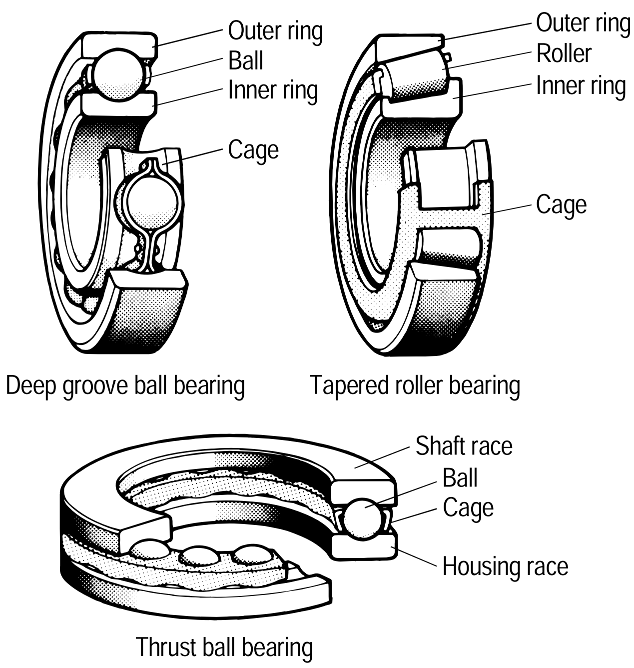
1. Rolling bearing structures and types

1-1 Structure

Rolling bearings (bearings hereinafter) normally comprise bearing rings, rolling elements and a cage. (see Fig. 1-1)

Rolling elements are arranged between inner and outer rings with a cage, which retains the rolling elements in correct relative position, so they do not touch one another. With this structure, a smooth rolling motion is realized during operation.

Bearings are classified as follows, by the number of rows of rolling elements :
single-row, double-row, or multi-row (triple- or four-row) bearings.



Note) In thrust bearings inner and outer rings and also called "shaft race" and "housing race" respectively. The race indicates the washer specified in JIS.

Fig. 1-1 Bearing structure

1) Bearing rings

The path of the rolling elements is called the raceway; and, the section of the bearing rings where the elements roll is called the raceway surface. In the case of ball bearings, since grooves are provided for the balls, they are also referred to as raceway grooves.

The inner ring is normally engaged with a shaft; and, the outer ring with a housing.

2) Rolling element

Rolling elements may be either balls or rollers. Many types of bearings with various shapes of rollers are available.

- Ball
- Cylindrical roller ($L_W \leqq 3\ D_W$)*
- Long cylindrical roller ($3D_W \leqq L_W \leqq 10D_W$, $D_W > 6$ mm)*
- Needle roller ($3D_W \leqq L_W \leqq 10D_W$, $D_W \leqq 6$ mm)*
- Tapered roller (tapered trapezoid)
- Convex roller (barrel shape)

* L_W : roller length (mm)
D_W : roller diameter (mm)

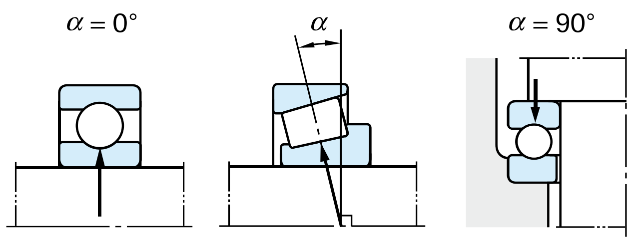
3) Cage

The cage guides the rolling elements along the bearing rings, retaining the rolling elements in correct relative position. There are various types of cages including pressed, machined, molded, and pin type cages.

Due to lower friction resistance than that found in full complement roller and ball bearings, bearings with a cage are more suitable for use under high speed rotation.

1-2 Type

The contact angle (α) is the angle formed by the direction of the load applied to the bearing rings and rolling elements, and a plan perpendicular to the shaft center, when the bearing is loaded.



Bearings are classified into two types in accordance with the contact angle (α).

· Radial bearings ($0° \leqq \alpha \leqq 45°$)
... designed to accommodate mainly radial load.
· Thrust bearings ($45° < \alpha \leqq 90°$)
... designed to accommodate mainly axial load.

Rolling bearings are classified in Fig. 1-2, and characteristics of each bearing type are described in Tables 1-1 to 1-13.

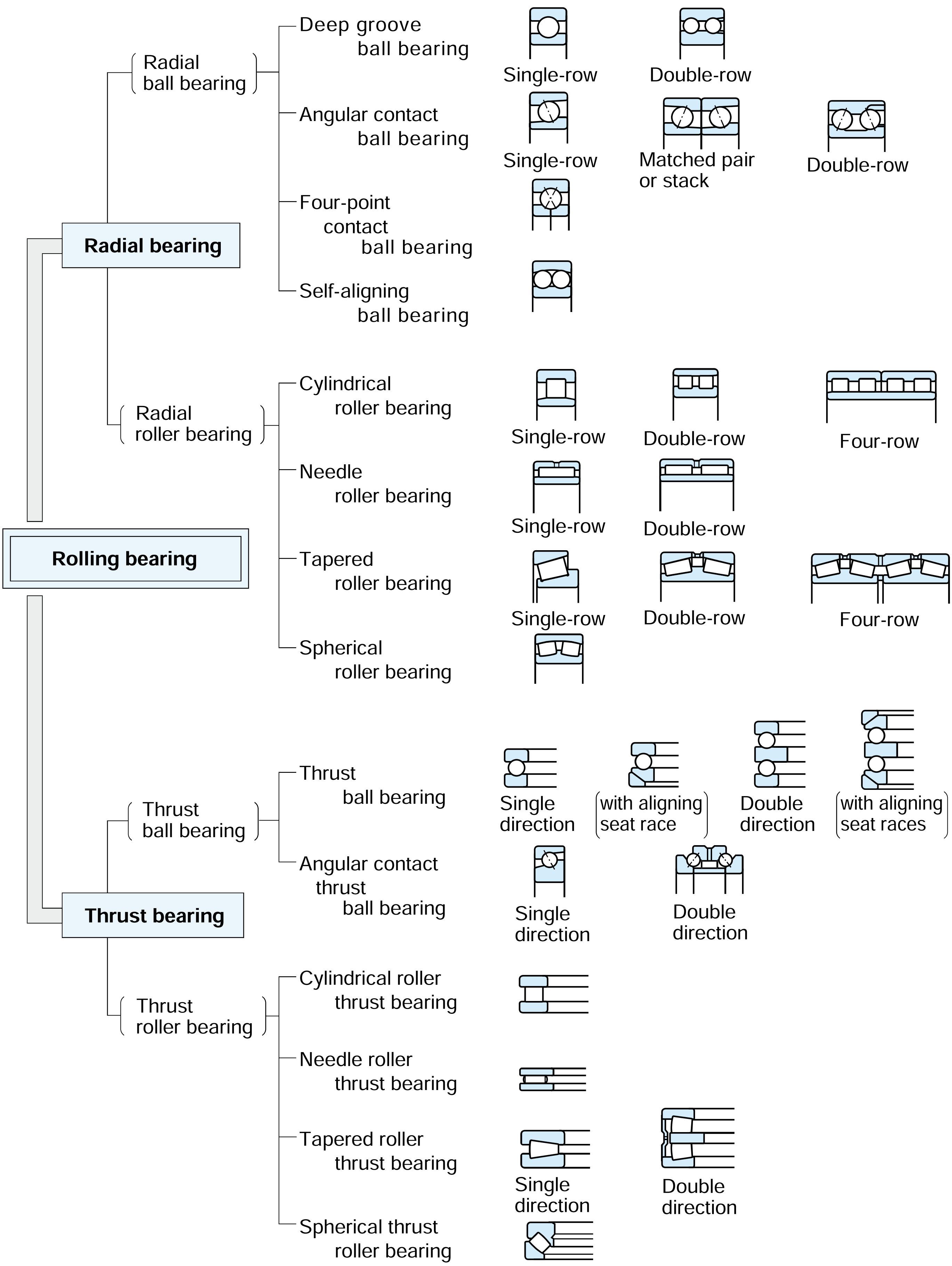


Fig. 1-2(1) Rolling bearings

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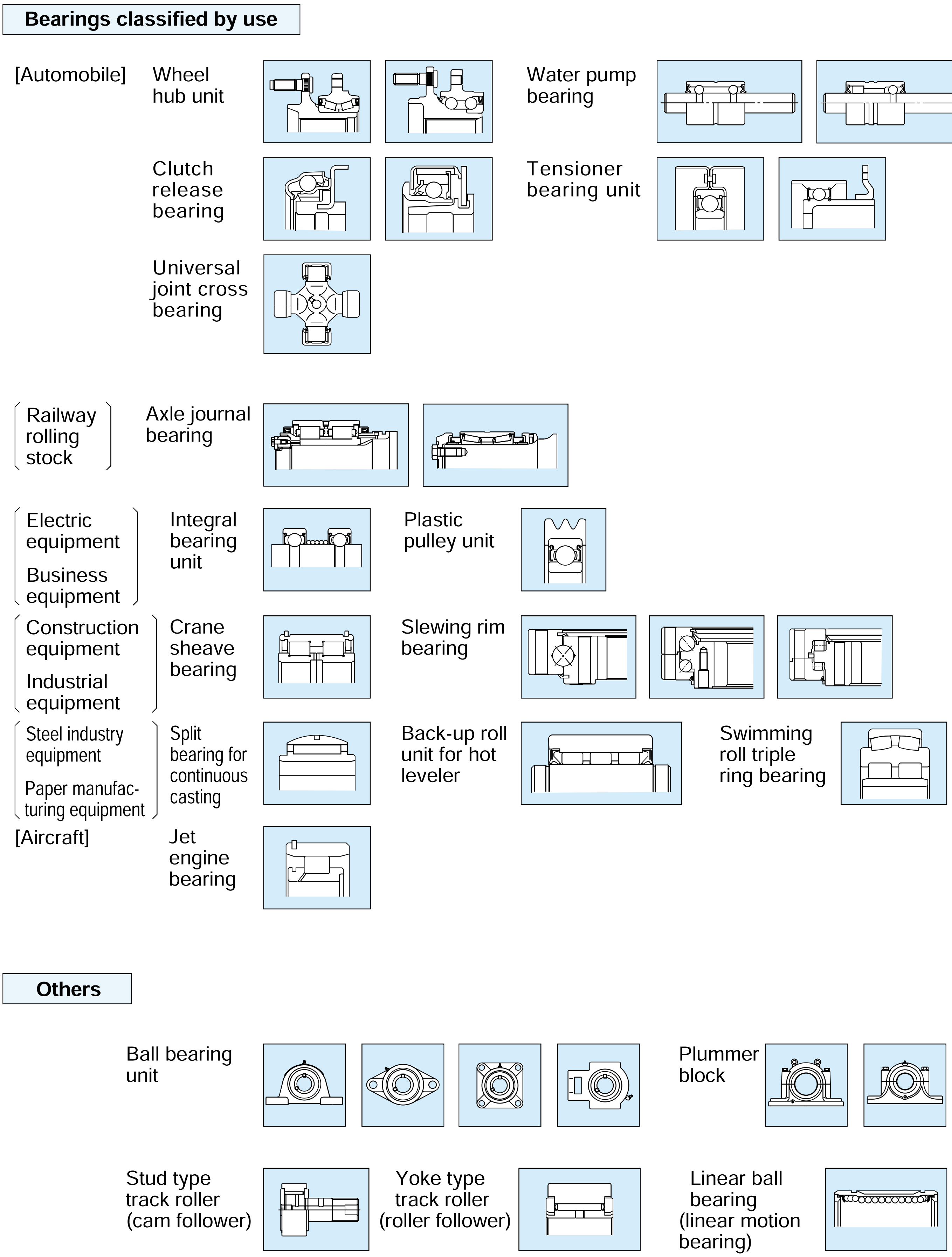
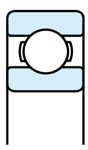
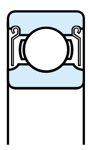
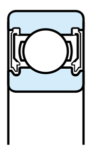
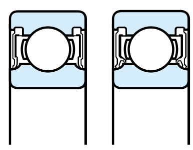
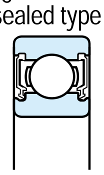
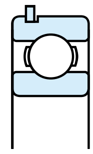
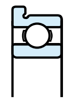
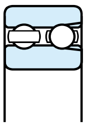


Fig. 1-2(2) Rolling bearings

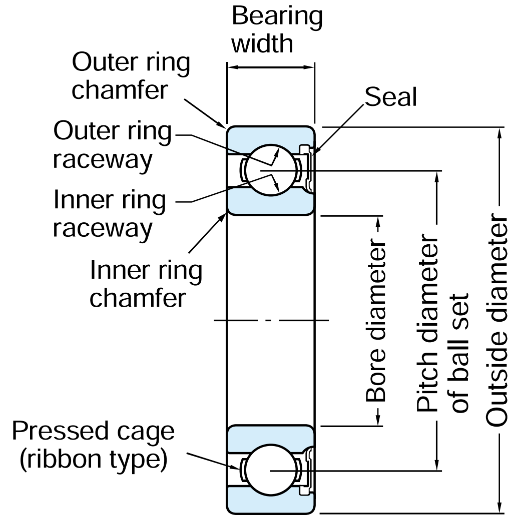
Table 1-1 Deep groove ball bearings

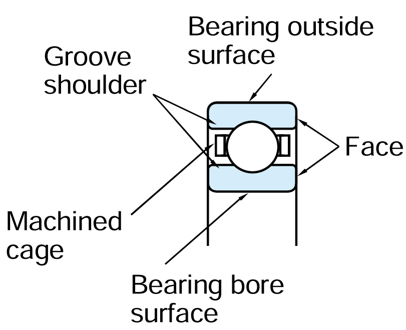
Single-row								Double-row
Open type	Shielded type	Non-contact sealed type	Contact sealed type		Extremely light contact sealed type	With locating snap ring	Flanged type	
								
	ZZ	2RU	2RS	2RK	2RD	NR	(Suitable for extra-small or miniature bearing)	
680, 690, 600, 620, 630, (ML) ···Extra-small, miniature bearing 6700, 6800, 6900, 16000, 6000, 6200, 6300, 6400								4200 4300

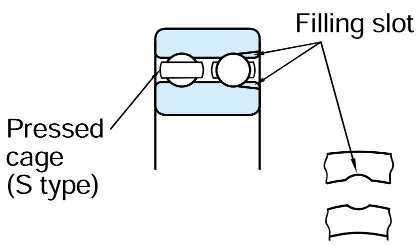
- The most popular types among rolling bearings, widely used in a variety of industries.
- Radial load and axial load in both directions can be accommodated.
- Suitable for operation at high speed, with low noise and low vibration.
- Sealed bearings employing steel shields or rubber seals are filled with the appropriate volume of grease when manufactured.
- Bearings with a flange or locating snap ring attached on the outer ring are easily mounted in housings for simple positioning of housing location.

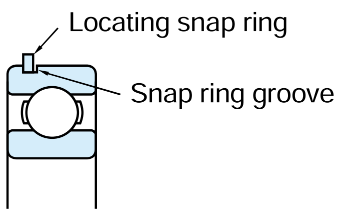
[Recommended cages] Pressed cage (ribbon type, snap type ··· single-row, S type ··· double-row), copper alloy or phenolic resin machined cage, synthetic resin molded cage

[Main applications]
Automobile : front and rear wheels, transmissions, electric devices
Electric equipment : standard motors, electric appliances for domestic use
Others : measuring instruments, internal combustion engines, construction equipment, railway rolling stock, cargo transport equipment, agricultural equipment, equipment for other industrial uses





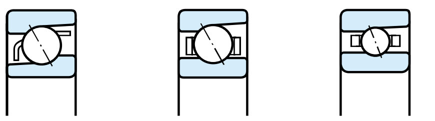
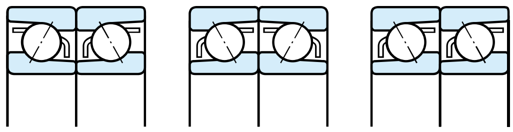
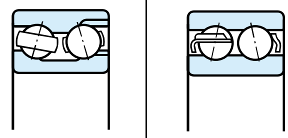




Bearing size (Reference) Unit : mm

Connotation	Bore diameter	Outside diameter
Miniature	–	Under 9
Extra-small	Under 10	9 or more
Small size	10 or more	80 or less
Medium size	–	80 – 180
Large size	–	180 – 800
Extra-large size	–	Over 800

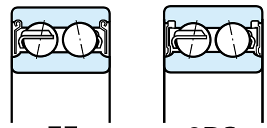
Table 1-2 Angular contact ball bearings

Single-row			Matched pair			Double-row	
		For high-speed use	Back-to-back arrangement	Face-to-face arrangement	Tandem arrangement		
							
(With pressed cage)	(With machined cage)	HAR	DB	DF	DT	(With filling slot)	
7000, 7200, 7300, 7400 ······ Contact angle 30°						3200	5200
7000B, 7200B, 7300B, 7400B ······ 40°						3300	5300
7900C, HAR900C, HAR000C, 7000C, 7200C, 7300C ······ 15°						Contact angle 32°	Contact angle 24°

- Bearing rings and balls possess their own contact angle which is normally 15°, 30° or 40°.
  - Larger contact angle ····· higher resistance against axial load
  - Smaller contact angle ··· more advantageous for high-speed rotation
- Single-row bearings can accommodate radial load and axial load in one direction.
- DB and DF matched pair bearings and double-row bearings can accommodate radial load and axial load in both directions.
  DT matched pair bearings are used for applications where axial load in one direction is too large for one bearing to accept.
- HAR type high speed bearings were designed to contain more balls than standard bearings by minimizing the ball diameter, to offer improved performance in machine tools.
- Angular contact ball bearings are used for high accuracy and high-speed operation.

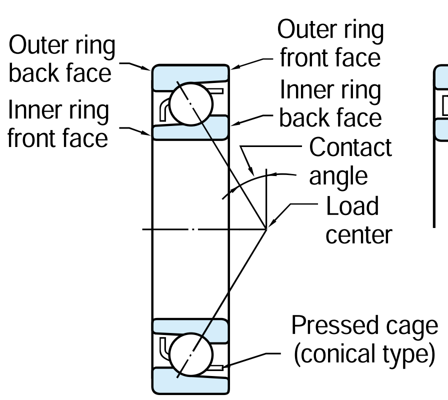
Double-row:

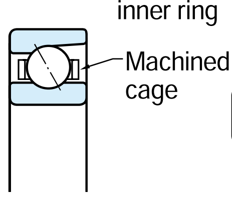
- Axial load in both directions and radial load can be accommodated by adapting a structure pairing two single-row angular contact ball bearings back to back.
- For bearings with no filling slot, the sealed type is available.

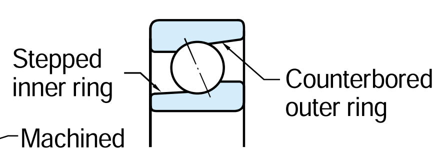


[Recommended cages] Pressed cage (conical type ··· single-row : S type, snap type ··· double-row), copper alloy or phenolic resin machined cage, synthetic resin molded cage

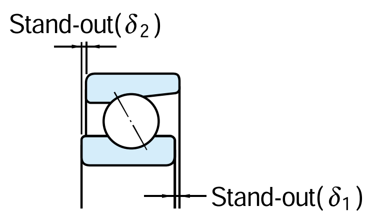
[Main applications] Single-row : machine tool spindles, high frequency motors, gas turbines, centrifugal separators, front wheels of small size automobiles, differential pinion shafts
Double-row : hydraulic pumps, roots blowers, air-compressors, transmissions, fuel injection pumps, printing equipment







(Ball and bearing ring are not separable.)

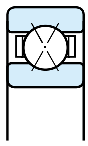
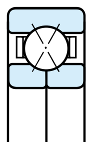
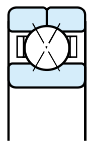


Contact angles (Reference)

Contact angle	Supplementary code
15°	C
20°	CA
25°	AC
30°	A (Omitted)
35°	E
40°	B

"G type" bearings are processed (with flush ground) such that the stand-out turns out to be $\delta_1 = \delta_2$.
The matched pair DB, DF, and DT, or stack are available.

Table 1-3 Four-point contact ball bearings

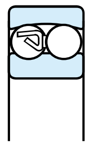
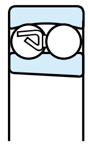
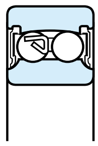
One-piece type	Two-piece inner ring	Two-piece outer ring
		
—	6200BI 6300BI	(6200BO) (6300BO)

- Radial load and axial load in both directions can be accommodated.
- A four-point contact ball bearing can substitute for a face-to-face or back-to-back arrangement of angular contact ball bearings.
- Suitable for use under pure axial load or combined radial and axial load with heavy axial load.
- This type of bearing possesses a contact angle (α) determined in accordance with the axial load direction. This means that the bearing ring and balls contact each other at two points on the lines forming the contact angle.

[Recommended cage] Copper alloy machined cage

[Main applications]
Motorcycle : Transmission, driveshaft pinion-side
Automobile : Steering, transmission

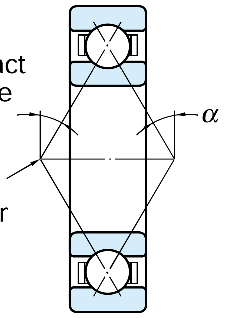
Table 1-4 Self-aligning ball bearings

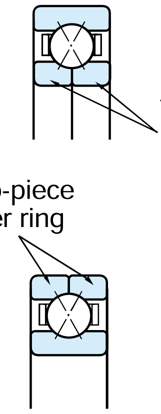
Cylindrical bore	Tapered bore	Sealed
	 K (Taper 1 : 12)	 2RS
120, 130 1200, 1300 2200, 2300	11200, 11300··· extended inner ring type	2200 2RS 2300 2RS

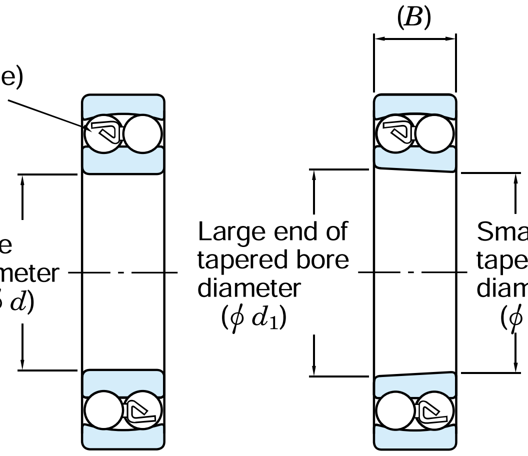
- Spherical outer ring raceway allows self-alignment, accommodating shaft or housing deflection and misaligned mounting conditions.
- Tapered bore design can be mounted readily using an adapter.

Pressed cage: staggered type···12, 13, 22...2RS, 23...2RS; snap type ··········22, 23

Power transmission shaft of wood working and spinning machines, plummer blocks







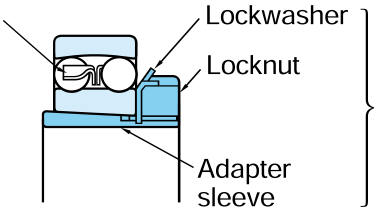
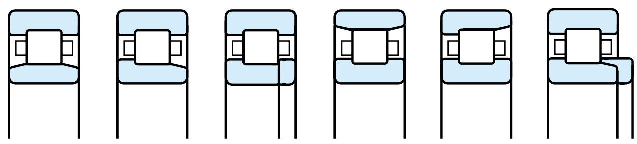
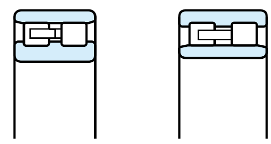
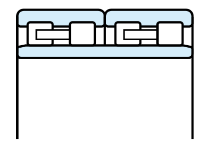


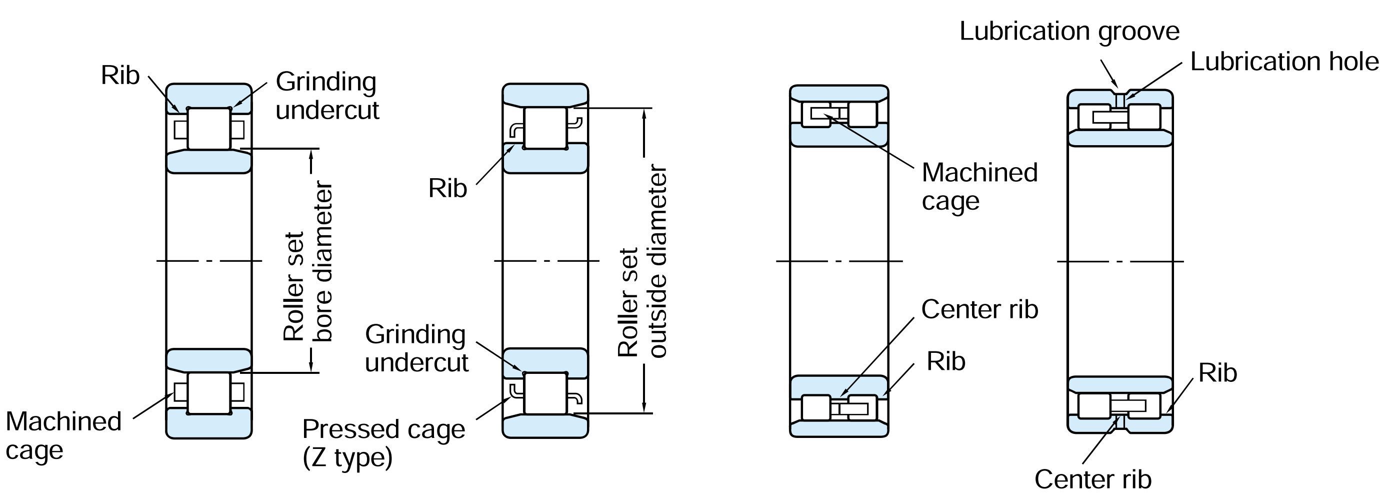
Table 1-5 Cylindrical roller bearings

Single-row	Double-row	Four-row
 NU NJ NUP N NF NH	 NN NNU	 (Mainly use on rolling mill roll neck)
NU1000, NU200 (R), NU300 (R), NU400 NU2200 (R), NU2300 (R) NU3200, NU3300	Cylindrical bore: NNU4900, NN3000 Tapered bore: NNU4900K, NN3000K	(FC) , (4CR)

- Since the design allowing linear contact of cylindrical rollers with the raceway provides strong resistance to radial load, this type is suitable for use under heavy radial load and impact load, as well as at high speed.
- N and NU types are ideal for use on the free side: they are movable in the shaft direction in response to changes in bearing position relative to the shaft or housing, which are caused by heat expansion of the shaft or improper mounting.
- NJ and NF types can accommodate axial load in one direction; and NH and NUP types can accommodate partial axial load in both directions.
- With separable inner and outer ring, this type ensures easy mounting.
- Due to their high rigidity, NNU and NN types are widely used in machine tool spindles.

[Recommended cages] Pressed cage (Z type), copper alloy machined cage, pin type cage, synthetic resin molded cage

[Main applications] Large and medium size motors, traction motors, generators, internal combustion engines, gas turbines, machine tool spindles, speed reducers, cargo transport equipment, and other industrial equipment



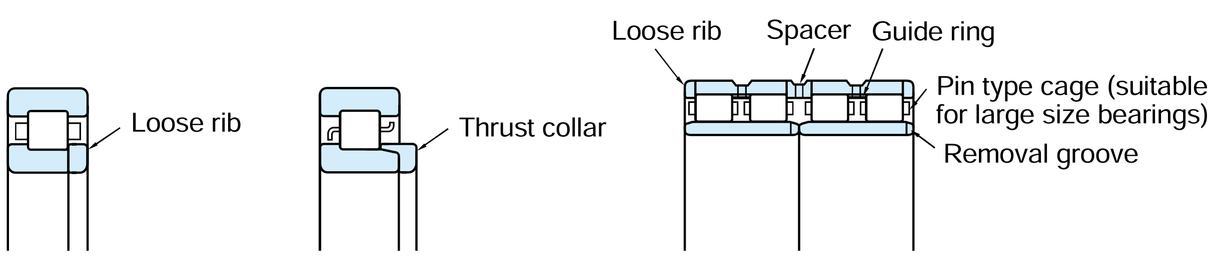
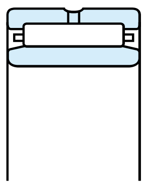
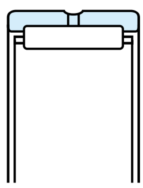
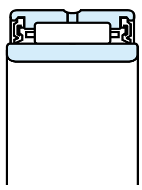
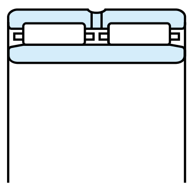
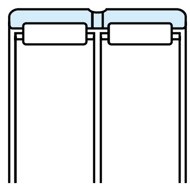


Table 1-6 Machined ring needle roller bearings

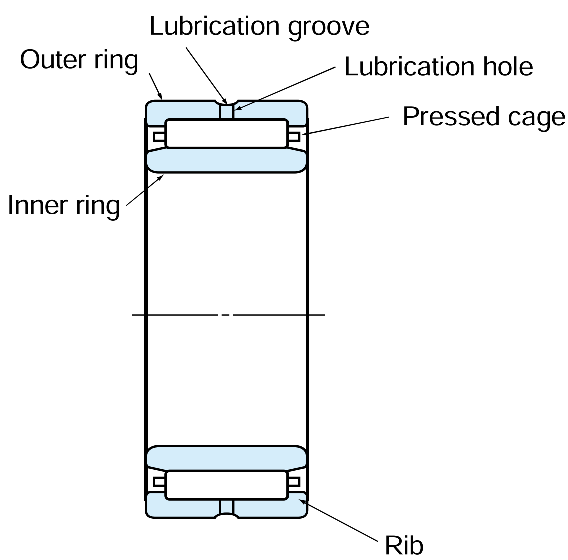
Single-row			Double-row	
				
With inner ring	Without inner ring	Sealed	With inner ring	Without inner ring
NA4800 NA4900 NA6900 (NKJ, NKJS)	RNA4800 RNA4900 RNA6900 (NK, NKS, HJ)	NA49002RS – (HJ.2RS)	NA6900 (d ≧ 32)	RNA6900 (Fw ≧ 40)

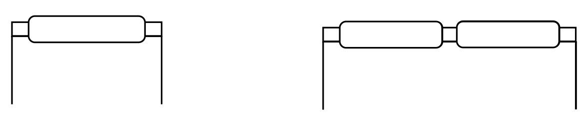
- In spite of their basic structure, which is the same as that of NU type cylindrical roller bearings, bearings with minimum ring sections offer space savings and greater resistance to radial load, by using needle rollers.
- Bearings with no inner rings function using heat treated and ground shafts as their raceway surface.

[Recommended cage] Pressed cage

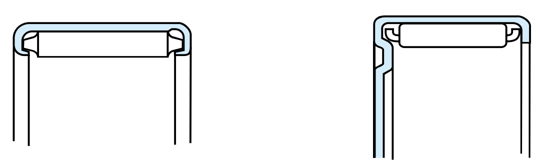
[Main applications] Automobile engines, transmissions, pumps, power shovel wheel drums, hoists, overhead traveling cranes, compressors

(Reference) Many needle roller bearings other than those with machined ring are available.
For details, refer to the pages for the needle roller bearing specification tables and the dedicated "Needle Roller Bearings" catalog (CAT No. B2020E), published separately.

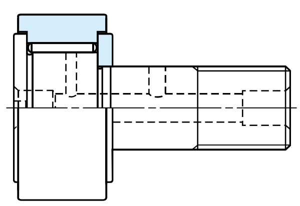




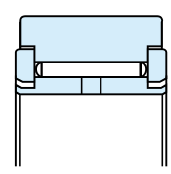
Needle roller and cage assemblies



Drawn cup needle roller bearings

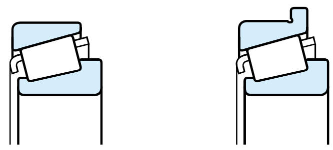
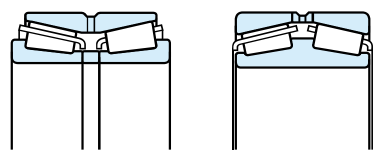
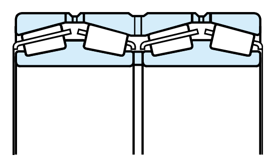


Stud type track roller
(cam follower)



Yoke type track roller
(roller follower)

Table 1-7 Tapered roller bearings

Single-row	Double-row	Four-row
 Flanged type	 TDO type TDI type	 (Mainly used on rolling mill roll necks)
(Standard contact angle) 32900JR 30200JR 32000JR 32200JR 33000JR 33200JR 33100JR 30300JR 32300JR (Inter mediate contact angle) 30200CR 32200CR 30300CR 32300CR (Steep contact angle) 30300DJ 30300DJR 31300JR	TDO type: 46200 46200A 46300 46300A (46T) TDI type: 45200 45300 (45T)	37200 47200 47300 (47T) (4TR)

- Tapered rollers assembled in the bearings are guided by the inner ring back face rib.
- The raceway surfaces of inner ring and outer ring and the rolling contact surface of rollers are designed so that the respective apexes converge at a point on the bearing center line.
- Single-row bearings can accommodate radial load and axial load in one direction, and double-row bearings can accommodate radial load and axial load in both directions.
- This type of bearing is suitable for use under heavy load or impact load.
- Bearings are classified into standard, intermediate and steep types, in accordance with their contact angle (α).
  The larger the contact angle is, the greater the bearing resistance to axial load.
- Since outer ring and inner ring assembly can be separated from each other, mounting is easy.
- Bearings designated by the suffix "J" and "JR" are interchangeable internationally.
- Items sized in inches are still widely used.

[Recommended cages] Pressed cage, synthetic resin molded cage, pin type cage

[Main applications] Automobile : front and rear wheels, transmissions, differential pinion
Others : machine tool spindles, construction equipment, large size agricultural equipment, railway rolling stock speed reduction gears, rolling mill roll necks and speed reducers, etc

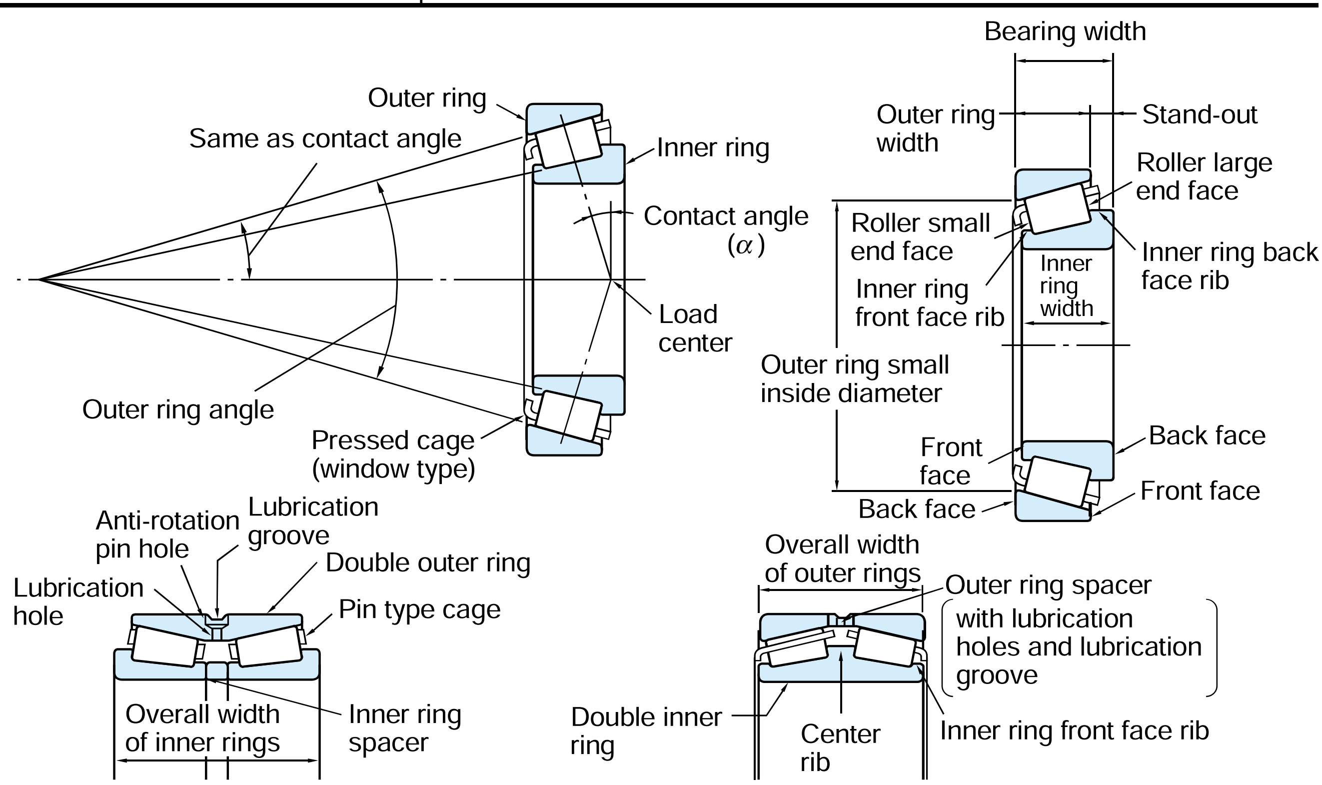
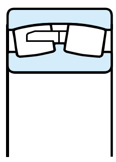
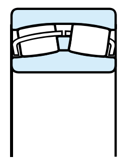
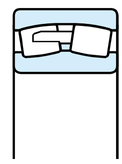
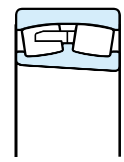


Table 1-8 Spherical roller bearings

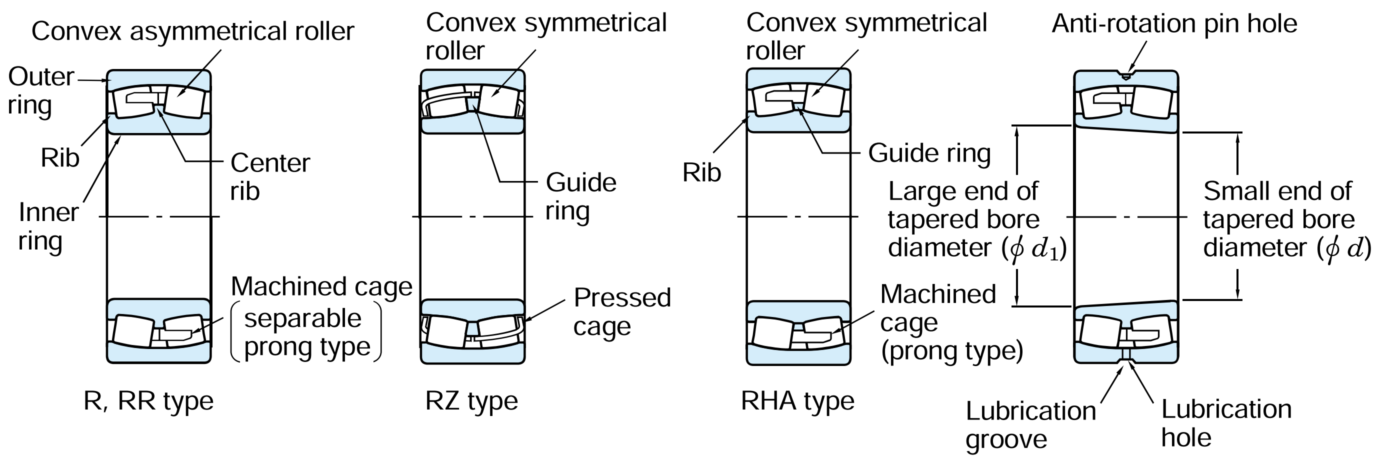
Cylindrical bore			Tapered bore
Convex asymmetrical roller type	Convex symmetrical roller type		
 R, RR	 RZ	 RHA	 K or K30

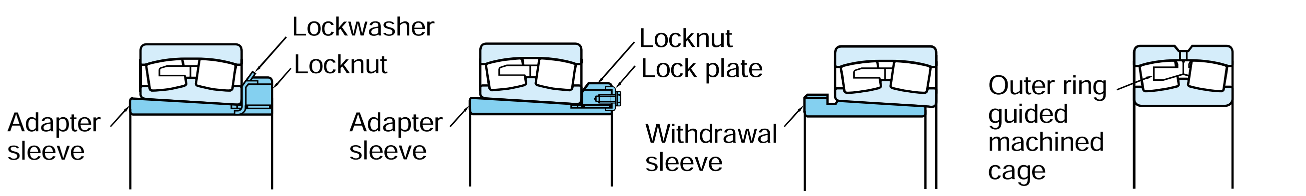
23900R, 23000R (RZ, RHA), 23100R (RZ, RHA), 22200R (RZ, RHA), 21300R (RZ)
24000R (RZ, RHA), 24100R (RZ, RHA), 23200R (RZ, RHA), 22300R (RZ, RHA)

- Spherical roller bearings comprising barrel-shaped convex rollers, double-row inner ring and outer ring are classified into three types : R(RR), RZ and RHA, according to their internal structure.
- With the bearing designed such that the circular arc center of the outer ring raceway matches with the bearing center, the bearing is self-aligning, insensitive to errors of alignment of the shaft relative to the housing, and to shaft bending.
- This type can accommodate radial load and axial load in both directions, which makes it especially suitable for applications in which heavy load or impact load is applied.
- The tapered bore type can be easily mounted/dismounted by using an adapter or withdrawal sleeve.
  There are two types of tapered bores (tapered ratio) :
  · 1 : 30 (supplementary code K30) ··· Suitable for series 240 and 241.
  · 1 : 12 (supplementary code K) ··· Suitable for series other than 240 and 241.
- Lubrication holes, a lubrication groove and anti-rotation pin hole can be provided on the outer ring. Lubrication holes and a lubrication groove can be provided on the inner ring, too.

[Recommended cages] Copper alloy machined cage, pressed cage

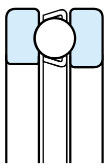
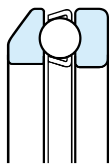
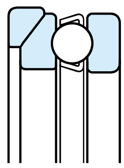
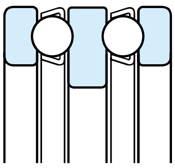
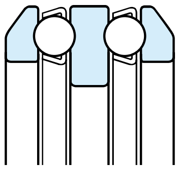
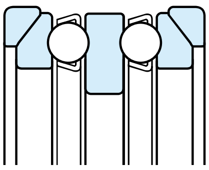
[Main applications] Paper manufacturing equipment, speed reducers, railway rolling stock axle journals, rolling mill pinion stands, table rollers, crushers, shaker screens, printing equipment, wood working equipment, speed reducers for various industrial uses, plummer blocks





(Shaft diameter ≦ 180 mm) (Shaft diameter ≧ 200 mm) (For shaker screen)

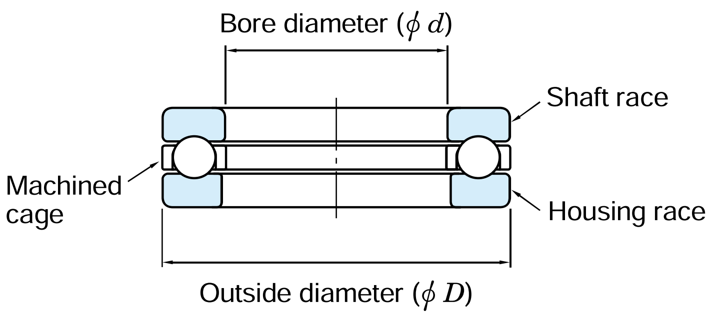
Table 1-9 Thrust ball bearings

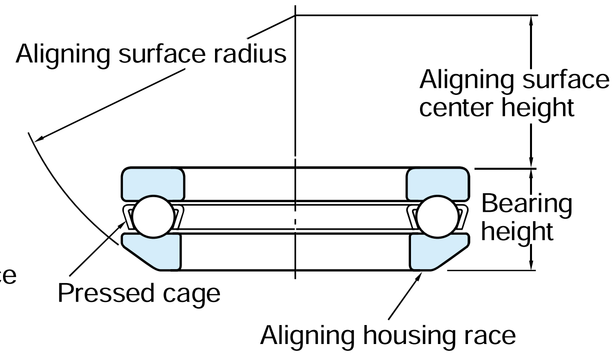
Single direction			Double direction		
With flat back faces	With spherical back face	With aligning seat race	With flat back faces	With spherical back faces	With aligning seat races
					
51100 51200 51300 51400	– 53200 53300 53400	– 53200U 53300U 53400U	– 52200 52300 52400	– 54200 54300 54400	– 54200U 54300U 54400U

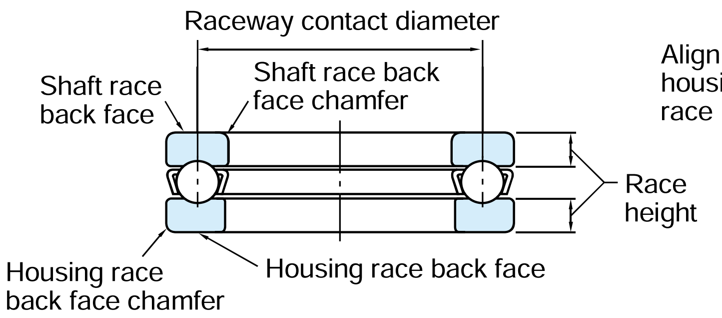
- This type of bearing comprises washer-shaped rings with raceway groove and ball and cage assembly.
- Races to be mounted on shafts are called shaft races (or inner rings); and, races to be mounted into housings are housing races (or outer rings).
  Central races of double direction bearings are mounted on the shafts.
- Single direction bearings accommodate axial load in one direction, and double direction bearings accommodate axial load in both directions. (Both of these bearings cannot accommodate radial loads.)
- Since bearings with a spherical back face are self- aligning, it helps to compensate for mounting errors.

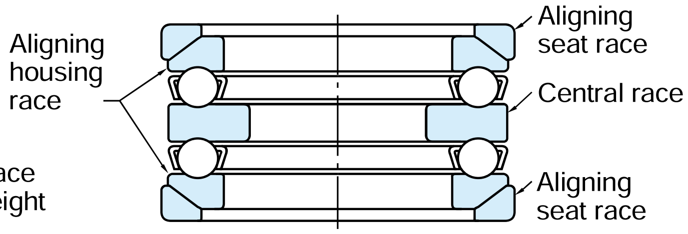
[Recommended cages] Pressed cage, copper alloy or phenolic resin machined cage, synthetic resin molded cage

[Main applications] Automobile king pins, machine tool spindles



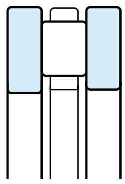






[Remark] The race indicates the washer specified in JIS.

Table 1-10 Cylindrical roller thrust bearings

Single direction

(811, 812, NTHA)
■ This type of bearing comprises washer-shaped rings (shaft and housing race) and cylindrical roller and cage assembly. Crowned cylindrical rollers produce uniform pressure distribution on roller/raceway contact surface. ■ Axial load can be accommodated in one direction. ■ Great axial load resistance and high axial rigidity are provided.
[Recommended cages] Copper alloy machined cage
[Main applications] Oil excavators, iron and steel equipment

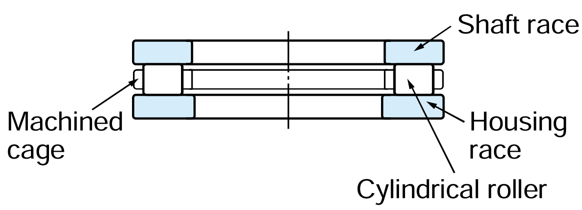
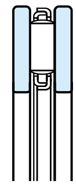
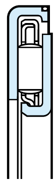
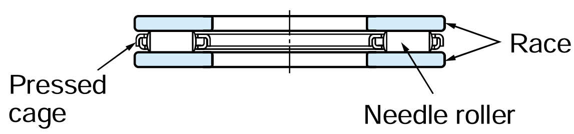
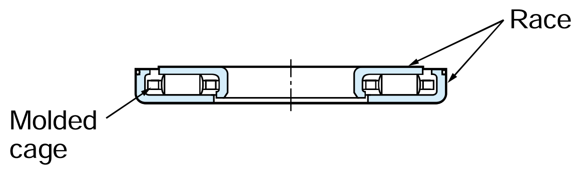


Table 1-11 Needle roller thrust bearings

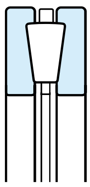
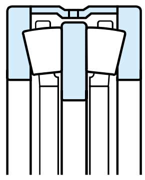
Separable	Non-separable
	
(AXK, FNT, NTA)	(FNTKF)
■ The separable type, comprising needle roller and cage thrust assembly and a race, can be matched with a pressed thin race (AS) or machined thick race (LS, WS.811, GS.811). ■ The non-separable type comprises needle roller and cage thrust assembly and a precision pressed race. ■ Axial load can be accommodated in one direction. ■ Due to the very small installation space required, this type contributes greatly to size reduction of application equipment. ■ In many cases, needle roller and cage thrust assembly function by using the mounting surface of the application equipment, including shafts and housings, as its raceway surface.	
Pressed cage, synthetic resin molded cage	
Transmissions for automobiles, cultivators and machine tools	





[Remark] The race indicates the thrust washer or washer specified in JIS.

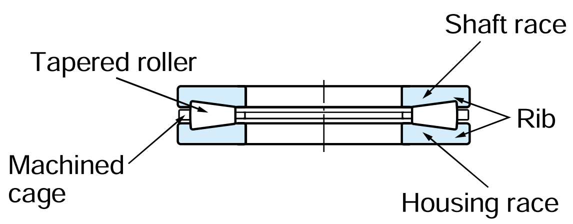
Table 1-12 Tapered roller thrust bearings

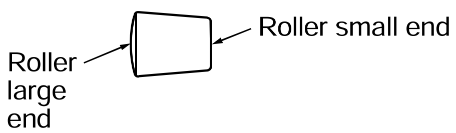
Single direction	Double direction
	
(T) (THR)	(2THR)

- This type of bearing comprises tapered rollers (with spherical large end), which are uniformly guided by ribs of the shaft and housing races.
- Both shaft and housing races and rollers have tapered surfaces whose apexes converge at a point on the bearing axis.
- Single direction bearings can accommodate axial load in one direction; and, double direction bearings can accommodate axial load in both directions.
- Double direction bearings are to be mounted such that their central race is placed on the shaft shoulder. Since this type is treated with a clearance fit, the central race must be fixed with a sleeve, etc.

[Recommended cages] Copper alloy machined cage

[Main applications]
Single direction : crane hooks, oil excavator swivels
Double direction : rolling mill roll necks





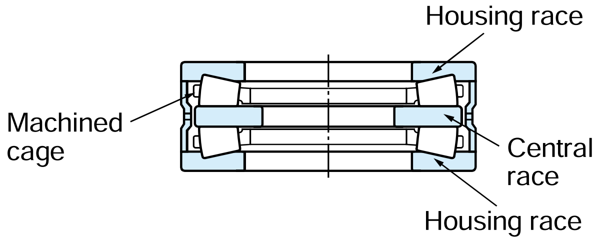
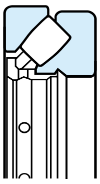


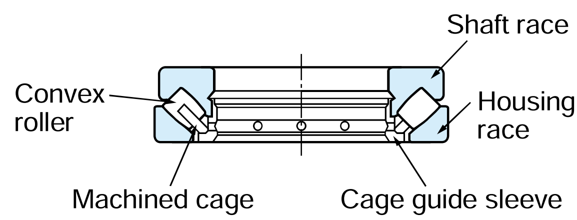
Table 1-13 Spherical thrust roller bearings


29200 29300 29400

- This type of bearing, comprising barrel-shaped convex rollers arranged at an angle with the axis, is self-aligning due to spherical housing race raceway; therefore, shaft inclination can be compensated for to a certain degree.
- Great axial load resistance is provided. This type can accommodate a small amount of radial load as well as heavy axial load.
- Normally, oil lubrication is employed.

Copper alloy machined cage

Hydroelectric generators, vertical motors, propeller shafts for ships, screw down speed reducers, jib cranes, coal mills, pushing machines, molding machines



2. Outline of bearing selection

Currently, as bearing design has become diversified, their application range is being increasingly extended. In order to select the most suitable bearings for an application, it is necessary to conduct a comprehensive study on both bearings and the equipment in which the bearings will be installed, including operating conditions, the performance required of the bearings, specifications of the other components to be installed along with the bearings, marketability, and cost performance, etc.

In selecting bearings, since the shaft diameter is usually determined beforehand, the prospective bearing type is chosen based upon installation space, intended arrangement, and according to the bore diameter required.

① Bearing type, arrangement → **② Bearing dimension** →

(Reference page No.)

(● Operating conditions to be considered)

- Installation space
- Load magnitude, types and direction of application
- Rotational speed
- Running accuracy
- Rigidity
- Misalignment
- Mounting ease ——— A 16
- Bearing arrangement ——— A 20
- Noise characteristics, friction torque
- Marketability, cost performance

(● Other data)

- Comparison of performance of bearing types ——— A 18
- Example of bearing arrangement ——— A 21

② Bearing dimension

- Specifications for installation
- Recommended service life ——— A 31
- Dynamic equivalent load ——— A 38
- Static equivalent load, safety coefficient ——— A 42
- Rotational speed

- Bearing boundary dimensions ——— A 52
- Basic dynamic load rating ——— A 24
- Basic static load rating ——— A 42
- Allowable axial load ——— A 44
  (for cylindrical roller bearing with rib)

→ **⑤ Cage type, material** → **⑥ Countermeasure for special environmental condition** →

⑤ Cage type, material

- Rotational speed
- Noise characteristics

⑥ Countermeasure for special environmental condition

- Conditions of application site
  (abnormal temperature, sea water, vacuum, chemical solution, dust, gas, magnetism)
- Lubrication

- Special bearing materials ——— A 130
- Special heat treatment ——— A 26
  (dimension stabilizing treatment)
- Special surface treatment
- Lubricant ——— A 124

(Reference) EXSEV & ceramic bearing series — C 57

Fig. 2-1(1) Bearing selection procedure

Next, from the bearing specifications are determined the service life required when compared to that of the equipment in which it is used, along with a calculation of the actual service life from operational loads.

Internal specifications including bearing accuracy, internal clearance, cage, and lubricant are also selected, depending on the application.

For reference, general selection procedure and operating conditions are described in Fig. 2-1. There is no need to follow a specific order, since the goal is to select the right bearing to achieve optimum performance.

③ Tolerance class

- Running accuracy (runout)
- Noise characteristics, friction torque
- Rotational speed

- Bearing tolerances ——— A 58

④ Fit and internal clearance

- Load magnitude, types
- Operational temperature distribution
- Materials, size and tolerances of shaft and housing
- Fit ——— A 86
- Difference in temperature of inner and outer rings
- Rotational speed
- Preload ——— A 112

- Bearing tolerances ——— A 58
- Bearing internal clearance ——— A 99

⑦ Lubrication, lubricant, sealing device

- Operating temperature
- Rotational speed
- Lubrication ——— A 117
- Lubricant ——— A 124
- Sealing device ——— A 137

- Limiting speed ——— A 84
- Grease service life ——— A 119

⑧ Mounting and dismounting, mounting dimension

- Mounting and dismounting — A 141

- Mounting dimensions — A 134

Final determination of bearing and associated aspect

Fig. 2-1(2) Bearing selection procedure

3. Selection of bearing type

In selecting bearings, the most important thing is to fully understand the operating conditions of the bearings.

The main factors to be considered are listed in Table 3-1, while bearing types are listed in Table 3-2.

Table 3-1 (1) Selection of bearing type

Items to be considered		Selection method	Reference page No.
1) Installation space	Bearing can be installed in target equipment	• When a shaft is designed, its rigidity and strength are considered essential; therefore, the shaft diameter, i.e., bore diameter, is determined at start. For rolling bearings, since wide variety with different dimensions are available, the most suitable bearing type should be selected. (Fig. 3-1)	A 52
2) Load	Load magnitude, type and direction which applied (Load resistance of bearing is specified in terms of the basic load rating, and its value is specified in the bearing specification table.)	• Since various types of load are applied to bearings, load magnitude, types (radial or axial) and direction of application (both directions or single direction in the case of axial load), as well as vibration and impact must be considered in order to select the proper bearing. • The following is the general order for radial resistance ; (deep groove ball bearings < angular contact ball bearings < cylindrical roller bearings < tapered roller bearings < spherical roller bearings)	A 18 (Table 3-2) A 87
3) Rotational speed	Response to rotational speed of equipment in which bearings will be installed (The limiting speed for bearing is expressed as allowable speed, and this value is specified in the bearing specification table.)	• Since the allowable speed differs greatly depend-ing not only upon bearing type but on bearing size, cage, accuracy, load and lubrication, all factors must be considered in selecting bearings. • In general, the following bearings are the most widely used for high speed operation. (deep groove ball bearings, angular contact ball bearings, cylindrical roller bearings)	A 18 (Table 3-2) A 84
4) Running accuracy	Accurate rotation delivering required performance (Dimension accuracy and running accuracy of bearings are provided by JIS, etc.)	• Performance required differs depending on equipment in which bearings are installed : for instance, machine tool spindles require high running accuracy, gas turbines require high speed rotation, and control equipment requires low friction. In such cases, bearings of tolerance class 5 or higher are required. • The following are the most widely used bearings. (deep groove ball bearings, angular contact ball bearings, cylindrical roller bearings)	A 18 (Table 3-2) A 58
5) Rigidity	Rigidity that delivers the bearing performance required (When load is applied to a bearing, elastic deformation occurs at the point where its rolling elements contact the raceway surface. The higher the rigidity that bearings possess, the better they control elastic deformation.)	• In machine tool spindles and automobile final drives, bearing rigidity as well as rigidity of equipment itself must be enhanced. • Elastic deformation occurs less in roller bearings than in ball bearings. • Rigidity can be enhanced by providing preload. This method is suitable for use with angular contact ball bearings and tapered roller bearings.	A 18 (Table 3-2) A 112

Table 3-1 (2) Selection of bearing type

Items to be considered		Selection method	Reference page No.
6) Misalignment (aligning capability)	Operating conditions which cause misalignment (shaft deflection caused by load, inaccuracy of shaft and housing, mounting errors) can affect bearing performance (Allowable misalignment (in angle) for each bearing type is described in the section before the bearing specification table, to facilitate determination of the self-aligning capability of bearings.)	• Internal load caused by excessive misalignment damages bearings. Bearings designed to absorb such misalignment should be selected. • The higher the self-aligning capability that bearings possess, the larger the angular misalignment that can be absorbed. The following is the general order of bearings when comparing allowable angular misalignment : (cylindrical roller bearings $<$ tapered roller-bearings $<$ deep groove ball bearings, angular contact ball bearings $<$ spherical roller-bearings, self-aligning ball bearings)	A 18 (Table 3-2)
7) Mounting and dismounting	Methods and frequency of mounting and dismounting required for periodic inspection	• Cylindrical roller bearings, needle roller bearings and tapered roller bearings, with separable inner and outer rings, are recommended for applications in which mounting and dismounting is conducted frequently. • Use of sleeve eases the mounting of self-aligning ball bearings and spherical roller bearings with tapered bore.	A 18 (Table 3-2)

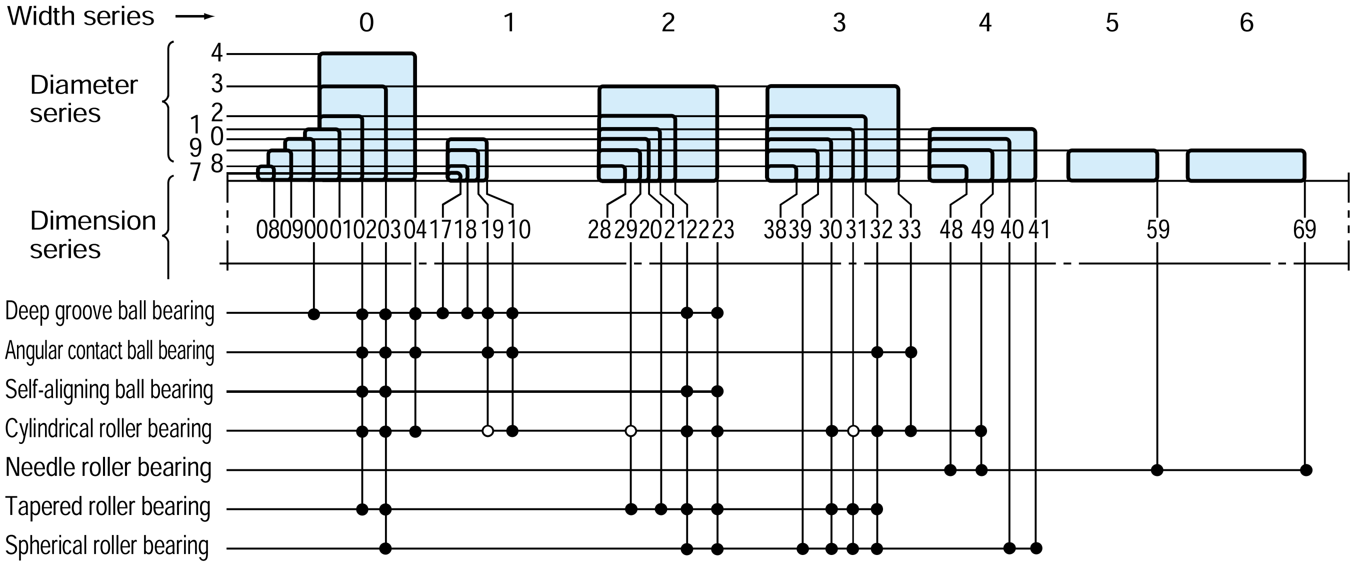
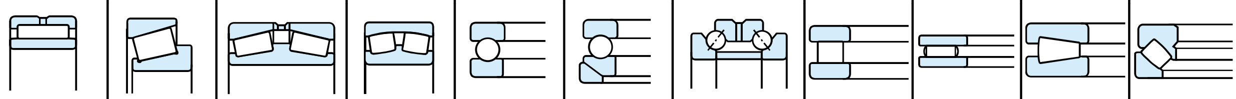


Fig. 3-1 Radial bearing dimension series

Table 3-2 Performance comparison of bearing type

		Deep groove ball bearing	Angular contact ball bearing: Single-row	Angular contact ball bearing: Matched pair or stack	Angular contact ball bearing: Double-row	Four-point contact ball bearing	Self-aligning ball bearing	Cylindrical roller bearing: NU · N	Cylindrical roller bearing: NJ · NF	Cylindrical roller bearing: NUP · NH	Cylindrical roller bearing: NN · NNU
Load resistance	Radial load	○	○	◎	◎	○	○	◎	◎	◎	◎
Load resistance	Axial load	○ ↔	◎ ←	◎ ↔ *	◎ ↔ *	◎ ↔	△ ↔	×	△ ←	△ ↔	×
Load resistance	Combined load radial and axial	○	○	◎	◎	○	△	×	△	△	×
Load resistance	Vibration or impact load	△	△	△	△	△	△	◎	◎	◎	◎
High speed adaptability		◎	◎	◎	○	◎	△	◎	◎	◎	◎
High accuracy		◎	◎	◎		◎		◎			◎
Low noise level/low torque		◎						○			
Rigidity				○		○		○	○	○	◎
Misalignment		○	△	×	×	×	◎	△	△	△	△
Inner and outer ring separability		×	×	×	×	■ *	×	■	■	■	■
Arrangement	Fixed side	■ ↔	■ ←	■ ↔	■ ↔ *	■ ↔	■ ↔	×	■ ←	■ ↔	×
Arrangement	Free side	□		□	□	□	□	■	□	□	■
Remarks			A pair of bearings mounted facing each other.	*DT arrangement is effective for one direction only.	*Filling slot type is effective for one direction only.	*Non-separable type is also available.					
Reference page No.		A4 B4	A5 B54			A6 —	A6 B124	A7 B138			

◎ Excellent ○ Good △ Fair × Unacceptable ↔ Both directions ← One direction only



	Needle roller bearing (machined ring type)	Tapered roller bearing		Spherical roller bearing	Thrust ball bearing		Double direction angular contact thrust ball bearing	Cylindrical roller thrust bearing	Needle roller thrust bearing	Tapered roller thrust bearing	Spherical thrust roller bearing	Reference page No.
		Single-row	Double-row, four-row		With flat back faces	With aligning seat race						
	◎	◎	◎	◎	×	×	×	×	×	×	△	—
	×	◎ ←	◎ ↔	△ ↔	○ ← *	○ ← *	◎ ↔	◎ ←	◎ ←	◎ ←	◎ ←	—
	×	◎	◎	△	×	×	×	×	×	×	△	—
	○	◎	◎	◎	△	△	△	○	○	◎	◎	—
	○	○	○	○	△	△	○	△	△	△	△	A16 A84
		○			○		◎					A16, 58 A117
												A16
	○	○	◎				○	◎	◎	◎		A16
	△	△	△	◎	×	◎	×	×	×	×	◎	A17 Description before specification table
	■	■	■	×	■	■	■	■	■ *	■	■	—
	×	■ ←	■ ↔	■ ↔								A20
	■		□	□								A20
		A pair of bearings mounted facing each other.			*Double direction bearings are effective for both directions.				*Non-separable type is also available.			—
	A8 B362	A9 B184		A10 B290	A11 B336		— —	A12 B448	A12 B444	A13 —	A13 B354	—

■ Acceptable □ Acceptable, but shaft shrinkage must be compensated for.

4. Selection of bearing arrangement

As bearing operational conditions vary depending on devices in which bearings are mounted, different performances are demanded of bearings. Normally, two or more bearings are used on one shaft.

In many cases, in order to locate shaft positions in the axial direction, one bearing is mounted on the fixed side first, then the other bearing is mounted on the free side.

Table 4-1 Bearings on fixed and free sides

	Features	Recommended bearing type	Example No.
Fixed side bearing	• This bearing determines shaft axial position. • This bearing can accommodate both radial and axial loads. • Since axial load in both directions is imposed on this bearing, strength must be considered in selecting the bearing for this side.	Deep groove ball bearing Matched pair or stack angular contact ball bearing Double-row angular contact ball bearing Self-aligning ball bearing Cylindrical roller bearing with rib (NUP and NH types) Double-row tapered roller bearing Spherical roller bearing	Examples 1–11
Free side bearing	• This bearing is employed to compensate for expansion or shrinkage caused by operating temperature change and to allow ajustment of bearing position. • Bearings which accommodate radial load only and whose inner and outer rings are separable are recommended as free side bearings. • In general, if non-separable bearings are used on free side, clearance fit is provided between outer ring and housing to compensate for shaft movement through bearings. In some cases, clearance fit between shaft and inner ring is utilized.	• Separable types Cylindrical roller bearing (NU and N types) Needle roller bearing (NA type, etc.) • Non-separable types Deep groove ball bearing Matched pair angular contact ball bearing (Back-to-back arrangement) Double-row angular contact ball bearing Self-aligning ball bearing Double-row tapered roller bearing (TDO type) Spherical roller bearing	
When fixed and free sides are not distinguished	• When bearing intervals are short and shaft shrinkage does not greatly affect bearing operation, a pair of angular contact ball bearings or tapered roller bearings is used in paired mounting to accommodate axial load. • After mounting, the axial clearance is adjusted using nuts or shims.	Deep groove ball bearing Angular contact ball bearing Self-aligning ball bearing Cylindrical roller bearing (NJ and NF types) Tapered roller bearing Spherical roller bearing	Examples 12–16
Bearings for vertical shafts	• Bearings which can accommodate both radial and axial loads should be used on fixed side. Heavy axial load can be accommodated using thrust bearings together with radial bearings. • Bearings which can accommodate radial load only are used on free side, compensating for shaft movement.	• Fixed side Matched pair angular contact ball bearing (Back-to-back arrangement) Double-row tapered roller bearing (TDO type) Thrust bearing + radial bearing	Examples 17 and 18

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Table 4-2 (1) Example bearing arrangements

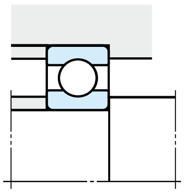
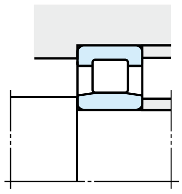
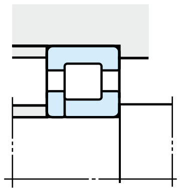
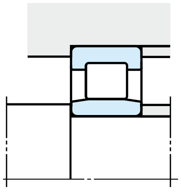
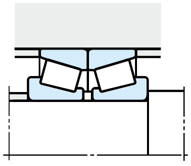
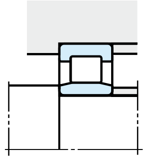
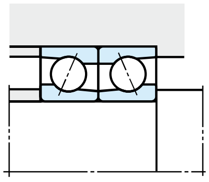
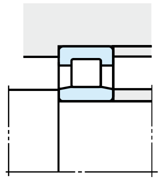
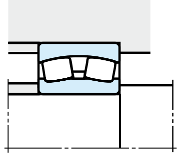
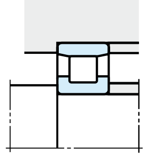
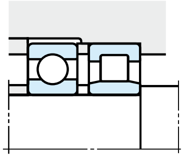
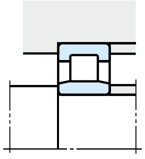
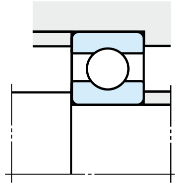
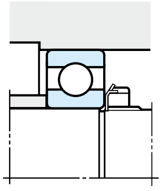
Example	Bearing arrangement		Recommended application	Application example
	Fixed side	Free side		
Ex. 1			○ Suitable for high-speed operation; used for various types of applications. ○ Not recommended for applications that have center displacement between bearings or shaft deflection.	Medium size motors, air blowers
Ex. 2			○ More suitable than Ex. 1 for operation under heavy load or impact load. Suitable also for high-speed operation. ○ Due to separability, suitable for applications requiring interference of both inner and outer rings. ○ Not recommended for applications that have center displacement between bearings or shaft deflection.	Traction motors for rail-way rolling stock
Ex. 3			○ Recommended for applications under heavier or greater impact load than those in Ex. 2. ○ This arrangement requires high rigidity from fixed side bearings mounted back to back, with preload provided. ○ Shaft and housing of accurate dimensions should be selected and mounted properly.	Steel manufac-turing table rollers, lathe spindles
Ex. 4			○ This is recommended for operation at high speed or axial load lighter than in Ex. 3. ○ This is recommended for applications requiring interference of both inner and outer rings. ○ Some applications use double-row angular con-tact ball bearings on fixed side instead of matched pair angular contact ball bearings.	Motors
Ex. 5			○ This is recommended for operations under rela-tively small axial load. ○ This is recommended for applications requiring interference of both inner and outer rings.	Paper manufac-turing calender rollers, diesel locomotive axle journals
Ex. 6			○ This is recommended for operations at high speed and heavy radial load, as well as normal axial load. ○ When deep groove ball bearings are used, clear-ance must be provided between outside diameter and housing, to prevent application of radial load.	Diesel locomotive transmissions
Ex. 7			○ This arrangement is most widely employed. ○ This arrangement can accommodate partial axial load as well as radial load.	Pumps, automobile transmissions

Table 4-2 (2) Example bearing arrangements

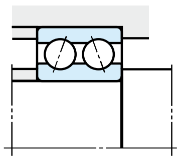
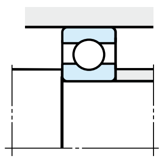
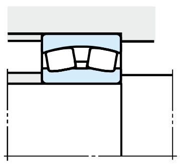
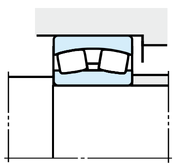
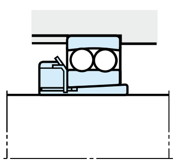
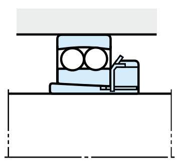
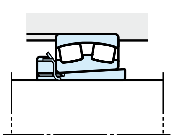
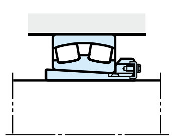
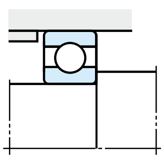
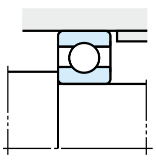
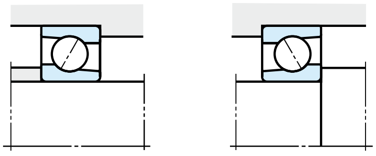
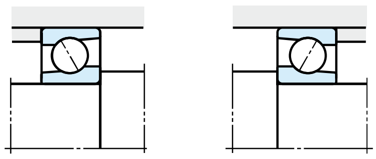
Example	Bearing arrangement: Fixed side	Bearing arrangement: Free side	Recommended application	Application example
Ex. 8			○ This is recommended for operations with relatively heavy axial load in both directions. ○ Some applications use matched pair angular contact ball bearings on fixed side instead of double-row angular contact ball bearings.	Worm gear speed reducers
Ex. 9			○ This is the optimum arrangement for applications with possible mounting errors or shaft deflection. ○ Bearings in this arrangement can accommodate partial axial load, as well as heavy radial load.	Steel manufacturing table roller speed reducers, overhead crane wheels
Ex. 10			○ This is optimum arrangement for applications with possible mounting errors or shaft deflection. ○ Ease of mounting and dismounting, ensured by use of adaptor, makes this arrangement suitable for long shafts which are neither stepped nor threaded. ○ This arrangement is not recommended for applications requiring axial load capability.	General industrial equipment counter shafts
Ex. 11			○ This is the optimum arrangement for applications with possible mounting errors or shaft deflection. ○ This is recommended for operations under impact load or radial load heavier than that in Ex. 10. ○ This arrangement can accommodate partial axial load as well as radial load.	Steel manufacturing table rollers
Arrangement in which fixed and free sides are not distinguished			**Recommended application**	**Application example**
Ex. 12			○ This arrangement is most popular when applied to small equipment operating under light load. ○ When used with light preloading, thickness-adjusted shim or spring is mounted on one side of outer ring.	Small motors, small speed reducers, small pumps
Ex. 13	 Back-to-back  Face-to-face		○ This is suitable for applications in which rigidity is enhanced by preloading. This is frequently employed in applications requiring high speed operation under relatively large axial load. ○ Back-to-back arrangement is suitable for applications in which moment load affects operation. ○ When preloading is required, care should be taken in preload adjustment.	Machine tool spindles

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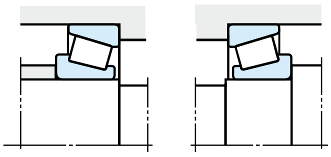
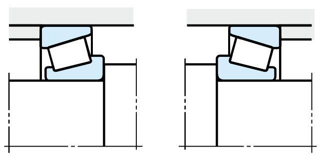
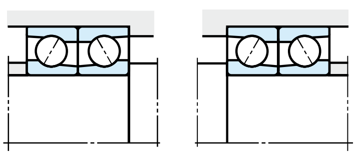
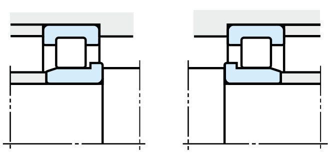
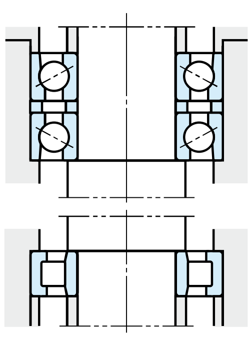
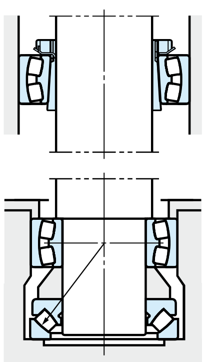
Table 4-2 (3) Example bearing arrangements

Example	Arrangement in which fixed and free sides are not distinguished	Recommended application	Application example
Ex. 14	 Back-to-back  Face-to-face	○ This is recommended for operation under impact load or axial load heavier than in Ex. 13. ○ This is suitable for applications in which rigidity is enhanced by preloading. ○ Back-to-back arrangement is suitable for applications in which moment load affects operation. ○ When interference is required between inner ring and shaft, face-to-face arrangement simplifies mounting. This arrangement is effective for applications in which mounting error is possible. ○ When preloading is required, care should be taken in preload adjustment.	Speed reducers, automobile wheels
Ex. 15		○ This is recommended for applications requiring high speed and high accuracy of rotation under light load. ○ This is suitable for applications in which rigidity is enhanced by preloading. ○ Tandem arrangement and face-to-face arrangement are possible, as is back-to-back arrangement.	Machine tool spindles
Ex. 16		○ This arrangement provides resistance against heavy radial and impact loads. ○ This is applicable when both inner and outer rings require interference. ○ Care should be taken not to reduce axial internal clearance a critical amount during operation.	Construction equipment final drive
Application to vertical shafts		**Recommended application**	**Application example**
Ex. 17	 Fixed side / Free side	○ This arrangement, using matched pair angular contact ball bearings on the fixed side and cylindrical roller bearings on the free side, is suitable for high speed operation.	Vertical motors, vertical pumps
Ex. 18	 Free side / Fixed side	○ This is recommended for operation at low speed and heavy load, in which axial load is heavier than radial load. ○ Due to self-aligning capability, this is suitable for applications in which shaft runout or deflection occurs.	Crane center shafts, vertical pumps

5. Selection of bearing dimensions

5-1 Bearing service life

When bearings rotate under load, material flakes from the surfaces of inner and outer rings or rolling elements by fatigue arising from repeated contact stress (ref. A 152).

This phenomenon is called flaking.

The total number of bearing rotations until flaking occurs is regarded as the bearing "(fatigue) service life".

"(Fatigue) service life" differs greatly depending upon bearing structures, dimensions, materials, and processing methods.
Since this phenomenon results from fatigue distribution in bearing materials themselves, differences in bearing service life should be statistically considered.

When a group of identical bearings are rotated under the same conditions, the total number of revolutions until 90 % of the bearings are left without flaking (i.e. a service life of 90 % reliability) is defined as the basic rating life. In operation at a constant speed, the basic rating life can be expressed in terms of time.

In actual operation, a bearing fails not only because of fatigue, but other factors as well, such as wear, seizure, creeping, fretting, brinelling, cracking etc (ref. A 152, 16. Examples of bearing failures).

These bearing failures can be minimized by selecting the proper mounting method and lubricant, as well as the bearing most suitable for the application.

5-2 Calculation of service life

5-2-1 Basic dynamic load rating C

The basic dynamic load rating is either pure radial (for radial bearings) or central axial load (for thrust bearings) of constant magnitude in a constant direction, under which the basic rating life of 1 million revolutions can be obtained, when the inner ring rotates while the outer ring is stationary, or vice versa. The basic dynamic load rating, which represents the capacity of a bearing under rolling fatigue, is specified as the basic dynamic radial load rating (C_r) for radial bearings, and basic dynamic axial load rating (C_a) for thrust bearings. These load ratings are listed in the specification table.

These values are prescribed by ISO 281/ 1990, and are subject to change by conformance to the latest ISO standards.

5-2-2 Basic rating life L_{10}

The basic rating life L_{10} is a service life of 90 % reliability when used under normal usage conditions for bearings of high manufacturing quality where the inside of the bearing is of a standard design made from bearing steel materials specified in JIS or equivalent materials.

The relationship between the basic dynamic load rating, dynamic equivalent load, and basic rating life of a bearing can be expressed using equation (5-1). This life calculation equation does not apply to bearings that are affected by factors such as plastic deformation of the contact surfaces of raceways and rolling elements due to extremely high load conditions (when P exceeds either the basic static load rating C_0 (refer to p. A 42) or 0.5C) or, conversely, to bearings that are affected by factors such as the contact surfaces of raceways and rolling elements slipping due to extremely low load conditions.

If conditions like these may be encountered, consult with JTEKT.

It is convenient to express the basic rating life in terms of time, using equation (5-2), when a bearing is used for operation at a constant speed; and, in terms of traveling distance (km), using equation (5-3), when a bearing is used in railway rolling stock or automobiles.

$$\begin{pmatrix}\text{Total}\\ \text{revolutions}\end{pmatrix} \quad L_{10} = \left(\frac{C}{P}\right)^{p} \quad \cdots\cdots\cdots\cdots\cdots(5\text{-}1)$$

$$(\text{Time}) \quad L_{10h} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^{p} \cdots\cdots\cdots(5\text{-}2)$$

$$\begin{pmatrix}\text{Running}\\ \text{distance}\end{pmatrix} \quad L_{10s} = \pi D L_{10} \quad \cdots\cdots\cdots\cdots\cdots(5\text{-}3)$$

where :

L_{10}	: basic rating life	10^6 revolutions
L_{10h}	: basic rating life	h
L_{10s}	: basic rating life	km
P	: dynamic equivalent load ·····················(refer to p. A 38.)	N
C	: basic dynamic load rating	N
n	: rotational speed	min^{-1}
p	: for ball bearings········ $p = 3$ for roller bearings······ $p = 10/3$	
D	: wheel or tire diameter	mm

Accordingly, where the dynamic equivalent load is P, and rotational speed is n, equation (5-4) can be used to calculate the basic dynamic load rating C; the bearing size most suitable for a specified purpose can then be selected, referring to the bearing specification table.

The recommended bearing service life differs depending on the machines with which the bearing is used, as shown in Table 5-5, p. A 31.

$$C = P\left(L_{10h} \times \frac{60n}{10^6}\right)^{1/p} \quad \cdots\cdots\cdots\cdots\cdots (5\text{-}4)$$

[Reference]

The equations using a service life coefficient (f_h) and rotational speed coefficient (f_n) respectively, based on equation (5-2), are as follows :

$$L_{10h} = 500\, f_h^{\,p} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (5\text{-}5)$$

Coefficient of service life :

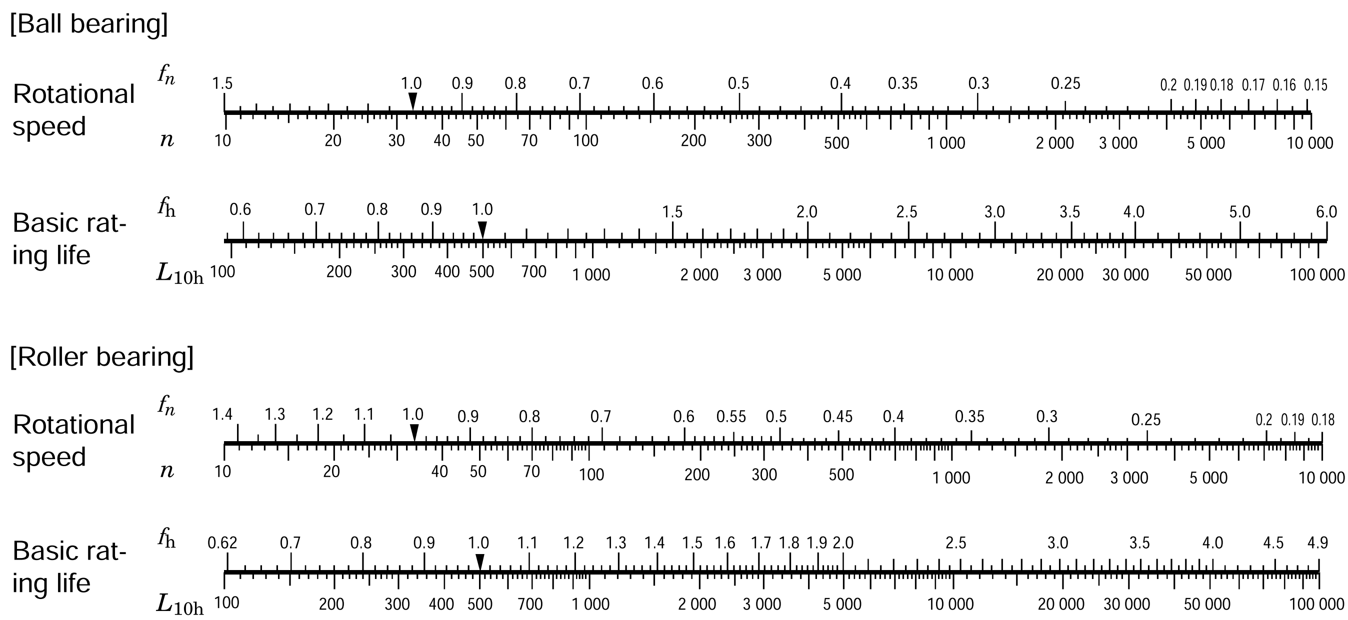
$$f_h = f_n \frac{C}{P} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (5\text{-}6)$$

Coefficient of rotational speed :

$$f_n = \left(\frac{10^6}{500 \times 60n}\right)^{1/p}$$

$$= (0.03n)^{-1/p} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (5\text{-}7)$$

For reference, the values of f_n, f_h, and L_{10h} can be easily obtained by employing the nomograph attached to this catalog, as an abbreviated method.



[Reference] Rotational speed (n) and its coefficients (f_n), and service life coefficient (f_h) and basic rating life (L_{10h})

5-2-3 Correction of basic dynamic load rating for high temperature use and dimension stabilizing treatment

In high temperature operation, bearing material hardness deteriorates, as material compositions are altered. As a result, the basic dynamic load rating is diminished. Once altered, material composition is not recovered, even if operating temperatures return to normal.

Therefore, for bearings used in high temperature operation, the basic dynamic load rating should be corrected by multiplying the basic dynamic load rating values specified in the bearing specification table by the temperature coefficient values in Table 5-1.

Table 5-1 Temperature coefficient values

Bearing temperature, °C	125	150	175	200	250
Temperature coefficient	1	1	0.95	0.90	0.75

Since normal heat treatment is not effective in maintaining the original bearing size in extended operation at 120 °C or higher, dimension stabilizing treatment is necessary. Dimension stabilizing treatment codes and their effective temperature ranges are described in Table 5-2.

Since dimension stabilizing treatment diminishes material hardness, the basic dynamic load rating may be reduced for some types of bearings.

Table 5-2 Dimension stabilizing treatment

Dimension stabilizing treatment code	Effective temperature range
S0	Over 100°C, up to 150°C
S1	150°C 200°C
S2	200°C 250°C

5-2-4 Modified rating life L_{nm}

The life of rolling bearings was standardized as a basic rating life in the 1960s, but in actual applications, sometimes the actual life and the basic rating life have been quite different due to the lubrication status and the influence of the usage environment. To make the calculated life closer to the actual life, a corrected rating life has been considered since the 1980s. In this corrected rating life, bearing characteristic factor a_2 (a correction factor for the case in which the characteristics related to the life are changed due to the bearing materials, manufacturing process, and design) and usage condition factor a_3 (a correction factor that takes into account usage conditions that have a direct influence on the bearing life, such as the lubrication) or factor a_{23} formed from the interdependence of these two factors, are considered with the basic rating life. These factors were handled differently by each bearing manufacturer, but they have been standardized as a modified rating life in **ISO 281** in 2007. In 2013, **JIS B 1518** (dynamic load ratings and rating life) was amended to conform to the **ISO**.

The basic rating life (L_{10}) shown in equation (5-1) is the (fatigue) life with a dependability of 90 % under normal usage conditions for rolling bearings that have standard factors such as internal design, materials, and manufacturing quality. **JIS B 1518**:2013 specifies a calculation method based on **ISO 281**:2007. To calculate accurate bearing life under a variety of operating conditions, it is necessary to consider elements such as the effect of changes in factors that can be anticipated when using different reliabilities and system approaches, and interactions between factors. Therefore, the specified calculation method considers additional stress due to the lubrication status, lubricant contamination, and fatigue load limit C_u (refer to p. A 29) on the inside of the bearing. The life that uses this life modification factor a_{ISO}, which considers the above factors, is called modified rating life L_{nm} and is calculated with the following equation (5-8).

$$L_{nm} = a_1\, a_{ISO}\, L_{10} \quad \cdots\cdots\cdots (5\text{-}8)$$

In this equation,

L_{nm} : Modified rating life 10^6 rotations
(This rating life has been modified for one of or a combination of the following: reliability of 90 % or higher, fatigue load limit, special bearing characteristics, lubrication contamination, and special operating conditions.)

L_{10} : Basic rating life 10^6 rotations
(reliability: 90 %)

a_1 : Life modification factor for reliability
··············· refer to section (1)

a_{ISO} : Life modification factor
··············· refer to section (2)

[Remark]
When bearing dimensions are to be selected given L_{nm} greater than 90 % in reliability, the strength of shaft and housing must be considered.

(1) Life modification factor for reliability a_1

The term "reliability" is defined as "for a group of apparently identical rolling bearings, operating under the same conditions, the percentage of the group that is expected to attain or exceed a specified life" in **ISO 281**:2007. Values of a_1 used to calculate a modified rating life with a reliability of 90 % or higher (a failure probability of 10 % or less) are shown in Table 5-3.

Table 5-3 Life modification factor for reliability a_1

Reliability, %	L_{nm}	a_1
90	L_{10m}	1
95	L_{5m}	0.64
96	L_{4m}	0.55
97	L_{3m}	0.47
98	L_{2m}	0.37
99	L_{1m}	0.25
99.2	$L_{0.8m}$	0.22
99.4	$L_{0.6m}$	0.19
99.6	$L_{0.4m}$	0.16
99.8	$L_{0.2m}$	0.12
99.9	$L_{0.1m}$	0.093
99.92	$L_{0.08m}$	0.087
99.94	$L_{0.06m}$	0.080
99.95	$L_{0.05m}$	0.077

(Citation from **JIS B 1518**:2013)

(2) Life modification factor a_{ISO}

a) System approach

The various influences on bearing life are dependent on each other. The system approach of calculating the modified life has been evaluated as a practical method for determining life modification factor a_{ISO} (ref. Fig. 5-1). Life modification factor a_{ISO} is calculated with the following equation. A diagram is available for each bearing type (radial ball bearings, radial roller bearings, thrust ball bearings, and thrust roller bearings). (Each diagram (Figs. 5-2 to 5-5) is a citation from **JIS B 1518**:2013.)

Note that in practical use, this is set so that life modification factor $a_{ISO} \leqq 50$.

$$a_{ISO} = f\left(\frac{e_c\, C_u}{P},\ \kappa\right) \quad \cdots\cdots\cdots (5\text{-}9)$$

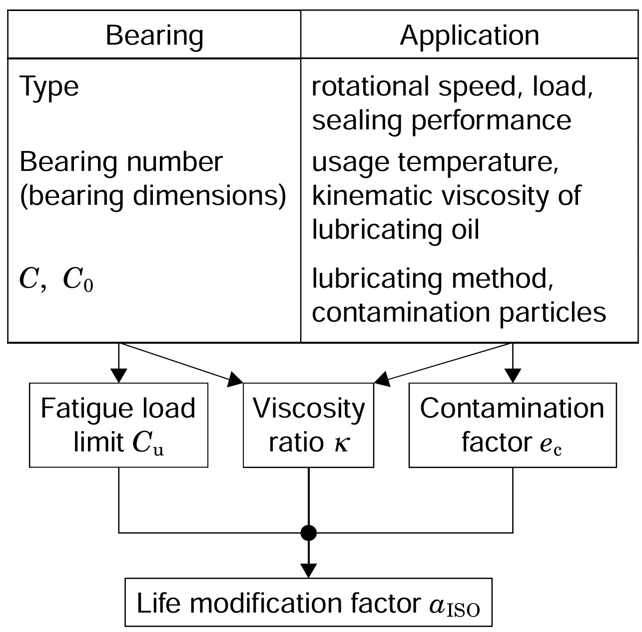


Fig. 5-1 System approach

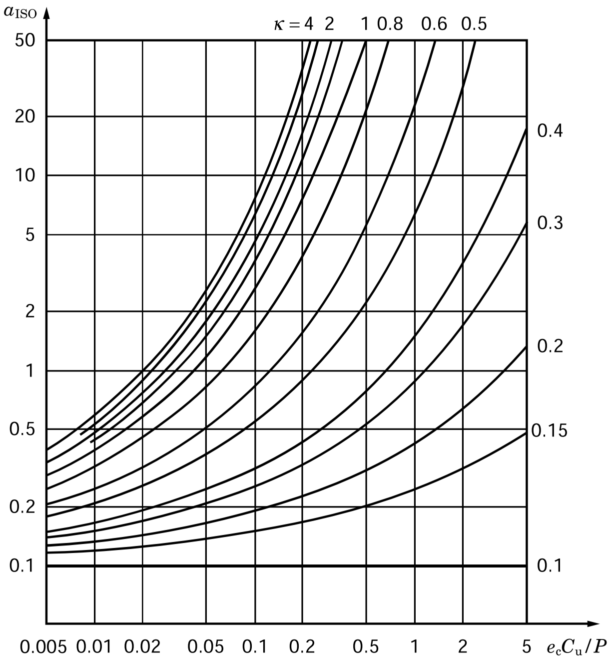


Fig. 5-2 **Life modification factor a_{ISO} (Radial ball bearings)**

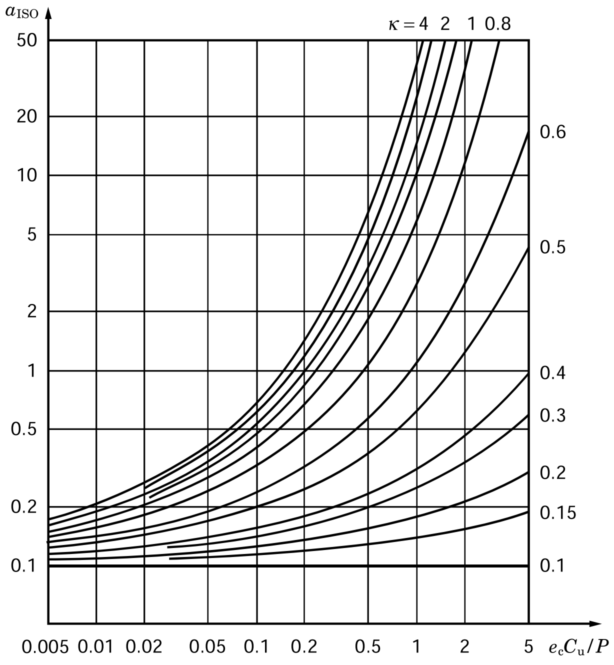


Fig. 5-3 **Life modification factor a_{ISO} (Radial roller bearings)**

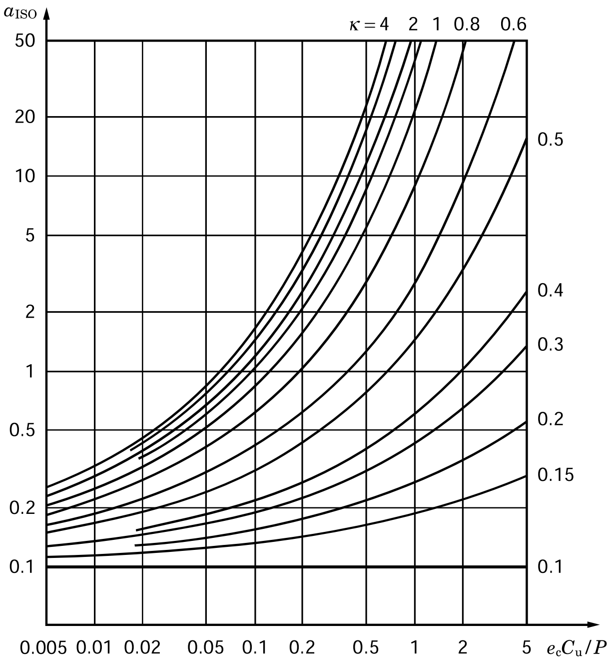


Fig. 5-4 **Life modification factor a_{ISO} (Thrust ball bearings)**

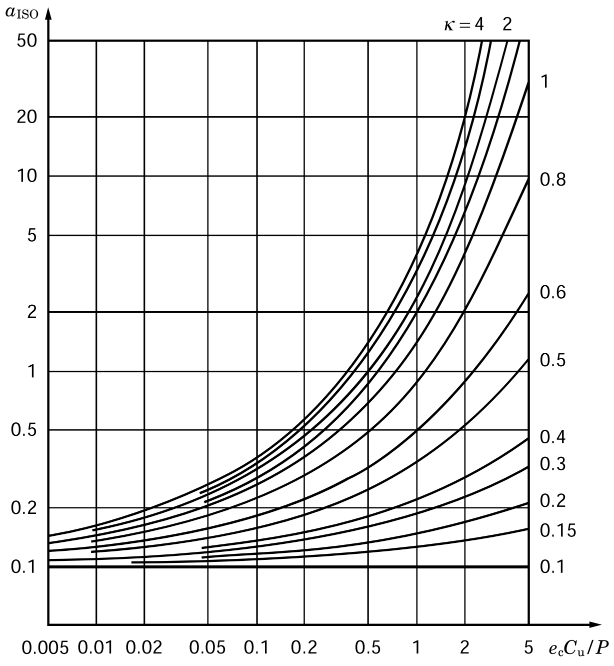


Fig. 5-5 **Life modification factor a_{ISO} (Thrust roller bearings)**

(Figs. 5-2 to 5-5 Citation from **JIS B 1518**:2013)

b) Fatigue load limit C_u

For regulated steel materials or alloy steel that has equivalent quality, the fatigue life is unlimited so long as the load condition does not exceed a certain value and so long as the lubrication conditions, lubrication cleanliness class, and other operating conditions are favorable. For general high-quality materials and bearings with high manufacturing quality, the fatigue stress limit is reached at a contact stress of approximately 1.5 GPa between the raceway and rolling elements. If one or both of the material quality and manufacturing quality are low, the fatigue stress limit will also be low.

The term "fatigue load limit" C_u is defined as "bearing load under which the fatigue stress limit is just reached in the most heavily loaded raceway contact" in **ISO 281**:2007. and is affected by factors such as the bearing type, size, and material.

For details on the fatigue load limits of special bearings and other bearings not listed in this catalog, contact JTEKT.

c) Contamination factor e_c

If solid particles in the contaminated lubricant are caught between the raceway and the rolling elements, indentations may form on one or both of the raceway and the rolling elements. These indentations will lead to localized increases in stress, which will decrease the life. This decrease in life attributable to the contamination of the lubricant can be calculated from the contamination level as contamination factor e_c.

D_{pw} shown in this table is the pitch diameter of ball/roller set, which is expressed simply as $D_{pw} = (D + d)/2$. (D: Outside diameter, d: Bore diameter)

For information such as details on special lubricating conditions or detailed investigations, contact JTEKT.

Table 5-4 Values of contamination factor e_c

Contamination level	e_c	
	$D_{pw} < 100$ mm	$D_{pw} \geqq 100$ mm
Extremely high cleanliness: The size of the particles is approximately equal to the thickness of the lubricant oil film, this is found in laboratory-level environments.	1	1
High cleanliness: The oil has been filtered by an extremely fine filter, this is found with standard grease-packed bearings and sealed bearings.	0.8 to 0.6	0.9 to 0.8
Standard cleanliness: The oil has been filtered by a fine filter, this is found with standard grease-packed bearings and shielded bearings.	0.6 to 0.5	0.8 to 0.6
Minimal contamination: The lubricant is slightly contaminated.	0.5 to 0.3	0.6 to 0.4
Normal contamination: This is found when no seal is used and a coarse filter is used in an environment in which wear debris and particles from the surrounding area penetrate into the lubricant.	0.3 to 0.1	0.4 to 0.2
High contamination: This is found when the surrounding environment is considerably contaminated and the bearing sealing is insufficient.	0.1 to 0	0.1 to 0
Extremely high contamination	0	0

(Table 5-4 Citation from **JIS B 1518**:2013)

d) Viscosity ratio κ

The lubricant forms an oil film on the roller contact surface, which separates the raceway and the rolling elements. The status of the lubricant oil film is expressed by viscosity ratio κ, the actual kinematic viscosity at the operating temperature ν divided by the reference kinematic viscosity ν_1 as shown in the following equation.

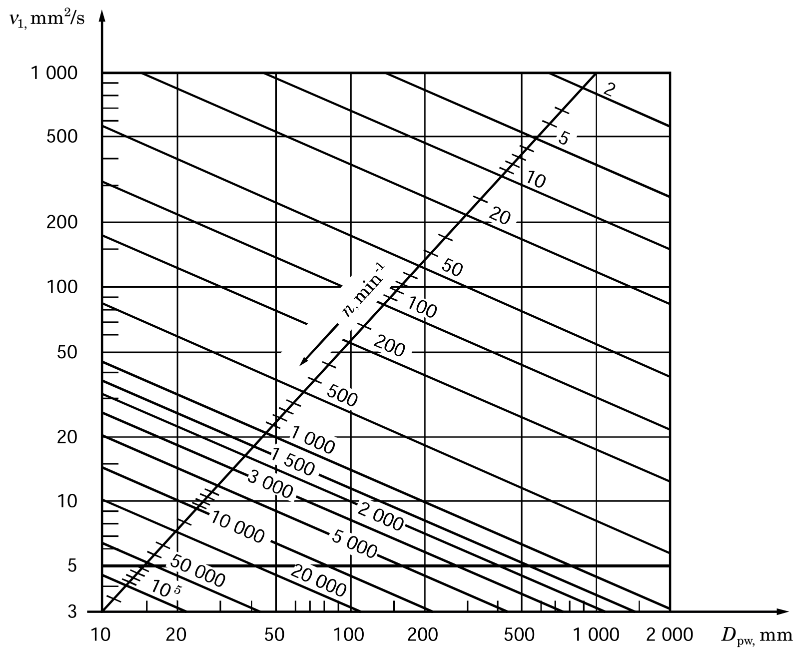
A κ greater than 4, equal to 4, or less than 0.1 is not applicable.

For details on lubricants such as grease and lubricants with extreme pressure additives, contact JTEKT.

$$\kappa = \frac{\nu}{\nu_1} \quad \cdots\cdots (5\text{-}10)$$

ν : Actual kinematic viscosity at the operating temperature; the viscosity of the lubricant at the operating temperature (refer to Fig. 12-3, p. A129)

ν_1 : Reference kinematic viscosity; determined according to the speed and pitch diameter of ball/roller set D_{pw} of the bearing (ref. Fig. 5-6)



(Fig. 5-6 Citation from **JIS B 1518**:2013)

Fig. 5-6 Reference kinematic viscosity ν_1

5-2-5 Service life of bearing system comprising two or more bearings

Even for systems which comprise two or more bearings, if one bearing is damaged, the entire system malfunctions.

Where all bearings used in an application are regarded as one system, the service life of the bearing system can be calculated using the following equation,

$$\frac{1}{L^e} = \frac{1}{L_1{}^e} + \frac{1}{L_2{}^e} + \frac{1}{L_3{}^e} + \cdots\cdots\cdots \quad (5\text{-}11)$$

where :

L : rating life of system

L_1, L_2, L_3…… : rating life of each bearing

e : constant

e = 10/9……ball bearing
e = 9/8……roller bearing
The mean value is for a system using both ball and roller bearings.

[Example]

When a shaft is supported by two roller bearings whose service lives are 50 000 hours and 30 000 hours respectively, the rating life of the bearing system supporting this shaft is calculated as follows, using equation (5-11) :

$$\frac{1}{L^{9/8}} = \frac{1}{50\,000^{9/8}} + \frac{1}{30\,000^{9/8}}$$

$$L \fallingdotseq 20\,000 \text{ h}$$

The equation suggests that the rating life of these bearings as a system becomes shorter than that of the bearing with the shorter life.

This fact is very important in estimating bearing service life for applications using two or more bearings.

5-2-6 Applications and recommended bearing service life

Since longer service life does not always contribute to economical operation, the most suitable service life for each application and operating conditions should be determined.

For reference, Table 5-5 describes recommended service life in accordance with the application, as empirically determined.

Table 5-5 Recommended bearing service life (reference)

Operating condition	Application	Recommended service life (h)
Short or intermittent operation	Household electric appliance, electric tools, agricultural equipment, heavy cargo hoisting equipment	4 000 – 8 000
Not extended duration, but stable operation required	Household air conditioner motors, construction equipment, conveyers, elevators	8 000 – 12 000
Intermittent but extended operation	Rolling mill roll necks, small motors, cranes	8 000 – 12 000
	Motors used in factories, general gears	12 000 – 20 000
	Machine tools, shaker screens, crushers	20 000 – 30 000
	Compressors, pumps, gears for essential use	40 000 – 60 000
Daily operation more than 8 hr. or continuous extended operation	Escalators	12 000 – 20 000
	Centrifugal separators, air conditioners, air blowers, woodworking equipment, passenger coach axle journals	20 000 – 30 000
	Large motors, mine hoists, locomotive axle journals, railway rolling stock traction motors	40 000 – 60 000
	Paper manufacturing equipment	100 000 – 200 000
24 hr. operation (no failure allowed)	Water supply facilities, power stations, mine water discharge facilities	100 000 – 200 000

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5-3 Calculation of loads

Loads affecting bearings includes force exerted by the weight of the object the bearings support, transmission force of devices such as gears and belts, loads generated in equipment during operation etc.

Seldom can these kinds of load be determined by simple calculation, because the load is not always constant.

In many cases, the load fluctuates, and it is difficult to determine the frequency and magnitude of the fluctuation.

Therefore, loads are normally obtained by multiplying theoretical values with various coefficients obtained empirically.

5-3-1 Load coefficient

Even if radial and axial loads are obtained through general dynamic calculation, the actual load becomes greater than the calculated value due to vibration and impact during operation.

In many cases, the load is obtained by multiplying theoretical values by the load coefficient.

$$F = f_w \cdot F_c \quad \cdots\cdots\cdots (5\text{-}12)$$

where :
- F : measured load N
- F_c : calculated load N
- f_w : load coefficient (ref. Table 5-6)

Table 5-6 Values of load coefficient f_w

Operating condition	Application example	f_w
Operation with little vibration or impact	Motors Machine tools Measuring instrument	1.0 – 1.2
Normal operation (slight impact)	Railway rolling stock Automobiles Paper manufacturing equipment Air blowers Compressors Agricultural equipment	1.2 – 2.0
Operation with severe vibration or impact	Rolling mills Crushers Construction equipment Shaker screens	2.0 – 3.0

5-3-2 Load generated through belt or chain transmission

In the case of belt transmission, the theoretical value of the load affecting the pulley shafts can be determined by obtaining the effective transmission force of the belt.

For actual operation, the load is obtained by multiplying this effective transmission force by the load coefficient (f_w) considering vibration and impact generated during operation, and the belt coefficient (f_b) considering belt tension.

In the case of chain transmission, the load is determined using a coefficient equivalent to the belt coefficient.

This equation (5-13) is as follows ;

$$F_b = \frac{2\,M}{D_p} \cdot f_w \cdot f_b$$

$$= \frac{19.1\times10^6\,W}{D_p n} \cdot f_w \cdot f_b \quad \cdots\cdots (5\text{-}13)$$

where :
- F_b : estimated load affecting pulley shaft or sprocket shaft N
- M : torque affecting pulley or sprocket mN · m
- W : transmission force kW
- D_p : pitch circle diameter of pulley or sprocket mm
- n : rotational speed min^{-1}
- f_w : load coefficient (ref. Table 5-6)
- f_b : belt coefficient (ref. Table 5-7)

Table 5-7 Values of belt coefficient f_b

Belt type	f_b
Timing belt (with teeth)	1.3 – 2.0
V-belt	2.0 – 2.5
Flat belt (with tension pulley)	2.5 – 3.0
Flat belt	4.0 – 5.0
Chain	1.2 – 1.5

5-3-3 Load generated under gear transmission

(1) Loads affecting gear and gear coefficient

In the case of gear transmission, loads transmitted by gearing are theoretically classified into three types: tangential load (K_t), radial load (K_r) and axial load (K_a).

Those loads can be calculated dynamically (using equations ⓐ, ⓑ and ⓒ, described in section (2)).

To determine the actual gear loads, these theoretical loads must be multiplied by coefficients considering vibration and impact during operation (f_w) (ref. Table 5-6) and the gear coefficient (f_g) (ref. Table 5-8) considering the finish treatment of gears.

Table 5-8 Values of gear coefficient f_g

Gear type	f_g
Precision gears (both pitch error and tooth shape error less than 0.02 mm)	1.0 – 1.1
Normal gears (both pitch error and tooth shape error less than 0.1 mm)	1.1 – 1.3

(2) Calculation of load on gears

ⓐ Tangential load (tangential force) K_t

Spur gears, helical gears, double-helical gears, straight bevel gears, spiral bevel gears

$$K_t = \frac{2\,M}{D_p} = \frac{19.1\times10^6\ W}{D_p n} \quad \cdots\cdots (5\text{-}14)$$

ⓐ~ⓒ where :

K_t	: gear tangential load	N
K_r	: gear radial load	N
K_a	: gear axial load	N
M	: torque affecting gears	mN · m
D_p	: gear pitch circle diameter	mm
W	: transmitting force	kW
n	: rotational speed	min^{-1}
α	: gear pressure angle	deg
β	: gear helix (spiral) angle	deg
δ	: bevel gear pitch angle	deg

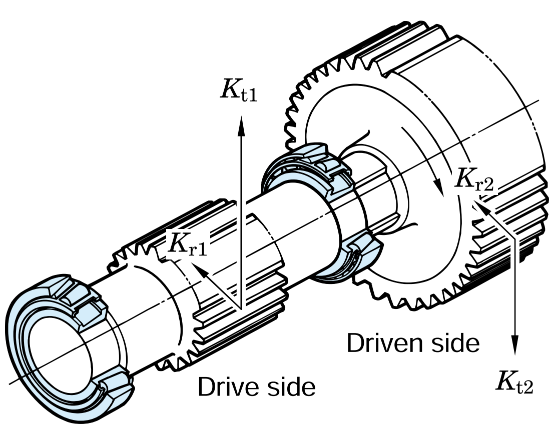


Fig. 5-7 Load on spur gears

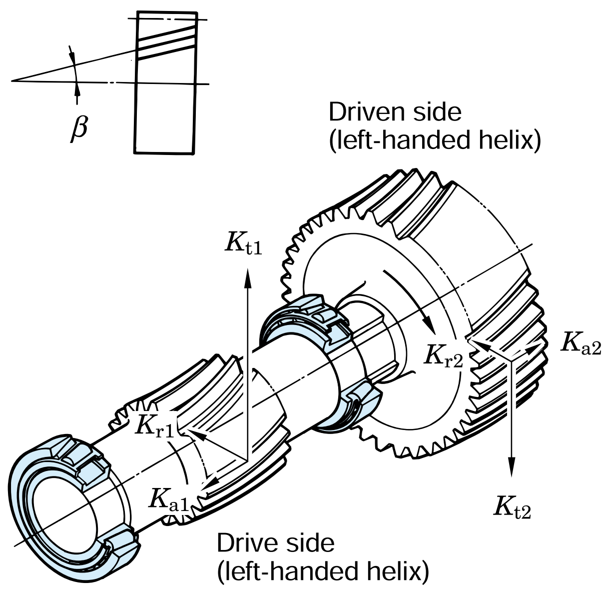


Fig. 5-8 Load on helical gears

		ⓑ Radial load (separating force) K_r	ⓒ Axial load (axial force) K_a
Spur gears		$K_r = K_t \tan\alpha$ ········ (5-15)	0
Helical gears		$K_r = K_t \dfrac{\tan\alpha}{\cos\beta}$ ········ (5-16)	$K_a = K_t \tan\beta$ ········ (5-22)
Double-helical gears		$K_r = K_t \dfrac{\tan\alpha}{\cos\beta}$ ········ (5-17)	0
Straight[1)] bevel gears	Drive side	$K_{r1} = K_t \tan\alpha \cos\delta_1$ ········ (5-18)	$K_{a1} = K_t \tan\alpha \sin\delta_1$ ········ (5-23)
	Driven side	$K_{r2} = K_t \tan\alpha \cos\delta_2$ ········ (5-19)	$K_{a2} = K_t \tan\alpha \sin\delta_2$ ········ (5-24)
Spiral[1), 2)] bevel gears	Drive side	$K_{r1} = \dfrac{K_t}{\cos\beta}\left(\tan\alpha \cos\delta_1 \pm \sin\beta \sin\delta_1\right)$ ········ (5-20)	$K_{a1} = \dfrac{K_t}{\cos\beta}\left(\tan\alpha \sin\delta_1 \mp \sin\beta \cos\delta_1\right)$ ········ (5-25)
	Driven side	$K_{r2} = \dfrac{K_t}{\cos\beta}\left(\tan\alpha \cos\delta_2 \mp \sin\beta \sin\delta_2\right)$ ········ (5-21)	$K_{a2} = \dfrac{K_t}{\cos\beta}\left(\tan\alpha \sin\delta_2 \pm \sin\beta \cos\delta_2\right)$ ········ (5-26)

[Notes] 1) Codes with subscript 1 and 2 shown in equations are respectively applicable to drive side gears and driven side gears.
2) Symbols (+) and (–) denote the following ;
- Symbols in upper row : clockwise rotation accompanied by right-handed spiral or counterclockwise rotation with left-handed spiral
- Symbols in lower row : counterclockwise rotation with right-handed spiral or clockwise rotation with left-handed spiral

[Remark] Rotating directions are described as viewed at the back of the apex of the pitch angle.

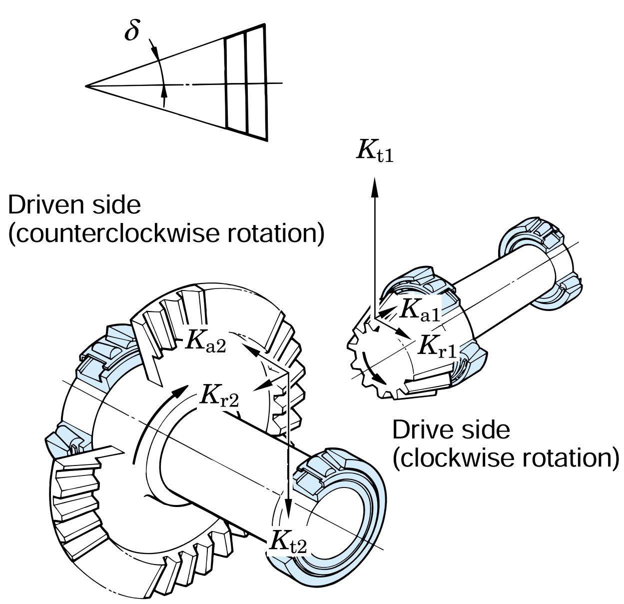


Fig. 5-9 Load on straight bevel gears

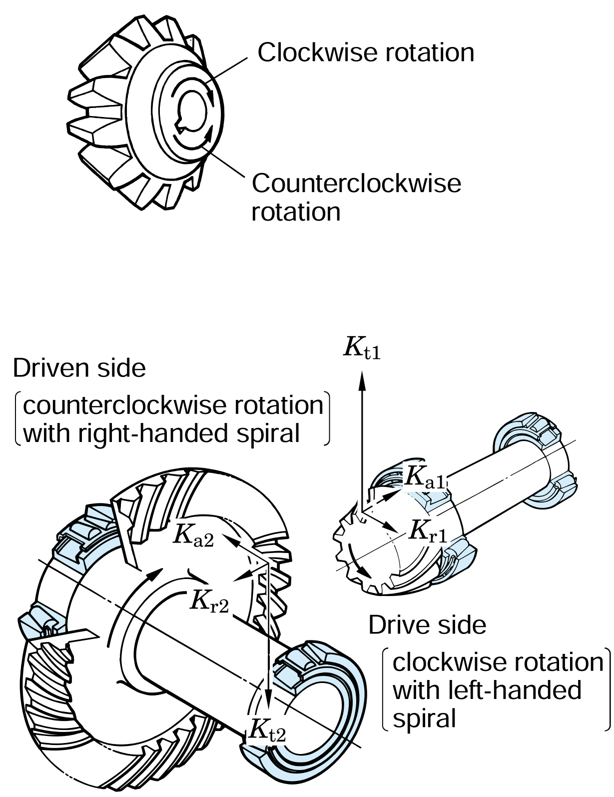


Fig. 5-10 Load on spiral bevel gears

5-3-4 Load distribution on bearings

The load distribution affecting bearings can be calculated as follows: first, radial force components are calculated, then, the sum of vectors of the components is obtained in accordance with the load direction.

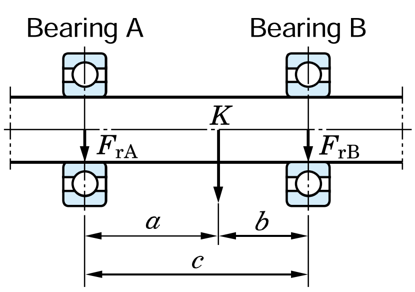
Calculation examples of radial load distribution are described in the following section.

[Remark]

Bearings shown in Exs. 3 to 5 are affected by components of axial force when these bearings accommodate radial load, and axial load (K_a) which is transferred externally, i.e. from gears.

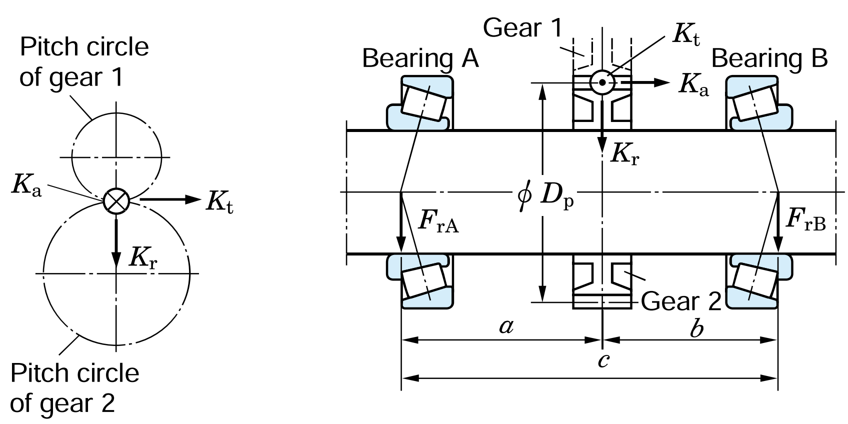
For calculation of the axial load in this case, refer to page A 38.

Example 1 Fundamental calculation (1)



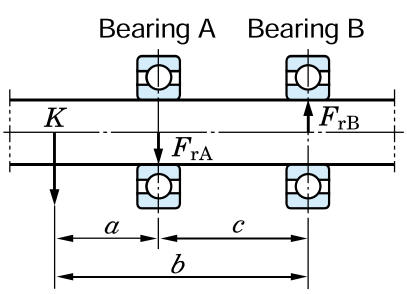
$$F_{rA} = \frac{b}{c}K$$
$$F_{rB} = \frac{a}{c}K \qquad \cdots\cdots (5\text{-}27)$$

Example 3 Gear load distribution (1)



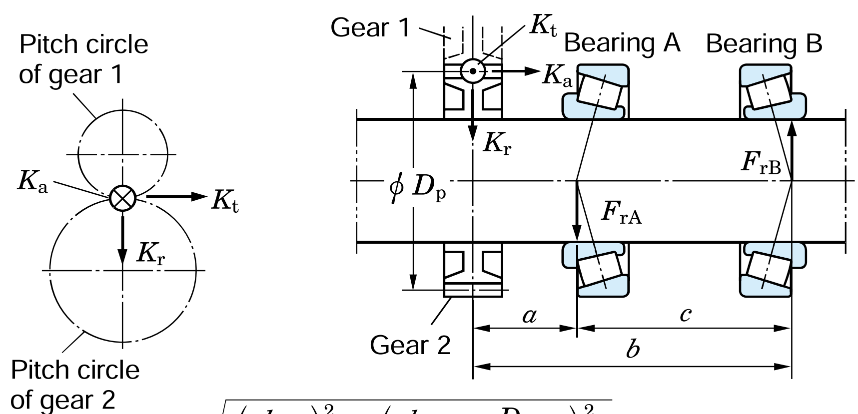
$$F_{rA} = \sqrt{\left(\frac{b}{c}K_t\right)^2 + \left(\frac{b}{c}K_r - \frac{D_p}{2c}K_a\right)^2}$$
$$F_{rB} = \sqrt{\left(\frac{a}{c}K_t\right)^2 + \left(\frac{a}{c}K_r + \frac{D_p}{2c}K_a\right)^2} \qquad \cdots\cdots (5\text{-}29)$$

Example 2 Fundamental calculation (2)



$$F_{rA} = \frac{b}{c}K$$
$$F_{rB} = \frac{a}{c}K \qquad \cdots\cdots (5\text{-}28)$$

Example 4 Gear load distribution (2)

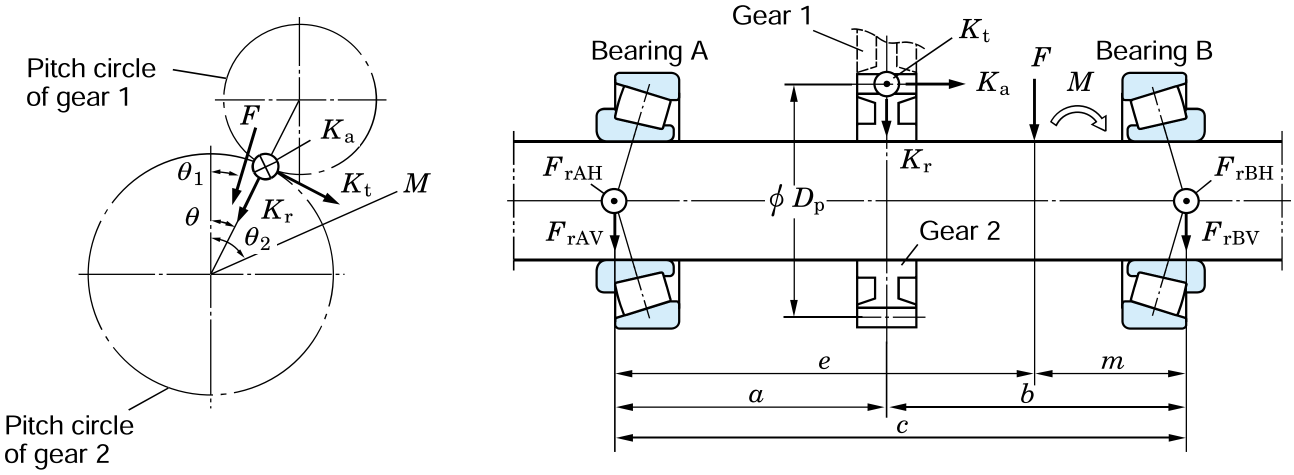


$$F_{rA} = \sqrt{\left(\frac{b}{c}K_t\right)^2 + \left(\frac{b}{c}K_r - \frac{D_p}{2c}K_a\right)^2}$$
$$F_{rB} = \sqrt{\left(\frac{a}{c}K_t\right)^2 + \left(\frac{a}{c}K_r - \frac{D_p}{2c}K_a\right)^2} \qquad \cdots\cdots (5\text{-}30)$$

Description of signs in Examples 1 to 5

F_{rA} : radial load on bearing A	N	D_p : gear pitch circle diameter	mm
F_{rB} : radial load on bearing B	N	⊙ : denotes load direction (upward perpendicular to paper surface)	
K : shaft load	N	⊗ : denotes load direction (downward perpendicular to paper surface)	
K_t, K_r, K_a : gear load (ref. A 34)	N		

Example 5 Simultaneous application of gear load and other load



(Gears 1 and 2 are engaged with each other at angle θ. External load F, moment M, are applied to these gears at angles θ_1 and θ_2.)

- Perpendicular radial component force (upward and downward along diagram)

$$F_{rAV} = \frac{b}{c}(K_r\cos\theta + K_t\sin\theta) - \frac{D_p}{2c}K_a\cos\theta + \frac{m}{c}F\cos\theta_1 - \frac{M}{c}\cos\theta_2$$

$$F_{rBV} = \frac{a}{c}(K_r\cos\theta + K_t\sin\theta) + \frac{D_p}{2c}K_a\cos\theta + \frac{e}{c}F\cos\theta_1 + \frac{M}{c}\cos\theta_2$$

- Horizontal radial component force (upward and downward perpendicular to diagram)

$$F_{rAH} = \frac{b}{c}(K_r\sin\theta - K_t\cos\theta) - \frac{D_p}{2c}K_a\sin\theta + \frac{m}{c}F\sin\theta_1 - \frac{M}{c}\sin\theta_2$$

$$F_{rBH} = \frac{a}{c}(K_r\sin\theta - K_t\cos\theta) + \frac{D_p}{2c}K_a\sin\theta + \frac{e}{c}F\sin\theta_1 + \frac{M}{c}\sin\theta_2$$

■ Combined radial force

$$F_{rA} = \sqrt{F_{rAV}^{\ 2} + F_{rAH}^{\ 2}}$$

$$F_{rB} = \sqrt{F_{rBV}^{\ 2} + F_{rBH}^{\ 2}} \quad \cdots\cdots\cdots\cdots (5\text{-}31)$$

(When θ, F, and M are zero, the same result as in Ex. 3 is obtained)

5-4 Dynamic equivalent load

Bearings are used under various operating conditions; however, in most cases, bearings receive radial and axial load combined, while the load magnitude fluctuates during operation.

Therefore, it is impossible to directly compare the actual load and basic dynamic load rating.

The two are compared by replacing the loads applied to the shaft center with one of a constant magnitude and in a specific direction, that yields the same bearing service life as under actual load and rotational speed.

This theoretical load is referred to as the dynamic equivalent load (P).

5-4-1 Calculation of dynamic equivalent load

Dynamic equivalent loads for radial bearings and thrust bearings ($\alpha \neq 90°$) which receive a combined load of a constant magnitude in a specific direction can be calculated using the following equation,

$$P = XF_r + YF_a \quad \cdots\cdots\cdots (5\text{-}32)$$

where :

- P : dynamic equivalent load N
  - for radial bearings, P_r : dynamic equivalent radial load
  - for thrust bearings, P_a : dynamic equivalent axial load
- F_r : radial load N
- F_a : axial load N
- X : radial load factor
- Y : axial load factor

(values of X and Y are listed in the bearing specification table.)

■ When $F_a/F_r \leqq e$ for single-row radial bearings, it is taken that $X = 1$, and $Y = 0$.
Hence, the dynamic equivalent load rating is $P_r = F_r$.

(Values of e, which designates the limit of F_a/F_r, are listed in the bearing specification table.)

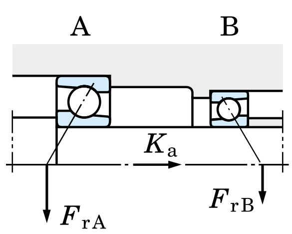
■ For single-row angular contact ball bearings and tapered roller bearings, axial component forces (F_{ac}) are generated as shown in Fig. 5-11, therefore a pair of bearings is arranged face-to-face or back-to-back.

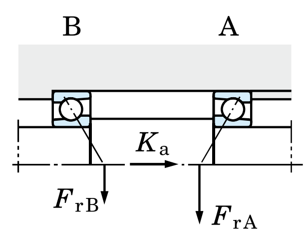
The axial component force can be calculated using the following equation.

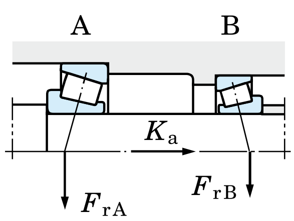
$$F_{ac} = \frac{F_r}{2\,Y} \quad \cdots\cdots\cdots (5\text{-}33)$$

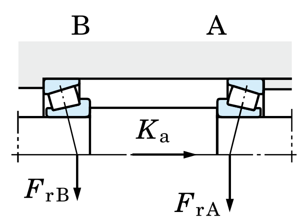
Table 5-9 describes the calculation of the dynamic equivalent load when radial loads and external axial loads (K_a) are applied to bearings.

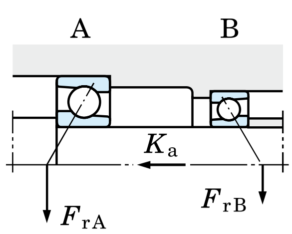
Paired mounting	
Back-to-back arrangement	Face-to-face arrangement

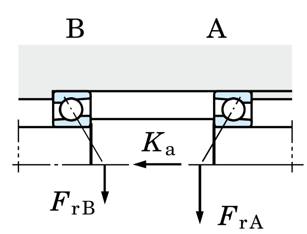


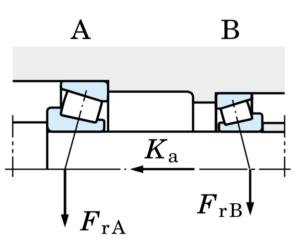


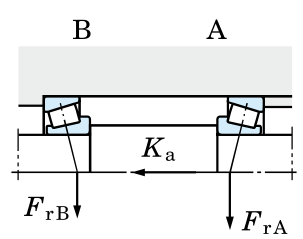


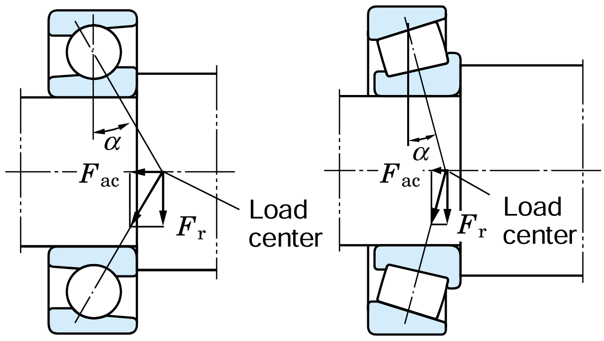












(Load center position is listed in the bearing specification table.)

Fig. 5-11 Axial component force

- For thrust ball bearings with contact angle $\alpha = 90°$, to which an axial load is applied, $P_a = F_a$.
- The dynamic equivalent load of spherical thrust roller bearing can be calculated using the following equation.

$$P_a = F_a + 1.2 F_r \quad \cdots\cdots\cdots\cdots \quad (5\text{-}34)$$

where : $F_r / F_a \leqq 0.55$

Table 5-9 Dynamic equivalent load calculation : when a pair of single-row angular contact ball bearings or tapered roller bearings is arranged face-to-face or back-to-back.

	Loading condition	Bearing	Axial load	Dynamic equivalent load
	$\frac{F_{rB}}{2Y_B} + K_a \geqq \frac{F_{rA}}{2Y_A}$	Bearing A	$\frac{F_{rB}}{2Y_B} + K_a$	$P_A = XF_{rA} + Y_A\left(\frac{F_{rB}}{2Y_B} + K_a\right)$ $P_A = F_{rA}$, where $P_A < F_{rA}$
		Bearing B	–	$P_B = F_{rB}$
	$\frac{F_{rB}}{2Y_B} + K_a < \frac{F_{rA}}{2Y_A}$	Bearing A	–	$P_A = F_{rA}$
		Bearing B	$\frac{F_{rA}}{2Y_A} - K_a$	$P_B = XF_{rB} + Y_B\left(\frac{F_{rA}}{2Y_A} - K_a\right)$ $P_B = F_{rB}$, where $P_B < F_{rB}$
	$\frac{F_{rB}}{2Y_B} \leqq \frac{F_{rA}}{2Y_A} + K_a$	Bearing A	–	$P_A = F_{rA}$
		Bearing B	$\frac{F_{rA}}{2Y_A} + K_a$	$P_B = XF_{rB} + Y_B\left(\frac{F_{rA}}{2Y_A} + K_a\right)$ $P_B = F_{rB}$, where $P_B < F_{rB}$
	$\frac{F_{rB}}{2Y_B} > \frac{F_{rA}}{2Y_A} + K_a$	Bearing A	$\frac{F_{rB}}{2Y_B} - K_a$	$P_A = XF_{rA} + Y_A\left(\frac{F_{rB}}{2Y_B} - K_a\right)$ $P_A = F_{rA}$, where $P_A < F_{rA}$
		Bearing B	–	$P_B = F_{rB}$

[Remarks] 1. These equations can be used when internal clearance and preload during operation are zero.
2. Radial load is treated as positive in the calculation, if it is applied in a direction opposite that shown in Fig. in Table 5-9.

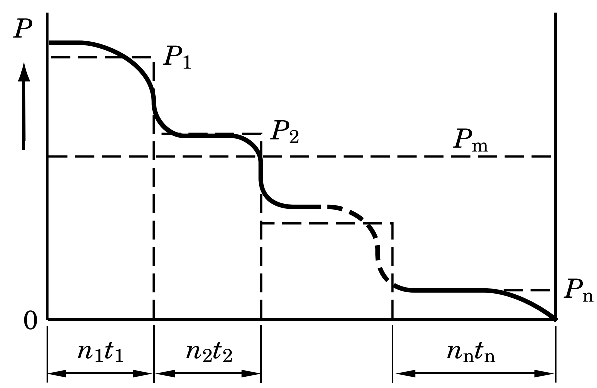
5-4-2 Mean dynamic equivalent load

When load magnitude or direction varies, it is necessary to calculate the mean dynamic equivalent load, which provides the same length of bearing service life as that under the actual load fluctuation.

The mean dynamic equivalent load (P_m) under different load fluctuations is described using Graphs (1) to (4).

As shown in Graph (5), the mean dynamic equivalent load under stationary and rotating load applied simultaneously, can be obtained using equation (5-39).

(1) Staged fluctuation



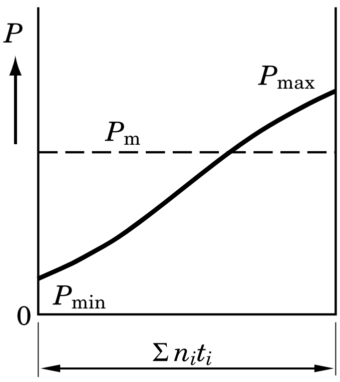
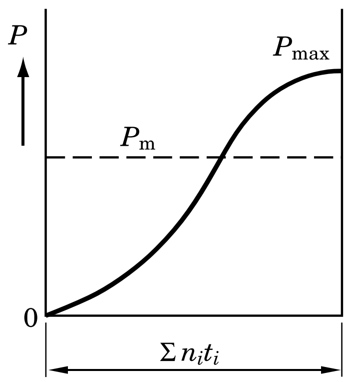
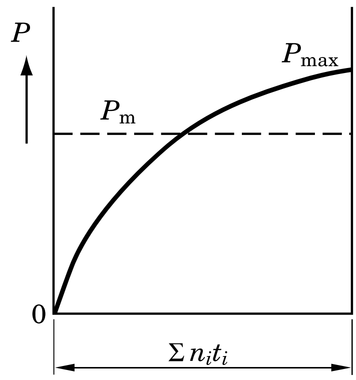
$$P_m = \sqrt[p]{\frac{P_1{}^p n_1 t_1 + P_2{}^p n_2 t_2 + \cdots\cdots + P_n{}^p n_n t_n}{n_1 t_1 + n_2 t_2 + \cdots\cdots\cdots + n_n t_n}} \quad \cdots\cdots (5\text{-}35)$$

Symbols for Graphs (1) to (4)

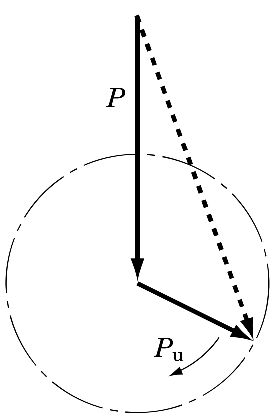
P_m	: mean dynamic equivalent load	N
P_1	: dynamic equivalent load applied for t_1 hours at rotational speed n_1	N
P_2	: dynamic equivalent load applied for t_2 hours at rotational speed n_2	N
⋮	⋮	⋮
P_n	: dynamic equivalent load applied for t_n hours at rotational speed n_n	N
P_{min}	: minimum dynamic equivalent load	N
P_{max}	: maximum dynamic equivalent load	N
$\Sigma n_i t_i$	: total rotation in (t_1 to t_i) hours	
p	: for ball bearings, $p = 3$ for roller bearings, $p = 10/3$	

[Reference] Mean rotational speed n_m can be calculated using the following equation :

$$n_m = \frac{n_1 t_1 + n_2 t_2 + \cdots\cdots\cdots + n_n t_n}{t_1 + t_2 + \cdots\cdots\cdots + t_n}$$

(2) Stageless fluctuation	(3) Fluctuation forming sine curve	(4) Fluctuation forming sine curve (upper half of sine curve)
		
$P_m = \frac{P_{min} + 2\,P_{max}}{3}$ .... (5-36)	$P_m = 0.68\,P_{max}$ .... (5-37)	$P_m = 0.75\,P_{max}$ .... (5-38)

(5) Stationary load and rotating load acting simultaneously



$$P_m = f_m\,(P + P_u) \quad \text{........ (5-39)}$$

where :

P_m : mean dynamic equivalent load N
f_m : coefficient (refer. Fig. 5-12)
P : stationary load N
P_u : rotating load N

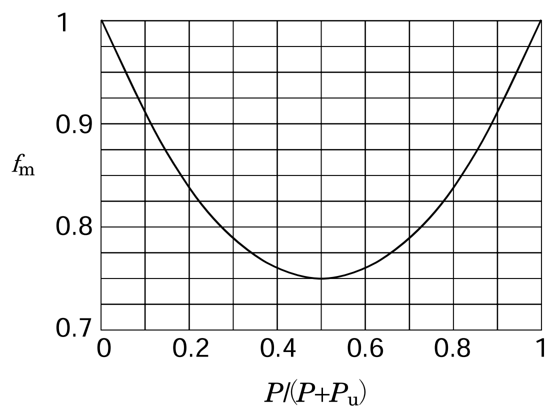


Fig. 5-12 Coefficient f_m

5-5 Basic static load rating and static equivalent load

5-5-1 Basic static load rating

Excessive static load or impact load even at very low rotation causes partial permanent deformation of the rolling element and raceway contacting surfaces. This permanent deformation increases with the load; if it exceeds a certain limit, smooth rotation will be hindered.

The basic static load rating is the static load which responds to the calculated contact stress shown below, at the contact center between the raceway and rolling elements which receive the maximum load.

- Self-aligning ball bearings ··· 4 600 MPa
- Other ball bearings ··············· 4 200 MPa
- Roller bearings ······················ 4 000 MPa

The total extent of contact stress-caused permanent deformation on surfaces of rolling elements and raceway will be approximately 0.000 1 times greater than the rolling element diameter.

The basic static load rating for radial bearings is specified as the basic static radial load rating, and for thrust bearings, as the basic static axial load rating. These load ratings are listed in the bearing specification table, using C_{0r} and C_{0a} respectively.

These values are prescribed by ISO 78/1987 and are subject to change by conformance to the latest ISO standards.

5-5-2 Static equivalent load

The static equivalent load is a theoretical load calculated such that, during rotation at very low speed or when bearings are stationary, the same contact stress as that imposed under actual loading condition is generated at the contact center between raceway and rolling element to which the maximum load is applied.

For radial bearings, radial load passing through the bearing center is used for the calculation; for thrust bearings, axial load in a direction along the bearing axis is used.

The static equivalent load can be calculated using the following equations.

[Radial bearings]
···The greater value obtained by the following two equations is used.

$$P_{0r} = X_0 F_r + Y_0 F_a \quad \cdots\cdots (5\text{-}40)$$

$$P_{0r} = F_r \quad \cdots\cdots (5\text{-}41)$$

[Thrust bearings]

($\alpha \neq 90°$)

$$P_{0a} = X_0 F_r + F_a \quad \cdots\cdots (5\text{-}42)$$

[When $F_a < X_0 F_r$, the solution becomes less accurate.]

($\alpha = 90°$)

$$P_{0a} = F_a \quad \cdots\cdots (5\text{-}43)$$

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
- X_0 : static radial load factor
- Y_0 : static axial load factor
  (values of X_0 and Y_0 are listed in the bearing specification table.)

5-5-3 Safety coefficient

The allowable static equivalent load for a bearing is determined by the basic static load rating of the bearing; however, bearing service life, which is affected by permanent deformation, differs in accordance with the performance required of the bearing and operating conditions.

Therefore, a safety coefficient is designated, based on empirical data, so as to ensure safety in relation to basic static load rating.

$$f_s = \frac{C_0}{P_0} \quad \cdots\cdots (5\text{-}44)$$

where :

f_s : safety coefficient (ref. Table 5-10)
C_0 : basic static load rating N
P_0 : static equivalent load N

Table 5-10 Values of safety coefficient f_s

Operating condition		f_s (min.) Ball bearing	f_s (min.) Roller bearing
With bearing rotation	When high accuracy is required	2	3
	Normal operation	1	1.5
	When impact load is applied	1.5	3
Without bearing rotation (occasional oscillation)	Normal operation	0.5	1
	When impact load or uneven distribution load is applied	1	2

[Remark] For spherical thrust roller bearings, $f_s \geqq 4$.

5-6 Allowable axial load for cylindrical roller bearings

Bearings whose inner and outer rings comprise either a rib or loose rib can accommodate a certain magnitude of axial load, as well as radial load. In such cases, axial load capacity is controlled by the condition of rollers, load capacity of rib or loose rib, lubrication, rotational speed etc.

For certain special uses, a design is available to accommodate very heavy axial loads. In general, axial loads allowable for cylindrical roller bearings can be calculated using the following equation, which are based on empirical data.

$$F_{ap} = 9.8\, f_a \cdot f_b \cdot f_p \cdot d_m^{\,2} \quad \cdots\cdots (5\text{-}45)$$

where :

F_{ap} : maximum allowable axial load N

f_a : coefficient determined from loading condition (Table 5-11)

f_b : coefficient determined from bearing diameter series (Table 5-12)

f_p : coefficient for rib surface pressure (Fig. 5-13)

d_m : mean value of bore diameter d and outside diameter D mm

$$\left(\frac{d+D}{2}\right)$$

Table 5-11 Values of coefficient determined from loading condition f_a

Loading condition	f_a
Continuous loading	1
Intermittent loading	2
Instantaneous loading	3

Table 5-12 Values of coefficient determined from bearing diameter series f_b

Diameter series	f_b
9	0.6
0	0.7
2	0.8
3	1.0
4	1.2

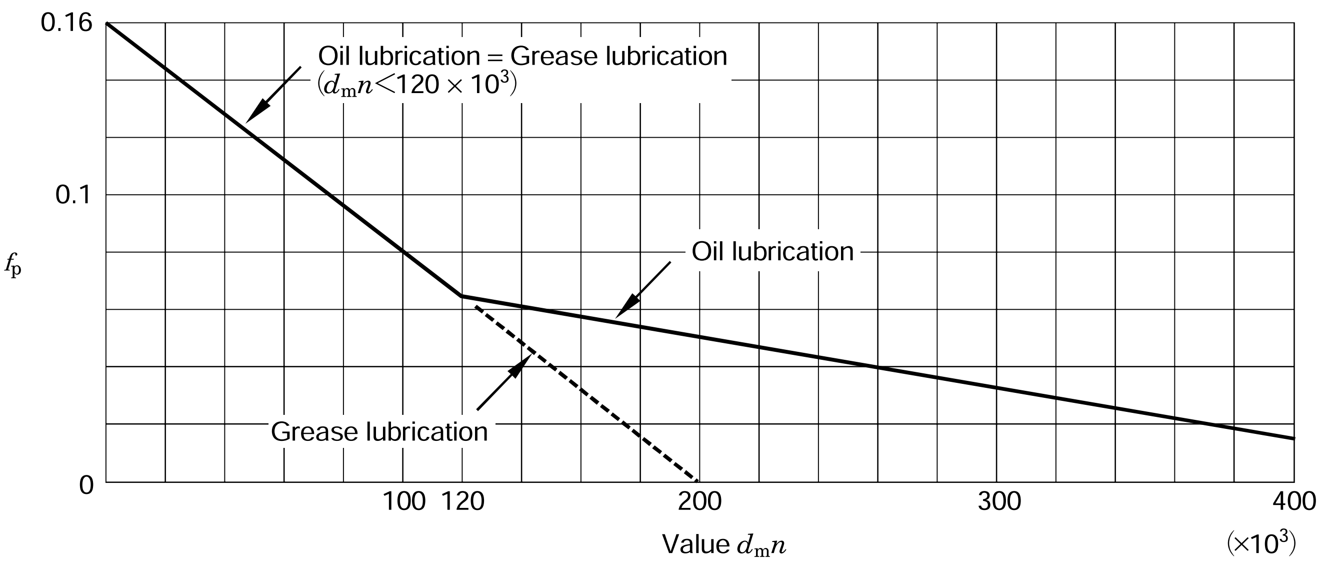


Fig. 5-13 Relationship between coefficient for rib surface pressure f_p and value $d_m n$ (n : rotational speed, $\min^{-1}$)

5-7 Applied calculation examples

[Example 1] Bearing service life (time) with 90 % reliability

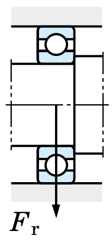
(Conditions)

Deep groove ball bearing : 6308

Radial load $F_r = 3\ 500$ N

Axial load not applied ($F_a = 0$)

Rotational speed $n = 800$ min^{-1}



① Basic dynamic load rating (C_r) is obtained from the bearing specification table.

$C_r = 50.9$ kN

② Dynamic equivalent radial load (P_r) is calculated using equation (5-32).

$P_r = F_r = 3\ 500$ N

③ Bearing sevice life (L_{10h}) is calculated using equation (5-2).

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

$$= \frac{10^6}{60 \times 800} \times \left(\frac{50.9 \times 10^3}{3\ 500}\right)^3 \fallingdotseq \underline{64\ 100\ \text{h}}$$

[Example 2] Bearing service life (time) with 96 % reliability

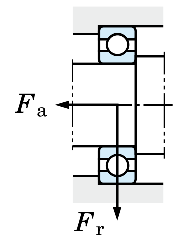
(Conditions)

Deep groove ball bearing : 6308

Radial load $F_r = 3\ 500$ N

Axial load $F_a = 1\ 000$ N

Rotational speed $n = 800$ min^{-1}



① From the bearing specification table ;

- Basic load rating (C_r , C_{0r}) f_0 factor is obtained.

$C_r = 50.9$ kN

$C_{0r} = 24.0$ kN

$f_0 = 13.2$

- Values X and Y are obtained by comparing value e, calculated from value $f_0 F_a / C_{0r}$ via proportional interpolation, with value $f_0 F_a / F_r$.

$$\frac{f_0 F_a}{C_{0r}} = \frac{13.2 \times 1\ 000}{24.0 \times 10^3} = 0.550$$

$$e = 0.22 + (0.26 - 0.22) \times \frac{(0.550 - 0.345)}{(0.689 - 0.345)}$$

$$= 0.24$$

$$\frac{F_a}{F_r} = \frac{1\ 000}{3\ 500} = 0.29 > e$$

The result is,

$$X = 0.56$$

$$Y = 1.99 - (1.99 - 1.71) \times \frac{(0.550 - 0.345)}{(0.689 - 0.345)}$$

$$= 1.82$$

② Dynamic equivalent load (P_r) is obtained using equation (5-32).

$$P_r = XF_r + YF_a$$

$$= (0.56 \times 3\ 500) + (1.82 \times 1\ 000) = 3\ 780\ \text{N}$$

③ Service life with 90 % reliability (L_{10h}) is obtained using equation (5-2).

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

$$= \frac{10^6}{60 \times 800} \times \left(\frac{50.9 \times 10^3}{3\ 780}\right)^3 \fallingdotseq \underline{50\ 900\ \text{h}}$$

[Example 3] Calculation of the a_{ISO} factor with the conditions in Example 2

(Conditions)
Oil lubrication
(Oil that has been filtered by a fine filter)
Operating temperature 70 °C
96 % reliability

④ Lubricating oil selection

From the bearing specification table, the pitch diameter $D_{\mathrm{pw}} = (40 + 90)/2 = 65$ is obtained.
$d_{\mathrm{mn}} = 65 \times 800 = 52\,000$. Therefore, select VG 68 from Table 12-8, p. A 129.

⑤ Calculating the a_{ISO} factor

The operating temperature is 70 °C, so according to Fig. 12-3, p. A 129, the viscosity when operating is $\nu = 20\ \mathrm{mm^2/s}$
According to Fig. A, $\nu_1 = 21.7\ \mathrm{mm^2/s}$

$$\kappa = \nu/\nu_1 = 20/21.7 = 0.92$$

The oil has been filtered by a fine filter, so Table 5-4 shows e_c is 0.5 to 0.6.
To stringently estimate the value, $e_c = 0.5$.

$$\frac{e_c \cdot C_u}{P} = \frac{0.5 \times 1\,850}{3\,780} = 0.24$$

Therefore, according to Fig. B

$$a_{\mathrm{ISO}} = 7.7$$

⑥ Service life with 96 % reliability (L_{nm}) is obtained using equation (5-8).

According to Table 5-3, $a_1 = 0.55$.

$$L_{4m} = a_1 a_{\mathrm{ISO}} L_{10} = 0.55 \times 7.7 \times 50\,900 \fallingdotseq 216\,000\ \mathrm{h}$$

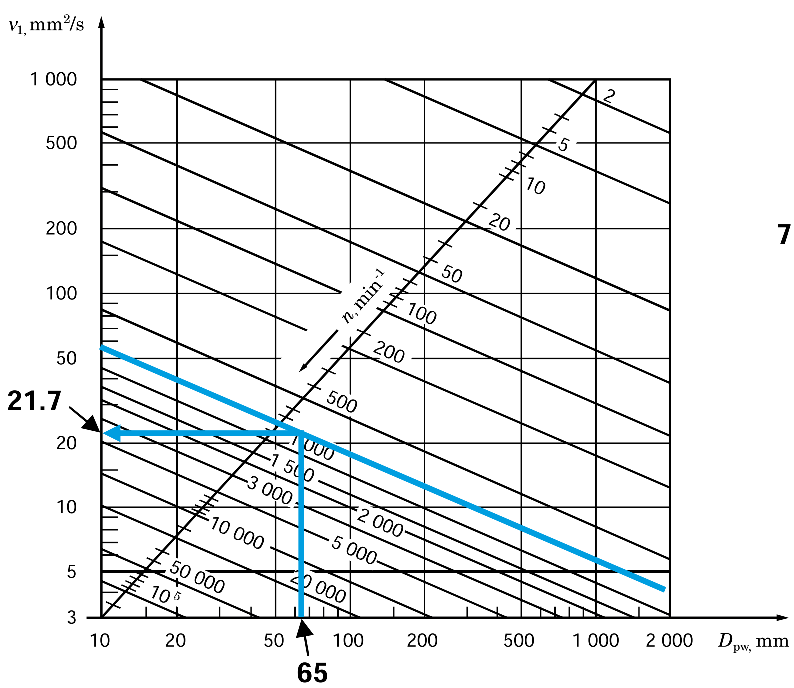


Fig. A

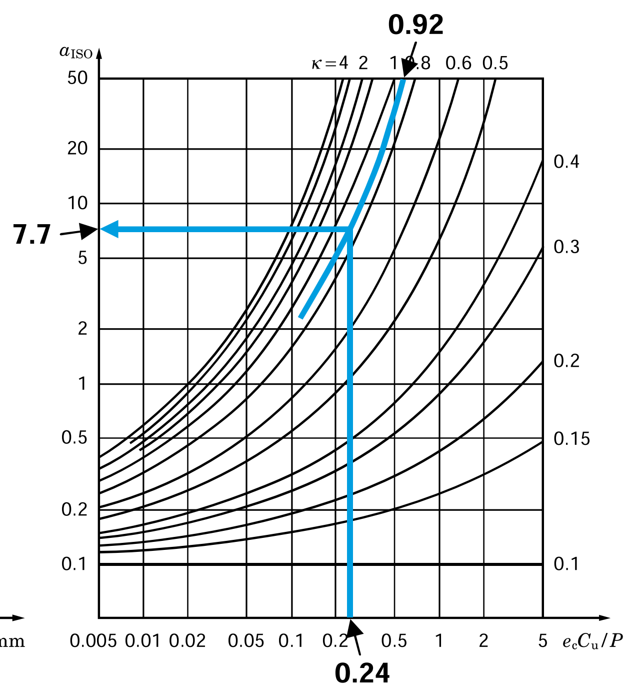
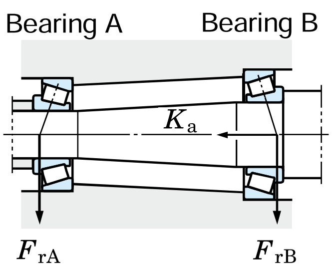


Fig. B

The a_{ISO} factor can also be calculated on our website.

[Example 4] Bearing service life (total revolution)

(Conditions)
Tapered roller bearing
Bearing A : 30207 JR
Bearing B : 30209 JR
Radial load F_{rA} = 5 200 N
F_{rB} = 6 800 N
Axial load K_a = 1 600 N



① From the bearing specification table, the following specifications are obtained.

	Basic dynamic load rating (C_r)	e	$X^{1)}$	$Y^{1)}$
Bearing A	68.8 kN	0.37	0.4	1.60
Bearing B	83.9 kN	0.40	0.4	1.48

[Note] 1) Those values are used, where $F_a / F_r > e$.
Where $F_a / F_r \leqq e$, $X = 1$, $Y = 0$.

② Axial load applied to shafts must be calculated, considering the fact that component force in the axial direction is generated when radial load is applied to tapered roller bearings. (ref. equation 5-33, Table 5-9)

$$\frac{F_{rA}}{2\,Y_A} + K_a = \frac{5\,200}{2 \times 1.60} + 1\,600 = 3\,225 \text{ N}$$

$$\frac{F_{rB}}{2\,Y_B} = \frac{6\,800}{2 \times 1.48} = 2\,297 \text{ N}$$

Consequently, axial load $\frac{F_{rA}}{2\,Y_A} + K_a$ is applied to bearing B.

③ Dynamic equivalent load (P_r) is obtained from Table 5-9.

$$P_{rA} = F_{rA} = 5\,200 \text{ N}$$

$$P_{rB} = XF_{rB} + Y_B \left(\frac{F_{rA}}{2\,Y_A} + K_a\right)$$

$$= 0.4 \times 6\,800 + 1.48 \times 3\,225 = 7\,493 \text{N}$$

④ Each bearing service life (L_{10}) is calculated using equation (5-1).

$$L_{10A} = \left(\frac{C_{rA}}{P_{rA}}\right)^{10/3} = \left(\frac{68.8 \times 10^3}{5\,200}\right)^{10/3}$$

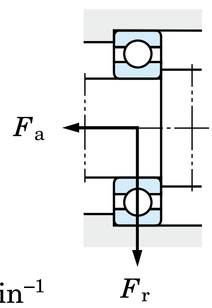
$$\fallingdotseq \underline{5\,480 \times 10^6 \text{ revolutions}}$$

$$L_{10B} = \left(\frac{C_{rB}}{P_{rB}}\right)^{10/3} = \left(\frac{83.9 \times 10^3}{7\,493}\right)^{10/3}$$

$$\fallingdotseq \underline{3\,140 \times 10^6 \text{ revolutions}}$$

[Example 5] Bearing size selection

(Conditions)
Deep groove ball bearing : 62 series
Required service life : more than 10 000 h
Radial load F_r = 2 000 N
Axial load F_a = 300 N
Rotational speed n = 1 600 min^{-1}



① The dynamic equivalent load (P_r) is hypothetically calculated.
The resultant value, F_a / F_r = 300/2 000 = 0.15, is smaller than any other values of e in the bearing specification table.
Hence, JTEKT can consider that $P_r = F_r$ = 2 000 N.

② The required basic dynamic load rating (C_r) is calculated according to equation (5-4).

$$C_r = P_r \left(L_{10h} \times \frac{60n}{10^6}\right)^{1/p}$$

$$= 2\,000 \times \left(10\,000 \times \frac{60 \times 1\,600}{10^6}\right)^{1/3}$$

$$= 19\,730 \text{ N}$$

③ Among those covered by the bearing specification table, the bearing of the 62 series with C_r exceeding 19 730 N is <u>6205 R</u>, with bore diameter for 25 mm.

④ The dynamic equivalent load obtained at step ① is confirmed by obtaining value e for 6205 R.
Where C_{0r} of 6205 R is 9.3 kN, and f_0 is 12.8

$$f_0 F_a / C_{0r} = 12.8 \times 300/9\,300 = 0.413$$

Then, value e can be calculated using proportional interpolation.

$$e = 0.22 + (0.26 - 0.22) \times \frac{(0.413 - 0.345)}{(0.689 - 0.345)}$$

$$= 0.23$$

As a result, it can be confirmed that

$$F_a / F_r = 0.15 < e.$$

Hence, $P_r = F_r$.

[Example 6] Bearing size selection

(Conditions)

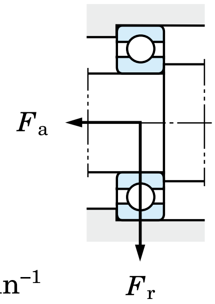
Deep groove ball bearing : 63 series

Required service life : more than 15 000 h

Radial load $F_r = 4\ 000$ N

Axial load $F_a = 2\ 400$ N

Rotational speed $n = 1\ 000\ \text{min}^{-1}$



① The hypothetic dynamic equivalent load (P_r) is calculated :

Since $F_a / F_r = 2\ 400/4\ 000 = 0.6$ is much larger than the value e specified in the bearing specification table, it suggests that the axial load affects the dynamic equivalent load.

Hence, assuming that $X = 0.56$, $Y = 1.6$ (approximate mean value of Y), using equation (5-32),

$$P_r = XF_r + XF_a = 0.56 \times 4\ 000 + 1.6 \times 2\ 400 = 6\ 080\ \text{N}$$

② Using equation (5-4), the required basic dynamic load rating (C_r) is :

$$C_r = P_r \left(L_{10h} \times \frac{60n}{10^6} \right)^{1/p}$$
$$= 6\ 080 \times \left(15\ 000 \times \frac{60 \times 1\ 000}{10^6} \right)^{1/3}$$
$$= 58\ 700\ \text{N}$$

③ From the bearing specification table, a 6309 with a bore diameter of 45 mm is selected as a 63 series bearing with C_r exceeding 58 700 N.

④ The dynamic equivalent load and basic rating life are confirmed, by calculating the value e for a 6309.

Values obtained using the proportional interpolation are :

where $f_0 F_a / C_{0r} = 13.3 \times 2\ 400/29\ 500 = 1.082$

$e = 0.283$, $Y = 1.54$.

Thus, $F_a / F_r = 0.6 > e$.

Using the resultant values, the dynamic equivalent load and basic rating life can be calculated as follows :

$$P_r = XF_r + YF_a$$
$$= 0.56 \times 4\ 000 + 1.54 \times 2\ 400 = 5\ 940\text{N}$$

$$L_{10h} = \frac{10^6}{60n} \left(\frac{C_r}{P_r} \right)^p$$
$$= \frac{10^6}{60 \times 1\ 000} \times \left(\frac{61.1 \times 10^3}{5\ 940} \right)^3 \fallingdotseq \underline{18\ 100\ \text{h}}$$

⑤ The basic rating life of the 6308, using the same steps, is :

$L_{10h} \fallingdotseq 11\ 500$ h, which does not satisfy the service life requirement.

[Example 7] Calculation of allowable axial load for cylindrical roller bearings

(Conditions)

Single-row cylindrical roller bearing : NUP 310

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

Oil lubrication

Axial load is intermittently applied.

① Using the bearing specification table, the value d_m for the NUP 310 can be calculated as follows :

$$d_m = \frac{d + D}{2} = \frac{50 + 110}{2} = 80\ \text{mm}$$

② Each coefficient used in equation (5-45).

From values listed in Table 5-11, coefficient f_a related to intermittent load is : $f_a = 2$

From values listed in Table 5-12, coefficient f_b related to diameter series 3 is : $f_b = 1.0$

According to Fig. 5-13, coefficient f_p for allowable rib surface pressure, related to

$d_m n = 80 \times 1\ 500 = 12 \times 10^4$, is : $f_p = 0.062$

③ Using equation (5-45), the allowable axial load F_{ap} is :

$$F_{ap} = 9.8\ f_a \cdot f_b \cdot f_p \cdot d_m{}^2$$
$$= 9.8 \times 2 \times 1.0 \times 0.062 \times 80^2$$
$$\fallingdotseq \underline{7\ 780\ \text{N}}$$

[Example 8] Calculation of service life of spur gear shaft bearings

(Conditions)
Tapered roller bearing
Bearing A : 32309 JR
Bearing B : 32310 JR
Gear type : spur gear (normally machined)
Gear pressure angle $\alpha_1 = \alpha_2 = 20°$
Gear pitch circle diameter $D_{p1} = 360$ mm
$D_{p2} = 180$ mm
Transmission power $W = 150$ kW
Rotational speed $n = 1\,000$ min^{-1}

Operating condition: accompanied by impact
Installation locations
$a_1 = 95$ mm, $a_2 = 265$ mm,
$b_1 = 245$ mm, $b_2 = 115$ mm,
$c = 360$ mm

① Using equations (5-14) and (5-15), theoretical loads applied to gears (tangential load, K_t; radial load, K_r) are calculated.

[Gear 1]

$$K_{t1} = \frac{19.1 \times 10^6 W}{D_p n} = \frac{19.1 \times 10^6 \times 150}{360 \times 1\,000}$$
$$= 7\,958 \text{ N}$$
$$K_{r1} = K_{t1} \tan \alpha_1 = 2\,896 \text{ N}$$

[Gear 2]

$$K_{t2} = \frac{19.1 \times 10^6 \times 150}{180 \times 1\,000} = 15\,917 \text{ N}$$
$$K_{r2} = K_{t2} \tan \alpha_2 = 5\,793 \text{ N}$$

② The radial load applied to the bearing is calculated, where the load coefficient is determined as $f_w = 1.5$ from Table 5-6, and the gear coefficient as $f_g = 1.2$ from Table 5-8.

[Bearing A]

- Load consisting of K_{t1} and K_{t2} is :

$$K_{tA} = f_w f_g \left(\frac{a_2}{c} K_{t1} + \frac{b_2}{c} K_{t2}\right)$$
$$= 1.5 \times 1.2 \times \left(\frac{265}{360} \times 7\,958 + \frac{115}{360} \times 15\,917\right) = 19\,697 \text{ N}$$

- Load consisting of K_{r1} and K_{r2} is :

$$K_{rA} = f_w f_g \left(\frac{a_2}{c} K_{r1} - \frac{b_2}{c} K_{r2}\right)$$
$$= 1.5 \times 1.2 \times \left(\frac{265}{360} \times 2\,896 - \frac{115}{360} \times 5\,793\right) = 506 \text{ N}$$

- Combining the loads of K_{tA} and K_{rA}, the radial load (F_{rA}) applied to bearing A can be calculated as follows :

$$F_{rA} = \sqrt{K_{tA}^2 + K_{rA}^2}$$
$$= \sqrt{19\,697^2 + 506^2} = 19\,703 \text{ N}$$

[Bearing B]

- Load consisting of K_{t1} and K_{t2} is :

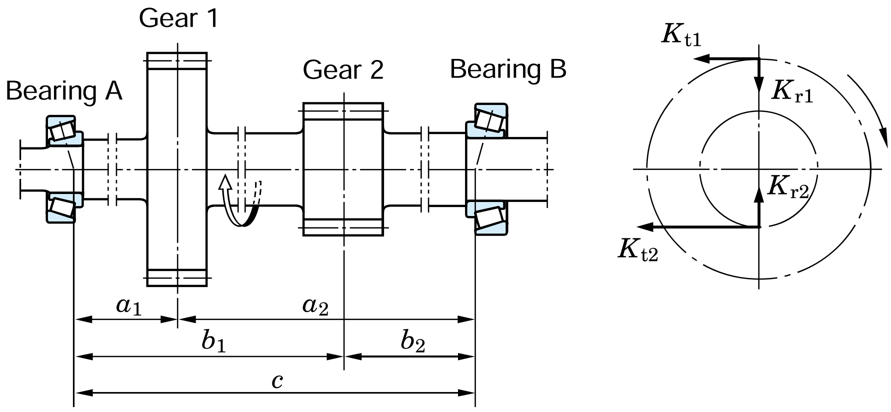
$$K_{tB} = f_w f_g \left(\frac{a_1}{c} K_{t1} + \frac{b_1}{c} K_{t2}\right)$$
$$= 1.5 \times 1.2 \times \left(\frac{95}{360} \times 7\,958 + \frac{245}{360} \times 15\,917\right) = 23\,278 \text{ N}$$

- Load consisting of K_{r1} and K_{r2} is :

$$K_{rB} = f_w f_g \left(\frac{a_1}{c} K_{r1} - \frac{b_1}{c} K_{r2}\right)$$
$$= 1.5 \times 1.2 \times \left(\frac{95}{360} \times 2\,896 - \frac{245}{360} \times 5\,793\right) = -5\,721 \text{ N}$$

- The radial load (F_{rB}) applied to bearing B can be calculated using the same steps as with bearing A.

$$F_{rB} = \sqrt{K_{tB}^2 + K_{rB}^2}$$
$$= \sqrt{23\,278^2 + (-5\,721)^2} = 23\,971 \text{ N}$$



③ The following specifications can be obtained from the bearing specification table.

	Basic dynamic load rating (C_r)	e	X 1)	Y 1)
Bearing A	183 kN	0.35	0.4	1.74
Bearing B	221 kN			

[Note] 1) Those values are used, where $F_a / F_r > e$.
Where $F_a / F_r \leqq e$, $X = 1$, $Y = 0$.

④ When an axial load is not applied externally, if the radial load is applied to the tapered roller bearing, an axial component force is generated.

Considering this fact, the axial load applied from the shaft and peripheral parts is to be calculated :
(Equation 5-33, Table 5-9)

$$\frac{F_{rB}}{2\,Y_B} = \frac{23\,971}{2 \times 1.74} > \frac{F_{rA}}{2\,Y_A} = \frac{19\,703}{2 \times 1.74}$$

According to the result, it is clear that the axial component force ($F_{rB}/2Y_B$) applied to bearing B is also applied to bearing A as an axial load applied from the shaft and peripheral parts.

⑤ Using the values listed in Table 5-9, the dynamic equivalent load is calculated, where $K_a = 0$:

$$P_{rA} = XF_{rA} + Y_A \frac{F_{rB}}{2\,Y_B}$$
$$= 0.4 \times 19\,703 \times 1.74 \times \frac{23\,971}{2 \times 1.74}$$
$$= 19\,867\ \mathrm{N}$$
$$P_{rB} = F_{rB} = 23\,971\ \mathrm{N}$$

⑥ Using equation (5-2), the basic rating life of each bearing is calculated :

[Bearing A]

$$L_{10hA} = \frac{10^6}{60n}\left(\frac{C_{rA}}{P_A}\right)^p$$
$$= \frac{10^6}{60 \times 1\,000} \times \left(\frac{183 \times 10^3}{19\,867}\right)^{10/3}$$
$$\fallingdotseq \underline{27\,300\ \mathrm{h}}$$

[Bearing B]

$$L_{10hB} = \frac{10^6}{60n}\left(\frac{C_{rB}}{P_B}\right)^p$$
$$= \frac{10^6}{60 \times 1\,000} \times \left(\frac{221 \times 10^3}{23\,971}\right)^{10/3}$$
$$\fallingdotseq \underline{27\,400\ \mathrm{h}}$$

Reference

Using equation (5-11), the system service life (L_{10hS}) using a pair of bearings is :

$$L_{10hS} = \frac{1}{\left(\frac{1}{L_{10hA}{}^e} + \frac{1}{L_{10hB}{}^e}\right)^{1/e}}$$
$$= \frac{1}{\left(\frac{1}{27\,300^{9/8}} + \frac{1}{27\,400^{9/8}}\right)^{8/9}}$$
$$\fallingdotseq \underline{14\,800\ \mathrm{h}}$$

6. Boundary dimensions and bearing numbers

6-1 Boundary dimensions

Bearing boundary dimensions are dimensions required for bearing installation with shaft or housing, and as described in Fig. 6-1, include the bore diameter, outside diameter, width, height, and chamfer dimension.

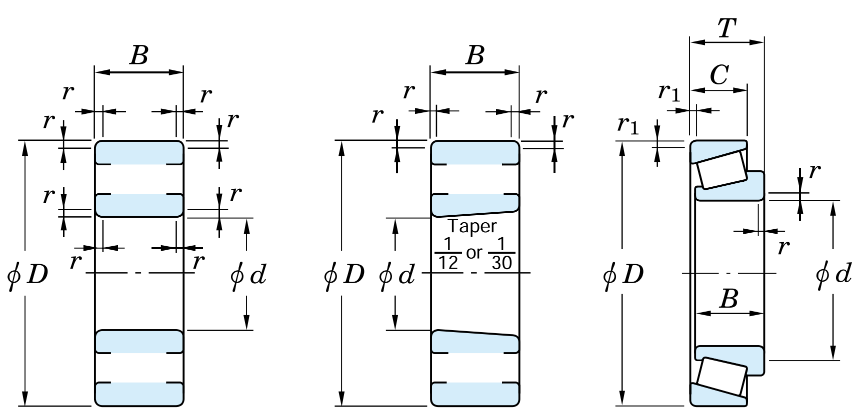
These dimensions are standardized by the International Organization for Standardization (ISO 15). JIS B 1512 "rolling bearing boundary dimensions" is based on ISO.

These boundary dimensions are provided, classified into radial bearings (tapered roller bearings are provided in other tables) and thrust bearings.

Boundary dimensions of each bearing are listed in Appendixes at the back of this catalog. In these boundary dimension tables, the outside diameter, width, height, and chamfer dimensions related to bearing bore diameter numbers and bore diameters are listed in diameter series and dimension series.

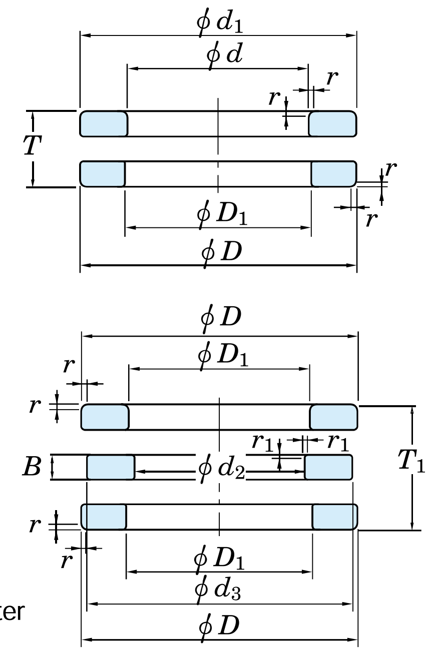
Reference

1) Diameter series is a series of nominal bearing outside diameters provided for respective ranges of bearing bore diameter; and, a dimension series includes width and height as well as diameters.
2) Tapered roller bearing boundary dimensions listed in the Appendixes are adapted to conventional dimension series (widths and diameters). Tapered roller bearing boundary dimensions provided in JIS B 1512-2000 are new dimension series based on ISO 355 (ref. descriptions before the bearing specification table); for reference, the bearing specification table covers numeric codes used in these dimension series.



(1) Radial bearing
(tapered roller bearings not included)

(2) Tapered roller bearing



(3) Thrust bearing
(single/double direction)

Radial bearing
(tapered roller bearings not included)
d : nominal bore diameter
D : nominal outside diameter
B : nominal assembled bearing width
r : inner/outer ring chamfer dimension[1)]

Tapered roller bearing
d : nominal bore diameter
D : nominal outside diameter
T : nominal assembled bearing width
B : nominal inner ring width
C : nominal outer ring width
r : inner ring chamfer dimension[1)]
r_1 : outer ring chamfer dimension[1)]

Thrust bearing
d : shaft race nominal bore diameter
d_1 : shaft race nominal outside diameter[2)]
d_2 : central race nominal bore diameter
d_3 : central race nominal outside diameter[2)]
D : housing race nominal outside diameter
D_1 : housing race nominal bore diameter[1)]
T : single direction nominal bearing height
T_1 : double direction nominal bearing height
B : central race nominal height
r : shaft/housing race chamfer dimension[1)]
r_1 : central race chamfer dimension[1)]

[Notes]
1) The bearing specification table includes the minimum value.
2) The bearing specification table includes the maximum value.

Fig. 6-1 Bearing boundary dimensions

Cross-section dimensions of radial bearings and thrust bearings expressed in dimension series can be compared using Figs. 6-2 and 6-3.

In this way, many dimension series are provided; however, not all dimensions are practically adapted.
Some of them were merely prescribed, given expected future use.

6-2 Dimensions of snap ring grooves and locating snap rings

JIS B 1509 "rolling bearing -radial bearing with locating snap ring-dimensions and tolerances" conforms to the dimensions of snap ring groove for fitting locating snap ring on the outside surface of bearing and the dimensions and tolerances of locating snap ring.

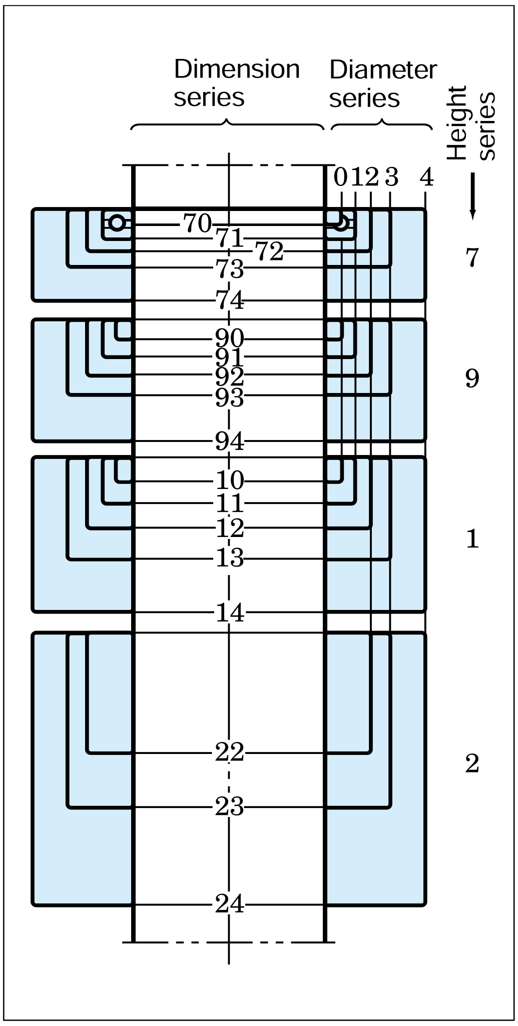


Fig. 6-3 Thrust bearing dimension series diagram (diameter series 5 omitted)

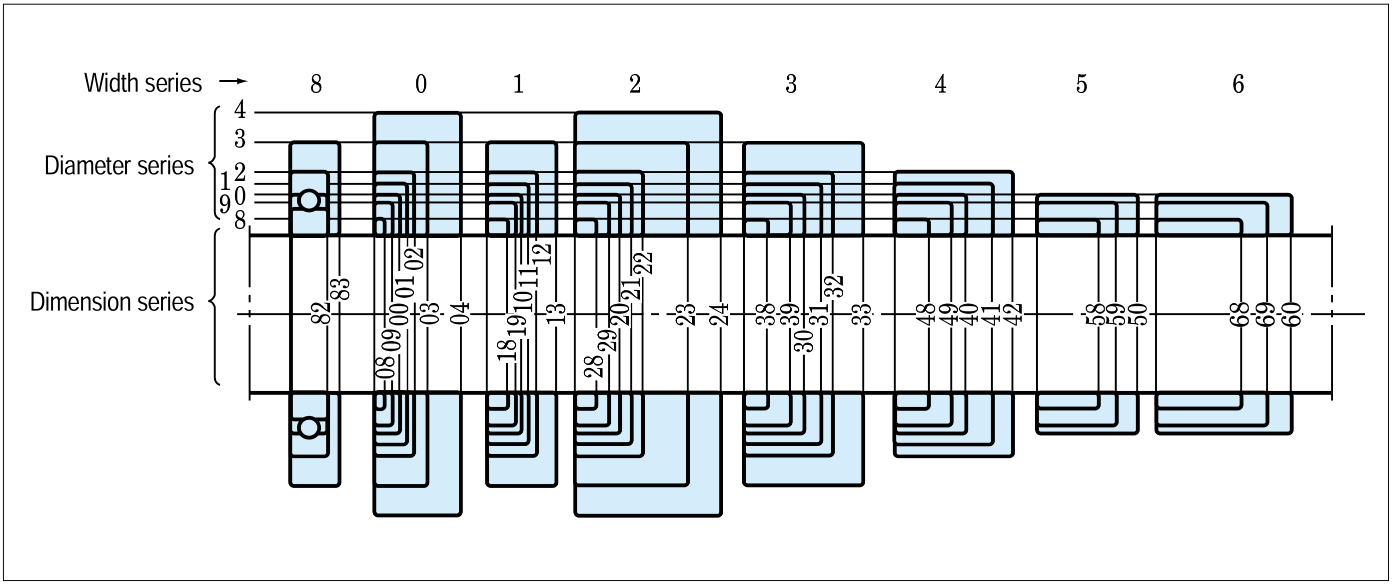


Fig. 6-2 Radial bearing dimension series diagram (diameter series 7 omitted)

6-3 Bearing number

A bearing number is composed of a basic number and a supplementary code, denoting bearing specifications including bearing type, boundary dimensions, running accuracy, and internal clearance.

Bearing numbers of standard bearings corresponding to JIS B 1512 "rolling bearing boundary dimensions" are prescribed in JIS B 1513.

As well as these bearing numbers, JTEKT uses supplementary codes other than those provided by JIS.

Among basic numbers, bearing series codes are listed in Table 6-1, and the composition of bearing numbers is described in Table 6-2, showing the order of arrangement of the parts.

[Examples of bearing numbers]

(Ex. 1)

62 03 ZZ C 2

- C 2 — Internal clearance code (clearance C2)
- ZZ — Shield code (both sides shielded)
- 03 — Bore diameter number (nominal bore diameter, 17 mm)
- 62 — Bearing series code (single-row deep groove ball bearing of dimension series 02)

(Ex. 2)

72 10 C DT P 5

- P 5 — Tolerance class code (class 5)
- DT — Matched pair or stack code (tandem arrangement)
- C — Contact angle code (nominal contact angle, 15°)
- 10 — Bore diameter number (nominal bore diameter, 50 mm)
- 72 — Bearing series code (single-row angular contact ball bearing of dimension series 02)

(Ex. 3)

NU 3 18 C 3 P 6

- P 6 — Tolerance class code (class 6)
- C 3 — Internal clearance code (clearance C3)
- 18 — Bore diameter number (nominal bore diameter, 90 mm)
- NU 3 — Bearing series code (single-row cylindrical roller bearing of dimension series 03)

(Ex. 4)

320 05 J R P 6 X

- P 6 X — Tolerance class code (class 6X)
- R — Internal design code (high load capacity)
- J — Code denoting that boundary dimensions and sub unit dimensions are based on ISO standards.
- 05 — Bore diameter number (nominal bore diameter, 25 mm)
- 320 — Bearing series code (single-row tapered roller bearing of dimension series 20)

(Ex. 5)

232/500 RZ K C 4

- C 4 — Internal clearance code (clearance C4)
- K — Bearing ring shape code (inner ring tapered bore (taper 1 : 12))
- RZ — Internal design code (with convex symmetric rollers, pressed cage)
- 500 — Bore diameter number (nominal bore diameter, 500 mm)
- 232 — Bearing series code (spherical roller bearing of dimension series 32)

(Ex. 6)

512 15

- 15 — Bore diameter number (nominal bore diameter, 75 mm)
- 512 — Bearing series code (single direction thrust ball bearing of dimension series 12)

Table 6-1 Bearing series code

Bearing type	Bearing series code	Type code	Dimension series code	
			Width series[1]	Diameter series
Single-row deep groove ball bearing	67	6	(1)	7
	68	6	(1)	8
	69	6	(1)	9
	160 [2]	6	(0)	0
	60	6	(1)	0
	62	6	(0)	2
	63	6	(0)	3
	64	6	(0)	4
Double-row deep groove ball bearing (with filling slot)	42	4	(2)	2
	43	4	(2)	3
Single-row angular contact ball bearing	79	7	(1)	9
	70	7	(1)	0
	72	7	(0)	2
	73	7	(0)	3
	74	7	(0)	4
Double-row angular contact ball bearing (with filling slot)	32	(0)	3	2
	33	(0)	3	3
Double-row angular contact ball bearing	52	5	(3)	2
	53	5	(3)	3
Self-aligning ball bearing	12	1	(0)	2
	22	2	(2)	2
	13	1	(0)	3
	23	2	(2)	3
	112 [2]	1	(0) [3]	2
	113 [2]	1	(0) [3]	3
Single-row cylindrical roller bearing	NU 10	NU [4]	1	0
	NU 2	NU [4]	(0)	2
	NU 22	NU [4]	2	2
	NU 32	NU [4]	3	2
	NU 3	NU [4]	(0)	3
	NU 23	NU [4]	2	3
	NU 4	NU [4]	(0)	4
Double-row cylindrical roller bearing	NNU 49	NNU	4	9
	NN 30	NN	3	0
Single-row needle roller bearing	NA 48	NA	4	8
	NA 49	NA	4	9
	NA 59	NA	5	9
Double-row needle roller bearing	NA 69	NA	6	9
Tapered roller bearing	329	3	2	9
	320	3	2	0
	330	3	3	0
	331	3	3	1
	302	3	0	2
	322	3	2	2
	332	3	3	2
	303	3	0	3
	313	3	1	3
	323	3	2	3
Spherical roller bearing	239	2	3	9
	230	2	3	0
	240	2	4	0
	231	2	3	1
	241	2	4	1
	222	2	2	2
	232	2	3	2
	213 [2]	2	0	3
	223	2	2	3
Single direction thrust ball bearing	511	5	1	1
	512	5	1	2
	513	5	1	3
	514	5	1	4
Single direction thrust ball bearing with spherical back face	532	5	3	2
	533	5	3	3
	534	5	3	4
Double direction thrust ball bearing	522	5	2	2
	523	5	2	3
	524	5	2	4
Double direction thrust ball bearing with spherical back faces	542	5	4	2
	543	5	4	3
	544	5	4	4
Spherical thrust roller bearing	292	2	9	2
	293	2	9	3
	294	2	9	4

[Notes]
1) Width series codes in parentheses are omitted in bearing series codes.
2) These are bearing series codes customarily used.
3) Nominal outer ring width series (inner rings only are wide).
4) Besides NU type, NJ, NUP, N, NF, and NH are provided.

Table 6-2 Bearing number configuration

	Basic number			Supplementary		
Order of arrengement	Bearing series code	Bore diameter No.	Contact angle code	Internal design code, cage guide code	Shield/seal code	Ring shape code, lubrication hole/groove code

(Codes and descriptions)

Bearing series code

68	Deep groove ball bearing
69	⋮
60	⋮
⋮	

(For standard bearing code, refer to Table 6-1)

Bore diameter No.

/0.6	0.6 mm (Bore diameter)	
1	1	
/1.5	1.5	
⋮	⋮	
9	9	
00	10	
01	12	
02	15	
03	17	
04	20	· Bore diameters (mm) of bearing in the bore diameter range 04 to 96 can be obtained by multiplying their bore diameter number by five.
/22	22	
05	25	
⋮	⋮	
96	480	
/500	500	
/2500	2500	

Contact angle code

A	(omitted)	30°	Angular contact ball bearing
AC		25°	
B		40°	
C		15°	
CA		20°	
E		35°	
B	(omitted)	Less than 17°	Tapered roller bearing
C		20°	
D		28° 30'	
DJ		28° 48' 39"	

Internal design code

R High load capacity
(Deep groove ball bearing, cylindrical roller bearing, tapered roller bearing)

G Equal stand-out is provided on both sides of the ring of angular contact ball bearing (In general, C2 clearance is used)

GST Angular contact ball bearing described above with standard internal clearance provided

J Tapered roller bearing, whose outer ring width, contact angle and outer ring small inside diameter conform to ISO standards

R	With convex asymmetric rollers and machined cage	Spherical roller bearings
RZ	With convex symmetric rollers and pressed cage	
RHA	With convex symmetric rollers and one-piece machined cage	

V Full complement type ball or roller bearing (with no cage)

Shield/seal code

one side	both sides	
Z	**ZZ**	Fixed shield
ZX	**ZZX**	Removable shield
ZU	**2ZU**	Non-contact seal
RU	**2RU**	
RS	**2RS**	Contact seal
RK	**2RK**	
U	**UU**	
RD	**2RD**	Extremely light contact seal

Ring shape code, lubrication hole/groove code

K Inner ring tapered bore provided (1 : 12)

K30 Inner ring tapered bore provided (1 : 30)

N Snap ring groove on outer ring outside surface provided

NR Snap ring groove and locating snap ring on outer ring outside surface provided

code						
Material code, special treatment code	Matched pair or stack code	Internal clearance code, preload code	Spacer code	Cage material/ shape code	Tolerance code	Grease code

(Codes and descriptions)

NY Creep prevention synthetic resin ring on outer ring outside surface provided
SG Spiral groove on inner ring bore surface provided
W Lubrication hole and lubrication groove on cylindrical roller bearing outer ring outside surface provided
W33 Lubrication hole and lubrication groove on spherical roller bearing outer ring outside surface provided

Material code, special treatment code

Code	Description	
Code not given	High carbon chrome bearing steel	
E, F, H, Y	Case carburizing steel	
ST	Stainless steel	
SH	Special heat treatment	
S0	Up to 150 °C	Dimension stabilizing treatment
S1	Up to 200 °C	Dimension stabilizing treatment
S2	Up to 250 °C	Dimension stabilizing treatment

Matched pair or stack code, cage guide code

DB	Back-to-back arrangement	Angular contact ball bearing
DF	Face-to-face arrangement	Angular contact ball bearing
DT	Tandem arrangement	Angular contact ball bearing
PA	With outer ring guide cage (Ball bearing)	
Q3	With roller guide cage (Roller bearing)	

Internal clearance code, preload code

C1	Smaller than C2	
C2	Smaller than standard clearance	Radial internal clearance for radial bearing
CN	Standard clearance	Radial internal clearance for radial bearing
C3	Greater than standard clearance	Radial internal clearance for radial bearing
C4	Greater than C3	Radial internal clearance for radial bearing
C5	Greater than C4	Radial internal clearance for radial bearing
M1 to M6	Radial internal clearance for extra-small/ miniature ball bearing	
CD2	Smaller than standard clearance	Radial internal clearance for double-row angular contact ball bearing
CDN	Standard clearance	Radial internal clearance for double-row angular contact ball bearing
CD3	Greater than standard clearance	Radial internal clearance for double-row angular contact ball bearing
CM	Radial internal clearance for electric motor bearing	Deep groove ball bearing
CT	Radial internal clearance for electric motor bearing	Cylindrical roller bearing
NA	Non-interchangeable cylindrical roller bearing radial internal clearance (C1NA to C5NA)	
S	Slight preload	Preload for angular contact ball bearing
L	Light preload	Preload for angular contact ball bearing
M	Medium preload	Preload for angular contact ball bearing
H	Heavy preload	Preload for angular contact ball bearing

Spacer code

Spacer width (mm) is affixed to the end of each code.

+	Inner and outer ring spacers provided	Deep groove ball bearing
/	Inner and outer ring spacers provided	Angular contact ball bearing
/P	Outer ring spacer provided	Angular contact ball bearing
/S	Inner ring spacer provided	Angular contact ball bearing
+DP	Inner and outer ring spacers provided	Cylindrical roller bearing, spherical roller bearing
+IDP	Inner ring spacer provided	Cylindrical roller bearing, spherical roller bearing
+ODP	Outer ring spacer provided	Cylindrical roller bearing, spherical roller bearing

Cage material/type code

/ /	Steel sheet	Pressed cage
YS	Stainless steel sheet	Pressed cage
FT	Phenol resin	
FY	High-tensile brass casting	Machined cage
FW	High-tensile brass casting (separable type)	Machined cage
MG, FG	Polyamide	(Molded cage)
FP	Carbon steel	(Pin type cage)

Tolerance code (JIS)

Omitted	Class 0
P6	Class 6
P6X	Class 6X
P5	Class 5
P4	Class 4
P2	Class 2

Grease code

A2	Alvania 2
AC	Andok C
B5	Beacon 325
SR	Multemp SRL