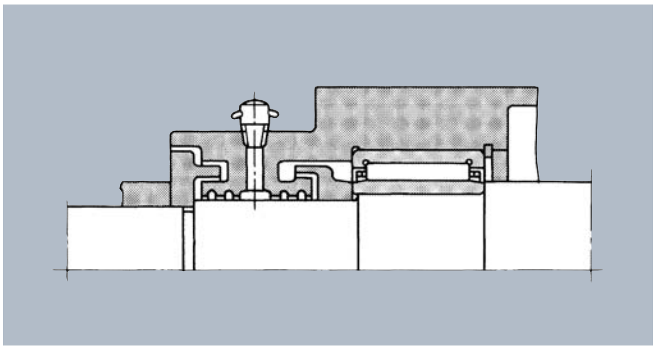


Fig. 10.11 Combined non-contact seal
Combination of labyrinth seal and oil groove seal

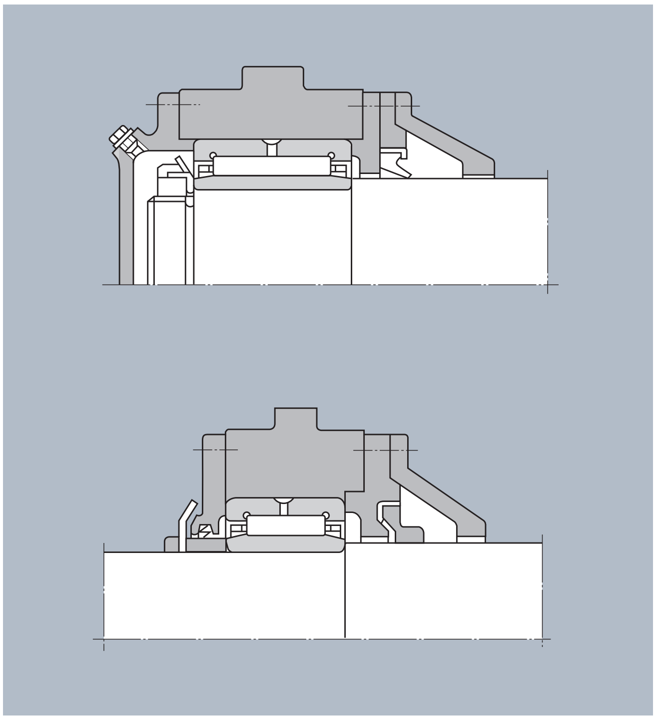


Fig. 10.12 Combined seal
Combination of contact seal and non-contact seal

10.3 Clearance setting

Oil groove seals and labyrinth seals have better sealing effects as the shaft - housing clearance gets smaller, but the actual clearance is generally selected from the following clearance values, under consideration of machining and assembling conditions, shaft deformation, etc.

Table 10.3 Clearances (Optional)

Seal type	Shaft diameter	Radial clearance	Axial clearance
Oil groove seal	50 or less	0.2～0.4	———
	Over 50 to 200	0.5～1.0	
Labyrinth seal	50 or less	0.2～0.4	1.0～2.0
	Over 50 to 200	0.5～1.0	3.0～5.0

10.4 NTN seals

Special-purposed **NTN** seals are available for needle roller bearings.(Refer to **Table 10.4** on page A-49.) For the more detailed information refer to the "Dimensions Table" on page B-273.

10.5 Seal materials and corresponding operating temperature ranges

The oil seal lip is ordinarily made of nitrile rubber, but acrylic rubber, silicone rubber and fluoro-rubber are used as the lip material depending on operating temperature, sealing objective, etc. **Table 10.5** shows the allowable operating temperature ranges available for the respective materials.

Table 10.5 Seal materials and corresponding operating temperature ranges (Reference)

Seal materials	Operating temperature ranges ˚C
Nitrile rubber	−25～+100
Acrylic rubber	−15～+130
Silicone rubber	−70～+150
Fluoro-rubber	−30～+180

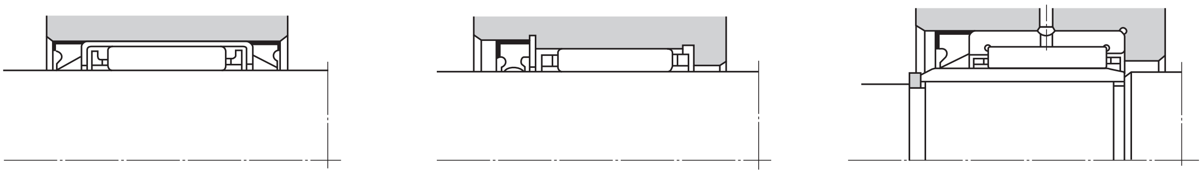
10.6 Seal types and allowable speed

The allowable speed for the contact seal type depends on the surface roughness, accuracy and lubrication properties of sliding surface, operating temperature, etc. **Table 10.6** shows the allowable speed for each seal type, as a guideline.

Table 10.6 Seal types and corresponding allowable speed (Reference)

Seal types	Allowable speed m/s
Oil seal (nitrile rubber)	16 or less
Oil seal (acrylic rubber)	26 or less
Oil seal (fluoro-rubber)	32 or less
Z-grease seal (nitrile rubber)	6 or less
V-ring seal (nitrile rubber)	40 or less

Table 10.4 Seals (NTN contact seals)

Seal type	**Contact seals (G type, GD type)**
	Seal using mainly direct contact
Features	This seal type is a special-purposed seal for needle roller bearings which was designed for smaller section height. This is a synthetic rubber contact seal reinforced with steel plate, for use in the operating temperature range of -25 to +120˚C and, under continuous running condition, used at 100˚C or less. For applications under special operating conditions of greater than 120˚C, please contact contact **NTN** engineering.
Application examples	 **Fig. 10.13 Bearing sealing by NTN seals (Example)**

10.7 Shaft surface roughness

Sealing performance and seal life depend on the surface roughness, accuracy and hardness of shaft sliding surface with which the seal lip comes in contact. **Table 10.7** shows the surface roughness as a guideline. For improved wear resistance of shaft surface it is desirable to maintain shaft surface hardness at least at HRC40 (HRC55 if possible) by heat treatment or hard chrome plating.

Table 10.7 Shaft surface hardness (Reference)

Speed m/s		Surface roughness
over	incl.	Ra
	5	0.8a
5	10	0.4a
10		0.2a

11. Bearing Handling

Bearings are precision parts. In order to preserve their accuracy and reliability, care must be exercised in their handling. In particular, bearing cleanliness must be maintained, sharp impacts avoided and rust prevented.

[1] Keep the bearing and other related parts clean
Foreign matters such as dust, moisture, etc. causes harmful effects on the life of the bearing. To avoid such harmful effects, bearings must be kept clean. In addition, tools, lubricants, washing oils, work environments, etc. must always be maintained in clean condition.

[2] Careful handling
Any shock to a bearing in handling could result in creating surface flaws and indentations of its raceway surface and rolling elements. In severe cases, cracking and chipping can occur. To avoid such defects, bearings must be handled with care.

[3] Use proper handling tools
Inappropriate tools should be avoided when installing and removing bearings.
Specific tools suited to the individual bearing types must be used. Special-purpose handling tools must be used particularly when installing a drawn cup needle roller bearing.

[4] Protect bearing from rusting
As a general rule, rust preventive oil is coated on all bearings. Direct handling of bearings should be avoided since the natural oil on hands can cause rusting of the bearings. To protect bearings from this type of rust, use a pair of gloves or coat mineral oil on the hands if directly handling the bearings with hands.

11.1 Bearing storage

Store bearings at room temperature with a relative humidity of 60% or less.

11.2 Washing

Never rotate a bearing with foreign matter within the raceway. This could result in damage to the raceway surfaces or rolling elements.

Therefore, any dismounted bearing is usually washed by light oil, kerosene or any other mild solvent to completely remove foreign matter.

In this case, two washing containers must be used: one for rough washing and another for finish washing.

Rough washing is done for removal of oil and foreign matter from bearings, while finish washing is done for fine washing of the roughly-washed bearing.

Further, any containers used for washing must be provided with a steel net in the center above the bottom of the container, as illustrated in **Fig. 11.1**, to prevent the bearing from coming in direct contact with the bottom of the container.

Furthermore, rust preventive treatment must be applied to the bearing immediately after washing, to protect it from corrosion.

Do not rinse grease-prefilled bearings (shielded bearings, sealed bearings, one-way clutches, etc.). Otherwise, prefilled grease can wash away or deteriorate.

In addition, follow all applicable legal requirements such as environmental preservation, industrial labor safety laws, etc. and use the washing instructions provided by the detergent manufacturer and washing tank manufacturer.

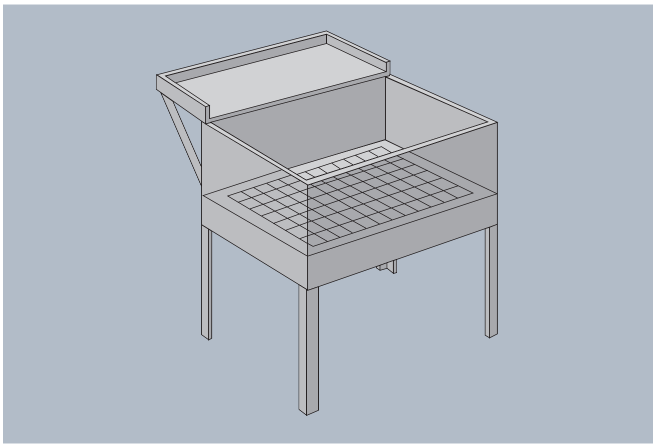


Fig. 11.1 Washing tank

11.3 Installation

Depending on bearing type and fitting conditions, the methods described below are used as a general method of installation. However, for installation of drawn cup needle roller bearings refer to Commentary given in the Dimensions Table.

(1) Preparations prior to installing
For the installation of bearings, it is desirable to prepare a clean and dry work place.
Contaminants, burrs, chips, etc. must be removed completely from all the parts related to a shaft and a housing before installing. Keep bearing in original packaging until ready for installation.
If the bearing is used in a grease-lubricated machine, it may be installed without removing the rust preventive oil coat on it. However, remove the rust preventive coat if the bearing is to be used with oil lubrication, or grease lubrication. Lubrication performance of the grease is jeopardized when mixed with the rust preventive agent. Use uncontaminated cleaning oil to remove the rust preventive agent coat and then allow the cleaning oil to dry or thoroughly wipe it away. Only then, install the bearing.
Do no to wash shield type and seal type bearings and one-way clutches.

(2) Interference-fit with a mechanical or hydraulic press

In general, the press-fit method using a press machine is used for the installation of bearings. The bearing ring (inner ring or outer ring) is press-fitted slowly via a backing strip as illustrated in **Fig. 11.2. Do not apply the press force to a bearing through its rolling elements. See example illustrated in Fig. 11.3.** Further, a small bearing with minimal interference may be installed by hammering the bearing ring with a plastic hammer or similar tool. **In that case, however, the uniform hammering force must be applied to the bearing side face via the backing strip as illustrated in Fig. 11.2, because direct hammering to the bearing end face or partial hammering by use of a punch could impair the specific bearing performance.**

While installing a bearing, NEVER hit the outer ring with a hard tool such as a hammer to fit the inner ring over the shaft. Never hammer the inner ring to install the bearing to the shaft. Otherwise, a flaw and/or dent mark may occur on the raceway surface and rolling elements of the bearing. Also, coating the fitting surfaces with high-viscosity oil will help reduce friction on the fitting surfaces.

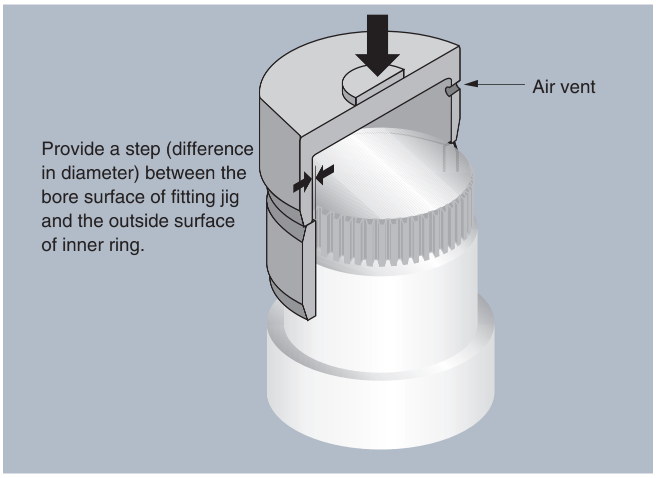


Fig. 11.2 Press-fitting of inner ring

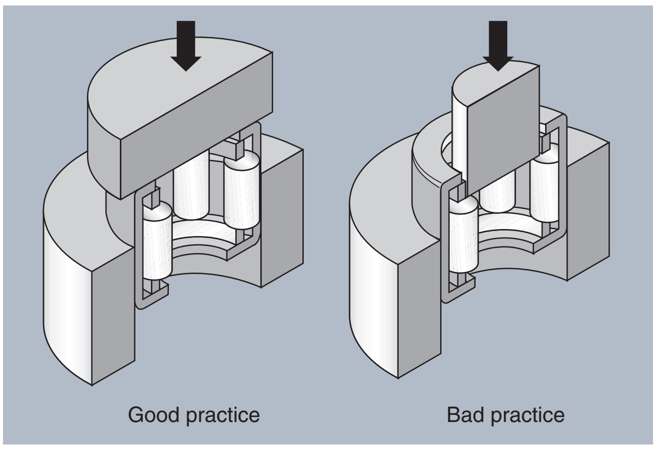


Fig.11.3 Good practice for press-fitting

(3) Shrink fit

This method too, is often used to install a bearing onto a shaft. The inner ring is heated in a medium such as a clean oil bath to expand its bore and is then fitted over the shaft. The oil used for this process should be pure mineral oil as it is less corrosive. The inner ring fitted onto the shaft is then allowed to stand to cool down. During the cool-down period, the inner ring shrinks in the axial direction too: therefore, the inner ring should be kept forced toward the shaft shoulder until it is fully cool in order to avoid a gap between it and the shaft shoulder. **Fig. 11.4** graphically illustrates the relationship between the expansion of the inner ring bore and the heating temperature. **Remember, however, that the inner ring must not be heated in excess of 120˚C. Also, do not apply shrink fit technique to a bearing with prefilled grease, or a bearing with a shield or seal.**

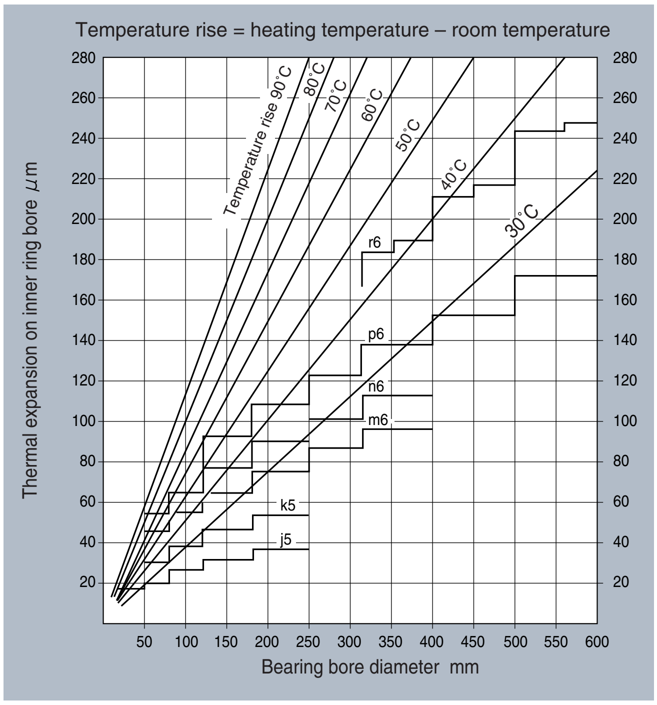


Fig. 11.4 Temperature rise needed for successful shrink fitting for inner ring

11.4 Bearing running test

To ensure that the bearing has been properly installed, a running test is performed after mounting.

Avoid running the bearing at its rated speed immediately after its installation. Otherwise, the bearing can fail if it has been incorrectly installed, or can seize if it is poorly lubricated. The shaft or housing should first be rotated by hand. If turning the shaft manually has proved to be problem-free, turn it at low speed with no load, and gradually increase the running speed and load while monitoring smoothness of bearing operation.

Carefully monitor noise and heat buildup on the running bearing. If any problem is detected, stop and

inspect the machine. If necessary, remove and inspect the bearing.

Sound level and tone of a running bearing can be checked by a sound scope held in contact with the bearing housing. The sound is normal if a pure sound is heard. A high metallic sound or irregular sounds from the bearing, indicates an error of function. In such a case, the possible cause of the failure can be measured by using a vibrometer to quantitatively determine vibration amplitude and frequency.

Generally, bearing temperature can be estimated from the housing surface temperature. However, if the bearing outer ring is accessible through oil holes, etc, the temperature can be more accurately measured.

Under normal conditions, bearing temperatures rise with rotation and then reach a stable operating temperature after a certain period of time. If the temperature does not level off and continues to rise, if there is a sudden temperature rise, or if the temperature is unusually high, the bearing must be inspected.

Table 11.1 shows the required check items.

Table 11.1

Hand operation	Variation in torque Over-torque Sticking Abnormal sound	Imperfect installation Under-clearance, great seal friction, etc. Indent and flaw on raceway surface Inclusion of dust and other foreign matter
Power operation	Abnormal noise and vibration Abnormal temperature	Inclusion of dust and other foreign matter, indent on raceway surface,over-clearance, inadequate lubrication Use of improper lubricant, incorrect installation, under-clearance

11.5 Bearing removal (dismounting)

Bearings are often removed as part of periodic inspection procedures or during the replacement of other parts. In this case, these bearings must be handled with the same care as when it was installed. Bearings, shafts, housings and other related parts must be designed to prevent damage during the dismounting procedure and the proper dismounting tools must be employed.

Regarding the dismounting method, generally the press method (**Fig. 11.5**) and the puller method (Fig.11.6) are used to dismount the inner ring depending on bearing type and fitting conditions.

Be sure to apply the extraction force to the inner ring or outer ring only when removing the bearing. Never attempt to extract the bearing ring by applying force through the rolling elements.

11.6 Force needed for press-fitting and extraction

The force needed for press-fitting or extracting a particular inner ring onto or from a shaft can be determined by **formula (11.1)** below:

$$K_a = f_K f_E \frac{d}{d+3} \Delta d_F \quad \cdots\cdots (11.1)$$

Where,

K_a : Force required for press-fitting or extraction N (kgf)

f_K : Resistance factor being determined by shaft to inner ring friction factor

For press-fitting ········ 40 (4)

For extraction ············ 60 (6)

f_E : Coefficient depending on inner ring dimension

$$f_E = B\left[1 - \left(\frac{d}{F_1}\right)^2\right]$$

B : Inner ring width mm

d : Inner ring bore diameter mm

F_1 : Mean outer diameter of inner ring mm

Δd_F : Apparent interference μm

Actual press-fit force and extraction force could eventually exceed the respective calculate value due to installing error. Hence, it is recommended to design the dismounting tools so as to have the strength (rigidity) resistible to a load 5 times as much as the calculated press-fit force and pull-out force.

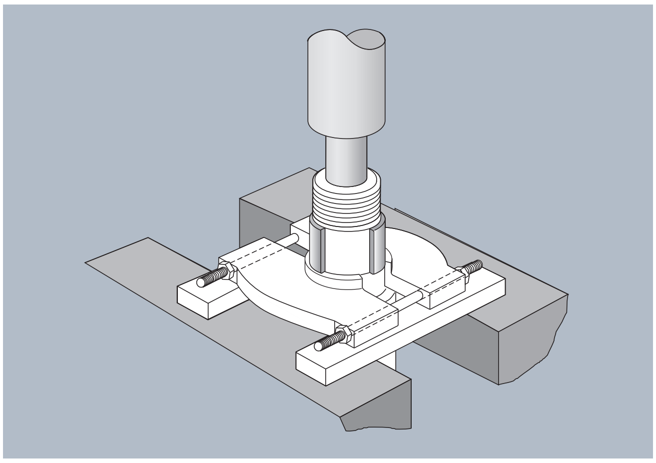


Fig. 11.5 Bearing removal by a press machine

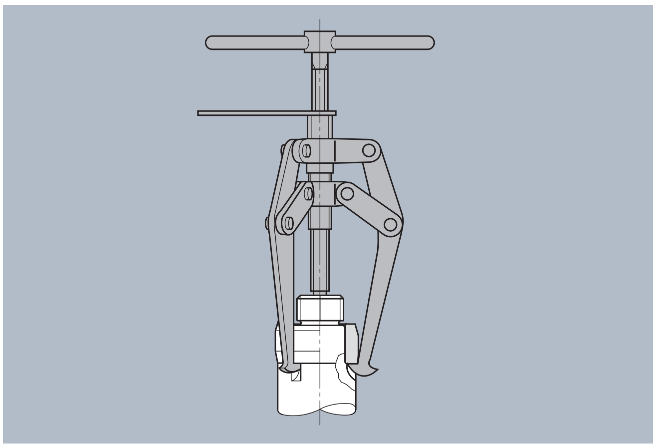


Fig. 11.6 Bearing removal by a puller

12. New Products Information

12. 1 HL Bearing

Bearing flaking can be categorized into two types: that which originates from inside of the bearing (subsurface flaking), and that which originates from the surface of the bearing.

Subsurface flaking usually occurs in areas where lubrication is considered to be good. This problem is believed to occur only when there exists a high level of contact stress. Present day steel is sufficiently clean so that cleanliness is not a contributing factor.

On the other hand, surface flaking is believed to be caused in areas where lubrication is insufficient. It is widely known that this problem is related to the oil film parameter (i.e. the ratio of oil film thickness at the point of contact to the combined surface roughness of the two objects in contact) which was derived from the elastohydrodynamic lubrication theory (EHL theory).

To reduce surface flaking, the oil film parameter needs to be increased. To do this, bearing manufactures have been working on both improving lubricants and surface roughness of the bearing raceway.

The EHL theory is based on the major premise that surface roughness of the contact surfaces is uniform. However, there are cases where the surface roughness determined in accordance with the EHL theory does not agree with the actual measured surface roughness.

In recent years a new theory has emerged. It contends that oil film formation in the contact areas can be improved by changing the character and direction of the machined parts surface finish.

NTN developed the long life HL (High Lubrication) bearing, based on the Micro EHL Theory, to reduce the problem of surface flaking.

12.1.1 Basic concept of HL bearing

The basic concept behind the development of the HL bearing is expressed by **Fig. 12.1.** These diagram are based on a flow model of the lubricant inside the contact area, developed by H.S. Cheng and his associates.The hatched areas in the diagrams are the contact points (elastically deformed) while the dotted lines show the flow of the lubricant.

The flow resistance of the lubricant is greater in (B) than in (A). This means that the volume of lubricant in each contact area increases, and accordingly the thickness of the oil film on the rolling contact surface also increases.

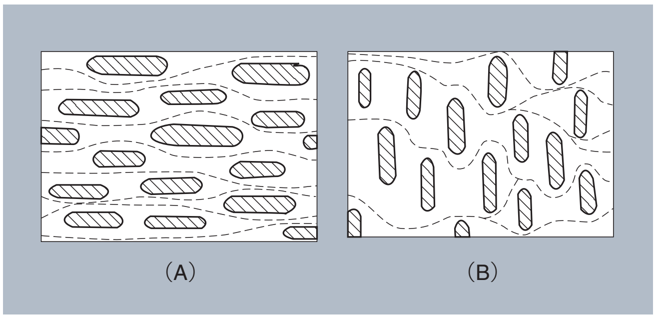


Fig. 12.1 Directional characteristics of finished surfaces and their effect on lubricant movement in a floe model

12.1.2 HL surface

As shown in **Fig. 12.2**, this newly developed surface (the HL surface : HL = High Lubrication) features a countless number of indentations (which are called micro oil pots) of about 10 μm which are produced at random. The black areas in the figure are the micro oil pots. This surface , featuring the desired size and number of micro oil pots, can be produced by changing the grinding conditions. Depth of the micro oil pots is about 1 μm.

12.1.3 HL bearing application examples

The HL surface-treated bearings are widely used in various fields. Such as car transmission, hydraulic devices, various reduction gears, etc.

HL surface treatment is applied to special applications, such as the rocker arm of a car engine where HL is on effective seizure preventive measure.

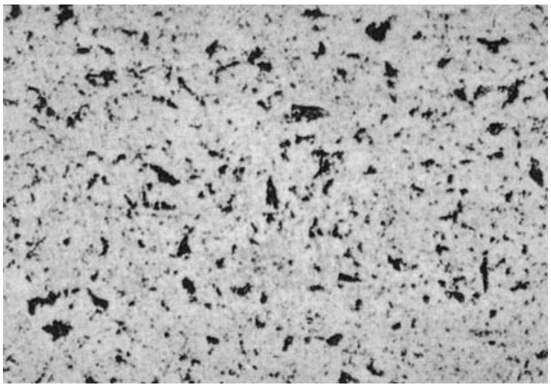


Fig. 12.2 Magnified photo showing HL roller surface

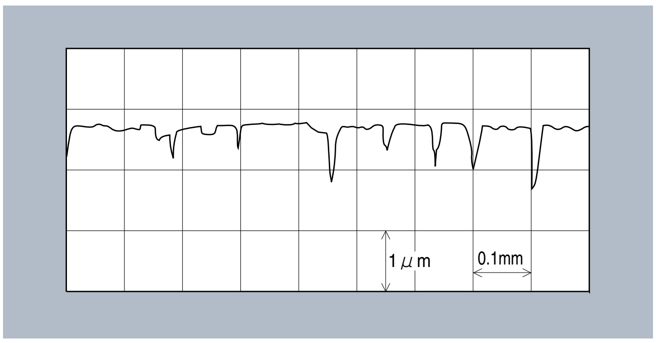


Fig. 12.3 HL surface roughness

12.2 Bearings with Solid Grease

"Solid grease" is a lubricant essentially composed of lubricating grease and ultra-high polymer polyethylene. Solid grease has the same viscosity as ordinary grease at normal temperature, but as a result of a special heat treatment process, this grease solidifies retaining a large proportion of the lubricant in it.

Thanks to this solidification, the grease does not easily leak from the bearing, even when the bearing is subjected to strong vibrations or centrifugal force, helping to extend bearing life.

All **NTN** needle roller bearings with Solid Grease are "full pack" products whose bearing space is nearly fully prefilled with solid grease.

12.2.1 Features of Bearings with Solid Grease

(1) Reduced lubricant leakage

Because the base oil is retained in a solid mixture, it is less likely to leak out of the bearing. During operation, temperature rise and/or centrifugal force will cause a gradual release of the base oil into the raceway groove. Eliminating grease leakage from the bearing ensures a consistent supply of lubricant and prevents contamination of the surrounding environment.

(2) Superior lubrication

Bearings with solid grease resist grease leakage prolonging bearing life in applications where high centrifugal force or vibration are present. The solid lubricant does not emulsify when exposed to water also extending both grease and bearing life.

(3) Sealing effect

Though solid grease protects a bearing against ingress of foreign matters (water, dust, etc.), it is not a sufficient means as a sealing device. Therefore, for applications that need reliable sealing performance, we recommend the use of contact type rubber seals.

12.2.2 Varieties of NTN needle roller bearings with Solid Grease

The **NTN** needle roller bearings with Solid Grease can be categorized into the general purpose group and the high-speed group (**Table 12.1**).

12.2.3 Precautions for using NTN needle roller bearing with Solid Grease

(1) Each **NTN** needle roller bearing type has unique set of available dimensions. For detailed information, contact **NTN** Engineering.

(2) A minimum radial load is required to prevent skidding of the rolling elements when using full-pack solid grease. The minimum load required is approximately 1% of the bearing dynamic load rating.

(3) Do not use any **NTN** needle roller bearing with Solid Grease in a situation where it will come into contact with organic solvents (acetone, petroleum benzene, refined kerosene, etc.).

12.2.4 Typical applications of bearings with Solid Grease

- Bearing for the paper feeder of a printing machine
- Bearing for the mast roller guide of a forklift
- Support bearing for the swing arm of a motorcycle
- Bearing for a machine tool
- Guide bearing for the guide unit of a press machine
- Bearing for the link mechanism of an automatic loom
- Bearing for the conveyor guide of a food packaging machine

For detailed information about **NTN** bearings with Solid Grease, refer to **NTN** CAT. NO. 3022 (Bearings with Solid Grease).

Table 12.1 Varieties of NTN needle roller bearings with Solid Grease

Type	General purpose group (LP03)	High-speed group (LP08)
Major components	(Resin) Super high-molecular weight polyethylene (Lubricant) Li-mineral oil based grease	(Resin) Super high-molecular weight polyethylene (Lubricant) Urea-synthetic oil based grease
Permissible temperature range (Bearing outer ring)	-20 – 80˚C 60˚C max. for prolonged operation	-20 – 100˚C 80˚C max. for prolonged operation
	For installing the bearing to a shaft or housing, shrink fit technique may be adopted. However, be sure to limit the maximum heating temperature to 100˚C and maximum retention time to 2 hours. During this process, be careful not to turn the bearing.	
Limiting speed F_w: Roller set bore diameter (mm) n : Operating running speed (min^{-1})	$F_w \cdot n$ value $\leqq 3\times10^4$	$F_w \cdot n$ value $\leqq 6\times10^4$

13. Bearing Type Symbols and Auxiliary Symbols

Table 13.1 Bearing Type Symbols

Type code	Bearing type
811	Single-direction thrust cylindrical roller bearing, dimension series 11
812	Single-direction thrust cylindrical roller bearing, dimension series 12
893	Single-direction thrust cylindrical roller bearing, dimension series 93
874	Single-direction thrust cylindrical roller bearing, dimension series 74
A	Needle roller, spherical type
ARA821	Double-direction thrust cylindrical roller bearing
ARB821	Double-direction thrust cylindrical roller bearing
ARN	Needle roller bearing with double-direction thrust cylindrical roller bearing
AS11	Steel plate thrust washer, dimension series 11
AXA21	Double-direction thrust needle roller bearing
AXB21	Double-direction thrust needle roller bearing
AXK11	Needle roller and cage thrust assembly, dimension series 11
AXN	Needle roller bearing with double-direction thrust needle roller bearing
BF	Metallic flat cage for linear flat rollers
BK	Drawn cup needle roller bearing with close end
BR	Housing snap ring
CR	Cam follower, inch series
CRV	Full complement roller for cam follower,
DCL	inch series
	Drawn cup needle roller bearing with open end, inch series
F	Needle roller, plane type
FF	Linear flat roller
FR	Bottom roller bearing, for drawing frame
FRIS	Bottom roller bearing, for fine spinning frame and flyer frame
G	Synthetic rubber seal, one-lip type
GD	Synthetic rubber seal, double-lip type
GK	Needle rollers with split type cage
GS811	Housing washer, dimension series 11
GS812	Housing washer, dimension series 12
GS893	Housing washer, dimension series 93
GS874	Housing washer, dimension series 74
HCK	Drawn cup needle roller bearing for universal joint
HF	One-way clutch
HFL	One-way clutch integral with bearing
HK	Drawn cup needle roller bearing with open end
HMK	Drawn cup needle roller bearing with open end, for heavy load application
IR	Inner ring
JF··S	Tension pulley holder
JPU··S	Tension pulley and jockey pulley
K	Needle rollers with cage
K811	Cylindrical roller and cage thrust assembly, dimension series 11
K812	Cylindrical roller and cage thrust assembly, dimension series 12
K893	Cylindrical roller and cage thrust assembly, dimension series 93
K874	Cylindrical roller and cage thrust assembly, dimension series 74
KBK	Needle roller and cage assembly, for small end
KD	Linear ball bearing, stroking type
KH	Linear ball bearing, drawn cup type
KJ··S	Needle roller and cage assembly
KLM	Linear ball bearing, machined ring type
KLM··S	Linear ball bearing, clearance-adjustable type
KLM··P	Linear ball bearing, open type
KMJ	Needle roller and cage assembly
KLJ··S	Needle roller and cage assembly
KR	Cam follower
KRM	Miniature cam follower
KRMV	Miniature cam follower, full complement roller type
KRT	Cam follower, w/ tapped hole
KRU	Cam follower, shaft eccentric type

Type code	Bearing type
KRVT	Cam follower, full complement roller type, w/ tapped hole
KRV	Cam follower, full complement roller type
KRVU	Cam follower, full complement roller and shaft eccentric type
KV··S	Needle roller and cage assembly
MI	Inner ring, inch series
MR	Machined ring needle roller bearing without inner ring, inch series
NA22	Roller follower with inner ring, dimension series 22
NA48	Machined ring needle roller bearing with inner ring, dimension series 48
NA49	Machined ring needle roller bearing with inner ring, dimension series 49
NA59	Machined ring needle roller bearing with inner ring, dimension series 59
NA69	Machined ring needle roller bearing with inner ring, dimension series 69
NA49···S	Clearance-adjustable needle roller bearing with inner ring
NAB2	Separable roller follower, w/ inner ring, diameter series2
NACV	Roller follower, full complement roller type, inch series
NAO	Machined ring needle roller bearing, separable type, with inner ring
NATR	Roller follower
NATV	Roller follower, full complement roller type
NIP	Grease nipple
NK	Machined ring needle roller bearing without inner ring
NKIA59	Complex bearing : Needle roller bearing with angular ball bearing dimension series 59
NKIB59	Complex bearing : Needle roller bearing with three-point contact type ball bearing dimension series 59
NKS	Machined ring needle roller bearing, w/o inner ring
NKX	Complex bearing : needle roller bearing with thrust ball bearing without dust-proof cover
NKX···Z	Complex bearing: Needle roller bearing with thrust ball bearing with dust-proof cover
NKXR	Complex bearing: Needle roller bearing with thrust cylindrical roller bearing without dust-proof cover
NKXR···Z	Complex bearing: Needle roller bearing with thrust cylindrical roller bearing with dust-proof cover
NUKR	Cam follower, full complement roller type
NUKRT	Cam follower, full complement roller type, w/ tapped hole
NUKRU	Cam follower, full complement roller type, w/ tapped hole, eccentric stud
NUTR2	Roller follower, diameter series 2
NUTR3	Roller follower, diameter series 3
NUTW	Roller follower, outer ring with center rib
PCJ	Needle roller and cage assembly, inch series
PK	Needle roller and cage assembly, for large end
PNA···R	Self-aligning needle roller bearing with inner ring
RF	Polyamide resin cage for linear flat rollers
RLM	Linear roller bearing
RNA22	Roller follower without inner ring, dimension series 22
RNA48	Machined ring needle roller bearing without inner ring, dimension series 48
RNA49	Machined ring needle roller bearing without inner ring, dimension series 49
RNA59	Machined ring needle roller bearing without inner ring, dimension series 59
RNA69	Machined ring needle roller bearing without inner ring, dimension series 69
RNA49···S	Clearance-adjustable needle roller bearing, without inner ring
RNAB2	Separable roller follower, w/o inner ring, diameter series 2
RNAO	Machined ring needle roller bearing, separable type, without inner ring
RPNA···R	Self-aligning needle roller bearing, w/o inner ring
WR	Snap ring for shaft
WS811	Thrust inner ring, dimension series 11
WS812	Thrust inner ring, dimension series 12
WS893	Thrust inner ring, dimension series 93
WS874	Thrust inner ring, dimension series 74
ZS	Thrust central ring

Table 13.2 Auxiliary symbols

	Symbol		Symbol representation
Initial symbols	Material heat-treatment symbols	TS-	Bearing for high temperature application which was heat-treated for dimensional stabilization
		E-	Bearing made of case-hardened steel
		8Q-	Nitro-carburized cage
		F-	Bearing made of stainless steel
		C-	Bearing made of carbon steel
	Expansion compensation	EC-	Expansion-compensated bearing
Basic symbols			
Suffix	Internal construction symbols	ZW	Double-row cage
		A,B,C	Internal construction change
	Cage symbols	J,JW	Steel plate punched cage
		L1	High strength brass cage
		T2	Polyamide resin cage
		S	Welded cage
	Seal symbol	L,LL	With synthetic rubber seal
	Bearing ring profile symbols	D	With oil hole
		D1	With oil hole and oil groove
		H	Cam follower with hexagon hole
	Roller symbol	T	Crowning and special heat treatment
	Combination symbols	D2,Dn	Complex bearing using two or more same bearings
	Clearance symbols	C2	Clearance smaller than ordinary clearance
		C3	Clearance larger than ordinary clearance
		C4	Radial clearance larger than C3
		NA	Non-interchangeable clearance
	Accuracy class symbols	P6	Bearing of JIS Class-6
		P5	Bearing of JIS Class-5
		P4	Bearing of JIS Class-4
	Lubrication symbols	/2AS	SHELL ALVANIA Grease 2
		/3AS	SHELL ALVANIA Grease 3
		/P03	Solid Grease
	Special symbols	V1～Vn	Special specification, requirements