General Explanation

Nippon Thompson Co., Ltd. is a bearing manufacturer that launched the technical development of needle roller bearings for the first time in Japan and is proud of the high quality level and abundant varieties of its products.

Needle roller bearings are bearings for rotary motion that incorporate needle-shaped thin rollers instead of ordinary bearing balls or rollers. Compared with other rolling bearings, they are small-sized and lightweight but have a large load capacity. They are widely used with high reliability in the fields of automobiles, industrial machinery, OA equipment, etc. as resource-saving type bearings that make the whole machine compact.

Characteristics of Needle Roller Bearings

Bearings can be classified into two main types, namely rolling bearings and sliding bearings. Rolling bearings can be subdivided further into ball bearings and roller bearings according to the rolling elements.
IKO Needle Roller Bearings are high-precision rolling bearings with a low sectional height, incorporating needle rollers as the rolling element. They have the following features.

Merits of Rolling Bearings

Compared with sliding bearings, rolling bearings have the following merits:

❶ Static and kinetic friction is low.

Since the difference between static friction and kinetic friction is small and the frictional coefficient is also small, drive units or machines can be made more compact and lightweight, saving machine costs and power consumption.

❷ Stable accuracy can be maintained for long periods.

Owing to less wear, stable accuracy can be maintained for long periods.

❸ Machine reliability is improved.

Since the bearing life can be estimated based on rolling fatigue, machine reliability is improved.

❹ Lubrication is simplified.

Since grease lubrication is sufficient in most cases, lubrication can be simplified for easy maintenance.

Merits of Needle Roller Bearings

Compared with other rolling bearings, IKO Needle Roller Bearings have the following advantages:

❶ With a low sectional height, they can withstand heavy loads.

Since they have a low sectional height compared with other rolling bearings and yet can withstand heavy loads, machines can be made more compact and lightweight, thus saving costs.

❷ Rotating torque is small, improving mechanical efficiency.

Since the rotating radius is small, the rotating torque is also small under the same frictional conditions, thus improving mechanical efficiency.

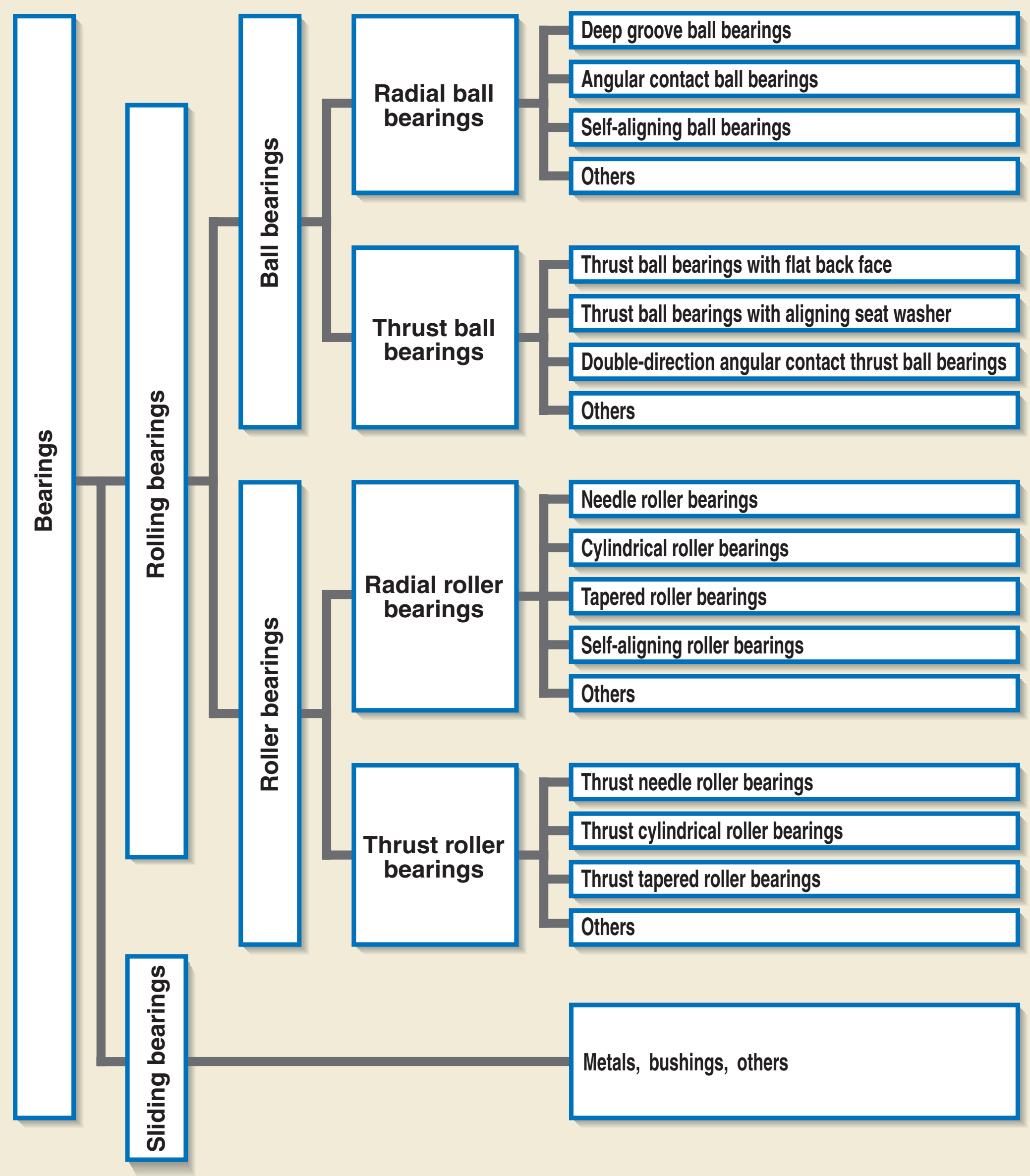
❸ Inertia is minimized.

Since the bearing volume and weight are small, the moment of inertia of the bearing is minimized when it is put in motion.

❹ Most suited to oscillating motions.

Many rolling elements are arranged at a small spacing pitch, and this configuration is most suited to oscillating motions.

Classification of bearings



Types and Features of Bearings

IKO Bearings can be roughly classified into radial bearings and thrust bearings according to applicable load direction. Radial Bearings are grouped into Shell Type Needle Roller Bearings, Machined Type Needle Roller Bearings, and various other types. Thrust Bearings are grouped into Thrust Needle Roller Bearings and Thrust Roller Bearings. Follower Bearings that are used for cam mechanisms and linear motion are grouped into Cam Followers and Roller Followers.

Classification of IKO Bearings

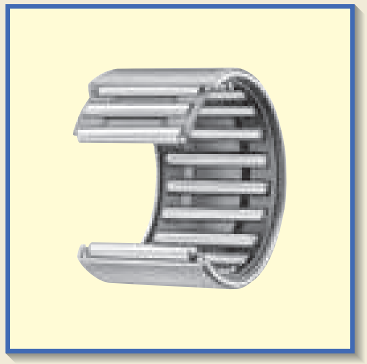
- **Radial Bearings**
  - **Shell Type Needle Roller Bearings**
    - TA、TAM
    - TLA、TLAM
    - BA、BAM
    - BHA、BHAM
    - YT
    - YTL
    - YB
    - YBH
  - **Needle Roller Cages**
    - Needle Roller Cages for General Usage
      - KT、KT…N
      - KTW
    - Needle Roller Cages for Engine Connecting Rods
      - KT…EG
      - KTV…EG
  - **Machined Type Needle Roller Bearings**
    - NA、RNA
    - TAFI、TAF
    - TRI、TR
    - BRI、BR
    - GTRI、GTR
    - C-Lube Machined Type Needle Roller Bearings
      - TAF…/SG
    - Needle Roller Bearings with Separable Cage
      - NAF、RNAF
      - NAFW、RNAFW
  - **Roller Bearings**
    - NAU
    - NAG
    - TRU
    - Roller Bearings for Sheaves
      - NAS
- **Thrust Bearings**
  - **Thrust Needle Roller Bearings**
    - NTB
  - **Thrust Roller Bearings**
    - AZK、AZ
- **Combined Type Bearings**
  - **Combined Type Needle Roller Bearings**
    - with Thrust Ball Bearing: NAXI、NAX
    - with Thrust Roller Bearing: NBXI、NBX
    - with Angular Contact Ball Bearing: NATA
    - with Three-point Contact Ball Bearing: NATB

Crossed Roller Bearings are special shape bearings that can simultaneously receive loads in all directions with a single bearing.
Bearings other than rolling bearings, such as self-aligning Spherical Bushings that can support radial loads and axial loads and PILLOBALLs and L-Balls that are used for link mechanisms, are also available.

- **Follower Bearings**
  - **Cam Followers**
    - Standard Type Cam Followers: CF
    - Double Hex Hole Cam Followers: CFKR
    - Solid Eccentric Stud Type Cam Followers: CFES
    - Eccentric Type Cam Followers: CFE
    - Thrust Disk Type Cam Followers: CF…W
    - Centralized Lubrication Type Cam Followers: CF-RU1、CF-FU1
    - Easy Mounting Type Cam Followers: CF-SFU
    - Cam Follower G: CF…G
    - C-Lube Cam Followers: CF…/SG
    - Miniature Type Cam Followers: CFS
    - Thrust Disk Type Miniature Cam Followers: CFS…W
    - Cylindrical Roller Cam Followers: NUCF
    - Inch Series Cam Followers: CR、CRH
  - **Roller Followers**
    - Separable Roller Followers: NAST、RNAST
    - Non-separable Roller Followers: NART、CRY
    - C-Lube Roller Followers: NART…/SG
    - Cylindrical Roller Followers: NURT
- **Crossed Roller Bearings**
  - **Crossed Roller Bearings**
    - High Rigidity Type Crossed Roller Bearings (V): CRBH(V)
    - Standard Type Crossed Roller Bearings: CRBC、CRB
    - Super Slim Type Crossed Roller Bearings: CRBT
    - Slim Type Crossed Roller Bearings: CRBS
    - Mounting Holed Type High Rigidity Crossed Roller Bearings (V): CRBF(V)
- **Spherical Sliding Bearings**
  - **Spherical Bushings**
    - SB
    - GE
    - SBB
  - **PILLOBALLs**
    - PB
    - PHS
    - POS
    - PHSA
  - **L-Balls**
    - LHSA
    - LHS

IKO

Shell Type Needle Roller Bearings



Shell Type Needle Roller Bearings are lightweight with the lowest sectional height among needle roller bearings with outer ring, because they employ a shell type outer ring made from a thin special-steel plate which is accurately drawn, carburized and quenched.
Since these bearings are press-fitted into the housing, no axial positioning fixtures are required. They are ideal for use in mass-produced articles that require economy.

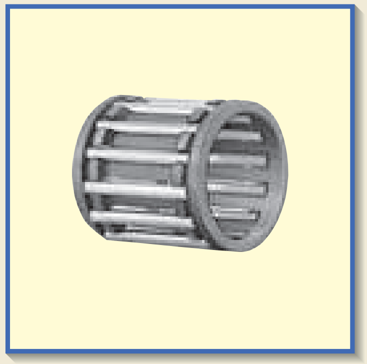
Radial Bearings Page B1

Needle Roller Cages for General Usage

Needle Roller Cages for General Usage are bearings that display excellent rotational performance. Their specially shaped cages with high rigidity and accuracy, precisely guide the needle rollers.
Since needle rollers with extremely small dimensional variations in diameter are incorporated and retained, Needle Roller Cages for General Usage are useful in small spaces when combined with shafts and housing bores that are heat treated and accurately ground as raceway surfaces.

Radial Bearing Page C1

Needle Roller Cages for Engine Connecting Rods



Needle Roller Gages for Engine Connecting Rods are used for motor cycles, small motor vehicles, outboard marines, snow mobiles, general-purpose engines, high-speed compressors, etc. that are operated under extremely severe and complex operating conditions such as heavy shock loads, high speeds, high temperatures, and stringent lubrication.
Needle Roller Cages for Engine Connecting Rods are lightweight and have high load ratings and high rigidity as well as superior wear resistance.

Radial Bearing Page C17

Machined Type Needle Roller Bearings

Machined Type Needle Roller Bearings have an outer ring made by machining, heat treatment, and grinding. The outer ring has stable high rigidity and can be easily used even for light alloy housings.
These bearings are available in various types and optimally selectable for different conditions such as heavy loads, high-speed rotation and low-speed rotation. They are most suitable for general-purpose applications.

Radial Bearing Page D1

Needle Roller Bearings with Separable Cage

In Needle Roller Bearings with Separable Cage, the inner ring, outer ring and Needle Roller Cage are combined, and they can be separated easily. This type has a simple structure with high accuracy. In addition, the radial clearance can be freely selected by choosing an assembly combination.
These bearings have excellent rotational performance, because Needle Roller Cages are used.

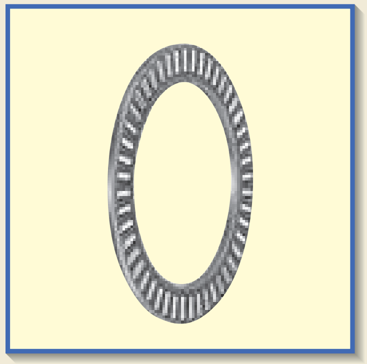
Radial Bearing Page D79

Roller Bearings

Roller Bearings, in which rollers are incorporated in double rows, are non-separable heavy-duty bearings. They can withstand not only radial loads but axial loads as well, which are supported at the contacts between the shoulders of inner and outer rings and the end faces of rollers. Therefore, they are most suitable for use at the fixing side of a shaft.

Radial Bearing Page E1

Thrust Bearings



Thrust Bearings consist of a precisely made cage and rollers, and can receive axial loads. They have high rigidity and high load capacities and can be used in small spaces.
Thrust Needle Roller Bearings use needle rollers, while Thrust Roller Bearings use cylindrical rollers.

Thrust Bearing Page F1

Combined Type Needle Roller Bearings

Combined Type Needle Roller Bearings are combinations of a radial bearing and a thrust bearing. Caged Needle Roller Bearings are used as radial bearings and Thrust Ball Bearings or Thrust Roller Bearings are used as thrust bearings.
They can be subjected to radial loads and axial loads simultaneously.

Combined Type Bearing Page G1

Inner Rings

Inner Rings are heat-treated and finished by grinding to a high degree of accuracy and are used for Needle Roller Bearings.
In the case of Needle Roller Bearings, normally the shafts are heat-treated and finished by grinding and used as raceway surfaces. However, when it is impossible to make shaft surfaces according to the specified surface hardness or surface roughness, Inner Rings are used.

Component part Page H1

IKO

Cam Followers

Cam Followers are bearings with a stud incorporating needle rollers in a thick walled outer ring.
They are designed for outer ring rotation, and the outer rings run directly on mating cam guide surfaces.
Various types of Cam Followers are available. They are widely used as follower bearings for cam mechanisms and for linear motions.

Follower Bearing Page I1

Roller Followers

Roller Followers are bearings in which needle rollers are incorporated in a thick walled outer ring.
These bearings are designed for outer ring rotation, and the outer rings run directly on mating cam guide surfaces.
They are used as follower bearings for cam mechanisms and for linear motions.

Follower Bearing Page I77

Crossed Roller Bearings

Crossed Roller Bearings are high-rigidity and compact bearings with their cylindrical rollers alternately crossed at right angles to each other between inner and outer rings. A single Crossed Roller Bearing can take loads from any directions at the same time such as radial, thrust, and moment loads.
These bearings are widely used in the rotating parts of industrial robots, machine tools, medical equipment, etc. which require compactness, high rigidity and high rotational accuracy.

Crossed Roller Bearing Page J1

Spherical Bushings

Spherical Bushings are self-aligning spherical plain bushings, which have inner and outer rings with spherical sliding surfaces. They can take a large radial load and a bi-directional axial load at the same time.
They are divided into steel-on-steel types that are suitable for applications where there are alternate loads or shock loads, and maintenance-free types which require no lubrication.

Spherical Sliding Bearing Page K1

PILLOBALLs

PILLOBALLs are compact self-aligning spherical plain bushings which can support a large radial load and a bi-directional axial load at the same time.
PILLOBALL Rod Ends have either a female thread in the body or a male thread on the body, so they can be easily assembled onto machines.
PILLOBALLs are used in control and link mechanisms in machine tools, textile machines, packaging machines, etc.

Spherical Sliding Bearing Page K29

L-Balls

L-Balls are self-aligning rod-ends consisting of a special zinc die-cast alloy body and a studded ball which has its axis at right-angles to the body.
They can perform tilting movement and rotation with low torque, and transmit power smoothly due to the uniform clearance between the sliding surfaces.
They are used in link mechanisms in automobiles, construction machinery, farm and packaging machines, etc.

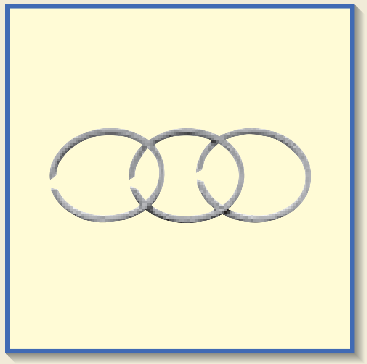
Spherical Sliding Bearing Page K45

Seals for Needle Roller Bearings

Seals for Needle Roller Bearings have a low sectional height and consist of a sheet metal ring and special synthetic rubber.
As these seals are manufactured to the same sectional height as Needle Roller Bearings, grease leakage and the penetration of foreign particles can be effectively prevented by fitting them directly to the sides of combinable bearings.

Component Part Page L1

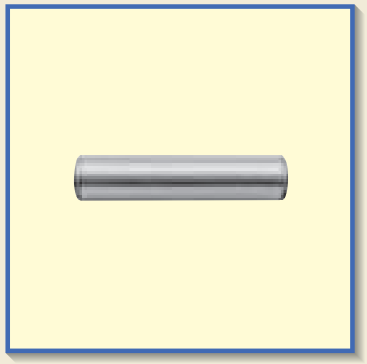
Cir-clips for Needle Roller Bearings



Cir-clips for Needle Roller Bearings have been specially designed for needle roller bearings on which, in many cases, generally available Cir-clips cannot be used. They have a low sectional height and are very rigid.
There are Cir-clips for shafts and for bores, and they are used for positioning to prevent bearing movement in the axial direction.

Component Part Page L17

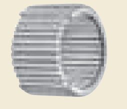
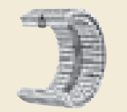
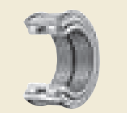
Needle Rollers



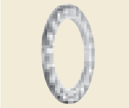
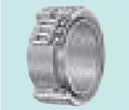
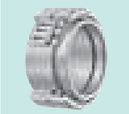
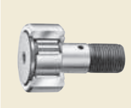
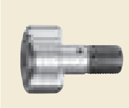
Needle Rollers are used for needle roller bearings and are rigid and highly accurate.
These needle rollers are widely used as rolling elements for bearings, and also as pins and shafts.

Component Part Page L23

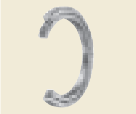
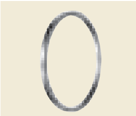
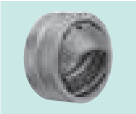
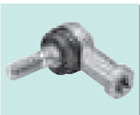
Features of IKO Bearings

Bearing series		Appearance	Direction of motion	Load direction and capacity	Allowable rotational speed	Friction	Sectional height	Reference page
Shell Type Needle Roller Bearings	Caged type		Rotation	Light radial load	○	○	◎	B1～
	Full complement type		Rotation	Light radial load	△	△	◎	
Needle Roller Cages	For general usage		Rotation	Medium radial load	◎	◎	◎	C1～
	For engine connecting rods		Rotation	Medium radial load	◎	◎	◎	C17～
Machined Type Needle Roller Bearings	Caged type		Rotation	Medium radial load	○	◎	○	D1～
	Full complement type		Rotation	Heavy radial load	△	△	○	
Needle Roller Bearings with Separable Cage	Caged type		Rotation	Medium radial load	◎	◎	○	D79～
Roller Bearings	Caged type		Rotation	Medium radial load, light axial load	○	○	○	E1～
	Full complement type		Rotation	Medium radial load, light axial load	△	△	○	
	For sheaves		Rotation	Heavy radial load, light axial load	△	△	△	

Symbol Rotation　Oscillating motion　Radial load　Axial load　Light load　Medium load　Heavy load　◎ Especially excellent　○ Excellent　△ Normal

Bearing series		Appearance	Direction of motion	Load direction and capacity	Allowable rotational speed	Friction	Sectional height	Reference page
Thrust Bearings	Needle roller bearings		Rotation	Medium axial load	○	○	◎	F1～
	Roller bearings		Rotation	Heavy axial load	○	○	○	
Combined Type Needle Roller Bearings	With thrust ball bearing		Rotation	Medium radial load, medium axial load	○	○	△	G1～
	With thrust roller bearing		Rotation	Medium radial load, heavy axial load	○	○	△	
	With angular contact ball bearing		Rotation	Medium radial load, light axial load	○	○	○	
	With three-point contact ball bearing		Rotation	Medium radial load, light axial load (both directions)	○	○	○	
Cam Followers	Caged type		Rotation	Medium radial load	○	○	△	I1～
	Full complement type		Rotation	Heavy radial load	△	△	△	
Roller Followers	Separable caged type		Rotation	Medium radial load	○	○	△	I77～
	Non-separable caged type		Rotation	Medium radial load	○	○	△	
	Non-separable full complement type		Rotation	Heavy radial load	△	△	△	

Features of IKO Bearings

Bearing series		Appearance	Direction of motion	Load direction and capacity	Allowable rotational speed	Friction	Sectional height	Reference page
Crossed Roller Bearings	Caged type, Separator type		Rotation	Medium radial load, Medium axial load	△	○	○	J1~
	Full complement type		Rotation	Heavy radial load, Heavy axial load	△	△	○	
	Slim type		Rotation	Light radial load, Light axial load	△	○	◎	
Spherical Bushings	Steel-on-steel type		Rotation, Oscillating motion	Heavy radial load, Medium axial load	△	△	△	K1~
	Maintenance-free type		Rotation, Oscillating motion	Medium radial load, Light axial load	△	△	△	
PILLOBALLs	Insert type, Lubrication type		Rotation, Oscillating motion	Medium radial load, Light axial load	△	△	△	K29~
	Die-casting type, Lubrication type		Rotation, Oscillating motion	Medium radial load, Light axial load	△	△	△	
	Maintenance-free type		Rotation, Oscillating motion	Medium radial load, Light axial load	△	△	△	
L-Balls	Lubrication type		Rotation, Oscillating motion	Medium radial load, Light axial load	△	△	△	K45~

Symbol Rotation · Oscillating motion · Radial load · Axial load · Light load · Medium load · Heavy load · ◎ Especially excellent · ○ Excellent · △ Normal

Outline of Bearing Selection

IKO Bearings are available in many types and sizes. To obtain satisfactory bearing performance in machines and equipment, it is essential to select the most suitable bearing by carefully studying the requirements for the application. Although there is no particular procedure or rule for bearing selection, an example of a commonly adopted procedure is shown in the figure below.

An example of procedure for bearing selection

1 Confirmation of requirements and operating conditions
- Identify the machine and place where the bearing is to be used.
- Confirm the requirements for bearings such as required bearing performance, and also confirm the operating conditions and special environment conditions.

2 Selection of bearing type
- Select the bearing type suitable for the operating conditions by considering load direction and magnitude, rigidity, friction, allowable rotational speed, bearing space, etc.

See page A5

3 Selection of bearing dimensions
- Select the bearing dimensions by calculating bearing load, life, static safety factor, etc.

See page A17

4 Selection of accuracy class, etc.
- Select the accuracy as required by the machine or equipment.

See page A30

5 Selection of radial clearance and fit
- Select the radial clearance considering the fit, temperature, rotational speed, inclination of the inner and outer rings, etc.

See page A37

6 Determination of bearing dimensions, accuracy, radial clearance and fit

7 Selection of lubrication and dust-proof methods
- Select oil or grease lubrication.
- After selection of lubricant, in case of oil lubrication, select the oil application method.
- Select the sealing method according to the lubricant.

See page A49

8 Design of surrounding part
- Design the surrounding part based on how to mount or dismount and based on mounting dimensions.

See page A57

9 Determination of final specifications of the bearing and the surrounding part

A B C D E F G H I J K L M

Basic Dynamic Load Rating and Life

Life

Rolling bearings will suffer damage due to various causes during service. Damage such as abnormal wear, seizure, and cracks is caused by improper use, including incorrect mounting, lack of oil, dust intrusion and so on, and can be avoided by remedying these causes. However, bearings will eventually be damaged due to fatigue-flaking even if used properly. When a bearing rotates under load, the raceways and the rolling elements are subjected to repeated stresses concentrated on the part close to the surface. Fatigue, therefore, occurs in the surface layer, producing damage in the form of scaling. This is called flaking (spalling). When this occurs, the bearing can no longer be used.

Bearing Life

Bearing life is defined as the total number of revolutions (or total service hours at a constant rotational speed) before a sign of the first flaking appears on the rolling surface of raceway or rolling elements. However, even when bearings of the same size, structure, material and heat treatment are subjected to the same conditions, the bearing lives will show variation (See Fig. 1.). This results from the statistical nature of the fatigue phenomenon.
In selecting a bearing, it is incorrect to take an average life for all bearings as the design standard. It is more practical to consider a bearing life that is reliable for the greater proportion of bearings used. Therefore, the basic rating life defined in the following is used.

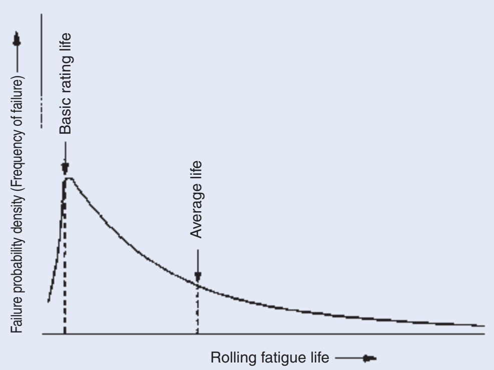


Fig. 1 Variation of rolling fatigue life

Basic rating life

The basic rating life is defined as the total number of revolutions that 90% of a group of identical bearings can be operated individually under the same conditions free from any material damage caused by rolling fatigue.
For rotation at a constant rotational speed, the basic rating life can be represented by the total service hours.

Basic dynamic load rating

The basic dynamic load rating is defined as the constant radial load (in the case of radial bearings) or the constant axial load acting along the bearing central axis (in the case of thrust bearings) that allows a basic rating life of 1,000,000 revolutions.

Calculation of rating life

The relationship among the basic rating life, basic dynamic load rating and dynamic equivalent load (bearing load) of rolling bearings is as follows:

$$L_{10}=\left(\frac{C}{P}\right)^{p} \cdots\cdots(1)$$

where, L_{10} : Basic rating life, 10^6 rev.
C : Basic dynamic load rating, N
P : Dynamic equivalent load, N
p : Exponent, Roller bearing: 10/3
Ball bearing: 3

Accordingly, when the rotational speed per minute is given, the basic rating life is represented as the total service hours according to the following equations:

$$L_h=\frac{10^6 L_{10}}{60n}=500 f_h^{\,p} \cdots\cdots(2)$$

$$f_h=f_n\frac{C}{P} \cdots\cdots(3)$$

$$f_n=\left(\frac{33.3}{n}\right)^{1/p} \cdots\cdots(4)$$

where, L_h : Basic rating life represented by service hours, h
n : Rotational speed, min^{-1}
f_h : Life factor
f_n : Velocity factor

In addition, the rating life can be calculated by obtaining f_h and f_n from the life calculation scales of Fig. 2.

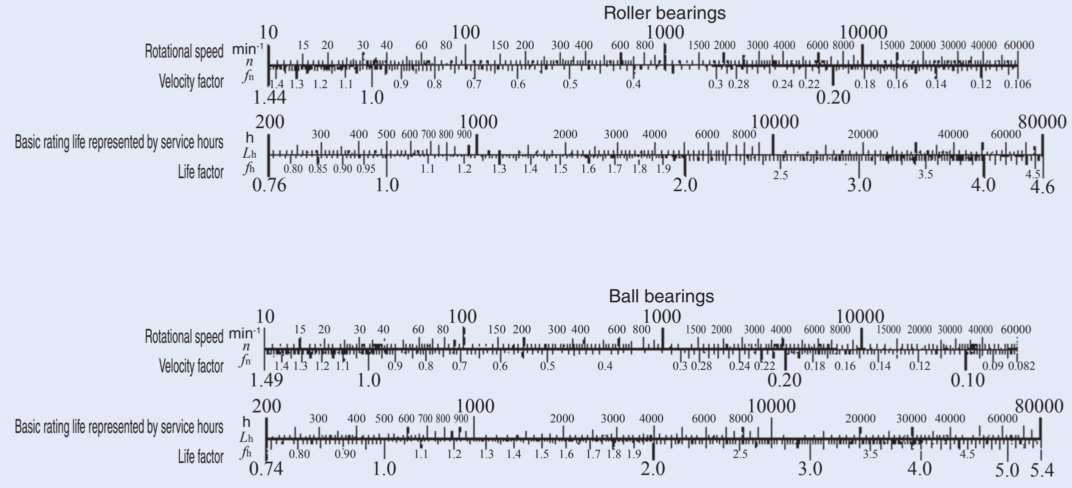


Fig. 2 Scales for rating life calculation

Bearing life factors for various machines

The required life of the bearing must be determined according to the machine in which the bearing is to be used and the operating conditions.
Table 1 shows reference values of life factors for selecting a bearing for each machine.

Table 1 Life factor of bearings f_h for various machines

Operating conditions	Machine and life factor f_h				
	～3	2～4	3～5	4～7	6～
Occasional or short term usage	・Power tools	・Agricultural machines			
Infrequent usage but requiring reliable operation		・Construction machinery	・Conveyors ・Elevators		
Intermittent operation but for comparatively long periods	・Roll neck of rolling mills	・Small motors ・Deck cranes ・General cargo cranes ・Passenger cars	・Factory motors ・Machine tools ・General gear units ・Printing machines	・Crane sheaves ・Compressors ・Important gear units	
Operated in excess of 8 hours per day or continuously for an extended time		・Escalators	・Centrifugal separators ・Blowers ・Wood working machines ・Plastic extruding machines		・Paper making machines
Continuous use for 24 hours and accidental stops not allowed					・Water supply equipment ・Power station equipment

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Life of oscillating bearing

The life of an oscillating bearing can be obtained from equation (5).

$$L_{OC} = \frac{90}{\theta}\left(\frac{C}{P}\right)^{p} \cdots\cdots\cdots (5)$$

where, L_{OC} : Basic rating life of oscillating bearing, 10^6 cycles
2θ : Oscillating angle, deg. (See Fig.3)
P : Dynamic equivalent load, N

Therefore, when the oscillating frequency n_1 min^{-1} is given, the basic rating life as represented by total oscillating hours can be obtained by substituting n_1 for n in equation (2) on page A17.
When 2θ is small, an oil film cannot be formed easily between the contact surfaces of the raceway and the rolling elements. This may cause fretting corrosion. In this case, please consult IKO.

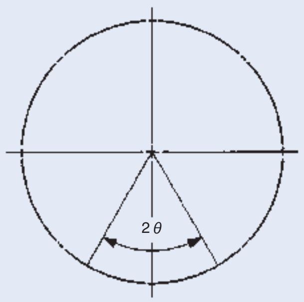


Fig. 3 Oscillating motion

Corrected rating life

When a rolling bearing is used in ordinary applications, the basic rating life can be calculated by equations (1) and (2) mentioned previously.
This basic rating life applies to bearings which require a reliability of 90%, have ordinary bearing properties being made of materials of ordinary quality for rolling bearings, and are used under ordinary operating conditions.
In some applications, however, it is necessary to obtain a rating life that applies to bearings which require high reliability, have special bearing properties or are used under special operating conditions. The corrected rating life for these special cases can be obtained from the following equation by using the bearing life adjustment factors a_1, a_2 and a_3, respectively.

$$L_{na} = a_1 a_2 a_3 L_{10} \cdots\cdots\cdots (6)$$

where, L_{na} : Corrected rating life, 10^6 rev.
a_1 : Life adjustment factor for reliability
a_2 : Life adjustment factor for special bearing properties
a_3 : Life adjustment factor for operating conditions

Life adjustment factor for reliability a_1

The reliability of rolling bearings is defined as the proportion of bearings having a life equal to or greater than a certain specified value when a group of identical bearings are operated under identical conditions. With respect to individual bearings, it refers to the probability of the life of a bearing being equal to or greater than a certain specified value.
The corrected rating life for a reliability of (100-n)% can be obtained using equation (6). Table 2 shows the values of the life adjustment factor a_1 for various reliabilities.

Table 2 Life adjustment factor for reliability a_1

Reliability %	L_n	a_1
90	L_{10}	1
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

Life adjustment factor for special bearing properties a_2

The bearing life is extended or shortened according to the quality of the material, the manufacturing technology of the bearing and its internal design. For these special bearing life properties, the life is corrected by the life adjustment factor for special bearing properties a_2.
The table of dimensions for IKO Bearings shows the values of the basic dynamic load rating which are determined taking into consideration the fact that bearing life has been extended by improved quality of materials and advances in manufacturing technologies. Therefore, the bearing life is calculated using equation (6) usually assuming $a_2 = 1$.

Life adjustment factor for operating conditions a_3

This factor helps take into account the effects of operating conditions, especially lubrication on the bearing.
The bearing life is limited by the phenomenon of fatigue which occurs, in general, beneath surfaces subjected to repeated stresses. Under good lubrication conditions where the rolling element and raceway surfaces are completely separated by an oil film and surface damage can be disregarded, a_3 is set to be 1.
However, when conditions of lubrication are not good, namely, when the viscosity of the lubricating oil is low or the peripheral speed of the rolling elements is especially low, and so on, $a_3 < 1$ is used.
On the other hand, when lubrication is especially good, a value of $a_3 > 1$ can be used. When lubrication is not good and $a_3 < 1$ is used, the life adjustment factor a_2 cannot generally exceed 1.

When selecting a bearing according to the basic dynamic load rating, it is recommended that a suitable value for reliability factor a_1 is chosen for each application. The selection should be made using the (C/P) or f_h values determined by machine type and based upon the actual conditions of lubrication, temperature, mounting, etc., which have already been experienced and observed in the same type of machines.

Limiting conditions

These bearing life equations are applicable only when the bearing is mounted and lubricated normally without intrusion of foreign materials and not used under extreme operating conditions.
Unless these conditions are satisfied, the life may be shortened. For example, it is necessary to separately consider the effects of bearing mounting errors, excessive deformation of housing and shaft, centrifugal force acting on rolling elements at high-speed revolution, excessive preload, especially large radial internal clearance of radial bearings, etc.
When the dynamic equivalent load exceeds 1/2 of the basic dynamic load rating, the life equations may not be applicable.

Correction of basic dynamic load rating for temperature and hardness

Temperature factor

The operating temperature for each bearing is determined according to its material and structure. If special heat treatment is performed, bearings can be used at temperatures higher than +150°C. As the allowable contact stress gradually decreases when the bearing temperature exceeds 150°C, the basic dynamic load rating is lowered and can be obtained by the following equation:

$$C_t = f_t\, C \cdots\cdots\cdots (7)$$

where, C_t : Basic dynamic load rating considering temperature rise, N
f_t : Temperature factor (See Fig. 4.)
C : Basic dynamic load rating, N

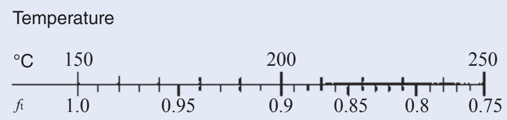


Fig. 4 Temperature factor

Further, if the bearing is used at high temperature, i.e. 120°C or above, the amount of dimensional displacement gets larger. So special heat treatment is necessary. If needed, please contact IKO.

Hardness factor

When the shaft or housing is used as the raceway surface instead of the inner or outer ring, the surface hardness of the part used as the raceway surface should be 58 ～ 64HRC.
If it is less than 58HRC, the basic dynamic load rating is lowered and can be obtained by the following equation:

$$C_H = f_H\, C \cdots\cdots\cdots (8)$$

where, C_H : Basic dynamic load rating considering hardness, N
f_H : Hardness factor (See Fig. 5.)
C : Basic dynamic load rating N

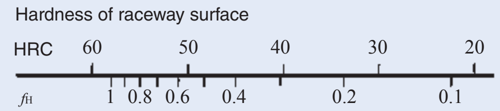


Fig. 5 Hardness factor

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Basic Static Load Rating and Static Safety Factor

Basic static load rating

When a bearing at rest sustains a heavy load or a bearing rotating at a relatively low speed receives a heavy shock load, the contact stress may exceed a certain limiting value, producing a local permanent deformation in the raceways or the rolling elements, and subsequently causing noise or vibration or lowering the rotating performance. The basic static load rating is, therefore, determined as a guideline for the maximum allowable load for the bearing at rest, under which the permanent deformation will not exceed a certain limit value, and the lowering of the rotating performance will not occur. Its definition is given as follows.

The basic static load rating is the static load that gives the contact stress shown in Table 3 at the center of the contact area of the rolling element and the raceway receiving the maximum load. A radial load constant in direction and magnitude is used in the case of radial bearings, while an axial load constant in magnitude acting along the bearing central axis is used in the case of thrust bearings.

Table 3

Type of bearing	Contact stress MPa
Roller bearings	4 000
Self-aligning ball bearings	4 600
Other ball bearings	4 200

Static safety factor

The basic static load rating gives the theoretical allowable limit of the static equivalent load. Normally, this limit is corrected by considering the operating conditions and the requirements for the bearing. The correction factor, namely, the static safety factor f_s is defined as in the following equation and its general values are shown in Table 4.

$$f_s = \frac{C_0}{P_0} \quad \cdots\cdots(9)$$

where, C_0 : Basic static load rating, N
P_0 : Static equivalent load, N

Table 4 Static safety factor

Operating conditions of the bearing	f_s
When high rotational accuracy is required	≧ 3
For ordinary operation conditions	≧ 1.5
For ordinary operation conditions not requiring very smooth rotation When there is almost no rotation	≧ 1

In case of Shell Type Needle Roller Bearings of which outer ring is drawn from a thin steel plate and then carburized and quenched, it is necessary to use a static safety factor of 3 or more.

Calculation of Bearing Loads

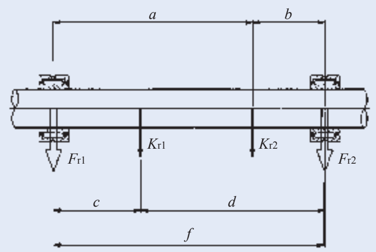
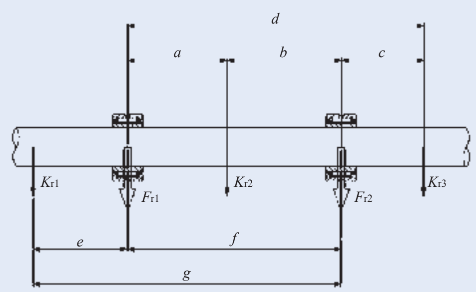
The loads acting on bearings include the weight of the machine parts supported by the bearings, the weight of the rotating body, loads produced when operating the machine, loads by belts or gears transmitting power, and various other loads.

These loads can be divided into radial loads perpendicular to the central axis of the bearings and axial loads parallel to the central axis, and they act independently or in combination with other loads. In addition, the magnitude of vibration or shocks on the bearings varies depending on the application of the machine. Thus, theoretically calculated loads may not always be accurate and have to be corrected by multiplying various empirical factors to obtain the actual bearing loads.

Load distribution to bearings

Table 5 shows examples of calculations where static loads are acting in radial direction.

Table 5 Load distribution to bearings

Example	Bearing load
	$F_{r1} = \dfrac{dK_{r1} + bK_{r2}}{f}$ $F_{r2} = \dfrac{cK_{r1} + aK_{r2}}{f}$
	$F_{r1} = \dfrac{gK_{r1} + bK_{r2} - cK_{r3}}{f}$ $F_{r2} = \dfrac{aK_{r2} + dK_{r3} - eK_{r1}}{f}$

Load factor

Although radial loads and axial loads can be obtained by calculation, it is not unusual for the actual bearing loads to exceed the calculated loads, due to vibration and shocks produced when operating the machine. The actual bearing load is obtained from the following equation, by multiplying the calculated load by the load factor:

$$F = f_w F_c \quad \cdots\cdots(10)$$

where, F : Bearing load, N
f_w : Load factor (See Table 6.)
F_c : Theoretically calculated load, N

Table 6 Load factor

Operating conditions	Example	f_w
Smooth operation without shocks	Electric motors, Air conditioning equipment, Measuring instruments, Machine tools	1 ～1.2
Ordinary operation	Reduction gearboxes, Vehicles, Textile machinery, Paper making machinery	1.2～1.5
Operation subjected to vibration and shocks	Rolling mills, Rock crushers, Construction machinery	1.5～3

Bearing loads in case of belt or chain transmission

When power is transmitted by a belt or chain, the load acting on the pulley or sprocket wheel is obtained from the following equations:

$$T = 9550000 \frac{H}{n} \quad \cdots\cdots (11)$$

$$K_t = \frac{T}{R} \quad \cdots\cdots (12)$$

where, T : Torque acting on pulley or sprocket wheel, N-mm
K_t : Effective transmitting force of belt or chain, N
H : Transmitting power, kW
n : Rotational speed, min^{-1}
R : Effective radius of pulley or sprocket wheel, mm

For belt transmission, the load K_r acting on the pulley shaft is obtained from the following equation, multiplying the effective transmitting force K_t by the belt factor f_b shown in Table 7.

$$K_r = f_b K_t \quad \cdots\cdots (13)$$

Table 7 Belt factor

Type of belt	f_b
V-belts	2 ～2.5
Timing belts	1.3～2
Plain belts (with tension pulley)	2.5～3
Plain belts	4 ～5

In the case of chain transmission, a value of 1.2 to 1.5 is taken as the chain factor corresponding to f_b. The load acting on the sprocket wheel shaft is obtained from equation (13) in the same manner as the belt transmission.

Bearing loads in case of gear transmission

When power is transmitted by gears, the force acting on the gears varies according to the type of gear. Spur gears produce radial loads only, but helical gears, bevel gears and worm gears produce axial loads in addition to radial loads. Taking the simplest case of spur gears as an example, the bearing load is obtained from the following equations:

$$T = 9550000 \frac{H}{n} \quad \cdots\cdots (14)$$

$$K_t = \frac{T}{R} \quad \cdots\cdots (15)$$

$$K_s = K_t \tan \theta \quad \cdots\cdots (16)$$

$$K_c = \sqrt{K_t^2 + K_s^2} = K_t \sec \theta \quad \cdots\cdots (17)$$

where, T : Torque applied to gear, N-mm
K_t : Tangential force acting on gear, N
K_s : Radial force acting on gear, N
K_c : Resultant normal force on gear tooth surface, N
H : Transmitting power, kW
n : Rotational speed, min^{-1}
R : Pitch circle radius of drive gear, mm
θ : Pressure angle of gear, deg.

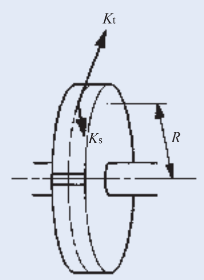


Fig. 6

In this case, the resultant normal force on the tooth surface acts as the radial force to the shaft and the magnitude of vibration or shocks varies depending on the accuracy and surface finish of the gear. Therefore, the radial load K_r applied to the shaft is obtained from the following equation, multiplying the resultant normal force K_c on gear tooth surface by the gear factor f_z shown in Table 8.

$$K_r = f_z K_c \quad \cdots\cdots (18)$$

Table 8 Gear factor

Type of gear	f_z
Precision gears (Pitch error and form error: Less than 0.02mm)	1.05～1.1
Ordinary machined gears (Pitch error and form error: 0.02～0.1mm)	1.1 ～1.3

Mean equivalent load corresponding to fluctuating load

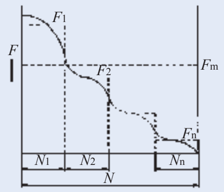
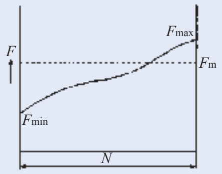
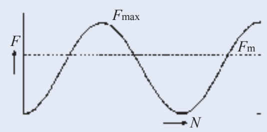
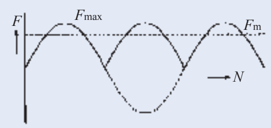
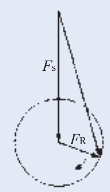
When the load applied to the bearing fluctuates, the bearing life is calculated by using the mean equivalent load F_m, which is a constant load that will give the bearing a life equal to that produced under the fluctuating load. The mean equivalent load is obtained from the following equation:

$$F_m = \sqrt[p]{\frac{1}{N}\int_0^N F_n^{\,p}\, dN} \quad \cdots\cdots (19)$$

where, F_m : Mean equivalent load, N
N : Total number of revolutions, rev.
F_n : Fluctuating load, N
p : Exponent, Roller bearing = 10/3
Ball bearing = 3

Table 9 shows examples of the calculation of mean equivalent loads for various fluctuating loads.

Table 9 Mean equivalent load for the fluctuation load

Type of fluctuating load		Mean equivalent load F_m
Step load		$F_m = \sqrt[p]{\frac{1}{N}(F_1^{\,p} N_1 + F_2^{\,p} N_2 + \cdots + F_n^{\,p} N_n)}$ where, N_1 : Total number of revolutions under load F_1 rev. N_2 : Total number of revolutions under load F_2 rev. N_n : Total number of revolutions under load F_n rev.
Monotonously changing load		$F_m = \frac{1}{3}(2F_{max} + F_{min})$ where, F_{max} : Maximum value of fluctuating load, N F_{min} : Minimum value of fluctuating load, N
Sinusoidally fluctuating load		$F_m \fallingdotseq 0.65 F_{max}$
		$F_m \fallingdotseq 0.75 F_{max}$
Stationary load plus rotating load		$F_m = F_S + F_R - \frac{F_S F_R}{F_S + F_R}$ where, F_S : Stationary load, N F_R : Rotating load, N

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Equivalent load

The loads applied to the bearing are divided into radial loads that are applied perpendicular to the central axis and axial loads that are applied in parallel to the central axis. These loads act independently or in combination with other loads.

Dynamic equivalent load

When both radial load and axial load are applied to the bearing simultaneously, the virtual load, acting on the center of the bearing, that will give a life equal to that under the radial load and the axial load is defined as a dynamic equivalent load.
In the case of needle roller bearings, radial bearings receive only radial loads and thrust bearings receive only axial loads. Accordingly, radial loads are directly used in the life calculation of the radial bearings, while axial loads are directly used for the thrust bearings.

[For radial bearings]

$$P_r = F_r \quad \cdots\cdots (20)$$

[For thrust bearings]

$$P_a = F_a \quad \cdots\cdots (21)$$

where, P_r : Dynamic equivalent radial load, N
P_a : Dynamic equivalent axial load, N
F_r : Radial load, N
F_a : Axial load, N

Static equivalent load

When both radial load and axial load are applied to the bearing simultaneously, the virtual load, acting on the center of the bearing, that will produce a maximum contact stress on the contact surface between the rolling element and the raceway equal to that given by the radial load and the axial load is defined as a static equivalent load.
In the case of needle roller bearings, radial bearings receive only radial loads and thrust bearings receive only axial loads. Accordingly, radial loads are directly used for the radial bearings, while axial loads are directly used for the thrust bearings.

[For radial bearings]

$$P_{0r} = F_r \quad \cdots\cdots (22)$$

[For thrust bearings]

$$P_{0a} = F_a \quad \cdots\cdots (23)$$

where, P_{0r} : Static equivalent radial load, N
P_{0a} : Static equivalent axial load, N
F_r : Radial load, N
F_a : Axial load, N

Boundary Dimensions and Identification Number

Boundary dimensions

Examples of symbols for quantities indicating the boundary dimensions of IKO Needle Roller Bearings are shown below. For details, see the table of dimensions for each model.

Machined Type Needle Roller Bearing

d : Nominal bearing bore diameter
D : Nominal bearing outside diameter
B : Nominal inner ring width
C : Nominal outer ring width
F_w : Nominal roller set bore diameter
r : Chamfer dimensions of inner and outer rings
$r_{s\,min}$: Smallest permissible single chamfer dimensions of inner and outer rings

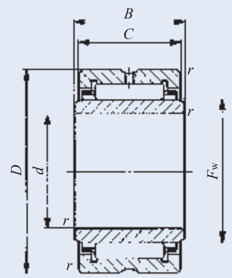


Fig. 7 Machined Type Needle Roller Bearing

Shell Type Needle Roller Bearing

D : Nominal bearing outside diameter
F_w : Nominal roller set bore diameter
C : Nominal outer ring width

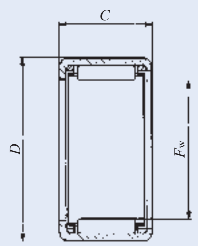


Fig. 8 Shell Type Needle Roller Bearing

Needle Roller Cage

E_w : Nominal roller set outside diameter
F_w : Nominal roller set bore diameter
B_c : Nominal cage width

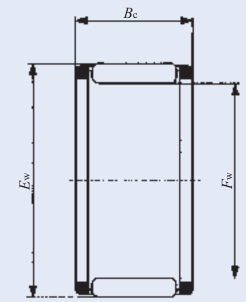


Fig. 9 Needle Roller Cage

Thrust Roller Bearing

D_c : Nominal cage outside diameter
d_c : Nominal cage bore diameter
D_w : Nominal roller diameter

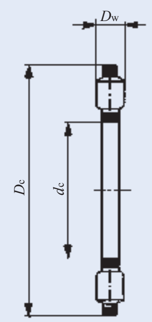


Fig. 10 Thrust Roller Bearing

Identification Number

The identification number of IKO Bearings consists of a model number and supplemental codes. The descriptions of typical codes and their arrangements are shown below. There are many codes other than those described. See the section of identification number of each bearing.

Table 10 Arrangement of identification number of bearing

Model number	Model code	❶
	Boundary dimensions	❷
Supplemental code	Material symbol	❸
	Cage symbol	❹
	Shield symbol Seal symbol,	❺
	Bearing ring shape symbol	❻
	Clearance symbol	❼
	Classification symbol	❽

❶Model code

The model code represents the bearing series. The features of each bearing series are shown on pages A5 to A15.

❷Boundary dimensions

One of the following four kinds of presentation methods is used for showing boundary dimensions in the identification number, which vary depending on the bearing series. Table 11 shows the presentation methods of boundary dimensions for each model code.

(a)Dimension series + Bore diameter number
(b)Bore diameter or roller set bore diameter + Outside diameter or roller set outside diameter + Width
(c)Bore diameter or roller set bore diameter + Width
(d)Basic diameter

❸Material symbol

Symbol	Type of material
F	Stainless steel for bearing rings and rolling elements

❹Cage symbol

Symbol	Descriptions
N	Made of synthetic resin
V	No cage or full complement

❺Seal or shield symbol

Symbol	Descriptions
Z	With dust cover
ZZ	With shields on both sides
U	With a seal on one side
UU	With seals on both sides
2RS	With seals on both sides

❻Bearing ring shape symbol

Symbol	Descriptions
NR	With stop ring on outer surface of outer ring
OH (1)	With oil hole in bearing ring
J	No oil hole

Note(1) This differs depending on the type of bearing. See the section of each bearing.

❼Clearance symbol

Symbol	Descriptions
C2	C2 clearance
(None)	CN clearance
C3	C3 clearance
C4	C4 clearance
C5	C5 clearance
T1	Special radial clearance (Applicable to Crossed Roller Bearings)
C1	
C2	

❽Classification symbol

Symbol	Descriptions
(None)	JIS Class 0
P6	JIS Class 6
P5	JIS Class 5
P4	JIS Class 4

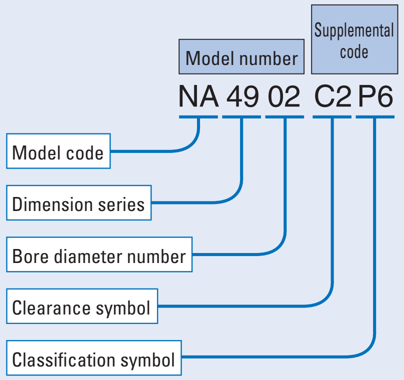
Table 11 Indication of boundary dimensions

Bearing type	Model number	
	Model code	Indication of boundary dimensions
Shell Type Needle Roller Bearings	TA, TLA, YT, YTL	Roller set bore diameter + Outer ring width
	BA, BHA, YB, YBH	Roller set bore diameter + Outer ring width (1)
Needle Roller Cages for General Usage	KT, KTW	Roller set bore diameter + Roller set outside diameter + Cage width
Needle Roller Cages for Engine Connecting Rods	KT···EG, KTV···EG	Roller set bore diameter + Roller set outside diameter + Cage width
Machined Type Needle Roller Bearings	NA, RNA	Dimension series + Bore diameter number
	TR, TAF, GTR	Roller set bore diameter + Bearing outside diameter + Bearing width
	TRI, TAFI, GTRI	Bearing bore diameter + Bearing outside diameter + Outer ring width
	BR	Roller set bore diameter + Bearing outside diameter + Bearing width (1)
	BRI	Bearing bore diameter + Bearing outside diameter + Outer ring width (1)
Needle Roller Bearings with Separable Cage	RNAF, RNAFW	Roller set bore diameter + Bearing outside diameter + Bearing width
	NAF, NAFW	Bearing bore diameter + Bearing outside diameter + Bearing width
Roller Bearings	NAU, NAG, NAS	Dimension series + Bore diameter number
	TRU	Bearing bore diameter + Bearing outside diameter + Bearing width
Thrust Bearings	NTB, AS, WS, GS	Bearing bore diameter + Bearing outside diameter
	AZ	Bearing bore diameter + Bearing outside diameter + Bearing height
	AZK	Bearing bore diameter + Bearing outside diameter + Roller diameter
Combined Type Needle Roller Bearings	NAX, NBX	Roller set bore diameter + Assembled bearing width
	NAXI, NBXI	Innerring bore diameter + Assembled bearing width
	NATA, NATB	Dimensional series + Bore diameter number
Cam Followers	CF, NUCF, CFS	Stud diameter
	CFKR	Bearing outside diameter
	CR, CRH	Bearing outside diameter (1)
Roller Followers	NAST, NART, NURT	Bearing bore diameter
	CRY	Bearing outside diameter (1)
Crossed Roller Bearings	CRBH(V), CRB, CRBS, CRBT	Bearing bore diameter + Bearing width
Spherical Bushings	SB···A, GE	Inner ring bore diameter
	SBB	Inner ring bore diameter (1)
PILLOBALLs	PB,PHS,POS,PHSB,POSB,PHSA	Inner ring bore diameter
L-Balls	LHSA, LHS	Screw size
Seals for Needle Roller Bearings	OS, DS	Shaft diameter + Seal outside diameter + Seal width
Cir-clips for Needle Roller Bearings	WR	Shaft diameter
	AR	Bore diameter

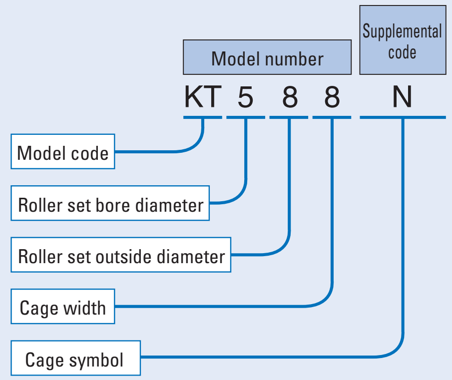
Note(1) The nominal dimensions of inch series bearings are indicated in units of 1/16 inch.

Example of identification number

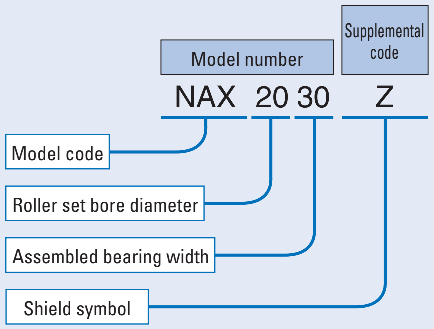
(a) Example of "Dimension series + Bore diameter number"



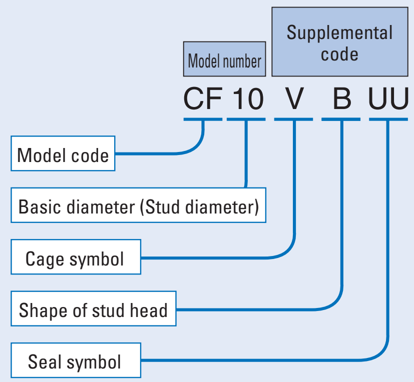
(b) Example of "Bore diameter or roller set bore diameter + Outside diameter or roller set outside diameter + width"



(c) Example of "Bore diameter or roller set bore diameter + width"



(d) Example of "Basic diameter"



Accuracy

The accuracy of IKO Needle Roller Bearings conforms to JIS B 1514-1～-3 (Rolling bearings - Tolerances of bearings), and the dimensional accuracy and rotational accuracy are specified. The specified items are shown in Fig. 11.

Needle Roller Bearings are classified into 4 classes of accuracy. These classes are represented by the numbers 0, 6, 5 and 4, written in order of increasing accuracy.

Table 12 shows the accuracy for the inner rings of radial bearings, Table 13 shows the accuracy for the outer rings of radial bearings, Table 14 shows the tolerances for the smallest single roller set bore diameter of radial bearings, and Table 15 shows the permissible limit values of chamfer dimensions of radial bearings. For thrust bearings, see the section on accuracy of Thrust Bearings. Note that the series of Shell Type Needle Roller Bearings, Roller Bearings, Cam Followers, Roller Followers, Combined Type Needle Roller Bearings, and Crossed Roller Bearings have special accuracy. For further details, see the section on accuracy of each bearing series.

Remarks

The meanings of the new symbols for quantities used for accuracy of radial bearings are as follows:

① Δ represents the deviation of a dimension from the specified value.

② V represents the variation of a dimension.

③ Suffixes $_s$, $_m$, and $_p$ represent a single (or actual) measurement, a mean measurement, and a measurement in a single radial plane, respectively.

[Example] V_{dp} means the difference between the largest and the smallest of the bore diameters in a single radial plane (circularity).

V_{dmp} means the difference between the largest and the smallest of the single plane mean bore diameters (cylindricity).

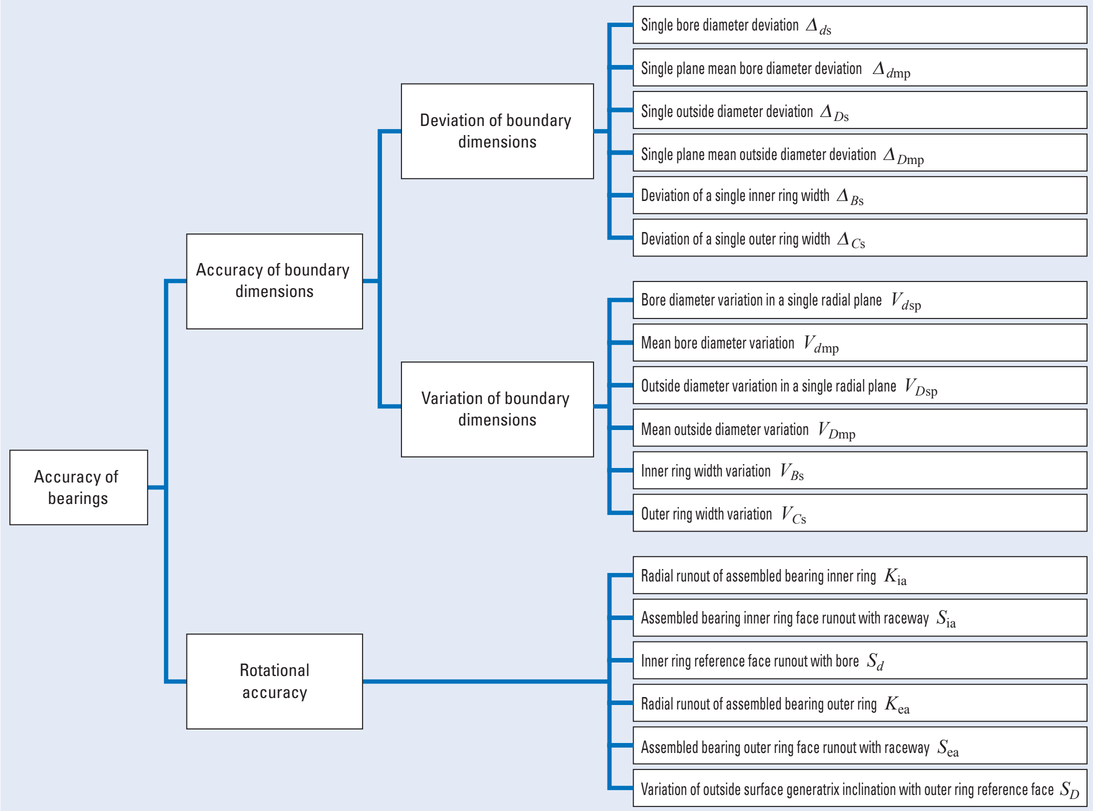


Fig. 11 Accuracy of bearings

Table 12 Tolerances for inner ring

unit: μm

d Nominal bearing bore diameter mm Over	Incl.	Δ_{dmp} Single plane mean bore diameter deviation Class 0 High	Class 0 Low	Class 6 High	Class 6 Low	Class 5 High	Class 5 Low	Class 4 High	Class 4 Low	Δ_{ds} Single bore diameter deviation Class 4 High	Class 4 Low	V_{dsp} Bore diameter variation in a single radial plane, Diameter series 8, 9(1) Class 0 Max.	Class 6	Class 5	Class 4	V_{dsp} Diameter series 0(2) Class 0 Max.	Class 6	Class 5	Class 4	V_{dmp} Mean bore diameter variation Class 0 Max.	Class 6	Class 5	Class 4
2.5	10	0	- 8	0	- 7	0	- 5	0	- 4	0	- 4	10	9	5	4	8	7	4	3	6	5	3	2
10	18	0	- 8	0	- 7	0	- 5	0	- 4	0	- 4	10	9	5	4	8	7	4	3	6	5	3	2
18	30	0	- 10	0	- 8	0	- 6	0	- 5	0	- 5	13	10	6	5	10	8	5	4	8	6	3	2.5
30	50	0	- 12	0	- 10	0	- 8	0	- 6	0	- 6	15	13	8	6	12	10	6	5	9	8	4	3
50	80	0	- 15	0	- 12	0	- 9	0	- 7	0	- 7	19	15	9	7	19	15	7	5	11	9	5	3.5
80	120	0	- 20	0	- 15	0	- 10	0	- 8	0	- 8	25	19	10	8	25	19	8	6	15	11	5	4
120	180	0	- 25	0	- 18	0	- 13	0	- 10	0	- 10	31	23	13	10	31	23	10	8	19	14	7	5
180	250	0	- 30	0	- 22	0	- 15	0	- 12	0	- 12	38	28	15	12	38	28	12	9	23	17	8	6
250	315	0	- 35	0	- 25	0	- 18					44	31	18		44	31	14		26	19	9	
315	400	0	- 40	0	- 30	0	- 23					50	38	23		50	38	18		30	23	12	
400	500	0	- 45	0	- 35							56	44			56	44			34	26		
500	630	0	- 50	0	- 40							63	50			63	50			38	30		
630	800	0	- 75																				
800	1000	0	- 100																				
1000	1250	0	- 125																				
1250	1600	0	- 160																				
1600	2000	0	- 200																				

K_{ia} Radial runout of assembled bearing inner ring Class 0 Max.	Class 6	Class 5	Class 4	S_d Inner ring reference face runout with bore Class 5 Max.	Class 4	S_{ia}(3) Assembled bearing inner ring face runout with raceway Class 5 Max.	Class 4	Δ_{Bs} Deviation of a single inner ring width Class 0 High	Class 0 Low	Class 6 High	Class 6 Low	Class 5 High	Class 5 Low	Class 4 High	Class 4 Low	V_{Bs} Inner ring width variation Class 0 Max.	Class 6	Class 5	Class 4	d Nominal bearing bore diameter mm Over	Incl.
10	6	4	2.5	7	3	7	3	0	- 120	0	- 120	0	- 40	0	- 40	15	15	5	2.5	2.5	10
10	7	4	2.5	7	3	7	3	0	- 120	0	- 120	0	- 80	0	- 80	20	20	5	2.5	10	18
13	8	4	3	8	4	8	4	0	- 120	0	- 120	0	- 120	0	- 120	20	20	5	2.5	18	30
15	10	5	4	8	4	8	4	0	- 120	0	- 120	0	- 120	0	- 120	20	20	5	3	30	50
20	10	5	4	8	5	8	5	0	- 150	0	- 150	0	- 150	0	- 150	25	25	6	4	50	80
25	13	6	5	9	5	9	5	0	- 200	0	- 200	0	- 200	0	- 200	25	25	7	4	80	120
30	18	8	6	10	6	10	7	0	- 250	0	- 250	0	- 250	0	- 250	30	30	8	5	120	180
40	20	10	8	11	7	13	8	0	- 300	0	- 300	0	- 300	0	- 300	30	30	10	6	180	250
50	25	13		13		15		0	- 350	0	- 350	0	- 350			35	35	13		250	315
60	30	15		15		20		0	- 400	0	- 400	0	- 400			40	40	15		315	400
65	35							0	- 450	0	- 450					50	45			400	500
70	40							0	- 500	0	- 500					60	50			500	630
80								0	- 750							70				630	800
90								0	- 1000							80				800	1000
100								0	- 1250							100				1000	1250
120								0	- 1600							120				1250	1600
140								0	- 2000							140				1600	2000

Note(1) Applicable to all series except NAS series
(2) Applicable to NAS series
(3) Applicable to NATA and NATB series

Table 13 Tolerances for outer ring

unit: μm

D Nominal bearing outside diameter mm Over	Incl.	Δ_{Dmp} Single plane mean outside diameter deviation Class 0 High	Class 0 Low	Class 6 High	Class 6 Low	Class 5 High	Class 5 Low	Class 4 High	Class 4 Low	Δ_{Ds} Single outside diameter deviation Class 4 High	Class 4 Low	V_{Dsp}(1) Outside diameter variation in a single radial plane, Open bearing, Diameter series 8, 9(2) Class 0 Max.	Class 6	Class 5	Class 4	Open bearing, Diameter series 0(3) Class 0 Max.	Class 6	Class 5	Class 4	Bearing with seal or shield, Diameter series 0(3) Class 6 Max.
2.5	6	0	- 8	0	- 7	0	- 5	0	- 4	0	- 4	10	9	5	4	8	7	4	3	9
6	18	0	- 8	0	- 7	0	- 5	0	- 4	0	- 4	10	9	5	4	8	7	4	3	9
18	30	0	- 9	0	- 8	0	- 6	0	- 5	0	- 5	12	10	6	5	9	8	5	4	10
30	50	0	- 11	0	- 9	0	- 7	0	- 6	0	- 6	14	11	7	6	11	9	5	5	13
50	80	0	- 13	0	- 11	0	- 9	0	- 7	0	- 7	16	14	9	7	13	11	7	5	16
80	120	0	- 15	0	- 13	0	- 10	0	- 8	0	- 8	19	16	10	8	19	16	8	6	20
120	150	0	- 18	0	- 15	0	- 11	0	- 9	0	- 9	23	19	11	9	23	19	8	7	25
150	180	0	- 25	0	- 18	0	- 13	0	- 10	0	- 10	31	23	13	10	31	23	10	8	30
180	250	0	- 30	0	- 20	0	- 15	0	- 11	0	- 11	38	25	15	11	38	25	11	8	
250	315	0	- 35	0	- 25	0	- 18	0	- 13	0	- 13	44	31	18	13	44	31	14	10	
315	400	0	- 40	0	- 28	0	- 20	0	- 15	0	- 15	50	35	20	15	50	35	15	11	
400	500	0	- 45	0	- 33	0	- 23					56	41	23		56	41	17		
500	630	0	- 50	0	- 38	0	- 28					63	48	28		63	48	21		
630	800	0	- 75	0	- 45	0	- 35					94	56	35		94	56	26		
800	1000	0	- 100	0	- 60							125	75			125	75			
1000	1250	0	- 125																	
1250	1600	0	- 160																	
1600	2000	0	- 200																	
2000	2500	0	- 250																	

V_{Dmp} Mean outside diameter variation Class 0 Max.	Class 6	Class 5	Class 4	K_{ea} Radial runout of assembled bearing outer ring Class 0 Max.	Class 6	Class 5	Class 4	S_D Variation of outside surface generatrix inclination with outer ring reference face Class 5 Max.	Class 4	S_{ea}(4) Assembled bearing outer ring face runout with raceway Class 5 Max.	Class 4	Δ_{Cs} Deviation of a single outer ring width Class 0, 6, 5, 4 High	Low	V_{Cs} Outer ring width variation Class 0 Max.	Class 6	Class 5	Class 4	D Nominal bearing outside diameter mm Over	Incl.
6	5	3	2	15	8	5	3	8	4	8	5	Same as the tolerance values of Δ_{Bs} for d of the same bearing		Same as the tolerance values of V_{Bs} for d of the same bearing		5	2.5	2.5	6
6	5	3	2	15	8	5	3	8	4	8	5					5	2.5	6	18
7	6	3	2.5	15	9	6	4	8	4	8	5					5	2.5	18	30
8	7	4	3	20	10	7	5	8	4	8	5					5	2.5	30	50
10	8	5	3.5	25	13	8	5	8	4	10	5					6	3	50	80
11	10	5	4	35	18	10	6	9	5	11	6					8	4	80	120
14	11	6	5	40	20	11	7	10	5	13	7					8	5	120	150
19	14	7	5	45	23	13	8	10	5	14	8					8	5	150	180
23	15	8	6	50	25	15	10	11	7	15	10					10	7	180	250
26	19	9	7	60	30	18	11	13	8	18	10					11	7	250	315
30	21	10	8	70	35	20	13	13	10	20	13					13	8	315	400
34	25	12		80	40	23		15		23						15		400	500
38	29	14		100	50	25		18		25						18		500	630
55	34	18		120	60	30		20		30						20		630	800
75	45			140	75													800	1000
				160														1000	1250
				190														1250	1600
				220														1600	2000
				250														2000	2500

Note(1) Classes 0 and 6 are applicable to outer rings without stop rings.
(2) Applicable to all series except NAS series
(3) Applicable to NAS series
(4) Applicable to NATA and NATB series

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Table 14 Tolerances for smallest single roller set bore diameter $F_{ws\,min}$(1) unit: μm

F_w Nominal roller set bore diameter mm		$\Delta_{Fws\,min}$ Deviation of smallest single roller set bore diameter	
Over	Incl.	High	Low
3	6	+ 18	+ 10
6	10	+ 22	+ 13
10	18	+ 27	+ 16
18	30	+ 33	+ 20
30	50	+ 41	+ 25
50	80	+ 49	+ 30
80	120	+ 58	+ 36
120	180	+ 68	+ 43
180	250	+ 79	+ 50
250	315	+ 88	+ 56
315	400	+ 98	+ 62
400	500	+ 108	+ 68

Note(1) This is the diameter of the cylinder used instead of the inner ring, where the radial clearance becomes 0 at least in one radial direction.

Table 15 Permissible limit values for chamfer dimensions of radial bearings unit: mm

$r_{s\,min}$ Smallest permissible single chamfer dimension	d Nominal bore diameter		$r_{s\,max}$ Largest permissible single chamfer dimension	
	Over	Incl.	Radial direction	Axial direction
0.1	—	—	0.55 (2)	0.55 (2)
0.15	—	—	0.6 (2)	0.6
0.2	—	—	0.7 (2)	0.8
0.3	—	40	0.8 (2)	1
	40	—	0.8	1
0.4 (1)	—	—	0.8	1.2
0.6	—	40	1.1 (2)	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 (1)	—	100	3.8	6
	100	280	4.5	6
	280	—	5	7
3	—	280	5	8
	280	—	5.5	8
4	—	—	6.5	9
5	—	—	8	10
6	—	—	10	13

Note(1) Not specified in JIS.
(2) The numeric value differs from JIS.
Remark Although the exact shape of the chamfer is not specified, its profile in the axial plane must not extend beyond the imaginary circular arc of radius $r_{s\,min}$ which is tangential to the inner ring side surface and bearing bore surface or to the outer ring side surface and bearing outside surface. (See Fig. 12.)

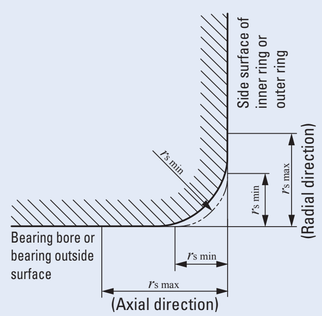
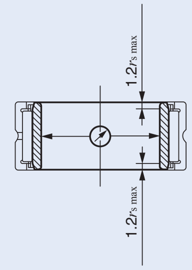
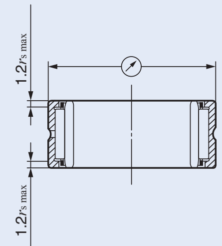


Fig. 12 Permissible values for chamfer dimensions

Methods of Measurement

Measurement of IKO Needle Roller Bearings is based on JIS B 1515-1, -2 (Rolling bearings-Tolerances). Tables 16 and 17 show some examples of the methods.
Special methods are used to measure Shell Type Needle Roller Bearings. Therefore, refer to the section on accuracy for these bearings on page B3.

Table 16 Measurement methods of accuracy of boundary dimensions

Measurement methods		Accuracy and definitions	
Single bore diameter	Zero the gauge indicator to the appropriate size using gauge blocks or a master ring. In several angular directions and in a single radial plane, measure and record the largest and the smallest single bore diameters, $d_{sp\,max}$ and $d_{sp\,min}$, within the measuring zone (excluding the zone 1.2 times the respective maximum allowable chamfer dimensions of the inner ring face). Repeat angular measurements and recordings in several radial planes to determine the largest and the smallest single bore diameter, $d_{s\,max}$ and $d_{s\,min}$. 	d_{mp} Mean bore diameter in a single plane	Calculated mean value of the maximum and minimum values of the single bore diameter within a radial plane. $d_{mp}=\frac{d_{sp\,max}+d_{sp\,min}}{2}$ d_{sp} : Single bore diameter in a single plane
		Δ_{dmp} Deviation of mean bore diameter in a single plane	Deviation of mean bore diameter in a single plane and nominal bore diameter. $\Delta_{dmp}=d_{mp}-d$ d : Nominal bearing bore diameter
		V_{dsp} Variation of bore diameter in a single plane	Deviation of the maximum and minimum values of the single bore diameter within a radial plane. $V_{dsp}=d_{sp\,max}-d_{sp\,min}$
		V_{dmp} Variation of mean bore diameter	Deviation of the maximum and minimum values of mean bore diameter in a single plane, for individual raceway rings with essentially cylindrical bore diameter surfaces. $V_{dmp}=d_{mp\,max}-d_{mp\,min}$
		Δ_{ds} Deviation of a single bore diameter	Deviation of single bore diameter and nominal bore diameter. $\Delta_{ds}=d_s-d$ d_s : Single bore diameter (distance between two parallel straight lines touching the intersection of the single bore diameter surface and the radial plane)
Single outside diameter	Zero the gauge indicator to the appropriate size using gauge blocks or a master. In several angular directions and in a single radial plane, measure and record the largest and the smallest single outside diameters, $D_{sp\,max}$ and $D_{sp\,min}$, within the measuring zone (excluding the zone 1.2 times the respective maximum allowable chamfer dimensions of the outer ring face). Repeat and record measurements in several radial planes to determine the largest and the smallest single outside diameter, $D_{s\,max}$ and $D_{s\,min}$. 	D_{mp} Mean outside diameter in a single plane	Calculated mean value of the maximum and minimum values of the single outside diameter within a radial plane. $D_{mp}=\frac{D_{sp\,max}+D_{sp\,min}}{2}$ D_{sp} : Single outside diameter in a single plane
		Δ_{Dmp} Deviation of mean outside diameter in a single plane	Deviation of mean outside diameter in a single plane and nominal outside diameter. $\Delta_{Dmp}=D_{mp}-D$ D : Nominal bearing outside diameter
		V_{Dsp} Variation of outside diameter in a single plane	Deviation of the maximum and minimum values of the single outside diameter within a radial plane. $V_{Dsp}=D_{sp\,max}-D_{sp\,min}$
		V_{Dmp} Variation of mean outside diameter	Deviation of the maximum and minimum values of mean outside diameter in a single plane, for individual raceway rings with essentially cylindrical outside diameter surfaces. $V_{Dmp}=D_{mp\,max}-D_{mp\,min}$
		Δ_{Ds} Deviation of a single outside diameter	Deviation of single outside diameter and nominal outside diameter. $\Delta_{Ds}=D_s-D$ D_s : Single outside diameter (distance between two parallel straight lines touching the intersection of the single outside diameter surface and the radial plane)

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

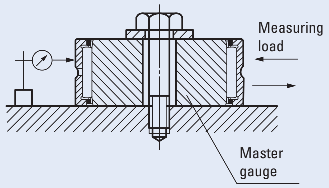
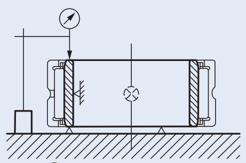
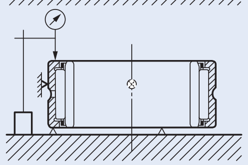
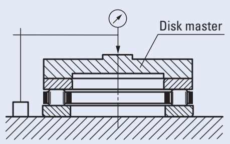
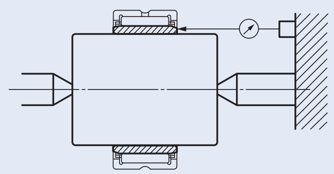
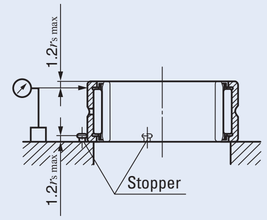
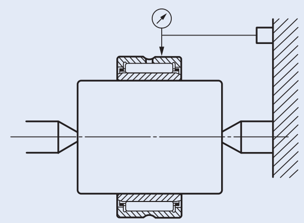
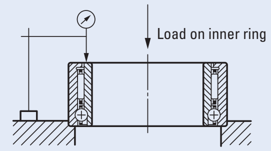
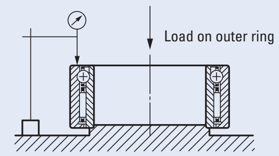
Measurement methods		Accuracy and definitions	
Single bore diameter of rolling element complement	Fasten the master gauge to a surface plate. Position the bearing on the master gauge and apply the indicator in the radial direction near the middle of the width on the ring outside surface. Measure the amount of movement of the outer ring in the radial direction by applying sufficient load on the outer ring in the same radial direction as that of the indicator and in the opposite radial direction. Record indicator readings at the extreme radial positions of the outer ring. Rotate the bearing and repeat the measurement in several different angular positions to determine the largest and the smallest readings, $F_{ws\,max}$ and $F_{ws\,min}$. 	F_{ws} Single bore diameter of roller complement	In radial bearings without inner rings, the distance between two parallel straight lines touching the intersection of the inscribed circle of roller complement and the radial plane.
		$F_{ws\,min}$ Smallest single bore diameter of roller complement	In radial bearings without inner rings, the minimum value of the single bore diameter of roller complement. Remark: Diameter of a cylinder where the smallest single bore diameter of roller complement has zero radial clearance in at least one radial direction.
Inner ring width	Zero the gauge indicator to the appropriate height from the reference surface using gauge blocks or a master. Support one face of the inner or outer ring on three equally spaced fixed supports of equal height and provide two suitable radial supports on the bore or outside surface set at 90° to each other to centre the inner or outer ring. Position the indicator against the other face of the ring opposite one fixed support. Rotate the inner or outer ring one revolution and measure and record the largest and the smallest single inner (outer) ring width, $B_{s\,max}$ and $B_{s\,min}$, ($C_{s\,max}$ and $C_{s\,min}$).	Δ_{Bs} Deviation of a single inner ring width	Deviation of single inner ring width and nominal inner ring width. $\Delta_{Bs}=B_s-B$
		V_{Bs} Variation of inner ring width	Deviation of the maximum and minimum values of the single inner ring width for individual inner rings. $V_{Bs}=B_{s\,max}-B_{s\,min}$
Outer ring width	 	Δ_{Cs} Deviation of a single outer ring width	Deviation of single outer ring width and nominal outer ring width. $\Delta_{Cs}=C_s-C$
		V_{Cs} Variation of outer ring width	Deviation of the maximum and minimum values of the single outer ring width for individual outer rings. $V_{Cs}=C_{s\,max}-C_{s\,min}$
Bearing height	Support the bearing on a surface plate. Zero the gauge indicator to an appropriate height from the surface plate using gauge blocks or a master. Place a plate of known thickness on the bearing assembly, apply a dynamically stable coaxial load, and position the indicator over the centre of the plate. Rotate the housing washer several times, to be sure to reach the smallest height, and take indicator readings. 	Δ_{Ts} Deviation of the actual bearing height	Deviation of actual bearing height and nominal bearing height of the thrust bearing. $\Delta_{Ts}=T_s-T$ T_s : Actual bearing height T : Nominal bearing height

Table 17 Measurement methods for rotational accuracy

Accuracy	Measurement methods	
S_d **Perpendicularity of inner ring face with respect to the bore**	Use a precision arbor having a taper of approximately 1 : 5 000 on diameter. Mount the bearing assembly on the tapered arbor and place the arbor between two centres so that it can be accurately rotated. Position the indicator against the reference face of the inner ring at a radial distance from the arbor axis of half the mean diameter of the face. Take indicator readings while rotating the inner ring one revolution.	
S_D **Perpendicularity of outer ring outside surface with respect to the face**	Support the reference face of the outer ring on a surface plate leaving the inner ring, if an assembled bearing, free. Locate the outer ring cylindrical outside surface against two supports set at 90° to each other to centre the outer ring. Position the indicator directly above one support. The indicator and the two supports are axially located at the extremes of the measurement zone (positions 1.2 times the respective maximum allowable chamfer dimensions of the outer ring face). Take indicator readings while rotating the outer ring one revolution.	
K_{ia} **Radial runout of inner ring of assembled bearing**	Use a precision arbor having a taper of approximately 1 : 5 000 on diameter. Mount the bearing assembly on the tapered arbor and place the arbor between two centres so that it can be accurately rotated. Position the indicator against the outside surface of the outer ring as close as possible to the middle of the outer ring raceway. Hold the outer ring to prevent rotation but ensure its weight is supported by the rolling elements. Take indicator readings while rotating the arbor one revolution.	
K_{ea} **Radial runout of outer ring of assembled bearing**	Use a precision arbor having a taper of approximately 1 : 5 000 on diameter. Mount the bearing assembly on the tapered arbor and place the arbor between two centres so that it can be accurately rotated. Position the indicator against the outside surface of the outer ring as close as possible to the middle of the outer ring raceway. Hold the inner ring stationary. Take indicator readings while rotating the outer ring one revolution.	
S_{ia} **Axial runout of inner ring of assembled bearing**	Support the reference face of the outer ring on a surface plate with a pilot for centering the outside diameter of the ring. Apply a dynamically stable coaxial load to the reference face of the inner ring in order to ensure contact between rolling elements and raceways. Position the indicator against the reference face of the inner ring and take indicator readings while rotating the inner ring one revolution.	
S_{ea} **Axial runout of outer ring of assembled bearing**	Support the reference face of the inner ring on a surface plate with a pilot for centering in the bore of the inner ring. Apply a dynamically stable coaxial load to the reference face of the outer ring in order to ensure contact between rolling elements and raceways. Position the indicator against the reference face of the outer ring and take indicator readings while rotating the outer ring one revolution.	

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Clearance

The clearances between the bearing rings and rolling elements are known as bearing clearances. When either the inner or outer ring is fixed and a specified measuring load is applied to the free bearing ring inward and outward alternately in the radial direction, the displacement of the free bearing is referred to as the radial internal clearance. The amount of measuring load in this case is extremely small, and its values are specified in JIS B 1515-2 (Rolling bearings-Tolerances-Part2:Measuring and gauging principles and methods).

❶Table 18 shows the radial internal clearances of Needle Roller Bearings with Inner Ring based on JIS B 1520 (Rolling bearings-Radial internal clearance). The radial internal clearances are classified into C2, CN, C3, C4, and C5, with clearances increasing in this order. CN is used under normal operating conditions. When a smaller range in radial internal clearance than the values shown in Table 18 is required, please consult IKO .

❷In the case of Shell Type Needle Roller Bearings, the correct dimensional accuracy is achieved only after the bearings are press-fitted into the specified housing bore. Therefore, the clearances shown in Table 18 are not applicable. See page B5.

❸For the radial internal clearances of Cam Followers, Roller Followers and Crossed Roller Bearings, see the relevant section for each bearing.

Table 18 Radial internal clearances of Needle Roller Bearings

unit: μm

d Nominal bore diameter mm		Classification of clearances									
		C2		CN		C3		C4		C5	
Over	Incl.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
—	10	0	25	20	45	35	60	50	75	—	—
10	24	0	25	20	45	35	60	50	75	65	90
24	30	0	25	20	45	35	60	50	75	70	95
30	40	5	30	25	50	45	70	60	85	80	105
40	50	5	35	30	60	50	80	70	100	95	125
50	65	10	40	40	70	60	90	80	110	110	140
65	80	10	45	40	75	65	100	90	125	130	165
80	100	15	50	50	85	75	110	105	140	155	190
100	120	15	55	50	90	85	125	125	165	180	220
120	140	15	60	60	105	100	145	145	190	200	245
140	160	20	70	70	120	115	165	165	215	225	275
160	180	25	75	75	125	120	170	170	220	250	300
180	200	35	90	90	145	140	195	195	250	275	330
200	225	45	105	105	165	160	220	220	280	305	365
225	250	45	110	110	175	170	235	235	300	330	395
250	280	55	125	125	195	190	260	260	330	370	440
280	315	55	130	130	205	200	275	275	350	410	485
315	355	65	145	145	225	225	305	305	385	455	535
355	400	100	190	190	280	280	370	370	460	510	600
400	450	110	210	210	310	310	410	410	510	565	665
450	500	110	220	220	330	330	440	440	550	625	735

Remark For bearings with CN clearance, no symbol is attached to the identification number. In the case of bearings with C2, C3, C4 and C5 clearances, these symbols are attached to the identification number.
Example NA 4905 C2

Selection of clearance

Radial clearances of needle roller bearings change according to bearing fit, temperature difference between bearing rings and rolling elements, loads, etc., and these factors greatly influence bearing life, accuracy, noise, generation of heat, etc. If radial clearances are too large, noise and vibration will increase, and if they are too small, abnormally great forces are exerted on the contact areas between raceways and rolling elements, resulting in abnormally high heat generation and a decrease in bearing life. Therefore, in the ideal case, the clearance provided before mounting should be such that it will become zero or slightly larger when the bearing has reached steady-state operation and the temperature has become constant (saturation temperature). However, it is difficult to achieve this ideal state for all bearings. Under general operating conditions, bearings with CN clearance are most widely used, and are manufactured to provide satisfactory performance when fitted according to Tables 21 and 22.
When radial internal clearances other than CN are used, refer to Table 19.

Table 19 Examples of selecting radial internal clearances other than CN clearance

Operating conditions	Selection of clearance
When heavy loads and shock loads are applied, and amount of interference is great.	C3 or larger clearance
When directionally indeterminate loads are applied, and a tight fit is required for both inner and outer rings.	
When temperature of inner ring is much higher than that of outer ring.	
When shaft deflection and/or mounting error to the housing are great.	
When less noise and vibration are required. When a loose fit is required for both inner and outer rings. When preload is required.	C2 or smaller clearance

Reduction of radial clearances by fit

When the inner or outer rings are interference fitted onto shafts and into housings, respectively, they expand or shrink due to elastic deformation. As the result, the radial clearances are reduced. These reduced radial clearances are called residual (internal) clearances.
The amount of reduction is obtained by the following equation, and it is generally 70 to 90% of the interference amount.

$$\Delta_C = \Delta_F + \Delta_E \quad \cdots\cdots (24)$$

where, Δ_C : Amount of reduction of the radial clearance, mm
Δ_F : Amount of expansion of the outside diameter of inner ring, mm
Δ_E : Amount of shrinkage of the bore diameter of outer ring, mm

❶**Amount of expansion of the outside diameter of inner ring**

· With solid shaft

$$\Delta_F = \Delta_{de}\frac{d}{F} \quad \cdots\cdots (25)$$

· With hollow shaft

$$\Delta_F = \Delta_{de}\frac{d}{F}\,\frac{1-(d_i/d)^2}{1-(d/F)^2\,(d_i/d)^2} \quad \cdots\cdots (26)$$

where, Δ_{de} : Effective interference of inner ring, mm
d : Bore diameter of inner ring, mm
F : Outside diameter of inner ring, mm
d_i : Bore diameter of hollow shaft, mm

❷**Amount of shrinkage of the bore diameter of outer ring**

· With steel housing $(D_0=\infty)$

$$\Delta_E = \Delta_{De}\frac{E}{D} \quad \cdots\cdots (27)$$

· With steel housing $(D_0 \neq \infty)$

$$\Delta_E = \Delta_{De}\frac{E}{D}\,\frac{1-(D/D_0)^2}{1-(E/D)^2\,(D/D_0)^2} \quad \cdots\cdots (28)$$

where, Δ_{De} : Effective interference of outer ring, mm
D : Outside diameter of outer ring, mm
E : Bore diameter of outer ring, mm
D_0 : Outside diameter of housing, mm

Reduction of radial clearances due to temperature differences between inner and outer rings

Frictional heat generated by rotation is dissipated through the shafts and housings as well as through oil and air. Under general operating conditions, heat dissipation is larger on the housing side compared with that on the shaft side, and the temperature of the outer ring is usually lower than that of the inner ring. During operation, the temperature of the rolling elements is the highest, followed by that of the inner ring and that of the outer ring. The amount of thermal expansion, therefore, varies, and the radial clearances are reduced. This reduced radial clearance is called the effective (internal) clearance, and the amount of reduction is obtained by the following equation:

$$\Delta\delta = \alpha\Delta_t E \quad \cdots\cdots (29)$$

where, $\Delta\delta$: Reduction of radial clearance, mm
α : Coefficient of linear expansion for bearing steel
$\fallingdotseq 12.5 \times 10^{-6}$ 1/°C
Δ_t : Temperature difference between the outer ring and the inner ring plus rolling elements considered as one unit, °C
E : Bore diameter of outer ring, mm

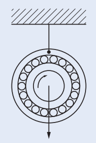
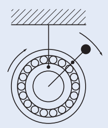
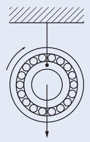
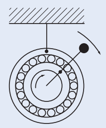
The temperature difference Δ_t is considered to be 5 ~ 10°C under normal operating conditions and 15 ~ 20°C at high rotational speeds. Therefore, when the temperature difference is great, a correspondingly larger radial internal clearance must be selected.

Fit

Purpose of fit

To achieve the best performance of needle roller bearings, it is important that the bearing rings are correctly fitted onto the shaft and into the housing.
The purpose of fit is to provide the appropriate amount of interference required between the inner ring and the shaft or between the outer ring and the housing, to prevent harmful mutual slippage.
If the interference is insufficient, it will cause a harmful relative displacement, known as creep, between the fitted surfaces in the circumferential direction. This may lead to abnormal wear of fitted surfaces, intrusion of wear particles into the bearing, generation of abnormal heat, vibration, etc. Therefore, a suitable fit must be selected.

Table 20 Nature of radial load and fit

Nature of the load		Rotating conditions	Fit: Inner ring	Fit: Outer ring
Rotating load on inner ring Stationary load on outer ring		Inner ring : Rotating Outer ring : Stationary Load direction : Fixed	Interference fit	Clearance fit
		Inner ring : Stationary Outer ring : Rotating Load direction : Rotating with outer ring		
Rotating load on outer ring Stationary load on inner ring		Inner ring : Stationary Outer ring : Rotating Load direction : Fixed	Clearance fit	Interference fit
		Inner ring : Rotating Outer ring : Stationary Load direction : Rotating with inner ring		
Directionally indeterminate load	The load direction is not fixed, including cases where the load direction is fluctuating or there is an unbalanced load.	Inner ring : Rotating or stationary Outer ring : Rotating or stationary Load direction : Not fixed	Interference fit	Interference fit

Conditions for determination of fit

When determining a suitable fit for a bearing, it is necessary to consider various conditions such as nature and magnitude of the load, temperature, required rotational accuracy, material/finish grade/thickness of the shaft and housing, ease of mounting and dismounting, etc.

❶ Nature of load and fit
Basically, the appropriate fit depends on whether the load direction is rotational or stationary in relation to the inner and outer rings.
The relationship between the nature of radial loads and the fit is, in general, based on Table 20.

❷ Load amount and interference
The greater the load, the larger the interference must be.
When selecting an interference between the inner ring and the shaft, it is necessary to estimate the reduction of interference due to the radial load. The amount of reduction of interference is obtained by the following equations.

· When $F_r \leqq 0.2C_0$

$$\Delta_{dF} = 0.08\sqrt{\frac{d}{B}F_r} \times 10^{-3} \quad \cdots\cdots (30)$$

· When $F_r > 0.2C_0$

$$\Delta_{dF} = 0.02\frac{F_r}{B} \times 10^{-3} \quad \cdots\cdots (31)$$

where, F_r : Radial load applied to bearing, N
C_0 : Basic static load rating, N
Δ_{dF} : Amount of reduction of inner ring interference, mm
d : Bore diameter of inner ring, mm
B : Width of inner ring, mm

❸ Temperature conditions and change of interference
The interference of fitted surfaces is also influenced by the temperature difference between the bearing and the shaft and housing. For example, when steam is flowing through a hollow shaft, or when the housing is made of light metal, it is necessary to take into consideration the differences in temperature, the coefficient of linear expansion and other such factors.
Usually, the interference of the inner ring decreases as the bearing temperature increases during operation. If the temperature difference between the inside of the bearing and the outside of the housing is taken as Δ_T, the temperature difference between the inner ring and the shaft can be estimated to be (0.1 ~ 0.15) Δ_T. Accordingly, the amount of reduction of the inner ring interference is obtained by the following equation.

$$\Delta_{dT} = (0.1 \sim 0.15)\Delta_T \alpha d \fallingdotseq 0.0015\Delta_T d \times 10^{-3} \quad \cdots (32)$$

where, Δ_{dT} : Reduction amount of inner ring interference due to temperature difference, mm
Δ_T : Temperature difference between the inside of the bearing and the outside of the housing, °C
α : Coefficient of linear expansion for bearing steel
$\fallingdotseq 12.5 \times 10^{-6}$ 1/°C
d : Bore diameter of inner ring, mm

❹ Shaft finish grade and interference
Since peaks of surface roughness of the fitted surface are crushed down when fitting the bearing, the effective interference becomes smaller than the apparent interference obtained by measurements, and it is generally obtained by the following equations.
· For ground shaft

$$\Delta_{de} = \frac{d}{d+2}\Delta_{df} \quad \cdots\cdots (33)$$

· For machined shaft

$$\Delta_{de} = \frac{d}{d+3}\Delta_{df} \quad \cdots\cdots (34)$$

where, Δ_{de} : Effective interference of inner ring, mm
d : Bore diameter of inner ring, mm
Δ_{df} : Apparent interference, mm

❺ Minimum interference and maximum interference
When the load direction is rotating in relation to the inner ring, the inner ring is fitted with interference to the shaft.
For solid ground steel shafts, the minimum interference (required apparent interference) Δ_{df} is expressed by the following equation which is deduced from equations (30) or (31), (32) and (33).

$$\Delta_{df} \geqq \frac{d+2}{d}(\Delta_{dF} + 0.0015\Delta_T d \times 10^{-3}) \quad \cdots (35)$$

It is desired that the maximum interference should be less than 1/1000 of the shaft diameter. In the case of the outer ring, the effective interference varies according to the housing material, thickness, shape, etc., so it is determined empirically.

Selection of fit

When selecting a suitable fit, in addition to the various conditions mentioned above, it is necessary to draw on experience and practical results.
Tables 21 and 22 show the most general fit data.
When a thin housing or a hollow shaft is used, the interference is made larger than an ordinary fit.
The fit between needle roller bearings without inner ring and shafts is based on Table 23.
For the fit between Shell Type Needle Roller Bearings and housing bores, see page B5.
For the fit between inner rings for Shell Type Needle Roller Bearings and shafts, see Table 22.

Table 21 Fit between needle roller bearings and housing bores (Not applicable to Shell Type Needle Roller Bearings)

Operating conditions		Tolerance class of housing bore (1)	Application examples (Reference)
Rotating load on outer ring	Heavy load on thin housing, large shock load	P7 (2)	Flywheels
	Heavy load, normal load	N7 (2)	Wheel bosses, transmission gears
	Light load, fluctuating load	M7	Pulleys, tension pulleys
Directionally indeterminate load	Large shock load	M7	Eccentric wheels, pumps
	Heavy load, normal load	K7	Compressors
	Normal load, light load	J7	Crankshafts, compressors
Stationary load on outer ring	Shock load, heavy load	J7	General bearing applications, gear shafts
	Normal load, light load	H7	General bearing applications
	With heat conduction through shaft	G7	Paper dryers
Light load, normal load, requirements of high-precision rotation and high rigidity		K6	Main spindles of machine tools

Notes(1) This table applies to steel or cast iron housings. For lighter metal, a tighter fit should be selected. For split housings, do not use a fit tighter than J7.
(2) Care should be taken so that the radial internal clearance is not too small.
Remark Light load, normal load and heavy load represent $P \leqq 0.06C$, $0.06C < P \leqq 0.12C$, and $0.12C < P$, respectively, where P is the dynamic equivalent radial load and C is the basic dynamic load rating of the bearing to be used.

Table 22 Fit between needle roller bearings with inner ring and shafts

Operating conditions		Shaft dia. mm Over	Shaft dia. mm Incl.	Tolerance class of shaft (1)	Application examples (Reference)
Stationary load on inner ring	Light load, normal load, low or medium rotating speed	All shaft diameters		g6	Wheels on dead axles Control lever gears Rope sheaves Tension pulleys
	Heavy load, medium rotating speed			h6	
	Especially smooth operation and accuracy are required.			h5	
Rotating load on inner ring or Directionally indeterminate load	Light load	— 50 100 200	50 100 200 —	j5 k5 m6 (2) n6 (3)	Electric appliances, Precision machinery Machine tools, Pumps Blowers, Transportation vehicles
	Normal load	— 50 150 200	50 150 200 —	k5 (4) m5, m6 (2) n6 (3) p6 (3)	General bearing applications Pumps, Transmission gearboxes, Wood working machinery, Internal combustion engines
	Heavy load Shock load	— 150	150 —	n6 (3) p6 (3)	Industrial vehicles, Construction machinery Crushers

Notes(1) This table applies to solid steel shafts.
(2) It is necessary to examine the reduction of radial internal clearances caused by the expansion of inner rings after mounting.
(3) It is necessary to use bearings with radial internal clearances greater than CN clearance.
(4) For NATA and NATB, do not use a tighter fit than k5.

Table 23 Tolerance class of shafts assembled with needle roller bearings without inner ring

F_w Nominal roller set bore diameter mm		Radial internal clearance		
		Smaller than CN clearance	CN clearance	Larger than CN clearance
Over	Incl.	Tolerance class of shaft (1)		
—	65	k5	h5	g6
65	80	k5	h5	f6
80	160	k5	g5	f6
160	180	k5	g5	e6
180	200	j5	g5	e6
200	250	j5	f6	e6
250	315	h5	f6	e6
315	—	g5	f6	d6

Note(1) When the housing bore fit is tighter than K7, the shaft diameter is made smaller by considering shrinkage of roller set bore diameter after mounting.

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Table 24 Fit values for radial bearings (JIS Class 0) (Fit with housing bore) unit: μm

D Nominal outside diameter mm		Δ_{Dmp} Single plane mean outside diameter deviation		G7	H7	J7	K6	K7	M7	N7	P7
Over	Incl.	High	Low								
3	6	0	- 8	- 24 ～ - 4	- 20 ～ 0	-14 ～ 6	-10 ～ 6	-11 ～ 9	- 8 ～ 12	- 4 ～ 16	0 ～ 20
6	10	0	- 8	- 28 ～ - 5	- 23 ～ 0	-16 ～ 7	-10 ～ 7	-13 ～ 10	- 8 ～ 15	- 4 ～ 19	1 ～ 24
10	18	0	- 8	- 32 ～ - 6	- 26 ～ 0	-18 ～ 8	-10 ～ 9	-14 ～ 12	- 8 ～ 18	- 3 ～ 23	3 ～ 29
18	30	0	- 9	- 37 ～ - 7	- 30 ～ 0	-21 ～ 9	-11 ～ 11	-15 ～ 15	- 9 ～ 21	- 2 ～ 28	5 ～ 35
30	50	0	-11	- 45 ～ - 9	- 36 ～ 0	-25 ～ 11	-14 ～ 13	-18 ～ 18	-11 ～ 25	- 3 ～ 33	6 ～ 42
50	80	0	-13	- 53 ～ -10	- 43 ～ 0	-31 ～ 12	-17 ～ 15	-22 ～ 21	-13 ～ 30	- 4 ～ 39	8 ～ 51
80	120	0	-15	- 62 ～ -12	- 50 ～ 0	-37 ～ 13	-19 ～ 18	-25 ～ 25	-15 ～ 35	- 5 ～ 45	9 ～ 59
120	150	0	-18	- 72 ～ -14	- 58 ～ 0	-44 ～ 14	-22 ～ 21	-30 ～ 28	-18 ～ 40	- 6 ～ 52	10 ～ 68
150	180	0	-25	- 79 ～ -14	- 65 ～ 0	-51 ～ 14	-29 ～ 21	-37 ～ 28	-25 ～ 40	-13 ～ 52	3 ～ 68
180	250	0	-30	- 91 ～ -15	- 76 ～ 0	-60 ～ 16	-35 ～ 24	-43 ～ 33	-30 ～ 46	-16 ～ 60	3 ～ 79
250	315	0	-35	-104 ～ -17	- 87 ～ 0	-71 ～ 16	-40 ～ 27	-51 ～ 36	-35 ～ 52	-21 ～ 66	1 ～ 88
315	400	0	-40	-115 ～ -18	- 97 ～ 0	-79 ～ 18	-47 ～ 29	-57 ～ 40	-40 ～ 57	-24 ～ 73	1 ～ 98
400	500	0	-45	-128 ～ -20	-108 ～ 0	-88 ～ 20	-53 ～ 32	-63 ～ 45	-45 ～ 63	-28 ～ 80	0 ～ 108

Remark The negative value denotes a clearance and the positive value denotes an interference.

Table 25 Fit values for radial bearings (JIS Class 0) (Fit with shaft) unit: μm

d Nominal bore diameter mm		Δ_{dmp} Single plane mean bore diameter deviation		g6	h5	h6	j5	k5	m5	m6	n6	p6
Over	Incl.	High	Low									
3	6	0	- 8	-12 ～ 4	- 5 ～ 8	- 8 ～ 8	- 2 ～ 11	1 ～ 14	4 ～ 17	4 ～ 20	8 ～ 24	12 ～ 28
6	10	0	- 8	-14 ～ 3	- 6 ～ 8	- 9 ～ 8	- 2 ～ 12	1 ～ 15	6 ～ 20	6 ～ 23	10 ～ 27	15 ～ 32
10	18	0	- 8	-17 ～ 2	- 8 ～ 8	-11 ～ 8	- 3 ～ 13	1 ～ 17	7 ～ 23	7 ～ 26	12 ～ 31	18 ～ 37
18	30	0	-10	-20 ～ 3	- 9 ～ 10	-13 ～ 10	- 4 ～ 15	2 ～ 21	8 ～ 27	8 ～ 31	15 ～ 38	22 ～ 45
30	50	0	-12	-25 ～ 3	-11 ～ 12	-16 ～ 12	- 5 ～ 18	2 ～ 25	9 ～ 32	9 ～ 37	17 ～ 45	26 ～ 54
50	80	0	-15	-29 ～ 5	-13 ～ 15	-19 ～ 15	- 7 ～ 21	2 ～ 30	11 ～ 39	11 ～ 45	20 ～ 54	32 ～ 66
80	120	0	-20	-34 ～ 8	-15 ～ 20	-22 ～ 20	- 9 ～ 26	3 ～ 38	13 ～ 48	13 ～ 55	23 ～ 65	37 ～ 79
120 140 160	140 160 180	0	-25	-39 ～ 11	-18 ～ 25	-25 ～ 25	-11 ～ 32	3 ～ 46	15 ～ 58	15 ～ 65	27 ～ 77	43 ～ 93
180 200 225	200 225 250	0	-30	-44 ～ 15	-20 ～ 30	-29 ～ 30	-13 ～ 37	4 ～ 54	17 ～ 67	17 ～ 76	31 ～ 90	50 ～ 109
250 280	280 315	0	-35	-49 ～ 18	-23 ～ 35	-32 ～ 35	-16 ～ 42	4 ～ 62	20 ～ 78	20 ～ 87	34 ～ 101	56 ～ 123
315 355	355 400	0	-40	-54 ～ 22	-25 ～ 40	-36 ～ 40	-18 ～ 47	4 ～ 69	21 ～ 86	21 ～ 97	37 ～ 113	62 ～ 138
400 450	450 500	0	-45	-60 ～ 25	-27 ～ 45	-40 ～ 45	-20 ～ 52	5 ～ 77	23 ～ 95	23 ～ 108	40 ～ 125	68 ～ 153

Remark The negative value denotes a clearance and the positive value denotes an interference.

IKO

Design of Shaft and Housing

Accuracy and roughness of shaft and housing

Accuracy and roughness of fitting surface

Since the bearing rings of needle roller bearings are thin, their performance is easily affected by poor accuracy of shafts or housings. Under general operating conditions, the fitting surfaces of shafts and housings can be finished by lathe turning. However, when the load is great and high accuracy and low noise are required, a grinding finish is required.
Table 26 shows the accuracy and roughness of fitting surfaces for general use.

Accuracy and roughness of raceway surface

In case of needle roller bearings unlike other bearings, mating surfaces such as shaft and housing bore surfaces can be used directly as the raceway surfaces. For such use, accuracy and roughness of the raceway surfaces are important because they will influence bearing life, noise and accuracy.
In general, accuracy and roughness of raceway surfaces are based on Table 26.

Inclination of shaft

Shafts and outer rings may have some inclination between them due to deflection of the shaft, machining accuracy of shafts and housings, errors in mounting, etc.
In this case, the use of two or more bearings in tandem arrangement on a single shaft should be avoided. Instead, a bearing with large load ratings should be used.
It is recommended that inclination of shafts be less than 1/1000.

Table 27 Tolerance class IT values for basic dimensions

Basic dimension mm		Tolerance class (1)		
		IT5	IT6	IT7
Over	Incl.	Tolerance μm		
–	3	4	6	10
3	6	5	8	12
6	10	6	9	15
10	18	8	11	18
18	30	9	13	21
30	50	11	16	25
50	80	13	19	30
80	120	15	22	35
120	180	18	25	40
180	250	20	29	46
250	315	23	32	52
315	400	25	36	57
400	500	27	40	63
500	630	30	44	70

Note(1) Based on JIS B 0401.

Table 26 Specifications of shafts and housings for radial needle roller bearings

Item	Shaft		Housing bore	
	Fitting surface	Raceway surface	Fitting surface	Raceway surface
Circularity	0.3 × IT6 (1) or 0.3 × IT5 (1)	0.3 × IT6 (1) or 0.3 × IT5 (1)	0.3 × IT7 (1) or 0.3 × IT6 (1)	0.3 × IT7 (1) or 0.3 × IT6 (1)
Cylindricity	0.5 × IT6 (2) or 0.5 × IT5 (2)	0.3 × IT6 (1) or 0.3 × IT5 (1)	0.5 × IT7 (2) or 0.5 × IT6 (2)	0.3 × IT7 (1) or 0.3 × IT6 (1)
Surface roughness μmR_a (μmR_y)	0.8 (3.2)	0.2 (3) (0.8)	1.6 (6.3)	0.2 (3) (0.8)
Hardness	–	58 ～ 64HRC (4)	–	58 ～ 64HRC (4)

Notes(1) 30% or less of the dimensional tolerance for shafts or housing bores is recommended.
(2) 50% or less of the dimensional tolerance for shafts or housing bores is recommended.
(3) When required accuracy is not critical, a surface roughness within 0.8 μmR_a (3.2 μmR_y) is allowable.
(4) An appropriate thickness of the hardened layer is required.
Remark For tolerance class IT, see Table 27.

Raceway materials and heat treatment

When using shafts and housings as raceways, the following materials are generally used.

High-carbon chromium bearing steel		
	SUJ2	JIS G 4805
Carburizing steel	SCM415～421	JIS G 4053
Carburizing steel	SNCM 220	JIS G 4053
Carburizing steel	SCr 420	JIS G 4053
Carburizing steel	SNC 415、815	JIS G 4053
Carburizing steel	S 15 CK	JIS G 4051

In addition, S50C and S55C (JIS G 4051) can be used after through hardening or induction hardening.
The hardened layer produced by tempering at +160～+180°C after hardening must have a fine uniform martensite microstructure.
When hardening the raceway surface by case hardening or induction hardening, a surface hardness of 58～64HRC and an appropriate thickness of the hardened layer must be ensured. The minimum effective thickness of the hardened layer after heat treatment and grinding is defined as the distance from the surface to the depth where the hardness is 550HV, and it is obtained by the following equation.

$$E_{ht} \geqq 0.8D_w(0.1+0.002D_w) \quad \cdots\cdots\cdots\cdots\cdots(36)$$

where, E_{ht} : Minimum effective thickness of the hardened layer, mm
D_w : Roller diameter, mm

Generally, the required effective thickness of the hardened layer is at least 0.3 mm.

Dimensions related to mounting of bearings

The dimensions of shaft and housing related to mounting of the needle roller bearings are shown in the table of dimensions for each bearing. (See Fig. 13.)
The minimum value of the shaft shoulder diameter d_a which receives the inner ring, and the maximum value of the housing shoulder diameter D_a which receives the outer ring, represent the effective shoulder diameters (excluding the chamfered part) which make proper contact with the side faces of the inner and outer rings respectively.
Also, the maximum value of the shaft shoulder (or inner ring retaining piece) diameter d_a is the dimension related to the ease of mounting/dismounting of the shaft and inner ring to/from the housing and outer ring.
The largest permissible single corner radius $r_{as\ max}$ of the shaft and housing must be smaller than the smallest permissible single chamfer dimension $r_{s\ min}$ of the bearing so that the side surface of the bearing can make proper contact with the shoulder. Table 28 shows the related dimensions.
For dimensions of the fillet relief when finishing the shaft or housing by grinding, the values shown in Table 29 are recommended.
For other dimensions related to mounting, see the related section for each bearing as required.
In addition, for ease in dismounting of bearings, it is convenient to make notches in the shoulder of the shaft or housing to allow the insertion of dismounting hooks.

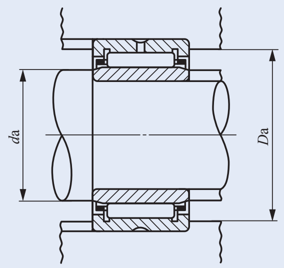


Fig. 13 Dimensions related to mounting

Table 28 Largest permissible single corner radius of shafts and housings $r_{as\ max}$

unit: mm

$r_{s\ min}$ Smallest permissible single chamfer dimension	$r_{as\ max}$ Largest permissible single corner radius of shafts and housings
0.1	0.1
0.15	0.15
0.2	0.2
0.3	0.3
0.4	0.4
0.6	0.6
1	1
1.1	1
1.5	1.5
2	2
2.1	2
2.5	2
3	2.5
4	3
5	4

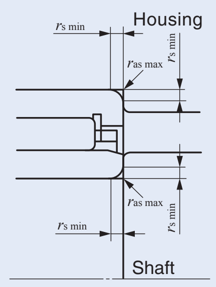
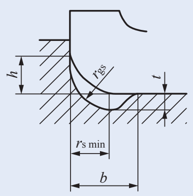


Table 29 Fillet relief dimensions for ground shafts and housings

unit: mm

$r_{s\ min}$ Smallest permissible single chamfer dimension	Fillet relief dimensions t	r_{gs}	b
1	0.2	1.3	2
1.1	0.3	1.5	2.4
1.5	0.4	2	3.2
2	0.5	2.5	4
2.1	0.5	2.5	4
3	0.5	3	4.7
4	0.5	4	5.9
5	0.6	5	7.4
6	0.6	6	8.6
7.5	0.6	7	10



Sealing

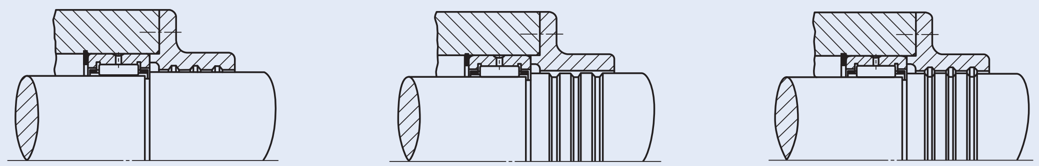
To obtain the best performance of rolling bearings, it is necessary to prevent leakage of lubricant and the entry of harmful foreign substances, such as dirt, dust and water. For this reason, sealing devices must always work effectively to seal and prevent against dust penetration under all operating conditions. Also, when selecting a suitable sealing method, it is necessary to consider such factors as the type of lubricant, peripheral speed of the seal, operating temperature, shaft eccentricity, seal friction, etc. as well as ease of assembly and disassembly.
Sealing methods are of the non-contact and contact types, and it is necessary to select the appropriate type depending on the application.

Non-contact type sealing method

There are many methods of non-contact type sealing, including the use of oil grooves, flingers and labyrinths, which utilize the centrifugal force and narrow gaps.
Since they do not make direct contact with the shaft or housing, it is unnecessary to consider friction and wear, and the non-contact sealing method is suitable for high speed rotation and high operating temperatures. However, because of gaps, this method is not always sufficient in preventing oil leakage and dust entry when the machine is not in operation.

❶ Oil groove

Oil grooves are provided on either the shaft or housing bore, or on both for more effective sealing (See Fig. 14.). The clearance between the shaft and the housing bore should be as small as possible, and the values shown in Table 30 are generally used, taking into consideration errors in machining and assembly, shaft deformation, etc. Three or more grooves are made with a width of 3～5 mm and a depth of 4～5 mm. If the grooves are filled with grease, it will be more effective for dust prevention.
As shown in Fig. 15, helical grooves are suitable for horizontal shafts which have a fixed direction of rotation. Right or left handed grooves are used according to the direction of rotation, and they are used for oil lubrication normally in conjunction with a suitable anti-dust device.



(1) (2) (3)

Fig. 14

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Table 30 Clearance between grooved shaft and housing bore unit: mm

Shaft dia.	Clearance
Incl. 50 mm	0.25～0.4
Over 50 mm	0.5 ～1

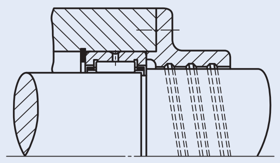


Fig. 15

❷ Flinger

The oil flinger is a disk attached to the shaft which throws off oil due to the centrifugal force of rotation and thus prevents oil leakage and the entry of foreign particles. Fig. 16 (1) shows an example in which the flinger is located inside the housing, mainly to prevent oil leakage. Since it sucks in dust and dirt, it should be used in a dust free environment. Fig. 16 (2) shows an example in which the flinger is located outside the housing, and is used in combination with another sealing device, to prevent entry of foreign particles.

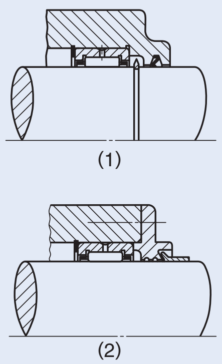


Fig. 16

❸ Labyrinth

Although it is a little difficult to make, the labyrinth is very effective in preventing oil leakage especially at high speeds. At low speeds, filling the labyrinth with grease is effective in preventing the entry of dust. In Fig. 17, it is necessary to split the housing or cover plate into two. In Fig. 18, it is easy to assemble, and if combined with an oil seal, it improves the sealing effect.
Table 31 shows the labyrinth clearances generally used.

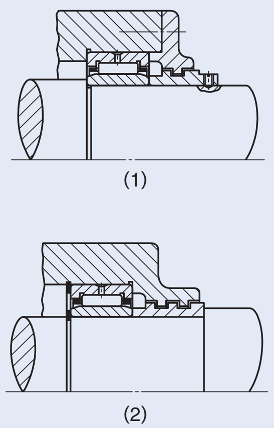


Fig. 17

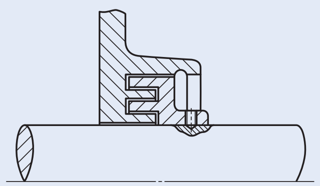


Fig. 18

Table 31 Labyrinth clearance unit: mm

Shaft dia.	Clearance	
	Radial direction	Axial direction
Incl. 50 mm	0.25～0.4	1～2
Over 50 mm	0.5 ～1	3～5

Contact type sealing method

In this type of sealing, the shaft is sealed by the application of pressure resulting from the elasticity of the seal material to the sealing surface of the shaft, which rotates, reciprocates or oscillates. Synthetic rubber, synthetic resin and felt are generally used as sealing materials.

❶ Oil seal

Synthetic rubber oil seals are the most general type of sealing used. The sealing effect is obtained when the elastic lip comes into contact with the shaft. Some lips are spring-loaded to maintain adequate pressing force.
The sliding surfaces of the lip and the shaft always show frictional behavior such that the boundary lubrication and fluid lubrication are mixed. If there is an insufficient amount of oil between the contact surfaces, it will cause heat generation, wear and seizure. Conversely, if the oil film is too thick, it may cause oil leakage.
General oil seals are specified in JIS B 2402-1~5. IKO Oil Seals for Needle Roller Bearings (See page L1.) have a low sectional height to match the Needle Roller Bearings.
Nitrile rubber is generally used as the material for oil seal lips. Table 32 shows the materials and their operating temperature ranges.
The finished surface of the shaft where the seal lip makes contact must have an appropriate surface roughness, as shown in Table 33, according to the peripheral speed. It must also have accurate circularity, and the shaft eccentricity should be less than 0.05 mm.
To increase wear resistance, the hardness of the sliding part of the shaft must be more than 40HRC. This can be achieved by hard-chrome plating or heat treatment.

Table 32 Seal materials and operating temperatures

Seal material		Operating temperature range ℃
Synthetic rubber	Nitrile rubber	−25～+120
	Acrylic rubber	−15～+130
	Silicon rubber	−50～+180
	Fluoro rubber	−10～+180
Tetrafluoethylene resin		−50～+220

Table 33 Peripheral speed and surface roughness of shaft

Peripheral speed m／s		Surface roughness
Over	Incl.	μmR_a (μmR_y)
—	5	0.8(3.2)
5	10	0.4(1.6)
10	—	0.2(0.8)

❷ Felt seal

Because of their simple structure, felt seals have long been used to protect grease lubrication from dust. Since felt absorbs some grease during operation, it hardly causes heat generation and seizure, but it cannot be used when the peripheral speed of the shaft is high (more than 4 m/s). Where there is a high concentration of dirt and dust, they may become attached to the contact surface of felt, sometimes scratching the shaft surface. To prevent this, two felt seals are placed apart from each other, or a felt seal is used together with a synthetic rubber seal.

Lubrication

Purpose of lubrication

The main purpose of bearing lubrication is to reduce friction and wear and to prevent heat generation and seizure. The lubricant and the lubricating method have a big influence on the operating performance of the bearing, and it is therefore necessary to select them suitably for the operating conditions.
The effects of lubrication are as follows.

❶ Reduction of friction and wear
At the contact surfaces between the race rings, rolling elements and cage of the bearing, lubrication prevents metal-to-metal contact, and reduces friction and wear due to sliding and rolling, in the latter of which micro-slips occur by differential slip, skew, spin, or elastic deformation.

❷ Elimination of frictional heat
The lubricant removes the heat generated by friction or transferred from outside, and prevents overheating of the bearing. Circulating lubrication is generally used for this purpose.

❸ Influence on bearing life
The bearing life is extended if the rolling contact surfaces between the race rings and rolling elements are separated by an oil film of adequate thickness, and is shortened if the oil film is inadequate due to low oil viscosity, etc.

❹ Rust prevention
The lubricant prevents rust formation on the inside and outside surfaces of the bearing.

❺ Dust prevention
Grease lubrication is particularly effective for dust prevention. Oil circulating or jet lubrication is effective in washing foreign particles away from the area around the bearing.

Methods of lubrication

Grease lubrication and oil lubrication are generally used for rolling bearings. In special cases, solid lubricants are also used.
In general, grease lubrication requires the simplest sealing device. It is therefore economical, and widely used. Also, once filled with grease, the bearing can be used for a long period without replenishing the grease. However, compared with oil, its heat removal properties and cooling capacity are inferior, since grease has high flow resistance, which causes high churning heat.
Oil has greater fluidity and superior heat removal properties. It is therefore suitable for high-speed operations. In addition, it is simple to filter out dust and dirt from oil. Thus it can prevent the generation of noise and vibration and increase bearing life.
Another advantage of oil lubrication is that it offers the possibility for selecting the appropriate method for particular operating conditions from among various available lubrication methods. However, measures to prevent oil leakage are required. As a guideline for selection, Table 34 compares grease and oil lubrication.
For the lubricants used for IKO Spherical Bushings, see page K8.

Table 34 Comparison between grease lubrication and oil lubrication

Item	Grease lubrication (1)	Oil lubrication
Sealing, Housing structure	Simple	Slightly complicated
Temperature	High temperature not allowed	High temperature allowed (Cooling effect by circulation)
Rotational speed	Low and medium speeds	High speed allowed
Load	Low and medium loads	High load allowed
Maintenance	Easy	Elaborate (Pay special attention to oil leaks.)
Lubricant replacement	Slightly complicated	Simple
Lubrication performance	Good	Very good
Dust filtration	Difficult	Simple
Entry of dust and dirt	Easy measures for protection	Dust and dirt can be removed by filtering in circulating lubrication.

Note(1) This represents bearing grease for general use.

Grease lubrication

❶ Amount of grease to be filled
The amount of grease to be filled depends on the housing structure, dimensions, type of grease used and atmosphere. Generally, filling about 1/3 to 1/2 of the free space inside of the bearing and the housing is considered to be appropriate. Too much will cause a rise in temperature, and care should be taken especially at high speed rotations.
In Fig. 19, several grease pockets are provided by the grease sectors on one side of the bearing. Even if the filled grease is dispersed by the centrifugal force at high rotational speeds, it is trapped by the grease pockets and diverted back into the bearing again. Old grease accumulates in the space on the opposite side of the bearing, and this can be removed periodically by taking off the cover.

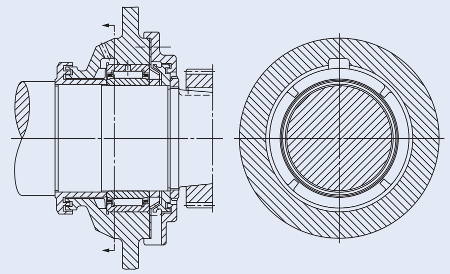
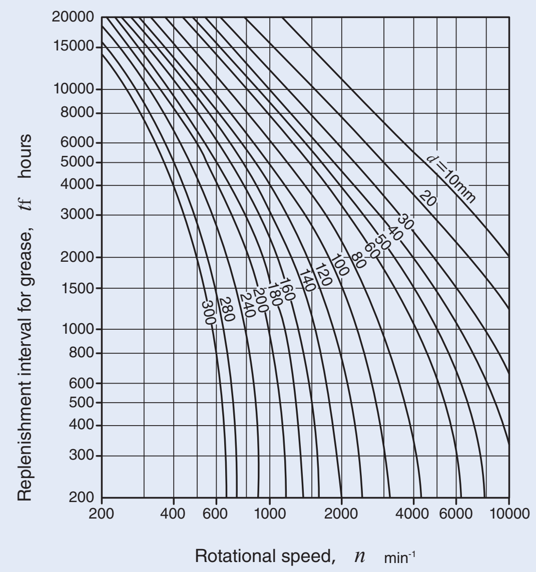


Fig. 19

❷ Replenishment of grease
The life of grease depends on its type and quality, the type and dimensions of the bearing, operating conditions, temperature, amount of wear, penetration of foreign particles and water, etc.
Fig. 20 shows the replenishment intervals for grease, and is used as a general guideline. The values obtained from this diagram apply to cases in which the load condition is normal, the machine body is stationary, and the operating temperature on the outer surface of bearing outer ring is less than +70°C. If the temperature exceeds +70°C, as a general rule, the replenishment interval is halved for every 15°C increase.



Remark
d is the bore diameter of the inner ring. For a bearing without an inner ring, the roller set bore diameter F_w is used for d.

Fig. 20 Interval for replenishment of grease

Oil lubrication

❶ Oil bath lubrication

This is the most commonly used oil lubrication method, and is used for medium and low speeds. If the amount of oil is too large, heat will be generated by churning, and if the amount is too small, seizure will occur. Therefore, the correct amount of oil must be maintained. When the machine is stationary, the correct oil level in the case of a bearing mounted on a horizontal shaft, is near the center of the lowest rolling element. In the case of a vertical shaft, about 50% of the surfaces of the rolling elements should be submerged in oil.

It is desirable to provide an oil gauge so that the oil level can be easily checked while the machine is stationary or running.

❷ Oil drip lubrication

Oil drips, which are fed down from a sight-feed oiler or along a fiber string, become an oil spray due to wind pressure generated by the rotating cage, shaft, nut, etc., or they strike the rotating parts and form an oil spray, which fills up the housing and every required part. Because oil spray removes frictional heat, this method has a more effective cooling effect than the oil bath method, and is widely used for high-speed rotation and medium load conditions.

In the case of the sight-feed oiler (Fig. 21), the number of drips can be adjusted. However, this is difficult using the string-feed method. The number of drips depends on the bearing type, rotational speed, etc., but 5 ～ 6 drips per minute is generally used.

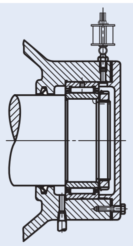


Fig. 21 Oil drip lubrication

❸ Oil splash lubrication

In this method, oil is splashed in all directions by the rotation of the gear or disk. This can be used for considerably high-speed rotations without soaking the bearing directly in oil.

In the gear case where shafts and bearings are lubricated with the same oil, wear particles may be introduced into the bearing as they might get mixed with the oil. In this case, a permanent magnet is provided at the bottom of the gear case to collect metal particles, or a shield plate is installed next to the bearing.

Fig. 22 shows another method in which the splashed oil flows along the grooves in the case and accumulates in the oil pockets, keeping the oil level constant. So the oil is steadily supplied to the bearing.

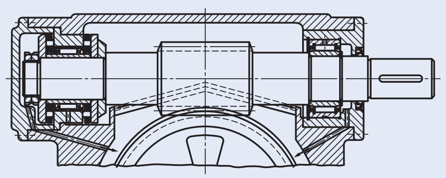


Fig. 22 Oil splash lubrication

❹ Oil circulating lubrication

When automatic lubrication is more economical because lubrication is required at many points, or when cooling is required for high rotational speed, this method is used. The oil is supplied with a pump, which can control the oil pressure, and a filter or cooler, etc. can be set up in the circulation system, making this an ideal method of lubrication. As shown in Fig. 23, the oil supply and discharge ports are located opposite to each other, and the discharge port is made large to prevent the accumulation of oil.

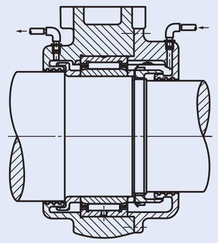


Fig. 23 Oil circulating lubrication

❺ Oil mist lubrication

After dirt and dust are removed by a filter, the oil is turned into a spray by dry compressed air, and this lubricates the bearing. When the air and oil pass through the bearing, the air cools the bearing and the oil lubricates it. In addition, because the air inside the housing is at a higher pressure than the outside air, the entry of water and foreign particles is prevented. There are many other advantages of this method, and it is suitable for high rotational speed applications such as high speed internal grinding spindles.

❻ Oil jet lubrication

This is a highly reliable lubrication method and is used under severe conditions such as ultra-high rotational speeds and high temperatures. The speed of the oil jet should be more than 20% of the peripheral speed of the inner ring raceway surface, since the air around the bearing rotates together with the bearing forming an air wall. As shown in Fig. 24, the jet from the nozzle blows directly into the space between the inner ring and the cage. Due to the large amount of oil being used, it is more effective to make the discharge port larger, and use the forced discharge.

When the $d_m n$ value (mean value of the bearing outside and bore diameters in millimeter x rotational speed in min^{-1}) is more than 1,000,000, the speed of the jet should be 10 ～ 20 m/s, the nozzle diameter should be about 1 mm, oil supply pressure should be 0.1 ～ 0.5 MPa, and the oil supply amount should be about 500 cc/min or greater. When the rotational speed is higher, the oil supply pressure and the oil amount should be higher.

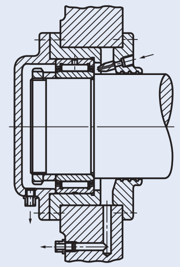


Fig. 24 Oil jet lubrication

Lubricants

For rolling bearings, lubricating grease or oil is generally used. For special applications, solid lubricants are used.

Lubricating grease

Grease is a semi-solid lubricant made by mixing base oil (liquid lubricant) and a thickener under heat and adding additives as required.

There are many types of grease according to various combinations of base oil, thickeners and additives. Grease is usually classified by thickeners and base oil. Table 35 shows the general properties of each type of grease.

Table 35 Properties of various types of grease

Item \ Name (Common name)	Calcium grease (Cup grease)	Sodium grease (Fiber grease)	Aluminum grease (Mobile grease)	Mixed base grease	Barium grease	Lithium grease			Non-soap base grease (Non-soap grease)	
							(Diester grease)	(Silicon grease)	(Bentone grease)	
Base oil	Mineral oil	Mineral oil	Mineral oil	Mineral oil	Mineral oil	Mineral oil	Diester oil	Silicon oil	Mineral oil	Synthetic oil
Thickener	Ca soap	Na soap	Al soap	Na + Ca soap, Li + Ca soap	Ba soap	Li soap	Li soap	Li soap	Bentone	Silica gel. Polyurea, etc.
Appearance	Buttery	Fibrous and buttery	Stringy and buttery	Fibrous and buttery	Fibrous and buttery	Buttery	Buttery	Buttery	Buttery	Buttery
Pour point ℃	80 ～ 90	150 ～ 180	70 ～ 90	160 ～ 190	150 ～ 180	170 ～ 190	170 ～ 190	200 ～ 250	200 ～	None
Operating temperature range ℃	− 10 ～+ 70	− 20 ～+ 120	− 10 ～+ 80	− 10 ～+ 100	− 10 ～+ 135	− 20 ～+ 120	− 50 ～+ 120	− 50 ～+ 180	− 10 ～+ 150	～+ 200
Pressure resistance	Strong to weak	Strong to medium	Strong	Strong	Strong to medium	Medium	Medium	Weak	Medium to weak	Medium
Water resistance	Good	Poor	Good	Good, poor for Na+Ca soap grease	Good	Good	Good	Good	Good	Good
Mechanical stability	Fair	Good	Poor	Good	Poor	Excellent	Excellent	Excellent	Good	Good to poor
Features and application	Contains about 1% water. When the temperature rises to more than +80°C, the water evaporates and the grease separates into oil and soap. This is used for medium loads.	Long fibrous grease cannot withstand high speeds, but has good pressure resistance properties. Short fibrous grease is comparatively good for high speeds.	It has water and rust resistant properties, and adheres easily to metal surface.	Usable at fairly high speeds.	It has water and heat resistant properties. This is an all-purpose grease.	This is the best all-purpose grease among soap based greases.	Excellent under low temperature conditions and has superior frictional properties. Suitable for small bearings used in measuring instruments.	Mainly used for high temperatures. Not suited to high speeds and heavy loads.	Generally good heat resistance. Grease having a mineral base oil is for general use. Grease having a synthetic base oil is suitable for special use where superior heat and chemical resistance properties are required.	

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

❶ Base oil
Petroleum lubricating oil is usually used as the base oil.
As the lubricating performance of grease depends mainly on that of base oil, the viscosity of the base oil is an important property. In general, low viscosity is suitable for light-load and high-speed rotations, and high viscosity for heavy-load and low-speed rotations. Synthetic lubricants of the diester or silicon series are used instead of lubricants of the petroleum series in consideration of the pour point and high temperature stability.

❷ Thickener
As shown in Table 35, metal soap bases are mostly used as thickeners. In particular, Na-soap is water-soluble and emulsifies easily, and it cannot be used in damp or wet areas. The type of thickener and the pour point of grease have a close relationship. In general, the higher the pour point, the higher the maximum usable temperature of grease. However, even when the grease uses a thickener having a high pour point, its upper operating temperature limit is low if its base oil has low heat resistance.

❸ Consistency
This represents the hardness grade of grease. Grease becomes harder in proportion to the amount of thickener if the same thickener is used.
Immediately after grease has been stirred (usually 60 times), a depression is formed in the grease in a specified time using a specified cone. The consistency (combined consistency) is expressed by the value of depth of depression (mm) multiplied by 10.
This value gives an estimate of the fluidity during operation with a greater value for softer grease.
Table 36 shows the consistency number of grease and the relationship between the consistency and operating conditions.

Table 36 Consistency and operating conditions of grease

NLGI consistency number	Combined consistency	Application
0	385 ~ 355	For centralized lubrication,
1	340 ~ 310	For oscillating motion
2	295 ~ 265	For general use
3	250 ~ 220	For general use, For high temperature
4	205 ~ 175	For sealing with grease

❹ Additives
Additives include various types of substances, which are added to grease in small quantities to improve its characteristics. For example, when a bearing is kept running for long periods of time, its temperature rises. This results in oxidation of the lubricant and formation of oxides, which lead to corrosion of the bearing.
Thus, when a bearing is to be operated for long periods of time without regreasing, antioxidants are added. In addition, grease containing extreme pressure additives is suitable for use in places that are subjected to heavy loads.

❺ Miscibility of different greases
In principle, it is desirable to use grease of the same brand. However, when the mixing of different greases is unavoidable, greases with the same type of thickener and with a similar type of base oil should be used.
It should be noted that if different types of grease are mixed, they may interact with each other and the consistency will become softer than that for the individual greases.

Lubricating oil

For rolling bearings, refined mineral oil or synthetic oil is used. To improve its properties, antioxidant additives, extreme pressure additives and detergent additives are added as required.
When selecting lubricating oil, it is important to select oil which has adequate viscosity under operating temperatures. If the viscosity is too low, the formation of the oil film will be insufficient, causing abnormal wear and seizure. On the other hand, if the viscosity is too high, it will generate excessive heat or increase power loss due to viscous resistance. As a general standard, oil having higher viscosity should be used for heavier loads and oil having lower viscosity should be used for higher rotational speeds.
Under conditions of normal use for various bearings, the values of viscosity shown in Table 37 will be a guideline.
The relationship between viscosity and temperature can be obtained from Fig. 25. Also, Table 38 shows examples of selecting lubricating oil according to the conditions of bearing use.

Table 37 Bearing series and required viscosity of lubricating oil

Bearing series	Kinematic viscosity at operating temperatures
Needle roller bearings Roller bearings	13 mm^2/s or more
Crossed roller bearings	20 mm^2/s or more
Thrust needle roller bearings Thrust roller bearings	32 mm^2/s or more

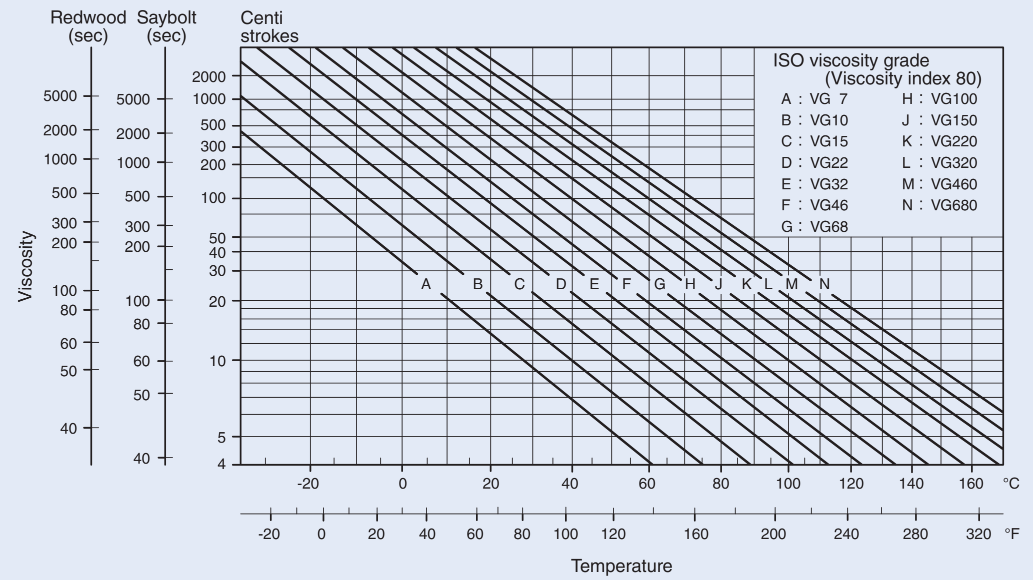


Fig. 25 Relationship between viscosity and temperature of lubricating oil

Table 38 Conditions of bearing use and examples of lubricating oil selection

Conditions		ISO viscosity grade(VG)											
		10	15	22	32	46	68	100	150	220	320	460	680
Operating temperature	−30 ~ 0℃		← Refrigerator oil	—	→								
	0 ~ 50℃	← Bearing oil	—	—	—	—	—	→					
					← Turbine oil	—	→						
	50 ~ 80℃				← Bearing oil	—	—	—	—	—	→		
					← Turbine oil	—	→						
	80 ~ 110℃						← Bearing oil	—	—	—	—	→	
							← Turbine oil →					← Gear oil	→
$d_{m}n$ value		Large ←	—	—	—	—	—	—	—	—	—	—	→ Small
Load		Small ←	—	—	—	—	—	—	—	—	—	—	→ Large

Remarks
· Lubricating oils are based on JIS K 2211 (Refrigerating machine oils), JIS K 2239 (Bearing Oil), JIS K 2213 (Turbine Oil), and JIS K 2219 (Gear Oil).
· The method of lubrication in these cases is mainly oil bath lubrication or circulating lubrication.
· When the temperature is on the high side within the operating temperature range, oils of high viscosity are used.
· $d_{m}n$ represents the mean value of the bore and outside diameters (mm) of the bearing multiplied by the rotational speed (min^{-1}).

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

C-Lube Bearing

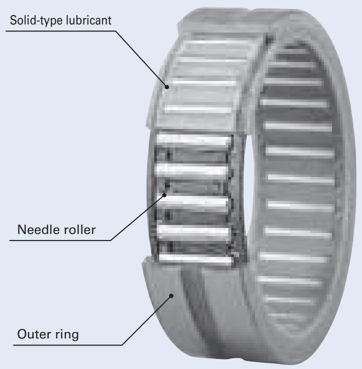
IKO C-Lube Bearing is a bearing that is lubricated with a newly developed thermosetting solid-type lubricant. A large amount of lubricating oil and fine particles of ultra high molecular weight polyolefin resin are solidified by heat treatment to fill the inner space of the bearing. As the bearing rotates, the lubricating oil oozes out onto the raceway in proper quantities, maintaining the lubrication performance for a long period of time.
The dimension tables for C-Lube Machined Type Needle Roller Bearings, C-Lube Cam Followers, and C-Lube Roller Followers are shown on pages D77, I49, and I95.
C-Lube Bearing is available in all Needle Roller Bearing series. Also C-Lube Bearings for food processing are available, using NSF H1-certified lubrication oil and resin compliant with FDA standards to mitigate any effect on human health. If needed, please contact IKO .

Features of C-Lube Bearing

- Most suitable for preventing grease dry-up in applications where lubrication is difficult.
- Great reduction of maintenance work by extending the lubrication interval.
- Elimination of oil contamination, making this bearing most suitable for applications that would be adversely affected by oil.

Cautions for using C-Lube Bearing

- Never wash C-Lube Bearing with organic solvent and/or white kerosene which have the ability to remove fat, or leave the bearing in contact with these agents.
- The operating temperature range is -15～+80°C. For continuous operation, the recommended operating temperature is +60°C or less.
- To ensure normal rotation of the bearing, apply a load of 1% or more of the basic dynamic load rating at use.
- The allowable rotational speed is different from that of the general needle roller bearings. For $d_m n$, $d_1 n$, and dn, use the values in Table 39 or less as guidelines.



Structure of C-Lube Bearing

Table 39 C-Lube Bearing $d_m n$, $d_1 n$, dn

Representative model		Allowable rotational speed
	Main model code	$d_m n$ (1)、$d_1 n$ (2)、dn (3)
C-Lube Machined Type Needle Roller Bearings	TAF･･･/SG	$d_m n$=20 000
C-Lube Cam Followers	CF･･･/SG	$d_1 n$ =10 000
C-Lube Roller Followers (4)	NART･･･/SG	dn = 8 000

Notes(1) $d_m n$ = (bore diameter of bearing [mm] + outside diameter of bearing [mm]) /2 × rotational speed [min⁻¹]
(2) $d_1 n$ = stud diameter [mm] ×rotational speed [min⁻¹]
(3) dn = inner ring bore diameter [mm] ×rotational speed [min⁻¹]
(4) The allowable rotational speed of C-Lube Roller Followers is applicable to use with oscillating rotation. For use with one-way or continuous rotation, please consult IKO.

Friction and Allowable Rotational Speed

Friction

Compared with sliding bearings, the starting (static) friction for rolling bearings is small, and the difference between the starting (static) friction and the kinetic friction is also small. The loss of power and temperature rise in machines are thus reduced, improving the mechanical efficiency.
Frictional torque is influenced by the bearing type, bearing load, rotational speed, lubricant characteristics, etc. It varies according to the lubricant when operated under light-loads and high-speed conditions, and according to the load when operated under heavy-loads and low-speed conditions.
Frictional torque of rolling bearings is complicated because it is influenced by various factors, but for convenience, it can be expressed approximately by the following equations.

· Radial bearings $$M = \mu P \frac{d}{2} \quad \cdots\cdots(37)$$

· Thrust bearings $$M = \mu P \frac{d_m}{2} \quad \cdots\cdots(38)$$

where, M : Frictional torque, N-mm
μ : Coefficient of friction
P : Bearing load, N
d : Bearing bore diameter, mm
d_m : Mean value of bearing bore and outside diameters, mm

The approximate coefficients of friction of IKO Bearings under operating conditions, in which lubrication and mounting are correct and where loads are relatively large and stable, are shown in Table 40.

Table 40 Coefficient of friction

Bearing series	μ
Needle roller bearings with cage	0.0010 ～ 0.0030
Full complement needle roller bearings	0.0030 ～ 0.0050
Thrust needle roller bearings	0.0030 ～ 0.0040
Thrust roller bearings	0.0030 ～ 0.0040

Allowable rotational speed

As the rotational speed of rolling bearings is increased, the bearing temperature also increases due to the heat generated at the contact surfaces between the cage, raceways and rolling elements, until it finally leads to bearing seizure. It is therefore necessary to maintain the rotational speed of a bearing below a certain limit value to ensure safe operation for long periods. This limit value is called the allowable rotational speed.
Since the amount of heat generated is approximately proportional to the sliding speed at the contact area, this sliding speed is an approximate guide indicating the limit of the bearing rotational speed.
The allowable rotational speed of bearings thus varies according to the bearing type, size, bearing load, method of lubrication, radial clearance, and other such factors.
The allowable rotational speeds shown in the table of dimensions are empirical values. They are not absolute values and can be changed according to the bearing use conditions. Depending on the structure and accuracy around the bearing, the lubricant and the lubrication method, it is possible for some bearings to be operated at more than twice the allowable rotational speed given in the table without trouble.

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Operating Temperature Range

The allowable operating temperature range for needle roller bearings is generally -20 ～ +120°C.
When operating at temperatures outside this range, the operation may be limited by the allowable temperature range of prepacked grease, seal, cage material, etc. Further, if the bearing is used at high temperature, i.e. 120°C or above, the amount of dimensional displacement gets larger. So special heat treatment is necessary.
The operating temperature range for some types of bearings is different from the above. See the section for each bearing.

Handling of Bearings

Precautions in handling

Since the bearing is a high-accuracy mechanical element, special attention must be paid to its handling. The following precautions should be noted when handling the bearings.

❶ Bearings and their surrounding parts should be kept clean.
Bearings and their surrounding parts must be kept clean paying special attention to dust and dirt. Tools and the working environment should also be cleaned.

❷ Bearings should be handled carefully.
A shock load during handling may cause scratches, indentations and even cracks or chips on the raceway surfaces and rolling elements.

❸ Bearings should be mounted or dismounted with proper tools.
When mounting and dismounting, tools suitable for the bearing type should be used.

❹ Bearings should be protected against corrosion.
Bearings are treated with anti-corrosive oil. However, when handling them with bare hands, sweat from the hands may result in future rust formation. Gloves should be worn, or hands should be dipped in mineral oil.

Mounting

Preparation

Before mounting the bearing, the dimensions and fillets of the shaft and housing should be checked to ensure that they conform to specifications.
Bearings should be unwrapped just before mounting. In case of grease lubrication, bearings should be filled with grease without cleaning the bearings. Even in the case of oil lubrication, it is normally unnecessary to clean the bearings. However, when high accuracy is required or when using at high speeds, the bearings should be cleaned using cleaning oil to remove thoroughly oily contents. The cleaned bearings should not be left alone without anti-corrosive precautions, because bearings can easily be corroded after anti-corrosive agents are removed.
Lubricating grease is prepacked in some types of bearings. Therefore, refer to the relevant section for each bearing.

Methods of mounting

Mounting methods of bearings are different according to the type of bearing and the fit. In general, mounting of needle roller bearings is comparatively easy. However, non-separable bearings with large interferences should be handled with great care.

❶ Mounting by press fit
Small and medium bearings with small interferences require a small pressing-in force for mounting, and they are mounted using a press at room temperature. The bearing should be pressed in carefully, applying a force evenly to the bearing with a fitting tool as shown in Fig. 26. For separable bearings, the inner and outer rings can be mounted separately, and the mounting work is simple. However, when installing the shaft and inner ring assembly into the outer ring, care should be taken not to damage the raceway surfaces and rolling elements.

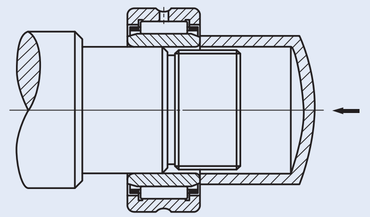


Fig. 26

When mounting non-separable bearings, the inner and outer rings are pressed in simultaneously by applying a cover plate as shown in Fig. 27. It must never happen that the inner ring is press-fitted to the shaft by striking the outer ring, or the outer ring by striking the inner ring, because the raceway surfaces and rolling elements will be scratched or indented.

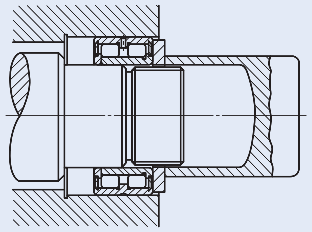


Fig. 27

When press fitting, the friction of the fitting surfaces can be reduced by applying high viscosity oil over the fitting surfaces.
The pressing-in or pulling-out force to be applied to the bearing is given on page A59.

❷ Mounting by shrink fitting
This method is used when the interference is great or when a large bearing is to be fitted. The housing is heated and thermally expanded when fitting the outer ring to the housing and the inner ring is heated and expanded when fitting it to the shaft allowing the bearing to be set easily within a short time. The maximum allowable temperature for the shrink fit is +120°C, and heating should be performed appropriately. Pure non-corrosive mineral oil is recommended as the heating oil for shrink fit, and insulation oil for transformers is considered to be the best. During cooling, the bearing also shrinks in the axial direction. Therefore, to ensure that there is no clearance between the bearing and the shoulder, an axial force must be applied continuously to the bearing until it has cooled.
When the interference between the outer ring and the housing is great, an expansion fit method in which the bearing is cooled using dry ice or other cooling agent before fitting can be used. Immediately after fitting, however, moisture from the air easily condenses on the bearing. Therefore, it is necessary to take preventive measures against corrosion.

A B C D E F G H I J K L M

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Pressing force and pulling force

Guidelines for the pressing force when pressing in the inner ring to the shaft and the pulling force when pulling it out are obtained from the following equation.

$$K = f_{\mathrm{k}}\frac{d}{d+2}\,\Delta_{df}B\left\{1-\left(\frac{d}{F}\right)^2\right\} \quad \cdots\cdots\cdots(39)$$

where,
- K : Pressing or pulling force, N
- f_{k} : Resistance factor determined by the coefficient of friction
  - When pressing in inner ring to shaft, $f_{\mathrm{k}}=4\times10^4$
  - When pulling out inner ring from shaft, $f_{\mathrm{k}}=6\times10^4$
- d : Bore diameter of inner ring, mm
- Δ_{df} : Apparent interference, mm
- B : Width of inner ring, mm
- F : Outside diameter of inner ring, mm

The actual pressing force or pulling force may be greater than the calculated value due to mounting errors. When designing a puller, it is necessary that the puller has the strength (rigidity) to withstand more than 5 times the calculated value.

Running test

After mounting the bearing, a running test is carried out to check whether the mounting is normal. Usually, it is first checked by manual turning. Then, it is operated by power gradually from no-load and low-speed up to normal operating conditions to check for abnormalities.
Noise can be checked by using a soundscope or similar instrument. In this test, checks are carried out for the following abnormalities.

❶ Manual turning
(a) Uneven torque ·················· Improper mounting
(b) Sticking and rattling ··· Scratches or indentations on the raceway surface
(c) Irregular noise ··· Penetration of dust or foreign particles

❷ Power running
(a) Abnormal noise or vibration ··· Indentations on the raceway surface, too great clearance
(b) Abnormal temperature ··· Unsuitable lubricant, improper mounting, too small clearance

Dismounting

Dismounting of the bearings is carried out for the periodic inspection or repairs of machines. By inspecting the bearing, related parts or mechanisms, lubrication, etc., important data is obtained. In the same manner as in mounting, care should be taken to prevent damage to the bearing or other parts.
A suitable dismounting method should be selected according to the type of the bearing, fit, etc. Bearings mounted by interference fit are especially difficult to dismount, and it is necessary to give due consideration to the structure around the bearing during the design stage.

Dismounting of outer ring

Outer rings mounted by interference fit are dismounted as shown in Fig. 28, by screwing in the push-out bolts evenly through several screw holes provided at places corresponding to the side face of the outer ring.

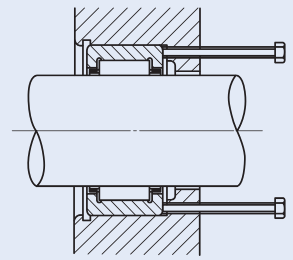


Fig. 28

Dismounting of inner ring

In the case of bearings such as needle roller bearings in which the inner and outer rings are separable, the simplest way to press out the inner ring is by using a press as shown in Fig. 29.
The puller shown in Fig. 30 is also generally used. This is designed according to the bearing size. In addition, there are a 3-hook puller (Fig. 31) and a 2-hook puller for wide-range use.

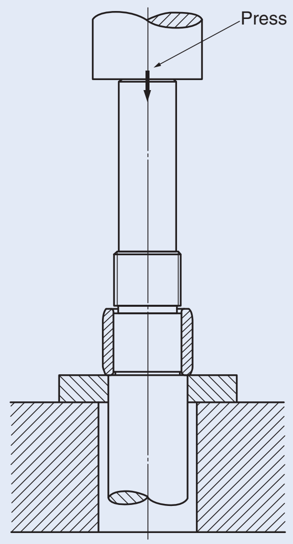


Fig. 29

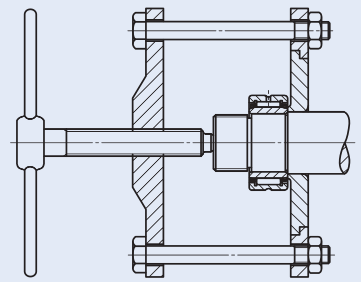


Fig. 30

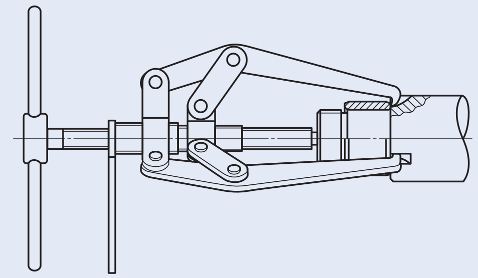


Fig. 31

In addition to these, when it is difficult to remove the inner ring due to high shoulders, several holes for removal pins are made through the shoulder, or several hook grooves are cut in the shoulder as shown in Fig. 32 and Fig. 33.
When a bearing is not to be used again after removal, it may be removed by heating with a torch lamp.

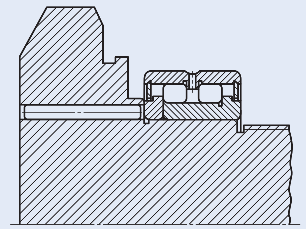


Fig. 32

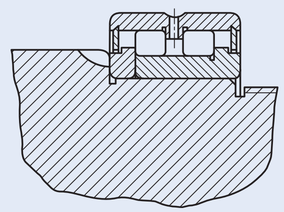


Fig. 33

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch

Inspection of bearing

Cleaning of bearing

When inspecting a bearing after removal, the appearance of the bearing should be recorded first. Then, after the residual amount of lubricant is checked and a sample of lubricant is collected, the bearing should be cleaned.
For cleaning, light oil or kerosene is commonly used. Cleaning is divided into rough cleaning and final cleaning, and wire gauze is set as a raised bottom in a container to prevent the bearing from touching the bottom of the container.
Lubricating grease and adhering substances such as foreign particles are removed with a brush, etc., using oil for rough cleaning. Care should be taken during this process, because if the bearing is turned with foreign particles attached, the raceway surfaces may be scratched.
Final cleaning is carried out by turning the bearing in cleaning oil. It is desirable that the cleaning oil is kept clean by filtering. Immediately after cleaning, the bearing must be protected against corrosion.

Inspection and evaluation of bearing

The judgement as to whether the removed bearing is reusable depends on the inspection after cleaning. Conditions of the raceway surfaces, rolling elements and fitting surfaces, wear condition of the cage, increase of bearing clearance, dimensions, rotational accuracy, etc. should be checked for damage and abnormalities.
The evaluation is performed based on the experience taking into consideration the degree of damage, machine performance, importance of the machine, operating conditions, period until the next inspection, and other such factors.

Maintenance and inspection

Maintenance and inspection

Maintenance and inspection are carried out to maintain good performance of bearings installed in the machine.
Maintenance is performed by checking the machine operating conditions, checking and replenishing or replacing the lubricant, checking the bearing and related parts by periodic disassembly and other such procedures.
Items for inspection of a running bearing in a machine include the bearing temperature, noise, vibration and condition of lubricant.
When any abnormality is found during operation, the cause should be investigated and measures taken by referring to the section on running test on page A59. When removing a bearing, refer to the section on dismounting on page A59.

Damage, causes and corrective action

Rolling bearings can generally be used fully up to their rolling fatigue life if they are properly selected, mounted, operated and maintained. However, they may actually be damaged earlier than their expected lifetimes creating problems or accidents. Common causes of damage include improper mounting or handling, insufficient lubrication and penetration of foreign particles.
It may be difficult to determine the exact cause of a problem by checking only the damaged bearing. The conditions of the machine before and after the occurrence of the damage, the location and the operating and ambient conditions of the bearing, the structure around the bearing, etc. should also be examined. It then becomes possible to assess the cause of the damage by linking the conditions of the damaged bearing to the probable causes arising from the machine operation, and to prevent the recurrence of similar problems.
Common types of damage, causes and corrective action are listed in Table 41.

Table 41 Damage, causes and corrective action

Condition of bearing damage		Cause	Corrective action
Flaking	Flakings at opposite circumferential positions on raceway surfaces	Improper roundness of housing bore	Correction of housing bore accuracy
	Flakings in the vicinity of raceway surface edges and roller ends	Improper mounting, Shaft deflection, Poor centering, Poor accuracy of shaft or housing	Careful mounting, Careful centering, Correction of shoulders of shaft and housing for right angles
	Flakings on raceway surfaces with an interval corresponding to roller pitch	Great shock load when mounting, Rusting during machine stoppage	Careful mounting, Protection against rust for long periods of machine stoppage
	Early flaking on raceway surfaces and rolling elements	Too small clearance, Too great load, Poor lubrication, Rusting, etc.	Correct selection of fit and clearance Correct selection of lubricant
Galling	Galling on raceway surfaces and rolling surfaces of rollers	Poor lubrication in early stage Grease consistency too hard High acceleration at start	Selection of softer grease, Avoiding quick acceleration
	Galling between roller end faces and collar guide surfaces	Poor lubrication, Poor mounting, Large axial load	Correct selection of lubricant Correct mounting
Breakage	Cracks in outer or inner ring	Excessive shock load, Too much interference. Poor cylindricity of shaft. Too large fillet radius, Development of thermal cracks, Development of flaking	Reevaluation of load conditions, Correction of fit, Correction of machining accuracy of shaft or sleeve, Making fillet radius smaller than the chamfer dimension of bearing
	Cracked rolling elements, broken collar	Development of flaking Shock to collar when mounting, Dropped by careless handling	Careful handling and mounting
	Broken cage	Abnormal load to cage by poor mounting, Poor lubrication	Minimizing mounting errors, Study of lubricating method and lubricant
Dent	Indentations on raceway surfaces at an interval corresponding to the pitch between rolling elements (brinelling)	Shock load applied when mounting, Excessive load while stopping	Careful handling
	Indentation on raceway surfaces and rolling surfaces of rollers	Biting of foreign substances such as metal chips and sands	Cleaning of housing, Improvement of sealing, Use of clean lubricant
Abnormal wear	False brinelling (Phenomenon like brinelling)	Vibration when the bearing is stationary such as during transportation, Oscillating motion with small amplitude	Fixing of shaft and housing, Use of lubricating oil, Application of preload to reduce vibration
	Fretting Localized wear of fitted surfaces accompanied by red-brown wear particles	Sliding between fitted surfaces	Increase of interference, Application of oil
	Wear on raceway surfaces, collar surfaces, rolling surfaces of rollers, cages, etc.	Penetration of foreign particles, Poor lubrication, Rust	Improvement of sealing, Cleaning of housing Use of clean lubricant
	Creep Wear on fitted surfaces	Sliding between fitted surfaces, Insufficient tightening of sleeve	Increase of interference, Correct tightening of sleeve
Seizure	Discoloration of rolling elements and/or raceway surfaces and/or flange surfaces, Adhesion and welding, Discoloration of cage	Poor lubrication, Too small clearance, Poor mounting	Supply of proper amount of proper lubricant, Rechecking of fit and bearing clearance Rechecking of mounting dimensions and related parts
Electric corrosion	Ripples on raceway surfaces	Melting by sparks due to electric current	Insulation of bearing, Grounding to avoid electric current
Rust, corrosion	Rust or corrosion on bearing inside surfaces or on fitted surfaces	Condensation of vapor in air, Penetration of corrosive substances	Careful storage if under high temperature and high humidity, Protection against rust, Improvement of sealing

1N=0.102kgf=0.2248lbs.
1mm=0.03937inch