NTN

Ball and Roller Bearings

Cat.No.2200/E

NTN corporation

Warranty

NTN warrants, to the original purchaser only, that the delivered product which is the subject of this sale (a) will conform to drawings and specifications mutually established in writing as applicable to the contract, and (b) be free from defects in material or fabrication. The duration of this warranty is one year from date of delivery. If the buyer discovers within this period a failure of the product to conform to drawings or specifications, or a defect in material or fabrication, they must promptly notify NTN in writing. In no event shall such notification be received by NTN later than 13 months from the date of delivery. Within a reasonable time after such notification, NTN will, at its option, (a) correct any failure of the product to conform to drawings, specifications or any defect in material or workmanship, with either replacement or repair of the product, or (b) refund, in part or in whole, the purchase price. Such replacement and repair, excluding charges for labor, is at NTN's expense. All warranty service will be performed at service centers designated by NTN. These remedies are the purchaser's exclusive remedies for breach of warranty.

NTN noes not warranty (a) any product, components or parts not manufactured by NTN, (b) defects caused by failure to provide a suitable installation environment for the product, (c) damage caused by use of the product for purposes other than those for which it was designed, (d) damage caused by disasters such as fire, flood, wind, and lightning, (e) damage caused by unauthorized attachments or modification, (f) damage during shipment, or (g) any other abuse or misuse by the purchaser.

THE FOREGOING WARRANTIES ARE IN LIEU OF ALL OTHER WARRANTIES, EXPRESS OR IMPLIED, INCLUDING BUT LIMITED TO THE IMPLIED WARRANTIES OF MERCHANTABILITY AND FITNESS FOR A PARTICULAR PURPOSE.

In no case shall NTN be liable for any special, incidental, or consequential damages based upon breach of warranty, breach of contract, negligence, strict tort, or any other legal theory, and in no case shall total liability of NTN exceed the purchase price of the part upon which such liability is based. Such damages include, but are not limited to, loss of profits, loss of savings or revenue, loss of use of the product or any associated equipment, cost of capital, cost of any substitute equipment, facilities or services, downtime, the claims of third parties including customers, and injury to property. Some states do not allow limits on warranties, or on remedies for breach in certain transactions. In such states, the limits in this paragraph and in paragraph (2) shall apply to the extent allowable under case law and statutes in such states.

Any action for breach of warranty or any other legal theory must be commenced within 15 months following delivery of goods.

Unless modified in writing signed by both parties, this agreement is understood to be the complete and exclusive agreement between the parties, superceding all prior agreements, oral or written, and all other communications between the parties relating to the subject matter of this agreement. No employees of NTN or any other party is authorized to make any warranty in addition to those made in this agreement.

This agreement allocates the risks of product failure between NTN and the purchaser. This allocation is recognized by both parties and is reflected in the price of the goods. The purchaser acknowledges that is has read this agreement, understands it, and is bound by its terms.

Issue of NTN General Catalog “Bearings”

We thank you for your favorable interest in NTN products. We have revised our general catalog “Bearings” to incorporate the latest revisions of JIS (Japanese Industrial Standards) and ISO (International Organization for Standardization) according to a new editing policy. We hope the catalog will be useful to you.

The expression method of chamfer dimensions, bearing accuracy, quantity symbols, and definitions have been revised by JIS and ISO. Our catalog has been revised in conformity with the JIS and ISO revisions. Basic load ratings have been revised with improvements in our materials and manufacturing techniques and according to JIS B 1518-1989. Further, the unit system has been revised entirely according to JIS Z 8202 and ISO 1000.

NTN Needle roller bearings, bearing units, pillow block precision ball screws, linear slides, and other linear motion bearings are not listed in this catalog. They are listed in a separate catalog.

A separate catalog also has been compiled for other NTN products such as constant velocity universal joints, high pressure pipe fittings, static air bearings and slides, magnetic bearings, and other precision and industrial equipment.

Contents

1. Classification and Characteristics of Rolling Bearings

1.1 Rolling bearing construction

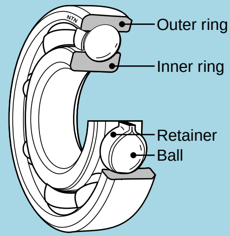
Most rolling bearings consist of rings with raceways (an inner ring and an outer ring), rolling elements (either balls or rollers) and a rolling element retainer. The retainer separates the rolling elements at regular intervals, holds them in place within the inner and outer raceways, and allows them to rotate freely. See figures 1.1-1.8.

Rolling elements come in two general shapes: ball or rollers. Rollers come in four basic styles: cylindrical, needle, tapered, and spherical.

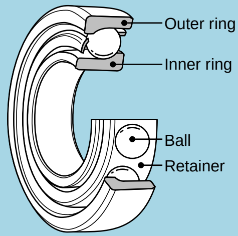
Balls geometrically contact the raceway surfaces of the inner and outer rings at “points”, while the contact surface of rollers is a “line” contact.

Theoretically, rolling bearings are so constructed as to allow the rollling elements to rotate orbitally while also rotating on their own axes at the same time.

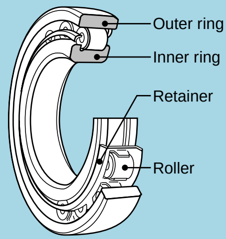
While the rolling elements and the bearing rings take any load applied to the bearings (at the contact point between the rolling elements and raceway surfaces), the retainer takes no direct load. The retainer only serves to hold the rollling elements at equal distances from each other and prevent them from falling out.



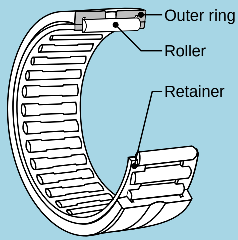
Deep groove ball bearing
Fig. 1.1



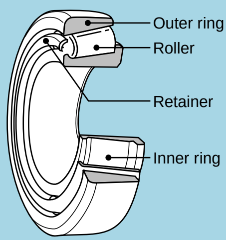
Angular contact ball bearing
Fig. 1.2



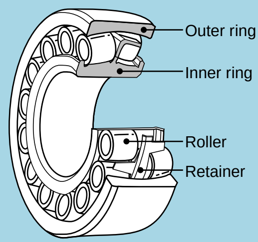
Cylindrical roller bearing
Fig. 1.3



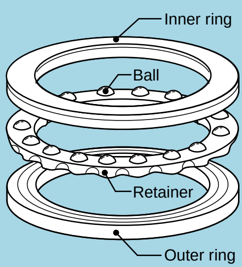
Needle roller bearing
Fig. 1.4



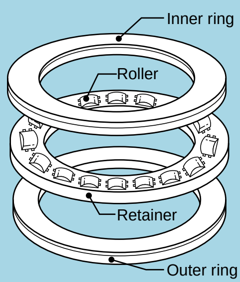
Tapered roller bearing
Fig. 1.5



Spherical roller bearing
Fig. 1.6



Thrust ball bearing
Fig. 1.7



Thrust roller bearing
Fig. 1.8

1.2 Classification of rolling bearings

Rolling element bearings fall into two main classifications: ball bearings and roller bearings. Ball bearings are classified according to their bearing ring configurations: deep groove, angular contact and thrust types. Roller bearings on the other hand are classified according to the shape of the rollers: cylindrical, needle, taper and spherical.

Rolling element bearings can be further classified according to the direction in which the load is applied; radial bearings carry radial loads and thrust bearings carry axial loads.

Other classification methods include: 1) number of rolling rows (single, multiple, or 4-row), 2) separable and non-separable, in which either the inner ring or the outer ring can be detached, 3) thrust bearings which can carry axial loads in only one direction, and double direction thrust bearings which can carry loads in both directions.

There are also bearings designed for special applications, such as: railway car journal roller bearings (RCT bearings), ball screw support bearings, turntable bearings, as well as rectilinear motion bearings (linear ball bearings, linear roller bearings and linear flat roller bearings).

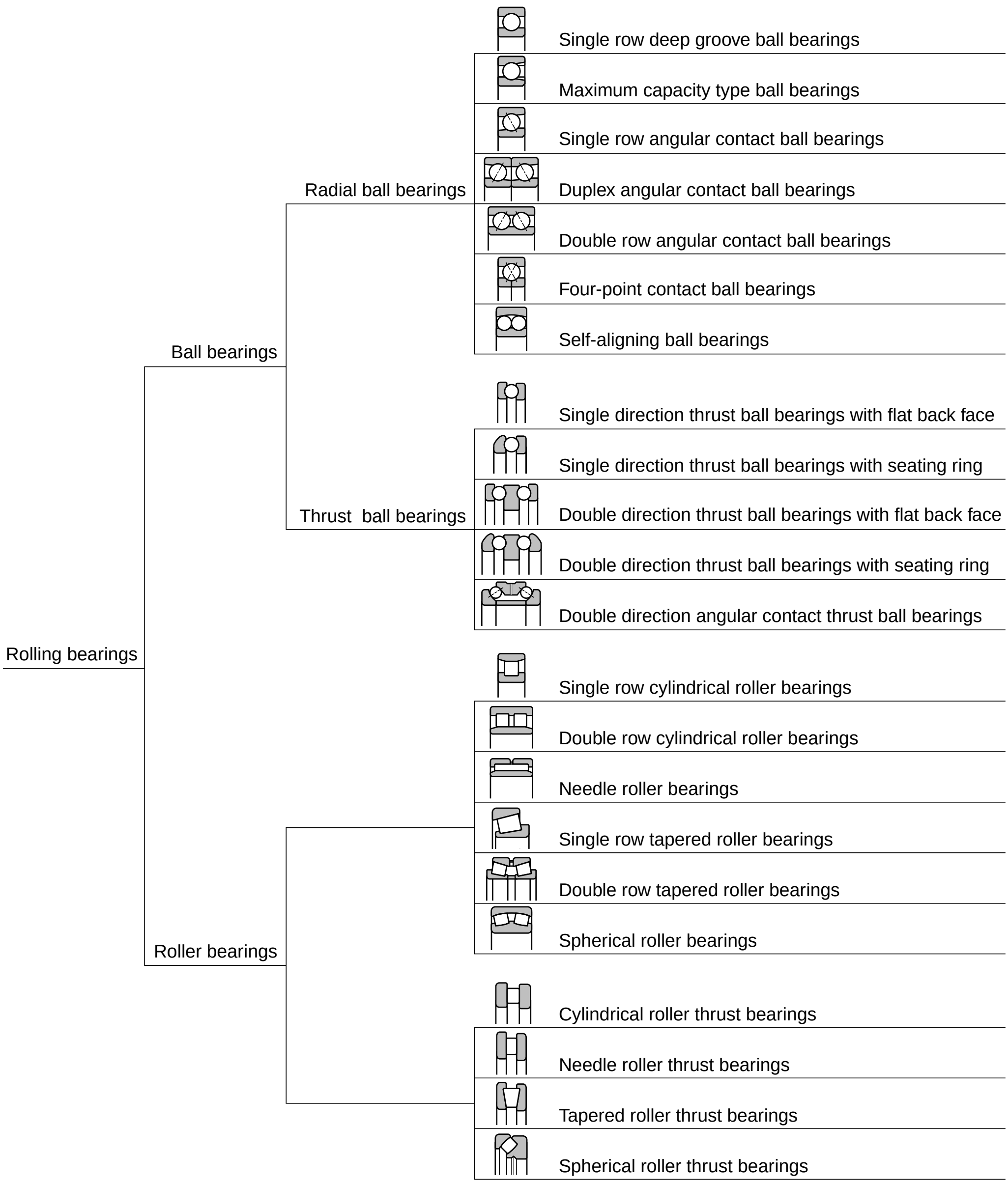


Fig. 1.9 Classification of rolling bearings

1.3 Characteristics of rolling bearings

1.3.1. Characteristics of rolling bearings

Rolling bearings come in many shapes and varieties, each with its own distinctive features.

However, when compared with sliding bearings, rolling bearings all have the followings advantages:

(1) The starting friction coefficient is lower and only a little difference between this and the dynamic friction coefficient is produced.

(2) They are internationally standardized, interchangeable and readily obtainable.

(3) Ease of lubrication and low lubricant consumption.

(4) As a general rule, one bearing can carry both radial and axial loads at the same time.

(5) May be used in either high or low temperature applications.

(6) Bearing rigidity can be improved by preloading.

Construction, classes, and special features of rolling bearings are fully described in the boundary dimensions and bearing numbering system section.

1.3.2. Ball bearings and roller bearings

Generally speaking, when comparing ball and roller bearings of the same dimensions, ball bearings exhibit a lower frictional resistance and lower face run-out in rotation than roller bearings.

This makes them more suitable for use in applications which require high speed, high precision, low torque and low vibration. Conversely, roller bearings have a larger load carrying capacity which makes them more suitable for applications requiring long life and endurance for heavy loads and shock loads.

1.3.3. Radial and thrust bearings

Almost all types of rollling bearings can carry both radial and axial loads at the same time.

Generally, bearings with a contact angle of less than 45° have a much greater radial load capacity and are classed as radial bearings; whereas bearings which have a contact angle over 45° have a greater axial load capacity and are classed as thrust bearings. There are also bearings classed as complex bearings which combine the loading characteristics of both radial and thrust bearings.

1.3.4. Standard bearings and special bearings

Bearings which are internationally standardized for shape and size are much more economical to use, as they are interchangeable and available on a worldwide basis.

However, depending on the type of machine they are to be used in, and the expected application and function, a non-standard or specially designed bearing may be best to use. Bearings that are adapted to specific applications, and “unit bearings” which are integrated (built-in) into a machine’s components, and other specially designed bearings are also available.

2. Bearing Selection

Rolling bearings come in a wide variety of types, shapes and dimensions. The most important factor to consider in bearing selection is a bearing that will enable the machine or part in which it is installed to satisfactorily perform as expected.

To facilitate the selection process and to be able to select the most suitable bearing for the job, it is necessary to analyze the prerequisites and examine them from various standpoints. While there are no hard-and-fast rules in selecting a bearing, the following list of evaluation steps is offered as a general guideline in selecting the most appropriate bearing.

(1) Thoroughly understand the type of machine the bearing is to be used in and the operating conditions under which it will function.

(2) Clearly define all demand factors.

(3) Select bearing shape.

(4) Select bearing arrangement.

(5) Select bearing dimensions.

(6) Select bearing specifications.

(7) Select mounting method, etc.

2.1 Operating conditions and environment

When selecting a bearing, having an accurate and comprehensive knowledge of which part of the machine or equipment it is to be installed in and the operating requirements and environment in which it will function, is the basis for selecting just the right bearing for the job. In the selection process, the following data is needed.

(1) The equipment's function and construction.

(2) Bearing mounting location (point).

(3) Bearing load (direction and magnitude).

(4) Bearing speed.

(5) Vibration and shock load.

(6) Bearing temperature (ambient and friction generated).

(7) Environment (corrosion, lubrication, cleanliness of the environment, etc.).

2.2 Demand factors

The required performance capacity and function demands are defined in accordance with the bearing application conditions and operating conditions. A list of general demand factors to be considered is shown in Table 2.1.

Table 2.1 Bearing Demand Factors

Demand factor	Ref. page
Dimension limitations	A-16
Durablility (life span)	A-40
Running accuracy	A-22
Allowable speed	A-77
Rigidity	A-74
Noise/vibration	—
Friction torque	A-78
Allowable misalignment for inner/outer rings	—
Requirements for mounting-dismounting	A-97
Bearing availability and economy	—

2.3 Design selection

By comparing bearing functions and performance demands with the characteristics of each bearing type, the most suitable bearing design can be selected. For easy reference, the characteristics of general bearing types are compared in Table 2.2 on page A-12.

2.4 Arrangement selection

Shaft assemblies generally require two bearings to support and locate the shaft both radially and axially relative to the stationary housing. These two bearings are called the fixed and floating bearings. The fixed bearing takes both radial and axial loads and "locates" or aligns the shaft axially in relation to the housing. Being axially "free", the floating bearing relieves stress caused by expansion and contraction of the shaft due to fluctuations in temperature, and can also allow for misalignment caused by fitting errors.

Bearings which can best support axial loads in both directions are most suitable for use as fixed bearings. In floating bearings the axial displacement can take place in the raceway (for example: cylindrical roller bearings) or along the fitting surfaces (for example: deep groove ball bearings). There is also the "cross location" arrangement in which both bearings (for example: angular contact ball bearings) act as fixing and non-fixing bearings simultaneously, each bearing guiding and supporting the shaft in one axial direction only. This arrangement is used mainly in comparatively short shaft applications.

These general bearing arrangements are shown in Table 2.3 on pages A-14 and A-15.

2.5 Dimension selection

Bearing dimension selection is generally based on the operating load and the bearing's life expectancy requirements, as well as the bearing's rated load capacity (P.A-40-A-53).

2.6 Specification determination

Specifications for rolling bearings which are designed for the widest possible use have been standardized. However, to meet the diversity of applications required, a bearing of non-standard design specifications may be selected. Items relating to bearing specification determination are given in Table 2.4.

Table 2.4 Bearing specifications

Specification item	Ref. page
Bearing tolerance (dimensional and running)	A-22
Bearing internal clearance and preload	A-64
Bearing material and heat treatment	A-92
Cage design and material	A-93

2.7 Handling methods

If bearings are to function as expected, appropriate methods of installation and handling must be selected and implemented. See Table 2.5.

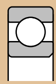
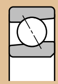
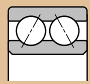
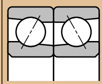
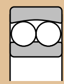
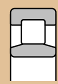
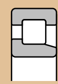
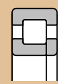
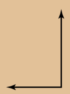
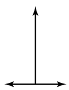
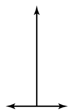
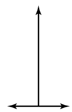
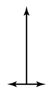
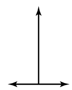
When selecting a bearing, frequently all the data required for the selection of the bearing is not necessarily clearly specified. Thus, some elements governing selection must be "factored in" on an estimated basis. Also, the order of priority and weight of each factor must be evaluated. For this reason it is essential to have ample experience as well as abundant, integrated, data base upon which the bearing selection can be based.

Over the years, NTN has gained considerable expertise in bearings selection. Please consult NTN for advice and assistance with any bearing selection problem.

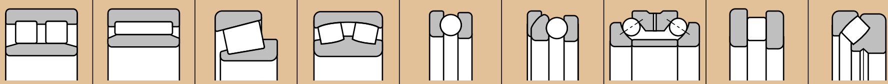
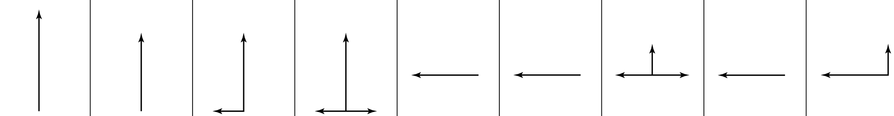
Table 2.5 Bearing handling

Treatment	Ref. page
Fitting methods	A-54
Lubrication methods and lubricants	A-79
Sealing methods and seals	A-88
Shaft and housing construction and dimensions	A-94

Table 2.2 Types and characteristics of rolling bearings

Bearing types / Characteristics	Deep groove ball bearings	Angular contact ball bearings	Double row angular contact ball bearings	Duplex angular contact ball bearings	Self-aligning ball bearings	Cylindrical roller bearings	Single-flange cylindrical roller bearings	Double-flange cylindrical roller bearings
								
Load Carrying Capacity 								
High speed[1]	☆☆☆☆	☆☆☆☆	☆☆	☆☆☆	☆☆	☆☆☆☆	☆☆☆	☆☆☆
High rotating accuracy[1]	☆☆☆	☆☆☆	☆☆	☆☆☆		☆☆☆	☆☆	☆
Low noise/vibration[1]	☆☆☆☆	☆☆☆		☆		☆	☆	☆
Low friction torque[1]	☆☆☆☆	☆☆☆		☆☆	☆	☆		
High rigidity[1]			☆☆	☆☆		☆☆	☆☆	☆☆
Vibration/shock resistance[1]			☆		★	☆☆	☆☆	☆☆
Allowable misalignment for inner/outer rings[1]	☆				☆☆☆	☆		
For fixed bearings[2]	◎	○	◎	◎ For DB and DF arrangment	◎		○	◎
For floating bearings[1]	○		○	○ For DB arrangment	○	◎		
Non-separable or separable[4]						○	○	○
Tapered bore bearings[5]					○			
Remarks		For duplex arrangment				NU, N type	NJ, NF type	NUP, NP, NH type
Reference page	**B-6**	**B-44**	**B-68**	**B-44**	**B-74**	**B-84**	**B-84**	**B-84**

Note 1) ☆The number of stars indicate the degree to which that bearing type displays that particular characteristi
★ Not applicable to that bearing type.
2) ◎ Indicates dual direction. ○ Indicates single direction axial movement only.

Double row cylindrical roller bearings	Needle roller bearings	Tapered roller bearings	Spherical roller bearings	Thrust ball bearings	Thrust ball bearings with seating ring	Double row angular contact thrust ball bearings	Cylindrical roller thrust bearings	Spherical roller thrust bearings	Reference page
									
									
☆☆☆	☆☆☆	☆☆☆	☆☆	☆	☆	☆☆☆	☆	☆	A-77
☆☆☆		☆☆☆		☆		☆☆☆			A-22
☆	☆								A-77
									A-78
☆☆☆	☆☆	☆☆	☆☆☆			☆☆	☆☆☆	☆☆☆	A-74
☆☆	☆☆	☆☆	☆☆☆			★	☆☆☆	☆☆☆	—
		☆	☆☆☆	★	☆☆☆	★	★	☆☆☆	—
		○	◎	○	○	◎	○	○	A-94
◎	◎		○						A-94
○	○	○		○	○	○	○	○	—
○			○						A-99
NNU, NN, type		For duplex arrangment					Including thrust needle roller bearings		
B-85	**B-112**	**B-118**	**B-186**	**B-218**	**B-218**	**B-218**	**B-218**	**B-218**	

c. 3) ◎ Indicates movement at raceway. ○ Indicates movement at mated surface of inner or outer ring.
4) ○ Indicates both inner ring and outer ring are detachable.
5) ○ Indicates inner ring with tapered bore is possible.

Table 2.3 (1) Bearing arrangement (Fixed and Floating)

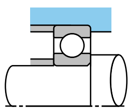
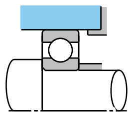
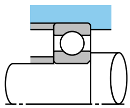
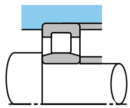
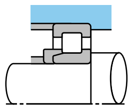
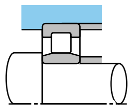
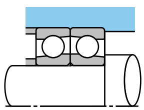
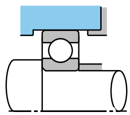
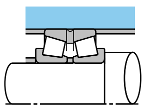
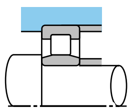
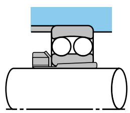
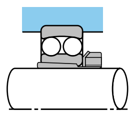
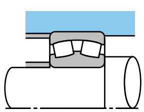
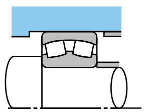
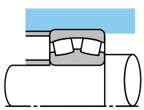
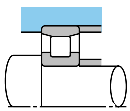
Arrangement		Comment	Application
Fixed	Floating		
		1. General arrangement for small machinery 2. For radial loads, but will also accept axial loads. 3. Preloading by springs or shims on outer ring face.	Small pumps, small electric motors, auto-mobile transmissions, etc.
		1. Suitable for high speed. Widely used. 2. Even with expansion and contraction of shaft, non-fixing side moves smoothly.	Medium-sized electric motors, ventilators, etc.
		1. Withstands heavy loading and some axial loading. 2. Inner and outer ring shrink-fit suitable. 3. Easy mounting and dismounting.	Railway vehicle electric motors, etc.
		1. Radial loading plus dual direction axial loading possible. 2. In place of duplex angular contact ball bearings, double-row angular contact ball bearings are also used.	Wormgear speed reducers, etc.
		1. Heavy loading capable. 2. Shafting rigidity increased by preloading the two back-to-back fixed bearings. 3. Requires high precision shafts and housings, and minimal fitting errors.	Machine tool spindles, etc.
		1. Allows for shaft deflection and fitting errors. 2. By using an adaptor on long shafts without screws or shoulders, bearing mounting and dismounting can be facilitated. 3. Not suitable for axial load applications.	Counter shafts for general industrial equipment, etc.
		1. Widely used in general industrial machinery with heavy and shock load demands. 2. Allows for shaft deflection and fitting errors. 3. Accepts radial loads as well as dual direction axial loads.	Reduction gears for general industrial equipment, etc.
		1. Widely used in general industrial machinery with heavy and shock loading. 2. Radial and dual directional axial loading.	Industrial machinery reduction gears, etc.

Table 2.3 (2) Bearing arrangement (Placed oppositely)

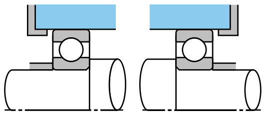
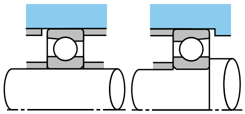
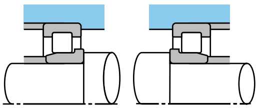
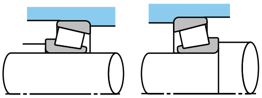
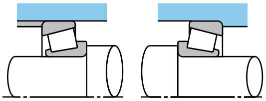
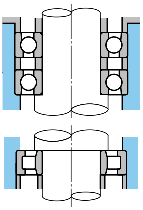
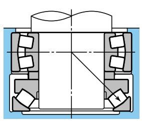
Arrangement	Comment	Application
	General arrangement for use in small machines.	Small electric motors, small reduction gears, etc.
	1. This type of back-to-back arrangement well suited for moment loads. 2. Preloading increases shaft rigidity. 3. High speed reliable.	Spindles of machine tools, etc.
	1. Accepts heavy loading. 2. Suitable if inner and outer ring shrink-fit is required. 3. Care must be taken that axial clearance does not become too small during operation.	Construction equipment, mining equipment sheaves, agitators, etc.
 Back-to-back arrangement  Face-to-face arrangement	1. Withstands heavy and shock loads. Wide range application. 2. Shafting rigidity increased by preloading. 3. Back-to-back arrangement for moment loads, and face-to-face arrangement to alleviate fitting errors. 4. With face-to-face arrangement, inner ring shrink-fit is facilitated.	Reduction gears, automotive axles, etc.

Table 2.3 (3) Bearing arrangement (Vertical shaft)

Arrangement	Comment	Application
	When fixing bearing is a duplex angular contact ball bearing, non-fixing bearing is a cylindrical rollerbearing.	Machine tool spindles, vertical mounted electric motors, etc.
	1. Most suitable arrangement for very heavy axial loads. 2. Depending on the relative alignment of the spherical surface of the rollers in the upper and lower bearings, shaft deflection and fitting errors can be absorbed. 3. Lower self-aligning spherical roller thrust bearing pre-load is possible.	Crane center shafts, etc.

3. Boundary Dimensions and Bearing Number Codes

3.1 Boundary dimensions

To facilitate international interchangeability and economic bearing production, the boundary dimensions of rolling bearings have been internationally standardized by the International Organization for Standardization (ISO) ISO 15 (radial bearings-except tapered roller bearings), ISO 355 (tapered roller bearings), and ISO 104 (thrust bearings).

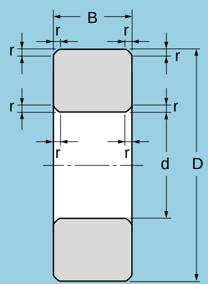
In Japan, standard boundary dimensions for rolling bearings are regulated by Japanese Industrial Standards (JIS B 1512) in conformity with the ISO standards.

Those boundary dimensions which have been standardized; i.e. bore diameter, outside diameter, width or height and chamfer dimensions are shown in cross-section in Figs. 3.1-3.4. However, as a general rule, bearing internal construction dimensions are not covered by these standards.

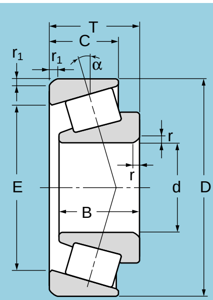
The 90 standardized bore diameters (*d*) for rolling bearings under the metric system range from 0.6 mm - 2500 mm and are shown in Table 3.1.

For all types of standard bearings there has been established a combined series called the dimension series. In all radial bearings (except tapered roller bearings) there are eight major outside diameters (*D*) for each standard bore diameter. This series is called the diameter series and is expressed by the number sequence (7, 8, 9, 0, 1, 2, 3, 4) in order of ascending magnitude (7 being the smallest and 4 being the largest).

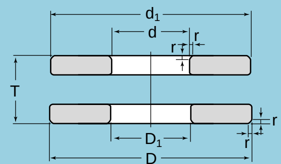
For the same bore and outside diameter combination there are eight width designations (*B*). This series is called the width series and is expressed by the number sequence (8, 0, 1, 2, 3, 4, 5, 6) in order of ascending size (i.e. 8 narrowest and 6 widest). The combination of these two series, the diameter series and the width series, forms the dimension series.



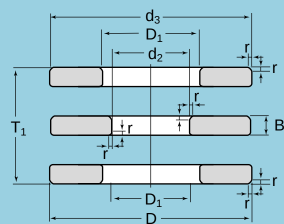
Boundary dimension of radial bearings
Fig. 3.1



Boundary dimension of tapered roller bearings
Fig. 3.2



Boundary dimension of single direction thrust bearings
Fig. 3.3



Boundary dimension of double direction of thrust bearings
Fig. 3.4

Table 3.1 Standardized bore diameter

Bore diameter for nominal bearing *d* mm over	include	Standardized bore diameter mm	Standard
—	1.0	0.6	—
1.0	3.0	1, 1.5, 2.5	Every 0.5 mm
3.0	10	3, 4,...9	Every 1 mm
10	20	10, 12, 15, 17	—
20	35	20, 22, 25, 28, 30, 32	Stanard number R20 series
35	110	35, 40, ...105	Every 5 mm
110	200	110, 120, ...190	Every 10 mm
200	500	200, 220, ...480	Every 20 mm
500	2500	500, 530, 2500	Standard number R40 series

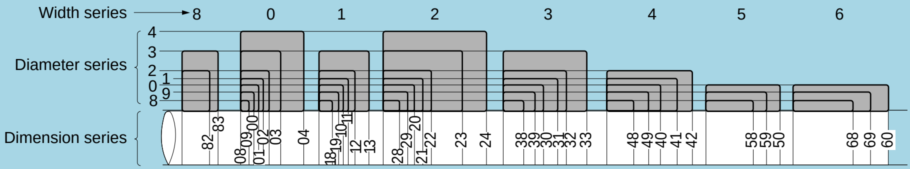


FIg. 3.5 Comparison of dimension series (Except tapered roller bearings) for radial bearings of same bore diameter

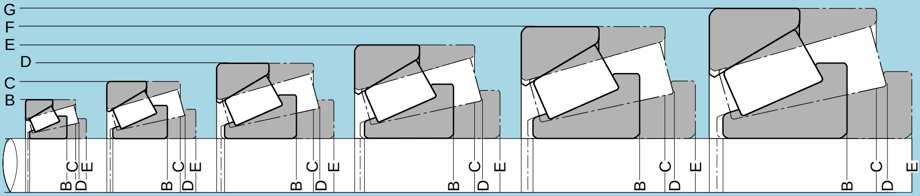


Fig. 3.6 Comparison of dimension series for tapered roller bearings

The relationship of these three series is illustrated in Fig. 3.5.

For tapered roller bearings, the standard bore (*d*) and outside diameter (*D*) combined series (i.e. diameter series) has six major divisions and is expressed by the letter sequence (B, C, D, E, F, G) in ascending order of the outside diameter size (B is the smallest outside diameter and G is the largest outside diameter). The width (*T*) is expressed in the width series by a four letter sequence (B, C, D, E) in ascending order; i.e. E being the widest.

The contact angle (∝) is shown by a six number contact angle series (2, 3, 4, 5, 6, 7) in ascending order (i.e. 2 being the smallest angle and 7 the largest angle). The combination of the contact angle series, the diameter series and the width series form the dimension series for tapered roller bearings (example: 2FB). This series relationship is shown in Fig. 3.6.

For thrust bearings, the standard bore diameter (*d*) and the outside diameter (*D*) relationship is expressed by the five major number diameter series (0, 1, 2, 3, 4). For the same bore and outside diameter combination, the height dimensions (*T*) is standardized into 4 steps and is expressed by the number sequence (7, 9, 1, 2). This relationship is shown in Fig. 3.7.

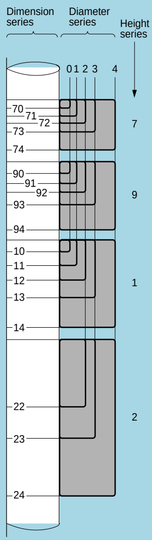


Fig. 3.7 Comparison of dimension series for thrust bearing of the same bore diameter

Chamfer dimensions (*r*) are covered by ISO standard 582 and JIS standard B1512 ($r_{s\ min}$: minimum allowable chamfer dimension). There are twenty-two standardized dimensions for chamfers ranging from 0.1 mm to 19 nn (0.05, 0.08, 0.1, 0.15, 0.2, 0.3, 0.6, 1, 1.1, 1.5, 2, 2.1, 2.5, 3, 4, 5, 6, 7.5, 9.5, 12, 15, 19).

Not all of the above mentioned standard boundary dimensions and size combinations (bore diameter, diameter series, width or height series) are standardized. Moreover, there are many standard bearing sizes which are not manufactured. Please refer to the bearing dimension tables in this catalog.

3.2 Bearing numbers

The bearing numbers indicate the bearing design, dimensions, accuracy, internal construction, etc.

The bearing number is derived from a series of number and letter codes, and is composed of three main groups of codes; i.e. two supplementary codes and a basic number code. The sequence and definition of these codes is shown in Table 3.2.

The basic number indicates general information such as bearing design, boundary dimensions, etc.: and is composed of the bearing series code, the bore diameter number and the contact angle code. These coded series are shown in Tables 3.4, 3.5, and 3.6 respectively.

The supplementary codes are derived from a prefix code series and a suffix code series. These codes designate bearing accuracy, internal clearance and other factors relating to bearing specifications and internal construction. These two codes are shown in Tables 3.3 and 3.7.

Table 3.2 Bearing number sequence

Number and code arrangement				TS2 - 7 3 05 B L1 DF+10 C3 P5
Supplementary prefix code	Special application code Material/heat treatment code			TS2
Basic number	Bearing series	Design code		7
		Dimension series code	Width/height series code	
			Diameter series code	3
	Bore diameter number			05
	Contact angle code			B
Supplementary suffix code	Internal modification code			
	Cage codes			L1
	Seal/Shield code			
	Ring configuration code			
	Duplex arrangement code			DF+10
	Internal clearance code			C3
	Tolerances code			P5
	Lubrication code			

Table 3.3 Supplementary prefix code

Code	Definition
TS-	Dimension stabilized bearing for high temperature use
M-	Hard chrome plated bearings
F-	Stainless steel bearings
H-	High speed steel bearings
N-	Special material bearings
TM-	Specially treated long-life bearings
EC-	Expansion compensation bearings
4T-	NTN 4 Top tapered roller bearings
ET-	ET Tapered roller bearings

Table 3.4 Bearing series symbol

Bearing series	Type symbol	Dimension series		Bearing type
		width series	diameter series	
67	6	(1)	7	Single row deep groove ball bearings
68		(1)	8	
69		(1)	9	
60		(1)	0	
62		(0)	2	
63		(0)	3	
78	7	(1)	8	Single row angular contact ball bearings
79		(1)	9	
70		(1)	0	
72		(0)	2	
73		(0)	3	
12	1	(0)	2	Self-aligning ball bearings
13	1	(0)	3	
22	2	(2)	2	
23	2	(2)	3	
NU10	NU	1	0	Cylindrical roller bearings
NU2		(0)	2	
NU22		2	2	
NU3		(0)	3	
NU23		2	3	
NU4		(0)	4	
N10	N	1	0	
N2		(0)	2	
N3		(0)	3	
N4		(0)	4	
NF2	NF	(0)	2	
NF3		(0)	3	
NA48	NA	4	8	Needle roller bearings
NA49		4	9	
NA59		5	9	
329X	3	2	9	Tapered roller bearings
320X		2	0	
302		0	2	
322		2	2	
303		0	3	
303D		0	3	
313X		1	3	
323		2	3	
239	2	3	9	Spherical roller bearings
230		3	0	
240		4	0	
231		3	1	
241		4	1	
222		2	2	
232		3	2	
213		0	3	
223		2	3	
511	5	1	1	Single-thrust ball bearings
512			2	
513			3	
514			4	
522	5	2	2	Double-thrust ball bearings
523			3	
524			4	
811	8	1	1	Cylindrical roller thrust bearings
812		1	2	
893		9	3	
292	2	9	2	Spherical roller thrust bearings
293			3	
294			4	

Table 3.6 Contact angle code

Code	Nominal contact angle		Bearing type
A[1)]	Standard 30°		Angular contact ball bearings
B	Standard 40°		
C	Standard 15°		
B[1)]	Over 10°	Incl. 17°	Tapered roller bearings
C	Over 17°	Incl. 24°	
D	Over 24°	Incl. 32°	

Note 1) A and B are not usually included in bearing numbers.

Table 3.5 Bore diameter number

Bore diameter number	Bore diameter *d* mm	Remark
/0.6 /1.5 /2.5	0.6 1.5 2.5	Slash (/) before bore diameter number
1 ⋮ 9	1 ⋮ 9	Bore diameter expressed in single digits without code
00 01 02 03	10 12 15 17	———
/22 /28 /32	22 28 32	Slash (/) before bore diameter number
04 05 06 ⋮ 88 92 96	20 25 30 ⋮ 440 460 480	Bore diameter number in double digits after dividing bore diameter by 5
/500 /530 /560 /2360 /2500	500 530 560 ⋮ 2360 2500	Slash (/) before bore diameter number

Table 3.7 Supplementary suffix code

	Code	Explanation
Internal modifications	U	Internationally interchangeable tapered roller bearings
	R	Non-internationally interchangeable tapered roller bearings
	ST	Low torque tapered roller bearings
	HT	High axial load use cylindrical roller bearings
Cage	L1	Machined Brass cage
	F1	Machined steel cage
	G1	Machined brass cage for cylindrical roller bearings, rivetless
	G2	Pin-type steel cage for tapered roller bearings
	J	Pressed steel cage
	T1	Phenolic cage
	T2	Plastic cage, nylon or teflon
Seal or shield	LLB	Synthetic rubber seal (non-contact type)
	LLU	Synthetic rubber seal (contact type)
	ZZ	Shield
	ZZA	Removable shield
Ring configuration	K	Tapered inner ring bore, taper 1 : 12
	K30	Tapered inner ring bore, taper 1 : 30
	N	Snap ring groove on outer ring, but without snap ring
	NR	Snap ring on outer ring
	D	Bearings with oil holes
Duplex arrangement	DB	Back-to-back arrangement
	DF	Face-to-face arrangement
	DT	Tandem arrangement
	D2	Two identical paired bearings
	G	Single bearings, flush ground side face for DB, DF and DT
	$+\alpha$	Spacer, (α=nominal width of spacer, mm)

	Code	Explanation
Internal clearance	C2	Radial internal clearance less than Normal
	C3	Radial internal clearance greater than Normal
	C4	Radial internal clearance greater than C3
	CM	Radial internal clearance for electric motor bearings
	NA	Non-interchangeable clearance (shown after clearance code)
	/GL	Light preload
	/GN	Normal preload
	/GM	Medium preload
	/GH	Heavy preload
Tolerance standard	P6	JIS standard Class 6
	P6X	JIS standard Class 6X (tapered roller brg.)
	P5	JIS standard Class 5
	P4	JIS standard Class 4
	P2	JIS standard Class 2
	2	Class 2 for inch series tapered roller bearings
	3	Class 3 for inch series tapered roller beaings
	0	Class 0 for inch series tapered roller bearings
	00	Class 00 for inch series tapered roller bearings
Lubrication	/2A	Shell Alvania 2 grease
	/5C	Chevron SRI 2
	/3E	ESSO Beacon 325 grease
	/5K	MUL-TEMP SRL

4. Bearing Tolerances

Bearing tolerances; i.e., dimensional accuracy, running accuracy, etc., are regulated by standards such as ISO and JIS. For dimensional accuracy these standards prescibe tolerances and allowable error limitations for those boundry dimensions (bore diameter, outside diameter, width, assembled bearing width, chamfer, and taper) necessary when installing bearings on shafts or in housings. For machining accuracy the standards provide allowable variation limits on bore, mean bore, outside diameter, mean outside diameter and raceway width or wall thickness (for thrust bearings). Running accuracy is defined as the allowable limits for bearing runout. Bearing runout tolerances are included in the standards for inner and outer ring radial and axial runout; inner ring side runout with bore; and outer ring outside surface runout with side.

Tolerances and allowable error limitations are established for each tolerance grade or class. For example, JIS standard B 1514 (tolerances for rolling bearings) establishes five tolerance classifications (classes 0, 6, 5, 4, 2).

Starting with class 0 (normal precision class bearings), the bearing precision becomes progressively greater as the class number becomes smaller.

A comparison of relative tolerance class standards between the JIS B1514 standard classes and other standards is shown in the comparative Table 4.1.

Table 4.2 indicates which standard and tolerance class is applicable to each bearing type.

Table 4.1 Comparison of tolerance classifications of national standards

Standard		Tolerance Class					Bearing Types
Japanese Industrial Standard	JIS B 1514	Class 0 Class 6X	Class 6	Class 5	Class 4	Class 2	All types
International Organization for Standardization	ISO 492	Normal class Class 6X	Class 6	Class 5	Class 4	Class 2	Radial bearings
	ISO 199	Normal class	Class 6	Class 5	Class 4	—	Thrust ball bearings
	ISO 578	Class 4	—	Class 3	Class 0	Class 00	Tapered roller bearings (Inch series)
	ISO 1224	—	—	Class 5A	Class 4A	—	Precision instrument bearings
Deutsches Institut fur Normung	DIN 620	P0	P6	P5	P4	P2	All types
American National Standards Institute (ANSI) Anti-Friction Bearing Manufacturers (AFBMA)	ANSI/AFBMA Std. 20[1]	ABEC-1 RBEC-1	ABEC-3 RBEC-3	ABEC-5 RBEC-5	ABEC-7	ABEC-9	Radial bearings (Except tapered roller bearings)
	ANSI/AFBMA Std. 19.1	Class K	Class N	Class C	Class B	Class A	Tapered roller bearings (Metric series)
	ANSI B 3.19 AFBMA Std. 19	Class 4	Class 2	Class 3	Class 0	Class 00	Tapered roller bearings (Inch series)
	ANSI/AFBMA Std. 12.1	—	Class 3P	Class 5P Class 5T	Class 7P Class 7T	Class 9P	Precision instrument ball bearings (Metric series)
	ANSI/AFBMA Sts. 12.2	—	Class 3P	Class 5P Class 5T	Class 7P Class 7T	Class 9P	Precision instrument ball bearings (Inch series)

1) "ABEC" is applied for ball bearings and "RBEC" for roller bearings.

Notes:
1. JIS B 1514, ISO 492 and 199, and DIN 620 have the same specification level.
2. The tolerance and allowance of JIS B 1514 are a little different from those of AFBMA standards.

Table 4.2 Bearing types and applicable tolerance

Bearing Type		Applicable standard	Applicable tolerance					Tolerance table
Deep groove ball bearing		ISO 492	class 0	class 6	class 5	class 4	class 2	Table 4.3
Angular contact ball bearings			class 0	class 6	class 5	class 4	class 2	
Self-aligning ball bearings			class 0	—	—	—	—	
Cylindrical roller bearings			class 0	class 6	class 5	class 4	class 2	
Needle roller bearings			class 0	class 6	class 5	class 4	—	
Spherical roller bearings			class 0	—	—	—	—	
Tapered	metric	ISO 492	class 0,6X	class 6	class 5	class 4	—	Table 4.4
roller	inch	AFBMA Std. 19	class 4	class 2	class 3	class 0	class 00	Table 4.5
bearings	J series	ANSI/AFBMA Std.19.1	class K	class N	class C	class B	class A	Table 4.6
Thrust ball bearings		ISO 199	class 0	class 6	class 5	class 4	—	Table 4.7
Thrust roller bearings		NTN standard	class 0	class 6	class 5	class 4	—	Page B-219 Table 2
Spherical roller thrust bearings		ISO 199	class 0	—	—	—	—	Table 4.8
Double direction angular contact thrust ball bearings		NTN standard	—	—	class 5	class 4	—	Table 4.9

The following is a list of codes and symbols used in the bearing tolerance standards tables. However, in some cases the code or symbol definition has been abbreviated.

(1) Dimension

- d : Nominal bore diameter
- d_2 : Nominal bore diameter (double direction thrust ball bearing)
- D : Nominal outside diameter
- B : Nominal inner ring width or nominal center washer height
- C : Nominal outer ring width[1)]
  Note 1) For radial bearings (except tapered roller bearings) this is equivalent to the nominal bearing width.
- T : Nominal bearing width of single row tapered roller bearing, or nominal height of single direction thrust bearing
- T_1 : Nominal height of double direction thrust ball bearing, or nominal effective width of inner ring and roller assembly of tapered roller bearing
- T_2 : Nominal height from back face of housing washer to back face of center washer on double direction thrust ball bearings, or nominal effective outer ring width of tapered roller bearing
- r : Chamfer dimensions of inner and outer rings (for tapered roller bearings, large end of inner rilng only)
- r_1 : Chamfer dimensions of center washer, or small end of inner and outer ring of angular contact ball bearing, and large end of outer ring of tapered roller bearing
- r_2 : Chamfer dimensions of small end of inner and outer rings of tapered roller bearing

(2) Dimension deviation

Δ_{ds} : Single bore diameter deviation
Δ_{dmp} : Single plane mean bore diameter deviation
Δ_{d2mp} : Single plane mean bore diameter deviation (double direction thrust ball bearing)
Δ_{Ds} : Single outside diameter deviation
Δ_{Dmp} : Single plane mean outside diameter deviation
Δ_{Bs} : Inner ring width deviation, or center washer height deviation
Δ_{Cs} : Outer ring width deviation
Δ_{Ts} : Overall width deviation of assembled single row tapered roller bearing, or height deviation of single direction thrust bearing
Δ_{T1s} : Height deviation of double direction thrust ball bearing, or effective width deviation of roller and inner ring assembly of tapered roller bearing
Δ_{T2s} : Double direction thrust ball bearing housing washer back face to center washer back face height deviation, or tapered roller bearing outer ring effective width deviation

(3) Chamfer boundry

r_s min : Minimum allowable chamfer dimension for inner/outer ring, or small end of inner ring on tapered roller bearing
r_s max : Maximum allowable chamfer dimension for inner/outer ring, or large end of inner ring on tapered roller bearing
r_{1s} min : Minimum allowable chamfer dimension for double direction thrust ball bearing center washer, small end of inner/outer ring of angular contact ball bearing, large end of outer ring of tapered roller bearing
r_{1s} max : Maximum allowable chamfer dimension for double direction thrust ball bearing center washer, small end of inner/outer ring of angular contact ball bearing, large end of outer ring of tapered roller bearing
r_{2s} min : Minimum allowable chamfer dimension for small end of inner/outer ring of tapered roller bearing
r_{2s} max : Maximum allowable chamfer dimension for small end of inner/outer ring of tapered roller bearing

(4) Dimension variation

V_{dp} : Single radial plane bore diameter variation
V_{d2p} : Single radial plane bore diameter variation (double direction thrust ball bearing)
V_{dmp} : Mean single plane bore diameter variation
V_{Dp} : Single radial plane outside diameter variation
V_{Dmp} : Mean single plane outside diameter variation
V_{Bs} : Inner ring width variation
V_{Cs} : Outer ring width variation

(5) Rotation tolerance

K_{ia} : Inner ring radial runout
S_{ia} : Inner ring axial runout (with side)
S_d : Face runout with bore
K_{ea} : Outer ring radial runout
S_{ea} : Outer ring axial runout
S_D : Outside surface inclination
S_i : Thrust beaing shaft washer raceway (or center washer raceway) thickness variation
S_e : Thrust bearing housing washer raceway thickness variation

Table 4.3 Tolerance for radial bearings (Except tapered roller bearings) max

Table 4.3 (1) Inner rings

Nominal bore diameter d (mm)		Δ_{dmp}										V_{dp}														
		class 0		class 6		class 5		class 4 [1]		class 2 [1]		diameter series 7,8,9					diameter series 0,1					diameter series 2,3,4				
												class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2
over	inc.	high	low	high	low	high	low	high	low	high	low	max					max					max				
0.6[1]	2.5	0	−8	0	−7	0	−5	0	−4	0	−2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
2.5	10	0	−8	0	−7	0	−5	0	−4	0	−2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
10	18	0	−8	0	−7	0	−5	0	−4	0	−2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
18	30	0	−10	0	−8	0	−6	0	−5	0	−2.5	13	10	6	5	2.5	10	8	5	4	2.5	8	6	5	4	2.5
30	50	0	−12	0	−10	0	−8	0	−6	0	−2.5	15	13	8	6	2.5	12	10	6	5	2.5	9	8	6	5	2.5
50	80	0	−15	0	−12	0	−9	0	−7	0	−4	19	15	9	7	4	19	15	7	5	4	11	9	7	5	4
80	120	0	−20	0	−15	0	−10	0	−8	0	−5	25	19	10	8	5	25	19	8	6	5	15	11	8	6	5
120	150	0	−25	0	−18	0	−13	0	−10	0	−7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
150	180	0	−25	0	−18	0	−13	0	−10	0	−7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
180	250	0	−30	0	−22	0	−15	0	−12	0	−8	38	28	15	12	8	38	28	12	9	8	23	17	12	9	8
250	315	0	−35	0	−25	0	−18	—	—	—	—	44	31	18	—	—	44	31	14	—	—	26	19	14	—	—
315	400	0	−40	0	−30	0	−23	—	—	—	—	50	38	23	—	—	50	38	23	—	—	30	23	18	—	—
400	500	0	−45	0	−35	—	—	—	—	—	—	56	44	—	—	—	56	44	—	—	—	34	26	—	—	—
500	630	0	−50	0	−40	—	—	—	—	—	—	63	50	—	—	—	63	50	—	—	—	38	30	—	—	—
630	800	0	−75	—	—	—	—	—	—	—	—	94	—	—	—	—	94	—	—	—	—	55	—	—	—	—
800	1000	0	−100	—	—	—	—	—	—	—	—	125	—	—	—	—	125	—	—	—	—	75	—	—	—	—
1000	1250	0	−125	—	—	—	—	—	—	—	—	155	—	—	—	—	155	—	—	—	—	94	—	—	—	—
1250	1600	0	−160	—	—	—	—	—	—	—	—	200	—	—	—	—	200	—	—	—	—	120	—	—	—	—
1600	2000	0	−200	—	—	—	—	—	—	—	—	250	—	—	—	—	250	—	—	—	—	150	—	—	—	—

1) The dimensional difference Δ_{ds} of bore diameter to be applied for classes 4 and 2 is the same as the tolerance of dimensional difference Δ_{dmp} of average bore diameter. However, the dimensional difference is applied to diameter series 0,1,2,3 and 4 against Class 4, and also to all the diameter series against Class 2.

Table 4.3 (2) Outer rings

Nominal outside diameter D (mm)		Δ_{Dmp}										V_{Dp} [6] open type														
		class 0		class 6		class 5		class 4 [5]		class 2 [5]		diameter series 7,8,9					diameter series 0,1					diameter series 2,3,4				
												class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2
over	inc.	high	low	high	low	high	low	high	low	high	low	max					max					max				
2.5[8]	6	0	−8	0	−7	0	−5	0	−4	0	−2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
6	18	0	−8	0	−7	0	−5	0	−4	0	−2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
18	30	0	−9	0	−8	0	−6	0	−5	0	−4	12	10	6	5	4	9	8	5	4	4	7	6	5	4	4
30	50	0	−11	0	−9	0	−7	0	−6	0	−4	14	11	7	6	4	11	9	5	5	4	8	7	5	5	4
50	80	0	−13	0	−11	0	−9	0	−7	0	−4	16	14	9	7	4	13	11	7	5	4	10	8	7	5	4
80	120	0	−15	0	−13	0	−10	0	−8	0	−5	19	16	10	8	5	19	16	8	6	5	11	10	8	6	5
120	150	0	−18	0	−15	0	−11	0	−9	0	−5	23	19	11	9	5	23	19	8	7	5	14	11	8	7	5
150	180	0	−25	0	−18	0	−13	0	−10	0	−7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
180	250	0	−30	0	−20	0	−15	0	−11	0	−8	38	25	15	11	8	38	25	11	8	8	23	15	11	8	8
250	315	0	−35	0	−25	0	−18	0	−13	0	−8	44	31	18	13	8	44	31	14	10	8	26	19	14	10	8
315	400	0	−40	0	−28	0	−20	0	−15	0	−10	50	35	20	15	10	50	35	15	11	10	30	21	15	11	10
400	500	0	−45	0	−33	0	−23	—	—	—	—	56	41	23	—	—	56	41	17	—	—	34	25	17	—	—
500	630	0	−50	0	−38	0	−28	—	—	—	—	63	48	28	—	—	63	48	21	—	—	38	29	21	—	—
630	800	0	−75	0	−45	0	−35	—	—	—	—	94	56	35	—	—	94	56	26	—	—	55	34	26	—	—
800	1000	0	−100	0	−60	—	—	—	—	—	—	125	75	—	—	—	125	75	—	—	—	75	45	—	—	—
1000	1250	0	−125	—	—	—	—	—	—	—	—	155	—	—	—	—	155	—	—	—	—	94	—	—	—	—
1250	1600	0	−160	—	—	—	—	—	—	—	—	200	—	—	—	—	200	—	—	—	—	120	—	—	—	—
1600	2000	0	−200	—	—	—	—	—	—	—	—	250	—	—	—	—	250	—	—	—	—	150	—	—	—	—
2000	2500	0	−250	—	—	—	—	—	—	—	—	310	—	—	—	—	310	—	—	—	—	190	—	—	—	—

5) The dimensional difference Δ_{Ds} of outer diameter to be applied for classes 4 and 2 is the same as the tolerance of dimensional difference Δ_{Dmp} of average outer diameter. However, the dimensional difference is applied to diameter series 0,1,2,3 and 4 against Class 4, and also to all the diameter series against Class 2.

Unit μm

V_{dp}					K_{ia}					S_d			S_{ia}[2]			Δ_{Bs}										V_{Bs}				
																normal						modified[3]								
class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2	class 5	class 4	class 2	class 0,6		class 5,4		class 2		class 0,6		class 5,4		class 0	class 6	class 5	class 4	class 2
max					max					max			max			high	low	high	low	high	low	high	low	high	low	max				
6	5	3	2	1.5	10	5	4	2.5	1.5	7	3	1.5	7	3	1.5	0	–40	0	–40	0	–40	—	—	0	–250	12	12	5	2.5	1.5
6	5	3	2	1.5	10	6	4	2.5	1.5	7	3	1.5	7	3	1.5	0	–120	0	–40	0	–40	0	–250	0	–250	15	15	5	2.5	1.5
6	5	3	2	1.5	10	7	4	2.5	1.5	7	3	1.5	7	3	1.5	0	–120	0	–80	0	–80	0	–250	0	–250	20	20	5	2.5	1.5
8	6	3	2.5	1.5	13	8	4	3	2.5	8	4	1.5	8	4	2.5	0	–120	0	–120	0	–120	0	–250	0	–250	20	20	5	2.5	1.5
9	8	4	3	1.5	15	10	5	4	2.5	8	4	1.5	8	4	2.5	0	–120	0	–120	0	–120	0	–250	0	–250	20	20	5	3	1.5
11	9	5	3.5	2	20	10	5	4	2.5	8	5	1.5	8	5	2.5	0	–150	0	–150	0	–150	0	–380	0	–250	25	25	6	4	1.5
15	11	5	4	2.5	25	13	6	5	2.5	9	5	2.5	9	5	2.5	0	–200	0	–200	0	–200	0	–380	0	–380	25	25	7	4	2.5
19	14	7	5	3.5	30	18	8	6	2.5	10	6	2.5	10	7	2.5	0	–250	0	–250	0	–250	0	–500	0	–380	30	30	8	5	2.5
19	14	7	5	3.5	30	18	8	6	5	10	6	4	10	7	5	0	–250	0	–250	0	–300	0	–500	0	–380	30	30	8	5	4
23	17	8	6	4	40	20	10	8	5	11	7	5	13	8	5	0	–300	0	–300	0	–350	0	–500	0	–500	30	30	10	6	5
26	19	9	—	—	50	25	13	—	—	13	—	—	15	—	—	0	–350	0	–350	—	—	0	–500	0	–500	35	35	13	—	—
30	23	12	—	—	60	30	15	—	—	15	—	—	20	—	—	0	–400	0	–400	—	—	0	–630	0	–630	40	40	15	—	—
34	26	—	—	—	65	35	—	—	—	—	—	—	—	—	—	0	–450	—	—	—	—	—	—	—	—	50	45	—	—	—
38	30	—	—	—	70	40	—	—	—	—	—	—	—	—	—	0	–500	—	—	—	—	—	—	—	—	60	50	—	—	—
55	—	—	—	—	80	—	—	—	—	—	—	—	—	—	—	0	–750	—	—	—	—	—	—	—	—	70	—	—	—	—
75	—	—	—	—	90	—	—	—	—	—	—	—	—	—	—	0	–1000	—	—	—	—	—	—	—	—	80	—	—	—	—
94	—	—	—	—	100	—	—	—	—	—	—	—	—	—	—	0	–1250	—	—	—	—	—	—	—	—	100	—	—	—	—
120	—	—	—	—	120	—	—	—	—	—	—	—	—	—	—	0	–1600	—	—	—	—	—	—	—	—	120	—	—	—	—
150	—	—	—	—	140	—	—	—	—	—	—	—	—	—	—	0	–2000	—	—	—	—	—	—	—	—	140	—	—	—	—

2) To be applied for deep groove ball bearings and angular contact ball bearings.
3) To be applied for individual raceway rings manufactured for combined bearing use.
4) Nominal bore diameter of bearings of 0.6 mm is included in this dimensional division.

Unit μm

V_{Dp}[6] capped bearings diameter series		V_{Dmp}					K_{ea}					S_D			S_{ea}			Δ_{Cs}	V_{Cs}			
class 6	class 0	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2	class 5	class 4	class 2	all type	class 0,6	class 5	class 4	class 2
max		max					max					max			max				max			
10	9	6	5	3	2	1.5	15	8	5	3	1.5	8	4	1.5	8	5	1.5	Identical to Δ_{Bs} of inner ring of same bearing	Identical to Δ_{Bs} and V_{Bs} of inner ring of same bearing	5	2.5	1.5
10	9	6	5	3	2	1.5	15	8	5	3	1.5	8	4	1.5	8	5	1.5			5	2.5	1.5
12	10	7	6	3	2.5	2	15	9	6	4	2.5	8	4	1.5	8	5	2.5			5	2.5	1.5
16	13	8	7	4	3	2	20	10	7	5	2.5	8	4	1.5	8	5	2.5			5	2.5	1.5
20	16	10	8	5	3.5	2	25	13	8	5	4	8	4	1.5	10	5	4			6	3	1.5
26	20	11	10	5	4	2.5	35	18	10	6	5	9	5	2.5	11	6	5			8	4	2.5
30	25	14	11	6	5	2.5	40	20	11	7	5	10	5	2.5	13	7	5			8	5	2.5
38	30	19	14	7	5	3.5	45	23	13	8	5	10	5	2.5	14	8	5			8	5	2.5
—	—	23	15	8	6	4	50	25	15	10	7	11	7	4	15	10	7			10	7	4
—	—	26	19	9	7	4	60	30	18	11	7	13	8	5	18	10	7			11	7	5
—	—	30	21	10	8	5	70	35	20	13	8	13	10	7	20	13	8			13	8	7
—	—	34	25	12	—	—	80	40	23	—	—	15	—	—	23	—	—			15	—	—
—	—	38	29	14	—	—	100	50	25	—	—	18	—	—	25	—	—			18	—	—
—	—	55	34	18	—	—	120	60	30	—	—	20	—	—	30	—	—			20	—	—
—	—	75	45	—	—	—	140	75	—	—	—	—	—	—	—	—	—			—	—	—
—	—	94	—	—	—	—	160	—	—	—	—	—	—	—	—	—	—			—	—	—
—	—	120	—	—	—	—	190	—	—	—	—	—	—	—	—	—	—			—	—	—
—	—	150	—	—	—	—	220	—	—	—	—	—	—	—	—	—	—			—	—	—
—	—	190	—	—	—	—	250	—	—	—	—	—	—	—	—	—	—			—	—	—

6) To be applied in case snap rings are not installed on the bearings.
7) To be applied for deep groove ball bearings and angular contact ball bearings.
8) Nominal outer diameter of bearings of 2.5 mm is included in this dimensional division.

Table 4.4 Tolerance for tapered roller bearings (Metric system)

Table 4.4 (1) Inner rings

Nominal bore diameter d (mm)		Δ_{dmp}						V_{dp}				V_{dmp}				K_{ia}				S_d	
		class 0,6X		class 5,6		class 4[1)]		class 0, 6X	class 6	class 5	class 4	class 0, 6X	class 6	class 5	class 4	class 0, 6X	class 6	class 5	class 4	class 5	class 4
over	incl.	high	low	high	low	high	low	max				max				max				max	
10	18	0	−12	0	−7	0	−5	12	7	5	4	9	5	5	4	15	7	5	3	7	3
18	30	0	−12	0	−8	0	−6	12	8	6	5	9	6	5	4	18	8	5	3	8	4
30	50	0	−12	0	−10	0	−8	12	10	8	6	9	8	5	5	20	10	6	4	8	4
50	80	0	−15	0	−12	0	−9	15	12	9	7	11	9	6	5	25	10	7	4	8	5
80	120	0	−20	0	−15	0	−10	20	15	11	8	15	11	8	5	30	13	8	5	9	5
120	180	0	−25	0	−18	0	−13	25	18	14	10	19	14	9	7	35	18	11	6	10	6
180	250	0	−30	0	−22	0	−15	30	22	17	11	23	16	11	8	50	20	13	8	11	7
250	315	0	−35	—	—	—	—	35	—	—	—	26	—	—	—	60	—	—	—	—	—
315	400	0	−40	—	—	—	—	40	—	—	—	30	—	—	—	70	—	—	—	—	—
400	500	0	−45	—	—	—	—	45	—	—	—	34	—	—	—	80	—	—	—	—	—
500	630	0	−50	—	—	—	—	50	—	—	—	38	—	—	—	90	—	—	—	—	—
630	800	0	−75	—	—	—	—	75	—	—	—	56	—	—	—	105	—	—	—	—	—
800	1000	0	−100	—	—	—	—	100	—	—	—	75	—	—	—	120	—	—	—	—	—

1) The dimensional difference Δ_{ds} of bore diameter to be applied for class 4 is the same as the tolerance of dimensional difference Δ_{dmp} of average bore diameter.

Table 4.4 (2) Outer rings

Nominal bore diameter D (mm)		Δ_{Dmp}						V_{Dp}				V_{Dmp}				K_{ea}				S_D	
		class 0,6X		class 5,6		class 4[2)]		class 0, 6X	class 6	class 5	class 4	class 0, 6X	class 6	class 5	class 4	class 0, 6X	class 6	class 5	class 4	class 5	class 4
over	incl.	high	low	high	low	high	low	max				max				max				max	
18	30	0	−12	0	−8	0	−6	12	8	6	5	9	6	5	4	18	9	6	4	8	4
30	50	0	−14	0	−9	0	−7	14	9	7	5	11	7	5	5	20	10	7	5	8	4
50	80	0	−16	0	−11	0	−9	16	11	8	7	12	8	6	5	25	13	8	5	8	4
80	120	0	−18	0	−13	0	−10	18	13	10	8	14	10	7	5	35	18	10	6	9	5
120	150	0	−20	0	−15	0	−11	20	15	11	8	15	11	8	6	40	20	11	7	10	5
150	180	0	−25	0	−18	0	−13	25	18	14	10	19	14	9	7	45	23	13	8	10	5
180	250	0	−30	0	−20	0	−15	30	20	15	11	23	15	10	8	50	25	15	10	11	7
250	315	0	−35	0	−25	0	−18	35	25	19	14	26	19	13	9	60	30	18	11	13	8
315	400	0	−40	0	−28	0	−20	40	28	22	15	30	21	14	10	70	35	20	13	13	10
400	500	0	−45	—	—	—	—	45	—	—	—	34	—	—	—	80	—	—	—	—	—
500	630	0	−50	—	—	—	—	50	—	—	—	38	—	—	—	100	—	—	—	—	—
630	800	0	−75	—	—	—	—	75	—	—	—	56	—	—	—	120	—	—	—	—	—
800	1000	0	−100	—	—	—	—	100	—	—	—	75	—	—	—	140	—	—	—	—	—
1000	1250	0	−125	—	—	—	—	125	—	—	—	84	—	—	—	165	—	—	—	—	—
1250	1600	0	−160	—	—	—	—	160	—	—	—	120	—	—	—	190	—	—	—	—	—

2) The dimensional difference Δ_{Ds} of outside diameter to be applied for class 4 is the same as the tolerance of dimensional difference Δ_{Dmp} of average outside diameter.

Unit μm

S_{ia}	Δ_{Bs}						Δ_{Ts}						$\Delta_{B1s}, \Delta_{C1s}$		$\Delta_{B2s}, \Delta_{C2s}$	
class 4	class 0,6		class 6X		class 4, 5		class 0,6		class 6X		class 4, 5		class 4, 5		class 4, 5	
max	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low
3	0	–120	0	–50	0	–200	+200	0	+100	0	+200	–200	—	—	—	—
4	0	–120	0	–50	0	–200	+200	0	+100	0	+200	–200	—	—	—	—
4	0	–120	0	–50	0	–240	+200	0	+100	0	+200	–200	+240	–240	—	—
4	0	–150	0	–50	0	–300	+200	0	+100	0	+200	–200	+300	–300	—	—
5	0	–200	0	–50	0	–400	+200	–200	+100	0	+200	–200	+400	–400	+500	–500
7	0	–250	0	–50	0	–500	+350	–250	+150	0	+350	–250	+500	–500	+600	–600
8	0	–300	0	–50	0	–600	+350	–250	+150	0	+350	–250	+600	–600	+750	–750
—	0	–350	0	–50	—	—	+350	–250	-200	0	—	—	+700	–700	+900	–900
—	0	–400	0	–80	—	—	+400	–400	+200	0	—	—	+800	–800	+1000	–1000
—	0	–450	—	—	—	—	—	—	—	—	—	—	+900	–900	+1200	–1200
—	0	–500	—	—	—	—	—	—	—	—	—	—	+1000	–1000	+1200	–1200
—	0	–750	—	—	—	—	—	—	—	—	—	—	+1500	–1500	+1500	–1500
—	0	–1000	—	—	—	—	—	—	—	—	—	—	+1500	–1500	+1500	–1500

S_{ea}	Δ_{Cs}			
class 4	class 0, 6, 5, 4		class 6X	
max	high	low	high	low
5	Identical to Δ_{Bs} of inner ring of same bearing		0	–100
5			0	–100
5			0	–100
6			0	–100
7			0	–100
8			0	–100
10			0	–100
10			0	–100
13			0	–100
—			0	–100
—			0	–100
—			—	—
—			—	—
—			—	—
—			—	—

Table 4.4 (3) Effective width of outer and inner rings with roller

Unit μm

Nominal bore diameter d (mm)		Δ_{T1s}				Δ_{T2s}			
		class 0		class 6X		class 0		class 6X	
over	incl.	high	low	high	low	high	low	high	low
10	18	+100	0	+50	0	+100	0	+50	0
18	30	+100	0	+50	0	+100	0	+50	0
30	50	+100	0	+50	0	+100	0	+50	0
50	80	+100	0	+50	0	+100	0	+50	0
80	120	+100	-100	+50	0	+100	-100	+50	0
120	180	+150	-150	+50	0	+200	-100	+100	0
180	250	+150	-150	+50	0	+200	-100	+100	0
250	315	+150	-150	+100	0	+200	-100	+100	0
315	400	+200	-200	+100	0	+200	-200	+100	0

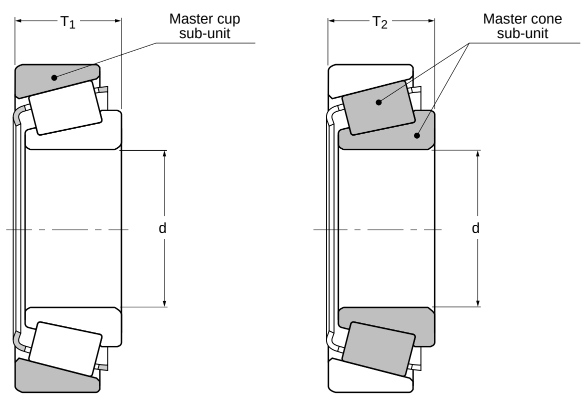


Table 4.5 Tolerance for tapered roller bearings of inch system

Table 4.5 (1) Inner rings

Unit μm
0.0001 inch

Nominal bore diameter d (mm, inch)		Δ_{ds}									
		Class 4		Class 2		Class 3		Class 0		Class 00	
over	incl.	high	low	high	low	high	low	high	low	high	low
—	76.2	+13	0	+13	0	+13	0	+13	0	+8	0
—	3	+5	0	+5	0	+5	0	+5	0	+3	0
76.2	304.8	+25	0	+25	0	+13	0	+13	0	+8	0
3	12	+10	0	+10	0	+5	0	+5	0	+3	0

Table 4.5 (2) Outer rings

Unit μm
0.0001 inch

Nominal outside diameter D (mm, inch)		Δ_{Ds}									
		Class 4		Class 2		Class 3		Class 0		Class 00	
over	incl.	over	incl.	over	incl.	over	incl.	over	incl.	over	incl.
—	304.8	+25	0	+25	0	+13	0	+13	0	+8	0
—	12	+10	0	+10	0	+5	0	+5	0	+3	0
304.8	609.6	+51	0	+51	0	+25	0	—	—	—	—
12	24	+20	0	+20	0	+10	0	—	—	—	—

Table 4.5 (3) Effective width of inner rings with roller and outer rings

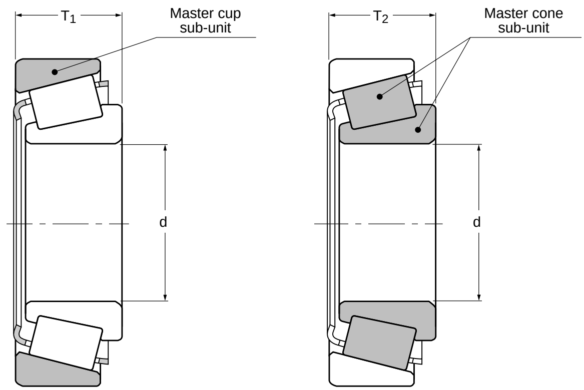
Unit μm
0.0001 inch

Nominal bore diameter d (mm, inch)		Nominal outside diameter D (mm, inch)		Δ_{Ts}								Δ_{B2s}, Δ_{C2s}	
				Class 4		Class 2		Class 3		Class 0, 00		Class 4, 2, 3, 0	
over	incl.	over	incl.	high	low	high	low	high	low	high	low	high	low
—	101.6			+203	0	+203	0	+203	–203	+203	–203	+1520	–1520
—	4			+80	0	+80	0	+80	–80	+80	–80	+599	–599
101.6	304.8			+356	–254	+203	0	+203	–203	+203	–203	+1520	–1520
4	12			+140	–100	+80	0	+80	–80	+80	–80	+599	–599
304.8	609.6	—	508.0	+381	–381	+381	–381	+203	–203	—	—	+1520	–1520
12	24	—	20	+150	–150	+150	–150	+80	–80	—	—	+599	–599
304.8	609.6	508.0	—	+381	–381	+381	–381	+381	–381	—	—	+1520	–1520
12	24	20	—	+150	–150	+150	–150	+150	–150	—	—	+599	–599

Table 4.5 (4) Radial deflection of inner and outer rings

Unit μm
0.0001 inch

Nominal outside diameter D (mm, inch)		K_{ia}, K_{ea}				
over	incl.	Class 4	Class 2	Class 3	Class 0	Class 00
—	304.8	51	38	8	4	2
—	12	20	15	3	1.5	0.75
304.8	609.6	51	38	18	—	—
12	24	20	15	7	—	—



Unit μm
0.0001 inch

Δ_{T1s}						Δ_{T2s}					
Class 4		Class 2		Class 3		Class 4		Class 2		Class 3	
high	low	high	low	high	low	high	low	high	low	high	low
+102	0	+102	0	+102	–102	+102	0	+102	0	+102	–102
+40	0	+40	0	+40	–40	+40	0	+40	0	+40	–40
+152	–152	+102	0	+102	–102	+203	–102	+102	0	+102	–102
+60	–60	+40	0	+40	–40	+80	–40	+40	0	+40	–40
—	—	+178	–178[1)]	+102	–102[1)]	—	—	+203	–203[1)]	+102	–102[1)]
—	—	+70	–70	+40	–40	—	—	+80	–80	+40	–40
—	—	—	—	—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—	—	—	—	—

1) To be applied for nominal bore diameters of 406.400 mm 16 inch or less.

Table 4.6 Tolerance of tapered roller bearings of J series (Metric system)

Table 4.6 (1) Inner rings

Nominal bore diameter d (mm)		Δ_{dmp}								V_{dp}				V_{dmp}			
		Class K		Class N		Class C		Class B		Class K	Class N	Class C	Class B	Class K	Class N	Class C	Class B
over	incl.	high	low	high	low	high	low	high	low	max				max			
10	18	0	–12	0	–12	0	–7	0	–5	12	12	4	3	9	9	5	4
18	30	0	–12	0	–12	0	–8	0	–6	12	12	4	3	9	9	5	4
30	50	0	–12	0	12	0	–10	0	–8	12	12	4	3	9	9	5	5
50	80	0	–15	0	–15	0	–12	0	–9	15	15	5	3	11	11	5	5
80	120	0	–20	0	–20	0	–15	0	–10	20	20	5	3	15	15	5	5
120	180	0	–25	0	–25	0	–18	0	–13	25	25	5	3	19	19	5	7
180	250	0	–30	0	–30	0	–22	0	–15	30	30	6	4	23	23	5	8

Note: Please consult NTN for bearings of Class A

Table 4.6 (2) Outer rings

Nominal outside diameter D (mm)		Δ_{Dmp}								V_{Dp}				V_{Dmp}			
		Class K		Class N		Class C		Class B		Class K	Class N	Class C	Class B	Class K	Class N	Class C	Class B
over	incl.	high	low	high	low	high	low	high	low	max				max			
18	30	0	–12	0	–12	0	–8	0	–6	12	12	4	3	9	9	5	4
30	50	0	–14	0	–14	0	–9	0	–7	14	14	4	3	11	11	5	5
50	80	0	–16	0	–16	0	–11	0	–9	16	16	4	3	12	12	6	5
80	120	0	–18	0	–18	0	–13	0	–10	18	18	5	3	14	14	7	5
120	150	0	–20	0	–20	0	–15	0	–11	20	20	5	3	15	15	8	6
150	180	0	–25	0	–25	0	–18	0	–13	25	25	5	3	19	19	9	7
180	250	0	–30	0	–30	0	–20	0	–15	30	30	6	4	23	23	10	8
250	315	0	–35	0	–35	0	–25	0	–18	35	35	8	5	26	26	13	9
315	400	0	–40	0	–40	0	–28	0	–20	40	40	10	5	30	30	14	10

Note: Please consult NTN for bearings of Class A

Table 4.6 (3) Effective width of inner and outer rings

Nominal bore diameter d (mm)		Δ_{T1s}								Δ_{T2s}							
		Class K		Class N		Class C		Class B		Class K		Class N		Class C		Class B	
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low
10	80	+100	0	+50	0	+100	–100	✻	✻	+100	0	+50	0	+100	–100	✻	✻
80	120	+100	–100	+50	0	+100	–100	✻	✻	+100	–100	+50	0	+100	–100	✻	✻
120	180	+150	–150	+50	0	+100	–100	✻	✻	+200	–100	+100	0	+100	–150	✻	✻
180	250	+150	–150	+50	0	+100	–150	✻	✻	+200	–100	+100	0	+100	–150	✻	✻

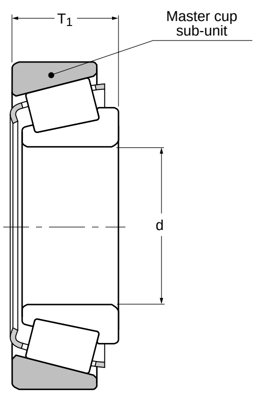
Note: 1) "✻" mark are to be manufactured only for combined bearings.
2) Please consult NTN for the bearings of Class A.

Unit μm

K_{ia}				K_{ia}	Δ_{T_s}							
Class K	Class N	Class C	Class B	Class B	Class K		Class N		Class C		Class B	
max				max	high	low	high	low	high	low	high	low
15	15	5	3	3	+200	0	+100	0	+200	–200	+200	–200
18	18	5	3	4	+200	0	+100	0	+200	–200	+200	–200
20	20	6	4	4	+200	0	+100	0	+200	–200	+200	–200
25	25	6	4	4	+200	0	+100	0	+200	–200	+200	–200
30	30	6	5	5	+200	–200	+100	0	+200	–200	+200	–200
35	35	8	6	7	+350	–250	+150	0	+200	–250	+200	–250
50	50	10	8	8	+350	–250	+150	0	+200	–300	+200	–300

Unit μm

K_{ea}				S_{ea}
Class K	Class N	Class C	Class B	Class B
max				max
18	18	5	3	3
20	20	6	3	3
25	25	6	4	4
35	35	6	4	4
40	40	7	4	4
45	45	8	4	5
50	50	10	5	6
60	60	11	5	6
70	70	13	5	6



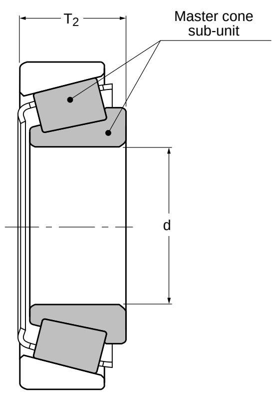


Table 4.7 Tolerance of thrust ball bearings

Table 4.7 (1) Inner rings

Unit μm

Nominal bore diameter d or d_2 (mm)		Δ_{dmp}, Δ_{d2mp}				V_{dp}, V_{d2p}		S_i 1)			
		Class 0, 6, 5		Class 4		Class 0, 6, 5	Class 4	Class 0	Class 6	Class 5	Class 4
over	incl.	high	low	high	low	max		max			
—	18	0	–8	0	–7	6	5	10	5	3	2
18	30	0	–10	0	–8	8	6	10	5	3	2
30	50	0	–12	0	–10	9	8	10	6	3	2
50	80	0	–15	0	–12	11	9	10	7	4	3
80	120	0	–20	0	–15	15	11	15	8	4	3
120	180	0	–25	0	–18	19	14	15	9	5	4
180	250	0	–30	0	–22	23	17	20	10	5	4
250	315	0	–35	0	–25	26	19	25	13	7	5
315	400	0	–40	0	–30	30	23	30	15	7	5
400	500	0	–45	0	–35	34	26	30	18	9	6
500	630	0	–50	0	–40	38	30	35	21	11	7

1) The division of double direction type bearings will be in accordance with division "d" of single direction type bearings corresponding to the identical nominal outer diameter of bearings, not according to division "d_2".

Table 4.7 (2) Outer rings

Unit μm

Nominal outside diameter D (mm)		Δ_{Dmp}				V_{Dp}		S_e 2)
		Class 0, 6, 5		Class 4		Class 0, 6, 5	Class 4	Class 0, Class 6, Class 5, Class 4
over	incl.	high	low	high	low	max		max
10	18	0	–11	0	–7	8	5	According to the tolerance of S_1 against "d" or "d_2" of the same bearings
18	30	0	–13	0	–8	10	6	
30	50	0	–16	0	–9	12	7	
50	80	0	–19	0	–11	14	8	
80	120	0	–22	0	–13	17	10	
120	180	0	–25	0	–15	19	11	
180	250	0	–30	0	–20	23	15	
250	315	0	–35	0	–25	26	19	
315	400	0	–40	0	–28	30	21	
400	500	0	–45	0	–33	34	25	
500	630	0	–50	0	–38	38	29	
630	800	0	–75	0	–45	55	34	

2) To be applied only for bearings with flat seats.

Table 4.7 (3) Height of bearings center washer

Unit μm

Nominal bore diameter d (mm) over	incl.	Single direction type Δ_{Ts} high	low	Double direction type Δ_{T1s} 3) high	low	Δ_{T2s} 3) high	low	Δ_{Cs} 3) high	low
—	30	0	–75	+50	–150	0	–75	0	–50
30	50	0	–100	+75	–200	0	–100	0	–75
50	80	0	–125	+100	–250	0	–125	0	–100
80	120	0	–150	+125	–300	0	–150	0	–125
120	180	0	–175	+150	–350	0	–175	0	–150
180	250	0	–200	+175	–400	0	–200	0	–175
250	315	0	–225	+200	–450	0	–225	0	–200
315	400	0	–300	+250	–600	0	–300	0	–250
400	500	0	–350	—	—	—	—	—	—
500	630	0	–400	—	—	—	—	—	—

3) To be in accordance with division "*d*" of single direction type bearings corresponding to the identical outer diameter of bearings in the same bearing series.

Note: The specifications will be applied for the bearings with flat seats of Class 0.

Table 4.8 Tolerance of spherical thrust roller bearings

Table 4.8 (1) Inner rings

Unit μm

Nominal bore diameter d (mm) over	incl.	Δ_{dmp} high	low	V_{dp} max	S_d max	Δ_{Ts} high	low
50	80	0	–15	11	25	+150	–150
80	120	0	–20	15	25	+200	–200
120	180	0	–25	19	30	+250	–250
180	250	0	–30	23	30	+300	–300
250	315	0	–35	26	35	+350	–350
315	400	0	–40	30	40	+400	–400
400	500	0	–45	34	45	+450	–450

Table 4.8 (2) Outer rings

Unit μm

Nominal bore diameter D (mm) over	incl.	Δ_{Dmp} high	low
120	180	0	–25
180	250	0	–30
250	315	0	–35
315	400	0	–40
400	500	0	–45
500	630	0	–50
630	800	0	–75
800	1000	0	–100

Table 4.9 Tolerance of double direction type angular contact thrust ball bearings

Table 4.9 (1) Inner rings and bearing height

Unit μm

Nominal bore diameter d (mm)		$\Delta_{dmp}, \Delta_{ds}$				S_d		S_{ia}		V_{Bs}		Δ_{Ts}	
		Class 5		Class 4		Class 5	Class 4	Class 5	Class 4	Class 5	Class 4	Class 5, Class 4	
over	incl.	high	low	high	low	max		max		max		high	low
18	30	0	–6	0	–5	8	4	5	3	5	2.5	0	–300
30	50	0	–8	0	–6	8	4	5	3	5	3	0	–400
50	80	0	–9	0	–7	8	5	6	5	6	4	0	–500
80	120	0	–10	0	–8	9	5	6	5	7	4	0	–600
120	180	0	–13	0	–10	10	6	8	6	8	5	0	–700
180	250	0	–15	0	–12	11	7	8	6	10	6	0	–800
250	315	0	–18	0	–15	13	8	10	8	13	7	0	–900
315	400	0	–23	0	–18	15	9	13	10	15	9	0	–1000

Table 4.9 (2) Outer rings

Unit μm

Nominal outside diameter D (mm)		$\Delta_{Dmp}, \Delta_{Ds}$		S_D		S_{ea}	V_{Cs}	
		Class 5	Class 4	Class 5	Class 4	Class 5 Class 4	Class 5	Class 4
over	incl.	high	low	max		max	max	
30	50	–30	–40	8	4	According to tolerance of S_{ia} against "*d*" of the same bearings	5	2.5
50	80	–40	–50	8	4		6	3
80	120	–50	–60	9	5		8	4
120	150	–60	–75	10	5		8	5
150	180	–60	–75	10	5		8	5
180	250	–75	–90	11	7		10	7
250	315	–90	–105	13	8		11	7
315	400	–110	–125	13	10		13	8
40	500	–120	–140	15	13		15	10

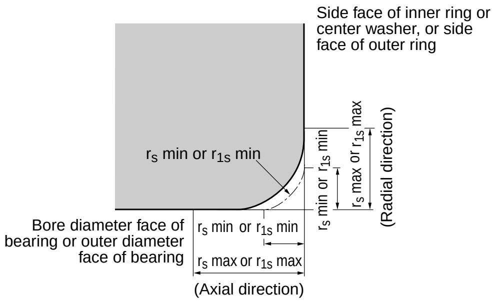


Table 4.10 Allowable critical-value of bearing chamfer

Table 4.10 (1) Radial bearings
(Except tapered roller bearings)

Unit mm

$r_{s\ min}$ [1]	Nominal bore diameter d		$r_{s\ max}$	
	over	incl.	Radial direction	Axial direction
0.05	—	—	0.1	0.2
0.08	—	—	0.16	0.3
0.1	—	—	0.2	0.4
0.15	—	—	0.3	0.6
0.2	—	—	0.5	0.8
0.3	—	40	0.6	1
	40	—	0.8	1
0.6	—	40	1	2
	40	—	1.3	2
1	—	50	1.5	3
	50	—	1.9	3
1.1	—	120	2	3.5
	120	—	2.5	4
1.5	—	120	2.3	4
	120	—	3	5
	—	80	3	4.5
2	80	220	3.5	5
	220	—	3.8	6
2.1	—	280	4	6.5
	280	—	4.5	7
	—	100	3.8	6
2.5	100	280	4.5	6
	280	—	5	7
3	—	280	5	8
	280	—	5.5	8
4	—	—	6.5	9
5	—	—	8	10
6	—	—	10	13
7.5	—	—	12.5	17
9.5	—	—	15	19
12	—	—	18	24
15	—	—	21	30
19	—	—	25	38

1) These are the allowable minimum dimensions of the chamfer dimension "*r*" and are described in the dimensional table.

Table 4.10 (2) Tapered roller bearings of metric system

Unit mm

$r_{s\,min}$ [2)] or $r_{1s\,min}$	Nominal bore diameter of bearing "*d*" or nominal outside diameter "*D*"		$r_{s\,max}$ or $r_{1s\,max}$	
	over	incl.	Radial direction	Axial direction
0.3	—	40	0.7	1.4
	40	—	0.9	1.6
0.6	—	40	1.1	1.7
	40	—	1.3	2
1	—	50	1.6	2.5
	50	—	1.9	3
1.5	—	120	2.8	4
	120	250	2.8	3.5
	250	—	3.5	4
2	—	120	2.8	4
	120	250	3.5	4.5
	250	—	4	5
2.5	—	120	3.5	5
	120	250	4	5.5
	250	—	4.5	6
3	—	120	4	5.5
	120	250	4.5	6.5
	250	400	5	7
	400	—	5.5	7.5
4	—	120	5	7
	120	250	5.5	7.5
	250	400	6	8
	400	—	6.5	8.5
5	—	180	6.5	8
	180	—	7.5	9
6	—	180	7.5	10
	180	—	9	11

2) These are the allowable minimum dimensions of the chamfer dimension "*r*" or "r_1" and are described in the dimensional table.

3) Inner rings shall be in accordance with the division of "*d*" and outer rings with that of "*D*".

Note: This standard will be applied to the bearings whose dimensional series (refer to the dimensional table) specified in the standard of ISO 355 or JIS B 1512. Further, please consult NTN for bearings other than those represented here.

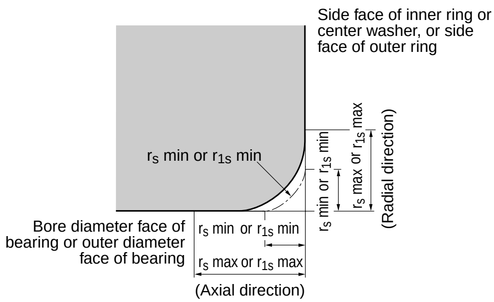
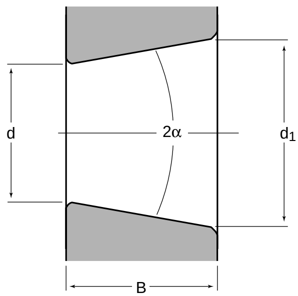


Table 4.10 (3) Thrust bearings

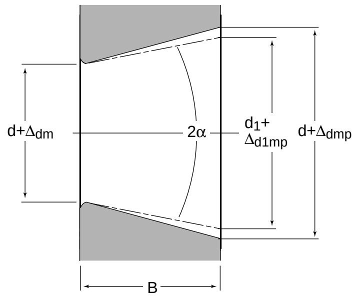
Unit mm

$r_{s\,min}$ or $r_{1s\,min}$ [4)]	$r_{s\,max}$ or $r_{1s\,max}$ Radial and axial direction
0.05	0.1
0.08	0.16
0.1	0.2
0.15	0.3
0.2	0.5
0.3	0.8
0.6	1.5
1	2.2
1.1	2.7
1.5	3.5
2	4
2.1	4.5
3	5.5
4	6.5
5	8
6	10
7.5	12.5
9.5	15
12	18
15	21
19	25

4) These are the allowable minimum dimensions of the chamfer dimension "*r*" or "r_1" and are described in the dimensional table.



Theoretical tapered hole



Tapered bore with dimensional within a flat plane tolerance

Table 4.11 Tolerance and allowable values (Class 0) of tapered bore of radial bearings

Unit μm

Nominal bore diameter d (mm) over	incl.	Δ_{dmp} high	low	$\Delta_{d1mp}-\Delta_{dmp}$ high	low	V_{dp} 1) max
—	10	+15	0	+15	0	10
10	18	+18	0	+18	0	10
18	30	+21	0	+21	0	13
30	50	+25	0	+25	0	15
50	80	+30	0	+30	0	19
80	120	+35	0	+35	0	25
120	180	+40	0	+40	0	31
180	250	+46	0	+46	0	38
250	315	+52	0	+52	0	44
315	400	+57	0	+57	0	50
400	500	+63	0	+63	0	56

1) To be applied for all radial flat surfaces of tapered bore.

Note: 1. To be applied for tapered bores of 1/12.

2. Symbols of quantity or values

d_1: Basic diameter at the theoretically large end of the tapered bore

$$d_1 = d + \frac{1}{12}B$$

Δdmp: Dimensional difference of the average bore diameter within the flat surface at the theoretical small-end of the tapered bore.

Δd1mp: Dimensional difference of the average bore diameter within the flat surface at the theoretical large-end of the tapered bore.

$V d$p: Inequality of the bore diameter within the flat surface

B: Nominal width of inner ring

α: Half of the nominal tapered angle of the tapered bore

α = 2°23'9.4"

= 2.38594°

= 0.041643 RAD

5. Load Rating and Life

5.1 Bearing life

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

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

5.2 Basic rated life and basic dynamic load rating

A group of seemingly identical bearings when subjected to indentical load and operating conditions will exhibit a wide diversity in their durability.

This “life” disparity can be accounted for by the difference in the fatigue of the bearing material itself. This disparity is considered statistically when calculating bearing life, and the basic rated life is defined as follows.

The basic rated life is based on a 90% statistical model which is expressed as the total number of revolutions 90% of the bearings in an identical group of bearings subjected to identical operating conditions will attain or surpass before flaking due to material fatigue occurs. For bearings operating at fixed constant speeds, the basic rated life (90% reliability) is expressed in the total number of hours of operation.

The basic dynamic load rating is an expression of the load capacity of a bearing based on a constant load which the bearing can sustain for one million revolutions (the basic life rating). For radial bearings this rating applies to pure radial loads, and for thrust bearings it refers to pure axial loads. The basic dynamic load ratings given in the bearing tables of this catalog are for bearings constructed of NTN standard bearing materials, using standard manufacturing techniques. Please consult NTN for basic load ratings of bearings constructed of special materials or using special manufacturing techniques.

The relationship between the basic rated life, the basic dynamic load rating and the bearing load is given in formula (5.1).

where, $$L_{10} = \left(\frac{C}{P}\right)^p \quad \cdots\cdots(5.1)$$

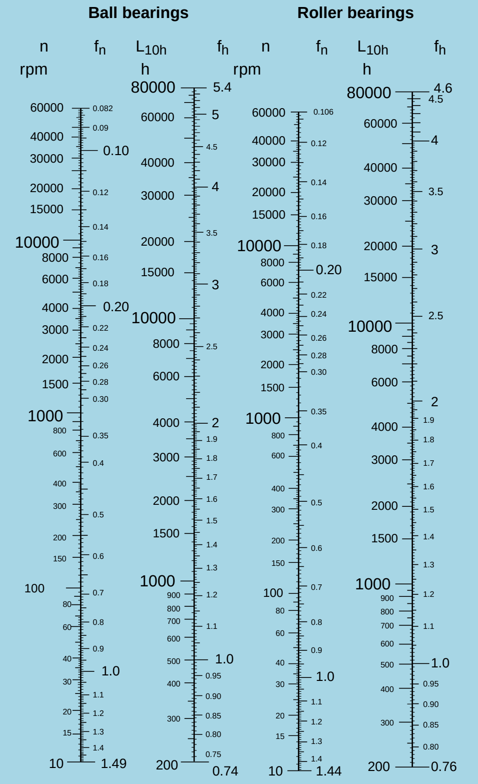


Fig. 5.1 Bearing life rating scale

p = 3 ·························· For ball bearings

p = 10/3 ·························· For roller bearings

L_{10} : Basic rated life 10 revolutions

C : Basic dynamic rated load N
(*C*r : radial bearings, *C*a : thrust bearings)

P : Equivalent dynamic load N
(*P*r : radial bearings, *P*a : thrust bearings)

The basic rated life can also be expressed in terms of hours of operation (revolution), and is calculated as shown in formula (5.2).

where,

$$L_{10\mathrm{h}} = 500\, f_{\mathrm{h}}^{\ p} \cdots\cdots\cdots (5.2)$$

$$f_{\mathrm{h}} = f_{\mathrm{n}} \frac{C}{P} \cdots\cdots\cdots (5.3)$$

$$f_{\mathrm{n}} = \left(\frac{33.3}{n}\right)^{1/p} \cdots\cdots\cdots (5.4)$$

L : Basic rated life h

*f*n : Life factor

*f*n : Speed

n : Rotational speed, r/min

Formula (5.2) can also be expressed as shown in formula (5.5).

$$L_{10\mathrm{h}} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^{p} \cdots\cdots\cdots (5.5)$$

The relation between Rotational speed n and speed factor *f*n as well as the relation between the basic rated life L10h and the life factor *f*h is shown in Fig. 5.1.

When several bearings are incorporated in machines or equipment as complete units, all the bearings in the unit are

Table 5.1 Machine application and requisite life

Service classification	Life factor f_h and machine application				
	~2.0	2.0~3.0	3.0~4.0	4.0~5.0	5.0~
Machines used for short periods or used only occasionally	Electric hand tools Household appliances	Farm machinery Office equipment			
Short period or intermittent use, but with high reliability requirements	Medical appliances Measuring instruments	Home air-conditioning motor Construction equipment Elevators Cranes	Crane (sheaves)		
Machines not in constant use, but used for long periods	Automobiles Two-wheeled vehicles	Small motors Buses/trucks Drivers Woodworking machines	Machine spindles Industrial motors Crushers Vibrating screens	Main gear drives Rubber/plastic Calender rolls Printing machines	
Machines in constant use over 8 hours a day		Rolling mills Escalators Conveyors Centrifuges	Railway vehicle axles Air conditioners Large motors Compressor pumps	Locomotive axles Traction motors Mine hoists Pressed flywheels	Papermaking machines Propulsion equipment for marine vessels
24 hour continuous operation, non-interruptable					Water supply equipment Mine drain pumps/ventilators Power generating equipment

considered as a whole when computing bearing life (see formula 5.6). The total bearing life of the unit is a life rating based on the viable lifetime of the unit before even one of the bearings fails due to rolling contact fatigue.

$$L = \frac{1}{\left(\frac{1}{L_1^{\,e}} + \frac{1}{L_2^{\,e}} + \cdots\cdots + \frac{1}{L_n^{\,e}}\right)^{1/e}} \cdots\cdots\cdots\cdots (5.6)$$

where,

$e = 10/9 \cdots\cdots\cdots\cdots$ For ball bearings

$e = 9/8 \cdots\cdots\cdots\cdots$ For roller bearings

L : Total basic rated life of entire unit h

$L_1, L_2 \cdots L_n$: Basic rated life of individual bearing 1, 2 $\cdots$ n h

When the load conditions vary at regular intervals, the life can be given by formula (5.7).

$$L_m = \left(\Sigma^{\phi_j}\!\big/ L_j\right)^{-1} \cdots\cdots\cdots\cdots\cdots\cdots (5.7)$$

where,

Φ_j : Frequency of individual load conditions

L_j : Life under individual conditions

5.3 Machine applications and requisite life

When selecting a bearing, it is essential that the requisite life of the bearing be established in relation to the operating conditions. The requisite life of the bearing is usually determined by the type of machine the bearing is to be used in, and duration of service and operational reliability requirements. A general guide to these requisite life criteria is shown in Table 5.1. When determining bearing size, the fatigue life of the bearing is an important factor; however, besides bearing life, the strength and rigidity of the shaft and housing must also be taken into consideration.

5.4 Adjusted life rating factor

The basic life rating (90% reliability factor) can be calculated through the formulas mentioned earlier in Section 5.2. However, in some applications a bearing life factor of over 90% reliability may be required. To meet these requirements, bearing life can be lengthened by the use of specially improved bearing materials or special construction techniques. Moreover, according to elastohydrodynamic lubrication theory, it is clear that the bearing operating conditions (lubrication, temperature, speed, etc.) all exert an effect on bearing life. All these adjustment factors are taken into consideration when calculating bearing life, and using the life adjustment factor as prescribed in ISO 281, the adjusted bearing life can be arrived at.

$$L_{na} = a_1 \quad a_2 \quad a_3\left(\frac{C}{P}\right)^p \cdots\cdots\cdots\cdots\cdots\cdots (5.8)$$

where,

L_{na} = Adjusted life rating in millions of revolutions (10^6) (adjusted for reliability, material and operating conditions)

a_1 = Reliability adjustment factor

a_2 = Material adjustment factor

a_3 = Operating condition adjustment factor

5.4.1. Life adjustment factor for reliability a_1

The values for the reliability adjustment factor a_1 (for a reliability factor higher than 90%) can be found in Table 5.2.

Table 5.2 Reliability adjustment factor values a_1

Reliability %	L_n	Reliabiltiy factor a1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

5.4.2. Life adjustment factor for material a_2

The values for the basic dynamic load ratings given in the bearing dimension tables are for bearings constructed from NTN's continued efforts at improving the quality and life of its bearings.

Accordingly, a_2=1 is used for the life adjustment factor in formula (5.8). For bearings constructed of specially improved materials or with special manufacturing methods, the life adjustment factor a_2 in formula (5.8) can have a value greater than one. Please consult NTN for special bearing materials or special construction requirements.

When high carbon chromium steel bearings, which have undergone only normal heat treatment, are operated for long periods of time at temperatures in excess of 120°C, considerable dimensional deformation may take place. For this reason, there are special high temperature bearings which have been treated for dimensional stability. This special treatment allows the bearing to operate at its maximum operational temperature without the occurrence of dimensional changes. However, these dimensionally stabilized bearings, designated with a "TS", prefix have a reduced hardness with a consequent decrease in bearing life. The adjusted life factor values used in formula (5.8) for such heat-stabilized bearings can be found in Table 5.3.

Table 5.3 Dimension stabilized bearings

Code	Max. operating temperature °C	Adjustment factor *a*
TS2	160	0.87
TS3	200	0.68
TS4	250	0.30

5.4.3. Life adjustment factor a_3 for operating conditions

The operating conditions life adjustment factor a_3 is used to adjust for such conditions as lubrication, operating temperature, and other operation factors which have an effect on bearing life.

Generally speaking, when lubricating conditions are satisfactory, the a_3 factor has a value on one; and when lubricating conditions are exceptionally favorable, and all other operating conditions are normal, a_3 can have a value greater than one.

However, when lubricating conditions are particularly unfavorable and the oil film formation on the contact surfaces of the raceway and rolling elements is insufficient, the value of a_3 becomes less than one. This insufficient oil film formation can be caused, for example, by the lubricating oil viscosity being too low for the operating temperature (below 13 mm^2/s for ball bearings; below 20 mm^2/s for roller bearings); or by exceptionally low rotational speed (n r/min x d_p mm less than 10,000). For bearings used under special operating conditions, please consult NTN.

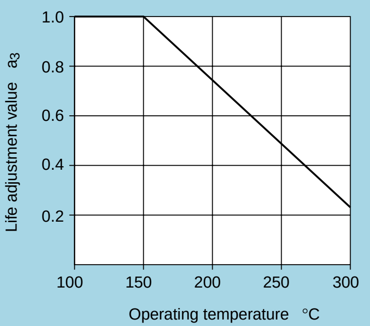


Fig. 5.2 Life adjustment value for operating temperature

As the operating temperature of the bearing increases, the hardness of the bearing material decreases. Thus, the bearing life correspondingly decreases. The operating temperature adjustment values are shown in Fig. 5.2.

5.5 Basic static load rating

When stationary rolling bearings are subjected to static loads, they suffer from partial permanent deformation of the contact surfaces at the contact point between the rolling elements and the raceway. The amount of deformity increases as the load increases, and if this increase in load exceeds certain limits, the subsequent smooth operation of the bearings is impaired.

It has been found through experience that a permanent deformity of 0.0001 times the diameter of the rolling element, occuring at the most heavily stressed contact point between the raceway and the rolling elements, can be tolerated without any impairment in running efficiency.

The basic rated static load refers to a fixed static load limit at which a specified amount of permanent deformation occurs. It applies to pure radial loads for radial bearings and to pure axial loads for contact stress occurring at the rollling element and raceway contact points are given below.

For ball bearings (except self-aligning ball bearings	4200 MP_a
For self-aligning ball bearings	4600MP_a
For roller bearings	4000MP_a

5.6 Allowable static equivalent load

Generally the static equivalent load which can be permitted (See Section 6.4.2. page A-50) is limited by the basic static rated load as stated in Section 5.5. However, depending on requirements regarding friction and smooth operation, these limits may be greater or lesser than the basic static rated load.

In the following formula (5.9) and Table 5.4 the safety factor S_o can be determined considering the maximum static equivalent load.

$$S_o = \frac{C_o}{P_{o\,max}} \cdots\cdots (5.9)$$

where,

S_o : Safety factor

C_o : Basic static rated load N (radial bearings: C_{or}, thrust bearings: C_{oa})

$P_{o\,max}$: Maximum static equivalent load N (radial: $P_{or\,max}$, thrust $P_{oa\,max}$)

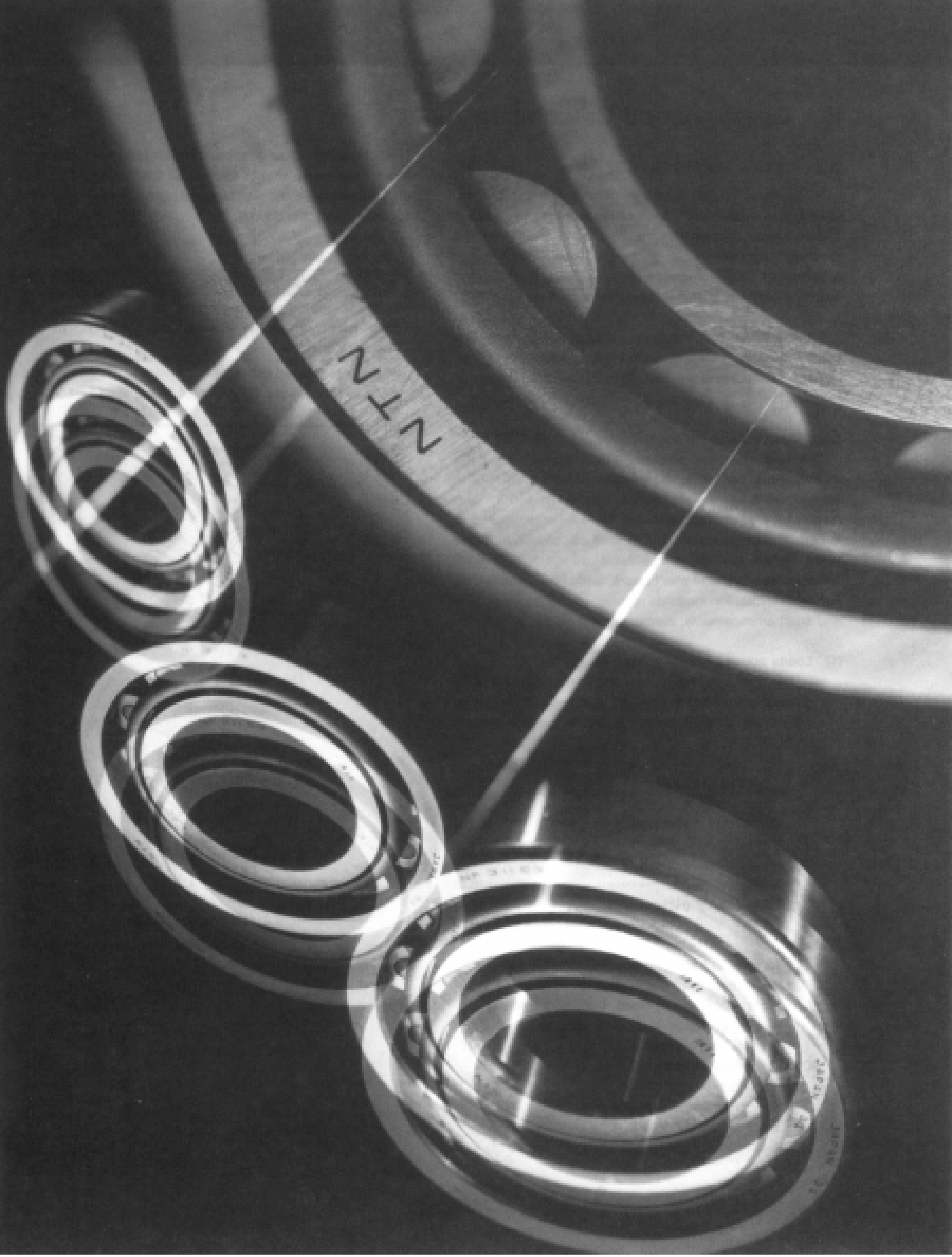
Table 5.4 Minimum safety factor values S_o

Operating conditions	Ball bearings	Roller bearings
High rotational accuracy demand	2	3
Normal rotating accuracy demand (Universal application)	1	1.5
Slight rotational accuracy deterioration permitted (Low speed, heavy loading, etc.)	0.5	1

Note 1. For spherical thrust roller bearings, min. S_o value = 4.

2. For shell needle roller bearings, min. S_o value = 3.

3. When vibration and/or shock loads are present, a load factor based on the shock load needs to be included in the $P_{o\,max}$ value.


NTN

6. Bearing Load Calculation

6.1 Loads acting on shafts

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

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

6.1.1. Gear load

The loads operating on gears can be divided into three main types according to the direction in which the load is applied; i.e. tangential (K_t), radial (K_s), and axial (K_a). The magnitude and direction of these loads differ according to the types of gears involved. The load calculation methods given herein are for two general-use gear and shaft arrangements: parallel shaft gears, and cross shaft gears. For load calculation methods regarding other types of gear and shaft arrangements, please consult NTN.

(1) Loads acting on parallel shaft gears

The forces acting on spur and helical parallel shaft gears are depicted in Figs. 6.1, 6.2, and 6.3. The load magnitude can be found by using formulas (6.1), through (6.4).

$$K_t = \frac{19.1 \times 10^6 \bullet HP}{D_p \bullet n} \cdots\cdots\cdots\cdots (6.1)$$

$$K_s = K_t \bullet \tan\alpha \,(\text{Spur gear}) \cdots\cdots\cdots (6.2a)$$

$$= K_t \bullet \frac{\tan\alpha}{\cos\beta} (\text{Helical gear}) \cdots\cdots (6.2b)$$

$$K_r = \sqrt{K_t^2 + K_s^2} \cdots\cdots\cdots\cdots (6.3)$$

$$K_a = K_t \bullet \tan\beta \,(\text{Helical gear}) \cdots\cdots (6.4)$$

where,

K_t : Tangential gear load (tangential force) N
K_s : Radial gear load (separating force) N
K_r : Right angle shaft load (resultant force of tangential force and separating force) N
K_a : Parallel load on shaft N
HP : Transmission force kW
n : Rotational speed, r/min
D_p : Gear pitch circle diameter mm
α : Gear pressure angle
β : Gear helix angle

Because the actual gear load also contains vibrations and shock loads as well, the theoretical load obtained by the above formula should also be adjusted by the gear factor fz as shown in Table 6.1.

Table 6.1 Gear factor f_z

Gear type	f_z
Precision ground gears (Pitch and tooth profile errors of less than 0.02 mm)	1.05~1.1
Ordinary machined gears (Pitch and tooth profile errors of less than 0.1 mm)	1.1~1.3

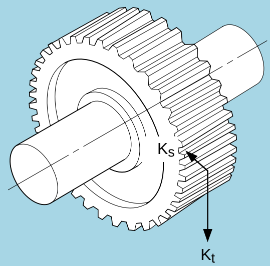


Fig. 6.1 Spur gear loads

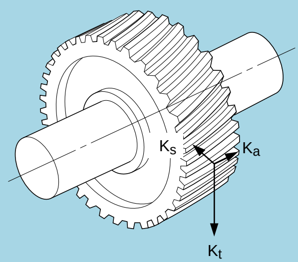


Fig. 6.2 Helical gear loads

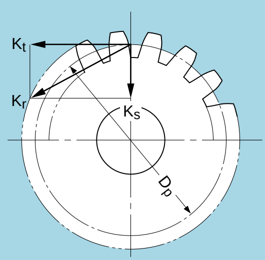


Fig. 6.3 Radial resultant forces

(2) Loads acting on cross shafts

Gear loads acting on straight tooth bevel gears and spiral bevel gears on cross shafts are shown in Figs. 6.4 and 6.5. The calculation methods for these gear loads are shown in Table 6.2. Herein, to calculate gear loads for straight bevel gears, the helix angle β = 0. The symbols and units used in Table 6.2 are as follows:

K_t : Tangential gear load (tangential force) N
K_s : Radial gear load (separating force) N
K_a : Parallel shaft load (axial load) N
HP : Transmission force kW
n : Speed in rpm
D_{pm} : Mean pitch circle diameter mm
α : Gear pressure angle
β : Helix angle
δ : Pitch cone angle

In general, the relationship between the gear load and the pinion gear load, due to the right angle intersection of the two shafts, is as follows:

$$K_{sp} = K_{ag} \quad \text{(6.5)}$$

$$K_{ap} = K_{sg} \quad \text{(6.6)}$$

where,

K_{sp}, K_{sg} : Pinion and gear separating force N
K_{ap}, K_{ag} : Pinion and gear axial load N

For spiral bevel gears, the direction of the load varies depending on the direction of the helix angle, the direction of rotation, and which side is the driving side or the driven side. The directions for the separating force (K_s) and axial load (K_a) shown in Fig. 6.5 are positive directions. The direction of rotation and the helix angle direction are defined as viewed from the large end of the gear. The gear rotation direction in Fig. 6.5 is assumed to be clockwise (right).

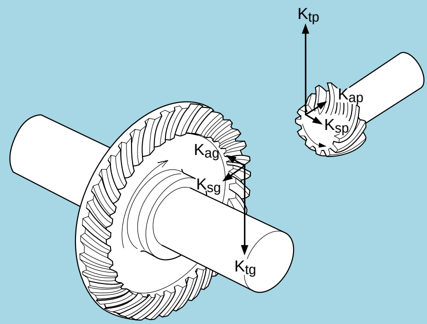


Fig. 6.4 Loads on bevel gears

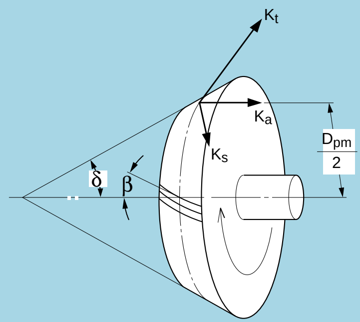


Fig. 6.5 Bevel gear diagram

Table 6.2 Loads acting on bevel gears Unit N

Pinion	Rotation direction	Clockwise	Counter clockwise	Clockwise	Counter clockwise
	Helix direction	Right	Left	Right	Left
Tangential load K_t		$K_t = \dfrac{19.1 \times 10^6 \bullet HP}{D_{pm} \bullet n}$			
Separating force K_s	Driving side	$K_s = K_t\left(\tan\alpha \dfrac{\cos\delta}{\cos\beta} + \tan\beta \sin\delta\right)$		$K_s = K_t\left(\tan\alpha \dfrac{\cos\delta}{\cos\beta} - \tan\beta \sin\delta\right)$	
	Driven side	$K_s = K_t\left(\tan\alpha \dfrac{\cos\delta}{\cos\beta} - \tan\beta \sin\delta\right)$		$K_s = K_t\left(\tan\alpha \dfrac{\cos\delta}{\cos\beta} + \tan\beta \sin\delta\right)$	
Axial load K_a	Driving side	$K_s = K_t\left(\tan\alpha \dfrac{\sin\delta}{\cos\beta} - \tan\beta \cos\delta\right)$		$K_a = K_t\left(\tan\alpha \dfrac{\sin\delta}{\cos\beta} + \tan\beta \cos\delta\right)$	
	Driven side	$K_a = K_t\left(\tan\alpha \dfrac{\sin\delta}{\cos\beta} + \tan\beta \cos\delta\right)$		$K_a = K_t\left(\tan\alpha \dfrac{\sin\delta}{\cos\beta} - \tan\beta \cos\delta\right)$	

6.1.2. Chain/belt shaft load

The tangential loads on sprockets or pulleys when power (load) is transmitted by means of chains or belts can be calculated by formula (6.7).

$$K_t = \frac{19.1\times10^6 \bullet HP}{D_p \bullet n} \cdots\cdots\cdots\cdots (6.7)$$

where,

K_t : Sprocket/pulley tangential load N
HP : Transmitted force kW
D_p : Sprocket/pulley pitch diameter mm

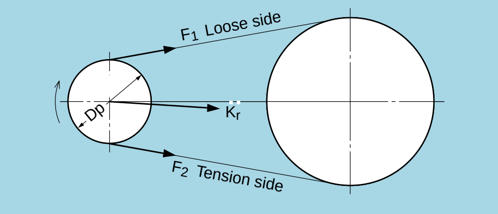


Fig. 6.6 Chain/belt loads

Table 6.3 Chain or belt factor f_b

Chain or belt type	f_b
Chain (single)	1.2~1.5
V-belt	1.5~2.0
Timing belt	1.1~1.3
Flat belt (w/ tension pulley)	2.5~3.0
Flat belt	3.0~4.0

For belt drives, and initial tension is applied to give sufficient constant operating tension on the belt and pulley. Taking this tension into account, the radial loads acting on the pulley are expressed by formula (6.8). For chain drives, the same formula can also be used if vibrations and shock loads are taken into consideration.

$$K_r = f_b \bullet K_t \cdots\cdots\cdots\cdots (6.8)$$

where,

K_r : Sprocket or pulley radial load N
f_b : Chain or belt factor (Table 6.3)

6.1.3 Load factor

There are many instances where the actual operational shaft load is much greater than the theoretically calculated load, due to machine vibration and/or shock. This actual shaft load can be found by using formula (6.9).

where, $$K = f_w \bullet K_c \cdots\cdots\cdots\cdots (6.9)$$

K : Actual shaft load N
K_c : Theoretically calculated value N
f_w : Load factor (Table 6.4)

Table 6.4 Load factor f_w

Amount of shock	f_w	Application
Very little or no shock	1.0 ~ 1.2	Electric machines, machine tools, measuring instruments
Light shock	1.2 ~ 1.5	Railway vehicles, automobiles, rolling mills, metal working machines, paper making machines, rubber mixing machines, printing machines, aircraft, textile machines, electrical units, office machines
Heavy shock	1.5 ~ 3.0	Crushers, agricultural equipment, constuction equipment, cranes

6.2 Bearing load distribution

For shafting, the static tension is considered to be supported by the bearings, and any loads acting on the shafts are distributed to the bearings.

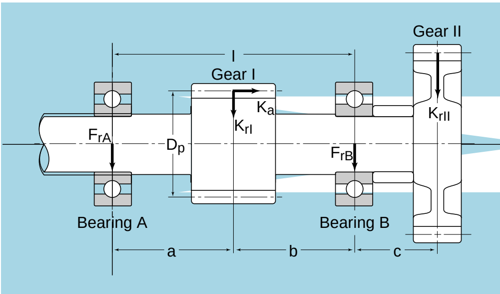
For example, in the gear shaft assembly depicted in Fig. 6.7, the applied bearing loads can be found by using formulas (6.10) and (6.11).

$$F_{rA} = K_{rI}\frac{b}{l} - K_{rII}\frac{c}{l} - K_a\frac{D_p}{2l} \cdots\cdots\cdots (6.10)$$

$$F_{rB} = K_{rI}\frac{a}{l} + K_{rII}\frac{a+b+c}{l} + K_a\frac{D_p}{2l} \cdots\cdots (6.11)$$

where,

F_{rA} : Radial load on bearing A N
F_{rB} : Radial load on bearings B N
K_{rI} : Radial load on gear I N
K_a : Axial load on gear I N
K_{rII} : Radial load on gear II N
D_p : Gear I pitch diameter mm
l : Distance between bearings mm



6.3 Mean load

The load on bearings used in machines under normal circumstances will, in many cases, fluctuate according to a fixed time period or planned operation schedule. The load on bearings operating under such conditions can be converted to a mean load (F_m), this is a load which gives bearings the same life they would have under constant operating conditions.

(1) Fluctuating stepped load

The mean bearing load, F_m, for stepped loads is calculated from formula (6.12). F_1, $F_2 \cdots F_n$ are the loads acting on the bearing; n_1, $n_2 \cdots .n_n$ and t_1, $t_2 \cdots t_n$ are the bearing speeds and operating times respectively.

$$F_m = \left[\frac{\Sigma\left(F_i^p n_i t_i\right)}{\Sigma\left(n_i t_i\right)} \right]^{1/p} \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (6.12)$$

where,

$p = 3$: For ball bearings
$p = 10/3$: For roller bearings

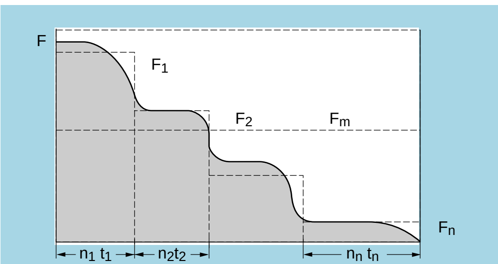


Fig. 6.8 Stepped load

(2) Consecutive series load

Where it is possible to express the function $F(t)$ in terms of load cycle to and time t, the mean load is found by using formula (6.13).

$$F_m = \left[\frac{1}{t_0} \int_0^{t_0} F(t)^p d_t \right]^{1/p} \cdots\cdots\cdots\cdots\cdots\cdots\cdots (6.13)$$

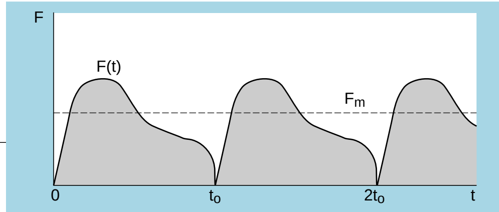


Fig. 6.9 Time function series load

(3) Linear fluctuating load

The mean load, F_m, can be approximated by formula (6.14).

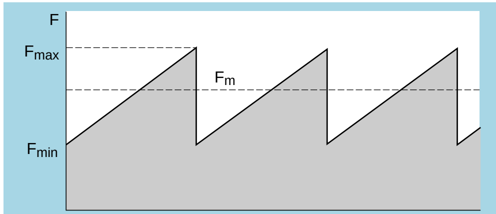


Fig. 6.10 Linear fluctuating load

$$F_m = \frac{F_{min} + 2F_{max}}{3} \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (6.14)$$

(4) Sinusoidal fluctuating load

The mean load, F_m, can be approximated by formula (6.15) and (6.16).

(a) $F_m = 0.75\ F_{max}$ .... (6.15)

(b) $F_m = 0.65\ F_{max}$ .... (6.16)

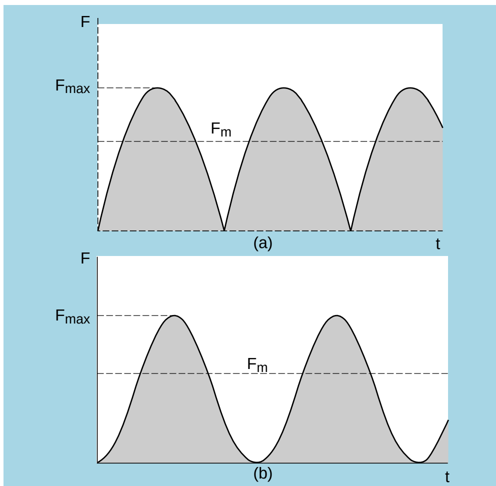


Fig. 6.11 Sinusoidal variable load

6.4 Equivalent load

6.4.1 Dynamic equivalent load

When both dynamic radial loads and dynamic axial loads act on a bearing at the same time, the hypothetical load acting on the center of the bearing which gives the bearings the same life as if they had only a radial load or only an axial load is called the dynamic equivalent load.

For radial bearings, this load is expressed as pure radial load and is called the dynamic equivalent radial load. For thrust bearings, it is expressed as pure axial load, and is called the dynamic equivalent axial load.

(1) Dynamic equivalent radial load

The dynamic equivalent radial load is expressed by formula (6.17).

$$P_r = XF_r + YF_a \cdots\cdots\cdots (6.17)$$

where,

P_r : Dynamic equivalent radial load N
F_r : Actual radial load N
F_a : Actual axial load N
X : Radial load factor
Y : Axial load factor

The values for X and Y are listed in the bearing tables.

(2) Dynamic equivalent axial load

As a rule, standard thrust bearings with a contact angle of 90° cannot carry radial loads. However, self-aligning thrust roller bearings can accept some radial load. The dynamic equivalent axial load for these bearings is given in formula (6.18).

$$P_a = F_a + 1.2F_r \cdots\cdots\cdots (6.18)$$

where,

P_a : Dynamic equivalent axial load N
F_a : Actual axial load N
F_r : Actual radial load N

Provided that $F_r / F_a \leq 0.55$ only.

6.4.2. Static equivalent load

The static equivalent load is a hypothetical load which would cause the same total permanent deformation at the most heavily stressed contact point between the rolling elements and the raceway as under actual load conditions; that is when both static radial loads and static axial loads are simultaneously applied to the bearing.

For radial bearings this hypothetical load refers to pure radial loads, and for thrust bearings it refers to pure centric axial loads. These loads are designated static equivalent radial loads and static equivalent axial loads respectively.

(1) Static equivalent radial load

For radial bearings the static equivalent radial load can be found by using formula (6.19) or (6.20). The greater of the two resultant values is always taken for P_{or}.

$$P_{or} = X_o F_r + Y_o F_a \cdots\cdots\cdots (6.19)$$

$$P_{or} = F_r \cdots\cdots\cdots (6.20)$$

where,

P_{or} : Static equivalent radial load N
X_o : Static radial load factor
Y_o : Static axial load factor
F_r : Actual radial load N
F_a : Actual axial load N

The values for X_o and Y_o are given in the respective bearing tables.

(2) Static equivalent axial load

For spherical thrust roller bearings the static equivalent axial load is expressed by formula (6.21).

$$P_{oa} = F_a + 2.7F_r \cdots\cdots\cdots (6.21)$$

where,

P_{oa} : Static equivalent axial load N
F_a : Actual axial load N
F_r : Actual radial load N

Provided that $F_r / F_a \leq 0.55$ only.

6.4.3 Load calculation for angular ball bearings and tapered roller bearings

For angular ball bearings and tapered roller bearings the pressure cone apex (load center) is located as shown in Fig. 6.12, and their values are listed in the bearing tables.

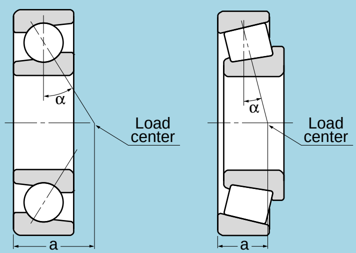


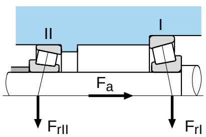
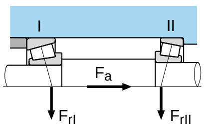
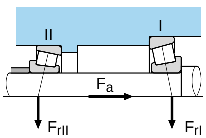
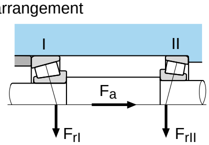
Fig. 6.12 Pressure cone apex

When radial loads act on these types of bearings the component force is induced in the axial direction. For this reason, these bearings are used in pairs (either DB or DF arrangements). For load calculation this component force must be taken into consideration and is expressed by formula (6.22).

$$F_a = \frac{0.5F_r}{Y} \cdots\cdots\cdots\cdots (6.22)$$

The equivalent radial loads for these bearing pairs are given in Table 6.5.

Table 6.5 Bearing arrangement and dynamic equivalent load

Bearing arrangement	Load condition	Axial load	Equivalent radial load
DB arrangement 	$\frac{0.5F_{rII}}{Y_{II}} \leq \frac{0.5F_{rI}}{Y_I} + F_a$	$F_{aI} = \frac{0.5F_{rI}}{Y_I}$ $F_{aII} = \frac{0.5F_{rI}}{Y_I} + F_a$	$P_{rI} = F_{rI}$ $P_{rII} = XF_{rII} + Y_{II}F_{aII}$
DF arrangement 	$\frac{0.5F_{rII}}{Y_{II}} > \frac{0.5F_{rI}}{Y_I} + F_a$	$F_{aI} = \frac{0.5F_{rII}}{Y_{II}} - F_a$ $F_{aII} = \frac{0.5F_{rII}}{Y_{II}}$	$P_{rI} = XF_{rI} + Y_IF_{aI}$ $P_{rII} = F_{rII}$
DB arrangement 	$\frac{0.5F_{rI}}{Y_I} \leq \frac{0.5F_{rII}}{Y_{II}} + F_a$	$F_{aI} = \frac{0.5F_{rII}}{Y_{II}} + F_a$ $F_{aII} = \frac{0.5F_{rII}}{Y_{II}}$	$P_{rI} = XF_{rI} + Y_IF_{aI}$ $P_{rII} = F_{rII}$
DF arrangement 	$\frac{0.5F_{rI}}{Y_I} > \frac{0.5F_{rII}}{Y_{II}} + F_a$	$F_{aI} = \frac{0.5F_{rI}}{Y_I}$ $F_{aII} = \frac{0.5F_{rI}}{Y_I} - F_a$	$P_{rI} = F_{rI}$ $P_{rII} = XF_{rII} + Y_{II}F_{aII}$

Note: 1) The above are valid when the bearing internal clearance and preload are zero.
2) Radial forces in the opposite direction to the arrow in the above illustration are also regarded as positive.

6.5 Bearing rated life and load calculation examples

In the examples given in this section, for the purpose of calculation, all hypothetical load factors as well as all calculated load factors may be presumed to be included in the resultant load values.

(Example 1)

What is the rating life in hours of operation (L_{10h}) for deep groove ball bearing 6208 operating at 650 r/min, with a radial load F_r of 3.2 kN?

For formula (6.17) the dynamic equivalent radial load *Pr* is:

$$P_r = F_r = 3.2 \text{ kN}$$

The basic dynamic rated load for bearing 6208 (from bearing table) is 29.1 kN, and the speed factor (*fn*)for ball bearings at 650 r/min (*n*) from Fig. 5.1 is 0.37. The life factor, *fh*, from formula (5.3) is:

$$f_h = f_n \frac{C_r}{P_r} = 0.37 \times \frac{29.1}{3.2} = 3.36$$

Therefore, with f_h=3.36 from Fig. 5.1 the rated life, L_{10h}, is approximately 19,000 hours.

(Example 2)

What is the life rating L_{10h} for the same bearing and conditions as in Example 1, but with an additional axial load F_a of 1.8 kN?

To find the dynamic equivalent radial load value for P_r, the radial load factor X and axial load factor Y are used. The basic static load rating, C_{or}, for bearing 6208 is 17.8 kN.

$$\frac{F_a}{C_{or}} = \frac{1.8}{17.8} = 0.10$$

Therefore, from the bearing tables e=0.29.
For the operating radial load and axial load:

$$\frac{F_a}{F_r} = \frac{1.8}{3.2} = 0.56 > e = 0.29$$

From the bearing tables X=0.56 and Y=1.48, and from formula (6.17) the equivalent radial load, P_r, is:

$$P_r = XF_r + YF_a = 0.56 \times 3.2 + 1.48 \times 1.8 = 4.46 \text{ kN}$$

From Fig. 5.1 and formula (5.3) the life factor, f_h, is:

$$f_h = f_n \frac{C_r}{P_r} = 0.37 \times \frac{29.1}{4.46} = 2.41$$

Therefore, with life factor f_h=2.41, from Fig. 5.1 the rated life, L_{10h}, is approximately 7,000 hours.

(Example 3)

Determine the optimum model number for a cylindrical roller bearing operating at 450 r/min, with a radial load F_r of 200 kN, and which must have a life of over 20,000 hours.

From Fig. 5.1 the life factor f_h=3.02 (L_{10h} at 20,000), and the speed factor f_n=0.46 (n=450 r/min). To find the required basic dynamic load rating, C_r, formula (5.3) is used.

$$C_r = \frac{f_h}{f_n} P_r = \frac{3.02}{0.46} \times 200 = 1313 \text{ kN}$$

From the bearing table, the smallest bearing that fulfills all the requirements is NU2336 (C_r=1,380 kN).

(Example 4)

What are the rated lives of the two tapered roller bearings supporting the shaft shown in Fig. 6.13?

Bearing II is an ET-32206 with a C_r=54.5 kN, and bearing I is an ET-32205 with a C_r=42.0 kN. The spur gear shaft has a pitch circle diameter D_p of 150 mm, and a pressure angle α of 20°. The gear transmitted force HP=150 kW at 2,000 r/min (speed factor n).

The gear load from formula (6.1), (6.2a) and (6.3) is:

$$K_t = \frac{19.1 \times 10^6 \bullet HP}{D_p \bullet n} = \frac{19\,100 \times 150}{150 \times 2\,000} = 9.55 \text{ kN}$$

$$K_s = K_t \bullet \tan\alpha = 9.55 \times \tan 20° = 3.48 \text{ kN}$$

$$K_r = \sqrt{K_t^2 + K_s^2} = \sqrt{9.55^2 + 3.48^2} = 10.16 \text{ kN}$$

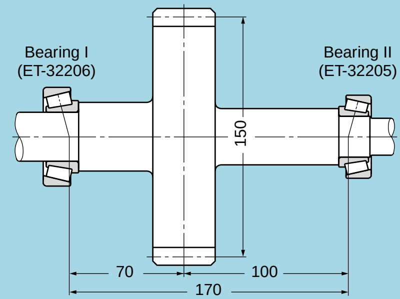


Fig. 6.13 Spur gear diagram

The radial loads for bearings I and II are:

$$F_{rI} = \frac{100}{170} K_r = \frac{100}{170} \times 10.16 = 5.98 \text{ kN}$$

$$F_{rII} = \frac{70}{170} K_r = \frac{70}{170} \times 10.16 = 4.18 \text{ kN}$$

$$\frac{0.5F_{rI}}{Y_I} = 1.87 > \frac{0.5F_{rII}}{Y_{II}} = 1.31$$

The equivalent radial load is:

$$P_{rI} = F_{rI} = 5.98 \text{ kN}$$

$$P_{rII} = XF_{rII} + Y_{II} \bullet \frac{0.5F_{rI}}{Y_I} = 0.4 \times 4.18 + 1.60 \times 1.87$$

$$= 4.66 \text{ kN}$$

From formula (5.3) and Fig. 5.1 the life factor, f_h, for each bearing is:

$$f_{hI} = f_n \frac{C_{rI}}{P_{rI}} = 0.293 \times \frac{54.5}{5.98} = 2.67$$

$$f_{hII} = f_n \frac{C_{rII}}{P_{rII}} = 0.293 \times \frac{42.0}{4.66} = 2.64$$

Therefore,

L_{hI} =13,200 hours

L_{hII} =12,700 hours

The combined bearing life, L_h, from formula (5.6) is:

$$L_h = \frac{1}{\left(\frac{1}{L_{hI}{}^e} + \frac{1}{L_{hII}{}^e}\right)^{1/e}} = \frac{1}{\left(\frac{1}{13\ 200^{9/8}} + \frac{1}{12\ 700^{9/8}}\right)^{8/9}}$$

$$= 6\ 990 \text{ hours}$$

(Example 5)

Find the mean load for spherical roller bearing 23932 (C_r=320 kN) when operated under the fluctuating conditions shown in Table 6.6.

Table 6.6

Condition No. *i*	Operating time % ϕ_i	radial load F_{ri} kN	axial load F_{ai} kN	revolution n_i r/min
1	5	10	2	1200
2	10	12	4	1000
3	60	20	6	800
4	15	25	7	600
5	10	30	10	400

The equivalent radial load, P_r, for each operating condition is found by using formula (6.17) and shown in Table 6.7. Because all the values for F_{ri} and F_{ai} from the bearing tables

are greater than $\frac{F_a}{F_r} > e = 0.18,\ X = 0.67$ and $Y_2 = 5.50.$

$$P_{ri} = XF_{ri} + Y_2F_{ai} = 0.67F_{ri} + 5.50F_{ai}$$

Table 6.7

Condition No. *i*	Equivalent radial load P_{ri} kN
1	17.7
2	30.0
3	46.4
4	55.3
5	75.1

From formula (6.12) the mean load, F_m, is:

$$F_m = \left[\frac{\Sigma\left(P_{ri}^{10/3} \bullet n_i \bullet \phi_i\right)}{\Sigma\left(n_i \bullet \phi_i\right)}\right]^{3/10} = 48.1 \text{ kN}$$

7. Bearing Fits

7.1 Interference

For rolling bearings the bearing rings are fixed on the shaft or in the housing so that slip or movement does not occur between the mated surface during operation or under load. This relative movement (sometimes called creep) between the fitted surfaces of the bearing and the shaft or housing can occur in a radial direction, or in an axial direction, or in the direction of rotation. This creeping movment under load causes damage to the bearing rings, shaft or housing in the form of abrasive wear, fretting corrosion or friction crack. This, in turn, can also lead to abrasive particles getting into the bearing, which can cause vibration, excessive heat, and lowered rotational efficiency. To ensure that slip does not occur between the fitted surfaces of the bearing rings and the shaft or housing, the bearing is usually installed with an interference fit.

The most effective interference fit is called a tight fit (or shrink fit). The advantage of this "tight fit" for thin walled bearings is that it provides uniform load support over the entire ring circumference without any loss in load carrying capacity.

However, with a tight interference fit, ease of mounting and dismounting the bearings is lost; and when using a non-separable bearing as a non-fixing bearing, axial displacement is impossible.

7.2 Calculation of interference

1) Load and interference

The minimum required amount of interference for inner rings mounted on solid shafts when acted on by radial loads, is found by formula (7.1) and (7.2).

When $F_r \leq 0.3C_{or}$

$$\Delta_{dF} = 0.08\sqrt{\frac{d \bullet F_r}{B}} \cdots\cdots\cdots\cdots\cdots (7.1)$$

When $F_r > 0.3C_{or}$

$$\Delta_{dF} = 0.02\frac{F_r}{B} \cdots\cdots\cdots\cdots\cdots (7.2)$$

where,

Δ_{dF} : Required effective interference (for load) μm
d : Nominal bore diameter mm
B : Inner ring width mm
F_r : Radial load N
C_{or} : Basic static rated load N

2) Temperature rise and interference

To prevent loosening of the inner ring on steel shafts due to temperature increases (difference between bearing temperature and ambient temperature) caused by bearing rotation, an interference fit must be given. The required amount of interference can be found by formula (7.3).

$$\Delta_{dT} = 0.0015 \bullet d \bullet \Delta T \cdots\cdots\cdots\cdots\cdots (7.3)$$

where,

Δ_{dT} : Required effective interference (for temperature) μm
ΔT : Difference between bearing temperature and ambient temperature °C
d : Bearing bore diameter mm

3) Effective interference and apparent interference

The effective interference (the actual interference after fitting) is different from the apparent interference derived from the dimensions measured value. This difference is due to the roughness or slight variations of the mating surfaces, and this slight flattening of the uneven surfaces at the time of fitting is taken into consideration. The relation between the effective and apparent interference, which varies according to the finish given to the mating surfaces, is expressed by formula (7.4).

$$\Delta d_{eff} = \Delta d_f - G \cdots\cdots\cdots\cdots\cdots (7.4)$$

where,

Δd_{eff} : Effective interference μm
Δd_f : Apparent interference μm
G =1.0~2.5 μm for ground shaft
=5.0~7.0 μm for turned shaft

4) Maximum interference

When bearing rings are installed with an interference fit on shafts or housings, the tension or compression stress may occur. If the interference is too large, it may cause damage to the bearing rings and reduce the fatigue life of the bearing. For these reasons, the maximum amount of interference should be less than 1/1000 of the shaft diameter, or outside diameter.

7.3 Fit selection

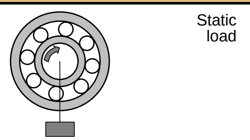
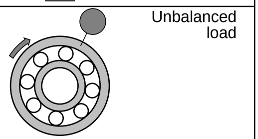
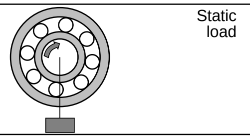
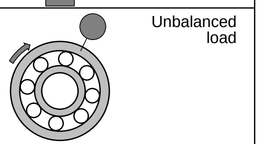
Selection of the proper fit is generally based on the following factors: 1) the direction and nature of the bearing load, 2) whether the inner ring or outer ring rotates, 3) whether the load on the inner or outer ring rotates or not, 4) whether there is static load or direction indeterminate load or not. For bearings under rotating loads or direction indeterminate loads, a tight fit is recommended; but for static loads, a transition fit or loose fit should be sufficient (see Table 7.1).

The interference should be tighter for heavy bearing loads or vibration and shock load conditions. Also, a tighter than normal fit should be given when the bearing is installed on hollow shafts or in housings with thin walls, or housings made of light allows or plastic.

In applications where high rotational accuracy must be maintained, high precision bearings and high tolerance shafts and housings should be employed instead of a tighter interference fit to ensure bearing stability. High interference fits should be avoided if possible as they cause shaft or housing deformities to be induced into the bearing rings, and thus reduce bearing rotational accuracy.

Because mounting and dismounting become very difficult when both the inner ring and outer ring of a non-separable bearing (for example a deep groove ball bearing) are given tight interference fits, one or the other rings should be given a loose fit.

Table 7.1 Radial load and bearing fit

Bearing rotation and load	Illustration	Ring load	Fit
Inner ring : Rotating Outer ring : Stationary Load direction : Constant		Rotating inner ring load	Inner ring : Tight fit
Inner ring : Stationary Outer ring : Rotating Load direction : Rotates with outer ring		Static outer ring load	Outer ring : Loose fit
Inner ring : Stationary Outer ring : Rotating Load direction : Constant		Static inner ring load	Inner ring : Loose fit
Inner ring : Rotating Outer ring : Stationary Load direction : Rotates with outer ring		Rotating outer ring load	Outer ring : Tight fit

7.4 Recommended fits

Metric size standard dimension tolerances for bearing shaft diameters and housing bore diameters are governed by ISO 286 and JIS B 0401 (dimension tolerances and fits). Accordingly, bearing fits are determined by the precision (dimensional tolerance) of the shaft diameter and housing bore diameter. Widely used fits for various shaft and housing bore diameter tolerances, and bearing bore and outside diameters are shown in Fig. 7.1.

Generally-used, recommended fits relating to the primary factors of bearing shape, dimensions, and load conditions are listed in Tables 7.2 through 7.5. Table 7.6 gives the numerical values for housing and shaft fits.

The bore and outside diameter tolerances and tolerance ranges for inch and metric tapered roller bearings are different. Recommended fits and numerical values for inch tapered roller bearings are shown in Table 7.8. For special fits or applications, please consult NTN.

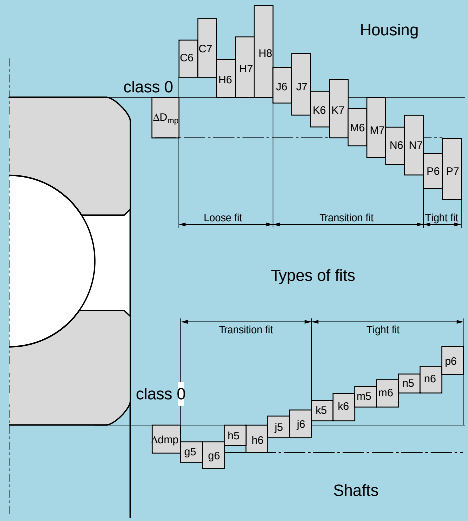


Fig. 7.1

Table 7.2 General standards for radial bearing fits (JIS class 0, 6, 6X)

Table 7.2 (1) Housing fits

Housing type	Load condition		Housing fits
Solid or split housing	Outer ring static load	All load conditions	H7
		Heat conducted through shaft	G7
Solid housing	Direction indeterminate load	Light to normal	JS7
		Normal to heavy load	K7
		Heavy shock load	M7
	Outer ring rotating load	Light or variable load	M7
		Normal to heavy load	N7
		Heavy load (thin wall housing) Heavy shock load	P7

Note: Fits apply to cast iron or steel housings. For light alloy housings, a tighter fit than listed is normally required.

Table 7.2 (2) Shaft fit

Bearing type	Load conditions		Ball bearings	Cylindrical and tapered roller bearings	Spherical roller bearings	Shaft fits
			Shaft diameter mm			
Cylindrical bore bearings	Rotating inner ring or indeterminate direction load	Light or fluctuating variable load	~ 18	—	—	h5
			18~100	~ 40	—	js6
			100~200	40~140	—	k6
			—	140~200	—	m6
		Normal to heavy load	~ 18	—	—	js5
			18~100	~ 40	~ 40	k5
			100~140	40~100	40~65	m5
			140~200	100~140	65~100	m6
			200~280	140~200	100~140	n6
			—	200~400	140~280	p6
			—	—	280~500	r6
		Very heavy or shock load	—	50~140	50~100	n6
			—	140~200	100~140	p6
			—	200~	140~	r6
	Static inner ring load	Inner ring axial displacement required	All shaft diameters			g6
		Easy axial displacement of inner ring not required	All shaft diameters			h6
Tapered bore bearings (With sleeve)	All load		All shaft diameters			h9/IT5

Note:
1. All values and fits listed are for solid steel shafts.
2. For radial bearings under axial loads, all shaft tolerance range classes are js6.
3. Load classifications are as follows:
   LIght load: $P_r \leq 0.06\ C_r$
   Normal load: $0.06\ C_r < P_r \leq 0.12\ C_r$
   Heavy load: $P_r > 0.12\ C_r$

   where,

   P_r: Bearing equivalent load
   C_r: Bearing basic dynamic load rating

Table 7.3 Solid type needle roller bearing fits

Table 7.3 (1) Shaft fit

Conditions			Shaft fits
Load type	Scale of load	Shaft diameter *d* mm	
Rotating inner ring or indeterminate direction load	Light load	~ 50	j5
	Normal load	~ 50	k5
		50~150	m5
		150~	m6
	Heavy load and shock load	~ 150	m6
		150~	n6
Static inner ring load	Medium & low speed revolution, light load	All sizes	g6
	General application	All sizes	h6
	When high rotation accuracy is required	All sizes	h5

Table 7.3 (2) Housing fit

Conditions		Housing fits
Static inner ring load	Normal to heavy load	J7
	Normal loads with split housings	H7
Outer ring rotating load	Light loads	M7
	Normal loads	N7
	Heavy and normal loads	P7
Direction indeterminate load	Light loads	J7
	Normal load	K7
	Very heavy or shock load	M7
High demands on running accuracy with light load		K6

Table 7.4 Standard fits for thrust bearings

Table 7.4 (1) Shaft fits

Load conditions		Shaft diameter	Shaft fits
"Pure" axial load (All thrust bearings)		All sizes	js6
Combined load: spherical roller thrust bearings	Static inner ring loads	All sizes	js6
	Inner ring rotating load or direction indeterminate	~200	k6
		200~400	m6
		400~	n6

Table 7.4 (2) Housing fits

Load conditions		Housing fits	Remarks
"Pure" axial load: All thrust bearings	When another bearing is used to support radial load	—	Clearance given between outer ring and housing
		H8	Accuracy required with thrust ball bearings
Combined load: spherical roller thrust bearings	Static outer ring load	H7	—
	Outer ring rotating load or direction indeterminate load	K7	Normal usage conditions
		M7	Relatively heavy

Table 7.5 Fits for electric motor bearings

Shaft or housing	Deep groove ball bearings			Cylindrical roller bearings		
	Shaft or housing bore diameter mm		Fits	Shaft or housing bore diameter mm		Fits
	over	incl.		over	incl.	
Shaft	—	18	j5	—	40	k5
	18	100	k5	40	160	m5
	100	160	m5	160	200	n5
Housing	All Sizes		H6 or J6	All sizes		H6 or J6

Table 7.6 Fitting values for radial bearings, Class 0

Table 7.6 (1) Shaft fit

Nominal bore diameter of bearing d (mm) over	incl.	Δ_{dmp} high	low	g5 bearing shaft	g6 bearing shaft	h5 bearing shaft	h6 bearing shaft	j5 bearing shaft	js5 bearing shaft	j6 bearing shaft
3	6	0	–8	4T~9L	4T~12L	8T~5L	8T~8L	11T~2L	10.5T~2.5L	14T~2L
6	10	0	–8	3T~11L	3T~14L	8T~6L	8T~9L	12T~2L	11T~3L	15T~2L
10	18	0	–8	2T~14L	2T~17L	8T~8L	8T~11L	13T~3L	12T~4L	16T~3L
18	30	0	–10	3T~16L	3T~20L	10T~9L	10T~13L	15T~4L	14.5T~4.5L	19T~4L
30	50	0	–12	3T~20L	3T~25L	12T~11L	12T~16L	18T~5L	17.5T~5.5L	23T~5L
50	80	0	–15	5T~23L	5T~29L	15T~13L	15T~19L	21T~7L	21.5T~6.5L	27T~7L
80	120	0	–20	8T~27L	8T~34L	20T~15L	20T~22L	26T~9L	27.5T~7.5L	33T~9L
120 140 160	140 160 180	0	–25	11T~32L	11T~39L	25T~18L	25T~25L	32T~11L	34T~9L	39T~11L
180 200 225	200 225 250	0	–30	15T~35L	15T~44L	30T~20L	30T~29L	37T~13L	40T~10L	46T~13L
250 280	280 315	0	–35	18T~40L	18T~49L	35T~23L	35T~32L	42T~16L	46.5T~11.5L	51T~16L
315 355	355 400	0	–40	22T~43L	22T~54L	40T~25L	40T~36L	47T~18L	52.5T~12.5L	58T~18L
400 450	450 500	0	–45	25T~47L	25T~60L	45T~27L	45T~40L	52T~20L	58.5T~13.5L	65T~20L

Table 7.6 (2) Housing fit

Nominal bore diameter of bearing D (mm) over	incl.	Δ_{dmp} high	low	G7 housing bearing	H6 housing bearing	H7 housing bearing	J6 housing bearing	J7 housing bearing	Js7 housing bearing	K6 housing bearing
6	10	0	–8	5L~28L	0~17L	0~23L	4T~13L	7T~16L	7.5T~15.5L	7T~10L
10	18	0	–8	6L~32L	0~19L	0~26L	5T~14L	8T~18L	9T~17L	9T~10L
18	30	0	–9	7L~37L	0~22L	0~30L	5T~17L	9T~21L	10.5T~19.5L	11T~11L
30	50	0	–11	9L~45L	0~27L	0~36L	6T~21L	11T~25L	12.5T~23.5L	13T~14L
50	80	0	–13	10L~53L	0~32L	0~43L	6T~26L	12T~31L	15T~28L	15T~17L
80	120	0	–15	12L~62L	0~37L	0~50L	6T~31L	13T~37L	17.5T~32.5L	18T~19L
120	150	0	–18	14L~72L	0~43L	0-~58L	7T~36L	14T~44L	20T~38L	21T~22L
150	180	0	–25	14L~79L	0~50L	0~65L	7T~43L	14T~51L	20T~45L	21T~29L
180	250	0	–30	15L~91L	0~59L	0~76L	7T~52L	16T~60L	23T~53L	24T~35L
250	315	0	–35	17L~104L	0~67L	0~87L	7T~60L	16T~71L	26T~61L	27T~40L
315	400	0	–40	18L~115L	0~76L	0~97L	7T~69L	18T~79L	28.5T~68.5L	29T~47L
400	500	0	–45	20L~128L	0~85	0~108	7T~78L	20T~88L	31.5T~76.5L	32T~53L

Unit μm

js6	k5	k6	m5	m6	n6	p6	r6
bearing shaft	bearing shaft	bearing shaft	bearing shaft	bearing shaft	bearing shaft	bearing shaft	bearing shaft
12T~4L	14T~1T	17T~1T	17T~4T	20T~4T	24T~8T	28T~12T	—
12.5T~4.5L	15T~1T	18T~1T	20T~6T	23T~6T	27T~10T	32T~15T	—
13.5T~5.5L	17T~1T	20T~1T	23T~7T	26T~7T	31T~12T	37T~18T	—
16.5T~6.5L	21T~2T	25T~2T	27T~8T	31T~8T	38T~15T	45T~22T	—
20T~8L	25T~2T	30T~2T	32T~9T	37T~9T	45T~17T	54T~26T	—
24.5T~9.5L	30T~2T	36T~2T	39T~11T	45T~11T	54T~20T	66T~32T	—
31T~11L	38T~3T	45T~3T	48T~13T	55T~13T	65T~23T	79T~37T	—
37.5T~12.5L	46T~3T	53T~3T	58T~15T	65T~15T	77T~27T	93T~43T	113T~63T 115T~65T 118T~68T
44.5T~14.5T	54T~4T	63T~4T	67T~17T	76T~17T	90T~31T	109T~50T	136T~77T 139T~80T 143T~84T
51T~16L	62T~4T	71T~4T	78T~20T	87T~20T	101T~34T	123T~56T	161T~94T 165T~98T
58T~18L	69T~4T	80T~4T	86T~21T	97T~21T	113T~37T	138T~62T	184T~108T 190T~114T
65T~20L	77T~5T	90T~4T	95T~23T	108T~23T	125T~40T	153T~68T	211T~126T 217T~132T

Unit μm

K7	M7	N7	P7
housing bearing	housing bearing	housing bearing	housing bearing
10T~13L	15T~8L	19T~4L	24T~1T
12T~14L	18T~8L	23T~3L	29T~3T
15T~15L	21T~9L	28T~2L	35T~5T
18T~18L	25T~11L	33T~3L	42T~6T
21T~22L	30T~13L	39T~4L	51T~8T
25T~25L	35T~15L	45T~5L	59T~9T
28T~30L	40T~18L	52T~6L	68T~10T
28T~37L	40T~25L	52T~13L	68T~3T
33T~43L	46T~30L	60T~16L	79T~3T
36T~51L	52T~35L	66T~21L	88T~1T
40T~57L	57T~40L	73T~24L	98T~1T
45T~63L	63T~45L	80T~28L	108T~0

Table 7.7 Fits for inch series tapered roller bearing (ANSI class 4)

Table 7.7 (1) Fit with shaft

Unit μm
0.0001 inch

Load conditions		Shaft diameter d mm, inch over	Shaft diameter d mm, inch incl.	Cone bore tolerance[2] Δ_{ds} high	Cone bore tolerance[2] Δ_{ds} low	Shaft tolerance high	Shaft tolerance low	Extreme fits[3] max~min
Rotating cone load	Normal loads, no shock	— —	76.200 3.0000	+13 +5	0 0	+38 +15	+26 +10	38T~13T 15T~5T
		76.200 3.0000	304.800 12.0000	+25 +10	0 0	+64 +25	+38 +15	64T~13T 25T~5T
	Heavy loads or shock loads	— —	76.200 3.0000	+13 +5	0 0	Use average tight cone fit of 0.5μm/mm, (0.0005 inch/inch) of cone bore, use a minimum fit of 25μm, 0.0010 inch tight.		
		76.200 3.0000	304.800 12.0000	+25 +10	0 0			
Stationary cone load	Cone axial displacement on shaft necessary[1]	— —	76.200 3.0000	+13 +5	0 0	0 0	–13 –5	0~26L 0~10L
		76.200 3.0000	304.800 12.0000	+25 +10	0 0	0 0	–25 –10	0~51L 0~20L
	Cone axial displacement on shaft unnecessary	— —	76.200 3.0000	+13 +5	0 0	+13 +5	0 0	13T~13L 5T~5L
		76.200 3.0000	304.800 12.0000	+25 +10	0 0	+25 +10	0 0	25T~25L 10T~10L

1) Applies only to ground shafts.
2) For bearings with negation deviation indicated in bearing tables, same fit applies.
3) T=tight, L=loose, *d*=cone bore, mm, inch
Note: For bearings higher than class 2, consult NTN.

Table 7.7 (2) Fit with housing

Unit μm
0.0001 inch

Load conditions		Housing bore diameter mm, inch over	Housing bore diameter mm, inch incl.	Cup O.D. tolerance[1] high	Cup O.D. tolerance[1] low	Housing bore tolerance high	Housing bore tolerance low	Extreme fits[2] max~min
Stationary cup load	Light and normal loads: cup easily axially displaceable	— —	76.200 3.0000	+25 +10	0 0	+76 +30	+50 +20	25L~76L 10L~30L
		76.200 3.0000	127.000 5.0000	+25 +10	0 0	+76 +30	+50 +20	25L~76L 10L~30L
		127.000 5.0000	304.800 12.0000	+25 +10	0 0	+76 +30	+50 +20	25L~76L 10L~30L
	Light and normal loads: cup axially adjustable	— —	76.200 3.0000	+25 +10	0 0	+25 +10	0 0	25T~25L 10T~10L
		76.200 3.0000	127.000 5.0000	+25 +10	0 0	+25 +10	0 0	25T~25L 10T~10L
		127.000 5.0000	304.800 12.0000	+25 +10	0 0	+51 +20	0 0	25T~51L 10T~20L
	Heavy loads: cup not axially displaceable	— —	76.200 3.0000	+25 +10	0 0	–13 –5	–39 –15	64T~13T 25T~5T
		76.200 3.0000	127.000 5.0000	+25 +10	0 0	–25 –10	–51 –20	76T~25T 30T~10T
		127.000 5.0000	304.800 12.0000	+25 +10	0 0	–25 –10	–51 –20	76T~25T 30T~10T
Rotating cup load	Cup not axially displaceable	— —	76.200 3.0000	+25 +10	0 0	–13 –5	–39 –15	64T~13T 25T~5T
		76.200 3.0000	127.000 5.0000	+25 +10	0 0	–25 –10	–51 –20	76T~25T 30T~10T
		127.000 5.0000	304.800 12.0000	+25 +10	0 0	–25 –10	–51 –20	76T~25T 30T~10T

1) For bearings with negation deviation indicated in bearing tables, same fit applies.
2) T=tight, L=loose
Note: For bearings higher than class 2, consult NTN.

Table 7.8 Fits for inch series tapered roller bearing (ANSI classes 3 and 0)

Table 7.8 (1) Fit with shaft

Unit μm
0.0001 inch

Load conditions		Shaft diameter mm, inch		Cone bore[2] tolerance		Shaft tolerance		Extreme fits[3]	
		over	incl.	high	low	high	low	max	min
Rotating cone load	precision machine tool spindles	— —	304.800 12.0000	+13 +5	0 0	+31 +12	+18 +7	31T~5T 12T~2T	
	heavy loads, or high speed or shock	— —	76.200 3.0000	+13 +5	0 0	Use minimum tight cone fit of 0.25μm/mm 0.00025 inch/inch) of cone bore.			
		76.200 3.0000	304.800 12.0000	+13 +5	0 0				
Stationary cone load	precision machine tool spindles	— —	304.800 12.0000	+13 +5	0 0	+31 +12	+18 +7	31T~5T 36T~2T	

Note: Must be applied for maximum bore dia. 241.300mm (9.500 inch) in case of class 0 product.

Note 1) T=tight, L=loose

2) Must be applied for maximum cup OD 304.800mm (12.000 inch) case of class 0 product.

Table 7.8 (2) Fit with housing

Unit μm
0.0001 inch

Load conditions		Housing bore diameter mm, inch		Cup O.D. tolerance		Housing bore tolerance		Extreme fits[2]	
		over	incl.	high	low	high	low	max	min
Stationary cup load	Floating	— —	152.400 6.0000	+13 +5	0 0	+38 +15	+26 +10	13L~38L 5L~15L	
		152.400 6.0000	304.800 12.0000	+13 +5	0 0	+38 +15	+26 +10	13L~38L 5L~14L	
	Clamped	— —	152.400 6.0000	+13 +5	0 0	+25 +10	+13 +5	0~25L 0~10L	
		152.400 6.0000	304.800 12.0000	+13 +5	0 0	+25 +10	+13 +5	0~25L 0~10L	
	Adjustable	— —	152.400 6.0000	+13 +5	0 0	+13 +5	0 0	13T~13L 5T~5L	
		152.400 6.0000	304.800 12.0000	+13 +5	0 0	+25 +10	0 0	13T~25L 5T~10L	
	Nonadjustable or in carriers	— —	152.400 6.0000	+13 +5	0 0	0 0	–12 –5	25T~0 10T~0	
		152.400 6.0000	304.800 12.0000	+13 +5	0 0	0 0	–25 –10	38T~0 15~0	
Rotating cup load	Nonadjustable or in carriers	— —	152.400 6.0000	+13 +5	0 0	–13 –5	–25 –10	38T~13T 15T~5T	
		152.400 6.0000	304.800 12.0000	+13 +5	0 0	–13 –5	–38 –15	51T~13T 20T~5T	

Note 1) T=tight, L=loose

2) Must be applied for maximum cup OD 304.800mm (12.000 inch) case of class 0 product.

8. Bearing Internal Clearance and Preload

8.1 Bearing internal clearance

Bearing internal clearance (initial clearance) is the amount of internal clearance a bearing has before being installed on a shaft or in a housing.

As shown in Fig. 8.1, when either the inner ring or the outer ring is fixed and the other ring is free to move, displacement can take place in either an axial or radial direction. This amount of displacement (radially or axially) is termed the internal clearance and, depending on the direction, is called the radial internal clearance or the axial internal clearance.

When the internal clearance of a bearing is measured, a slight measurement load is applied to the raceway so the internal clearance may be measured accurately. However, at this time, a slight amount of elastic deformation of the bearing occurs under the measurement load, and the clearance measurement value (measured clearance) is slightly larger than the true clearance. This discrepancy between the true bearing clearance and the increased amount due to the elastic deformation must be compensated for. These compensation values are given in Table 8.1. For roller bearings the amount of elastic deformation can be ignored.

The internal clearance values for each bearing class are shown in Tables 8.3 through 8.10.

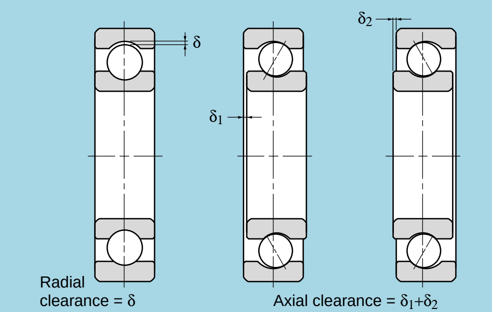


Fig. 8.1 Internal clearance

Table 8.1 Adjustment of radial internal clearance based on measured load — Unit μm

Nominal Bore Diameter d (mm) over	incl.	Measuring Load (N)	Radial Clearance Increase C2	Normal	C3	C4	C5
10	18	24.5	3~4	4	4	4	4
18	50	49	4~5	5	6	6	6
50	200	147	6~8	8	9	9	9

8.2 Internal clearance selection

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

Effective internal clearance:

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

$$\delta_{\text{eff}} = \delta_{\text{o}} - (\delta_{\text{f}} + \delta_{\text{t}}) \cdots\cdots\cdots (8.1)$$

where,

δ_{eff} : Effective internal clearance mm
δ_{o} : Bearing internal clearance mm
δ_{f} : Reduced amount of clearance due to interference mm
δ_{t} : Reduced amount of clearance due to temperature differential of inner and outer rings mm

Reduced clearance due to interference:

When bearings are installed with interference fits on shafts and in housings, the inner ring will expand and the outer ring will contract; thus reducing the bearings' internal clearance. The amount of expansion or contraction varies depending on the shape of the bearing, the shape of the shaft or housing, dimensions of the respective parts, and the type of materials used. The differential can range from approximately 70% to 90% of the effective interference.

$$\delta_{\text{f}} = (0.70 \sim 0.90) \bullet \Delta_{deff} \cdots\cdots\cdots (8.2)$$

where,

δ_{f} : Reduced amount of clearance due to interference mm
Δ_{deff} : Effective interference mm

Reduced internal clearance due to inner/outer ring temperature difference:

During operation, normally the outer ring will be from 5° to 10° cooler than the inner ring or rotating parts. However, if the cooling effect of the housing is large, the shaft is connected to a heat source, or a heated substance is conducted through the hollow shaft; the temperature difference between the two rings can be even greater. The amount of internal clearance is thus further reduced by the differential expansion of the two rings.

$$\delta_t = \alpha \bullet \Delta_T \bullet D_o \cdots\cdots\cdots\cdots\cdots (8.3)$$

where,

- δ_t : Amount of reduced clearance due to heat differential mm
- α : Bearing steel linear expansion coefficient 12.5×10^{-6}/°C
- Δ_T : Inner/outer ring temperature differential °C
- D_o : Outer ring raceway diameter mm

Outer ring raceway diameter, D_o, values can be approximated by using formula (8.4) or (8.5).

For ball bearings and self-aligning roller bearings,

$$D_o = 0.20(d + 4.0D) \cdots\cdots\cdots\cdots\cdots (8.4)$$

For roller bearings (except self-aligning),

$$D_o = 0.25(d + 3.0D) \cdots\cdots\cdots\cdots\cdots (8.5)$$

where,

- d : Bearing bore diameter mm
- D : Bearing outside diameter mm

8.3 Bearing internal clearance selection standards

Theoretically, to maximize life, the optimum operating internal clearance for any bearing would be a slight negative clearance after the bearing had reached normal operating temperature.

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

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

Table 8.2 lists non-standard clearance recommendations for various applications and operating conditions.

Table 8.2 Examples of applications where bearing clearances other than normal clearances are used

Operating conditions	Applications	Selected clearance
With heavy or shock load, clearance is great.	Railway vehicle axles	C3
	Vibration screens	C3
With direction indeterminate load, both inner and outer rings are tight-fitted.	Railway vehicle traction motors	C4
	Tractors and final speed regulators	C4
Shaft or inner ring is heated.	Paper making machines and driers	C3, C4
	Rolling mill table rollers	C3
Clearance fit for both inner and outer rings.	Rolling mill roll necks	C2
To reduce noise and vibration when rotating.	Micromotors	C2
To reduce shaft runout, clearance is adjusted.	Main spindles of lathes (Double-row cylindrical roller bearings)	C9NA, C0NA

Table 8.3 Radial internal clearance of deep groove ball bearings Unit μm

Nominal bore diameter *d* mm		C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
—	2.5	0	6	4	11	10	20	—	—	—	—
2.5	6	0	7	2	13	8	23	—	—	—	—
6	10	0	7	2	13	8	23	14	29	20	37
10	18	0	9	3	18	11	25	18	33	25	45
18	24	0	10	5	20	13	28	20	36	28	48
24	30	1	11	5	20	13	28	23	41	30	53
30	40	1	11	6	20	15	33	28	46	40	64
40	50	1	11	6	23	18	36	30	51	45	73
50	65	1	15	8	28	23	43	38	61	55	90
65	80	1	15	10	30	25	51	46	71	65	105
80	100	1	18	12	36	30	58	53	84	75	120
100	120	2	20	15	41	36	66	61	97	90	140
120	140	2	23	18	48	41	81	71	114	105	160
140	160	2	23	18	53	46	91	81	130	120	180
160	180	2	25	20	61	53	102	91	147	135	200
180	200	2	30	25	71	63	117	107	163	150	230
200	225	2	35	25	85	75	140	125	195	175	265
225	250	2	40	30	95	85	160	145	225	205	300
250	280	2	45	35	105	90	170	155	245	225	340
280	315	2	55	40	115	100	190	175	270	245	370
315	355	3	60	45	125	110	210	195	300	275	410
355	400	3	70	55	145	130	240	225	340	315	460
400	450	3	80	60	170	150	270	250	380	350	510
450	500	3	90	70	190	170	300	280	420	390	570
500	560	10	100	80	210	190	330	310	470	440	630
560	630	10	110	90	230	210	360	340	520	490	690
710	800	20	140	120	290	270	450	430	630	600	840
800	900	20	160	140	320	300	500	480	700	670	940
900	1000	20	170	150	350	330	550	530	770	740	1040
1000	1120	20	180	160	380	360	600	580	850	820	1150
1120	1250	20	190	170	410	390	650	630	920	890	1260

Table 8.5 Radial internal clearance of double row and duplex angular contact ball bearings Unit μm

Nominal bore diameter *d* mm		C1		C2		Normal		C3		C4	
over	incl.	min	max	min	max	min	max	min	max	min	max
—	10	3	8	6	12	8	15	15	22	22	30
10	18	3	8	6	12	8	15	15	24	30	40
18	30	3	10	6	12	10	20	20	32	40	55
30	50	3	10	8	14	14	25	25	40	55	75
50	80	3	11	11	17	17	32	32	50	75	95
80	100	3	13	13	22	22	40	40	60	95	120
100	120	3	15	15	30	30	50	50	75	110	140
120	150	3	16	16	33	35	35	55	80	130	170
150	180	3	18	18	35	35	60	60	90	150	200
180	200	3	20	20	40	40	65	65	100	180	240

Note: The clearance group in the table is applied only to contact angles in the table below.

Contact angle symbol	Nominal contact angle	Applicable clearance group
C	15°	C1, C2
A[1]	30°	C2, Normal, C3
B	40°	Normal, C3, C4

1) Usually not to be indicated

Table 8.6 Radial internal clearance of cylindrical roller bearings, needle roller bearings

Table 8.6 (1) Cylindrical Bore Interchangeable Radial Clearance ISO Cylindrical Roller Bearings ONLY

Unit μm

Nominal bore diameter d mm		C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
0	10	0	25	20	45	35	60	50	75	—	—
10	24	0	25	20	45	35	60	50	75	65	90
24	30	0	25	20	45	35	60	50	75	70	95
30	40	5	30	25	50	45	70	60	85	80	105
40	50	5	35	30	60	50	80	70	100	95	125
50	65	10	40	40	70	60	90	80	110	110	140
65	80	10	45	40	75	65	100	90	125	130	165
80	100	15	50	50	85	75	110	105	140	155	190
100	120	15	55	50	90	85	125	125	165	180	220
120	140	15	60	60	105	100	145	145	190	200	245
140	160	20	70	70	120	115	165	165	215	225	275
160	180	25	75	75	125	120	170	170	220	250	300
180	200	35	90	90	145	140	195	195	250	275	330
200	225	45	105	105	165	160	220	220	280	305	365
225	250	45	110	110	175	170	235	235	300	330	395
250	280	55	125	125	195	190	260	260	330	370	440
280	315	55	130	130	205	200	275	275	350	410	485
315	355	65	145	145	225	225	305	305	385	455	535
355	400	100	190	190	280	280	370	370	460	510	600
400	450	110	210	210	310	310	410	410	510	565	665
450	500	110	220	220	330	330	440	440	550	625	735
500	560	120	240	240	360	360	480	480	600	—	—
560	630	140	260	260	380	380	500	500	620	—	—
630	710	145	285	285	425	425	565	565	705	—	—
710	800	150	310	310	470	470	630	630	790	—	—
800	900	180	350	350	520	520	690	690	860	—	—
900	1000	200	390	390	580	580	770	770	960	—	—
1000	1120	220	430	430	640	640	850	850	1060	—	—
1120	1250	230	470	470	710	710	950	950	1190	—	—
1250	1400	270	530	530	790	790	1050	1050	1310	—	—

Table 8.6 (2) Tapered Bore Interchangeable Radial Clearance

Unit μm

Nominal bore diameter d mm		C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
0	24	15	40	30	55	40	65	50	75	—	—
24	30	20	45	35	60	45	70	55	80	—	—
30	40	20	45	40	65	55	80	70	95	—	—
40	50	25	55	45	75	60	90	75	105	—	—
50	65	30	60	50	80	70	100	90	120	—	—
65	80	35	70	60	95	85	120	110	145	—	—
80	100	40	75	70	105	95	130	120	155	—	—
100	120	50	90	90	130	115	155	140	180	—	—
120	140	55	100	100	145	130	175	160	205	—	—
140	160	60	110	110	160	145	195	180	230	—	—
160	180	75	125	125	175	160	210	195	245	—	—
180	200	85	140	140	195	180	235	220	275	—	—
200	225	95	155	155	215	200	260	245	305	—	—
225	250	105	170	170	235	220	285	270	335	—	—
250	280	115	185	185	255	240	310	295	365	—	—
280	315	130	205	205	280	265	340	325	400	—	—
315	355	145	225	225	305	290	370	355	435	—	—
355	400	165	225	255	345	330	420	405	495	—	—
400	450	185	285	285	385	370	470	455	555	—	—
450	500	205	315	315	425	410	520	505	615	—	—

Table 8.7 Radial internal clearance of cylindrical roller bearings, needle roller bearings (Non-interchangeable)

Nominal bore diameter *d* mm		Bearing with cylindrical bore											
		C1NA		C2NA		NA[1)]		C3NA		C4NA		C5NA	
over	incl.	min	max	min	max	min	max	min	max	min	max	min	max
—	10	5	10	10	20	20	30	35	45	45	55	—	—
10	18	5	10	10	20	20	30	35	45	45	55	65	75
18	24	5	10	10	20	20	30	35	45	45	55	65	75
24	30	5	10	10	25	25	35	40	50	50	60	70	80
30	40	5	12	12	25	25	40	45	55	55	70	80	95
40	50	5	15	15	30	30	45	50	65	65	80	95	110
50	65	5	15	15	35	35	50	55	75	75	90	110	130
65	80	10	20	20	40	40	60	70	90	90	110	130	150
80	100	10	25	25	45	45	70	80	105	105	125	155	180
100	120	10	25	25	50	50	80	95	120	120	145	180	205
120	140	15	30	30	60	60	90	105	135	135	160	200	230
140	160	15	35	35	65	65	100	115	150	150	180	225	260
160	180	15	35	35	75	75	110	125	165	165	200	250	285
180	200	20	40	40	80	80	120	140	180	180	220	275	315
200	225	20	45	45	90	90	135	155	200	200	240	305	350
225	250	25	50	50	100	100	150	170	215	215	265	330	380
250	280	25	55	55	110	110	165	185	240	240	295	370	420
280	315	30	60	60	120	120	180	205	265	265	325	410	470
315	355	30	65	65	135	135	200	225	295	295	360	455	520
355	400	35	75	75	150	150	225	255	330	330	405	510	585
400	450	45	85	85	170	170	255	285	370	370	455	565	650
450	500	50	95	95	190	190	285	315	410	410	505	625	720

1) For bearings with normal clearance, only NA is added to bearing numbers. Ex. NU310NA, NN03020KNAP5

Table 8.4 Radial internal clearance of self-aligning ball bearings

Nominal bore diameter *d* mm		Bearing with cylindrical bore									
		C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
2.5	6	1	8	5	15	10	20	15	25	21	33
6	10	2	9	6	17	12	25	19	33	27	42
10	14	2	10	6	19	13	26	21	35	30	48
14	18	3	12	8	21	15	28	23	37	32	50
18	24	4	14	10	23	17	30	25	39	34	52
24	30	5	16	11	24	19	35	29	46	40	58
30	40	6	18	13	29	23	40	34	53	46	66
40	50	6	19	14	31	25	44	37	57	50	71
50	65	7	21	16	36	30	50	45	69	62	88
65	80	8	24	18	40	35	60	54	83	76	108
80	100	9	27	22	48	42	70	64	96	89	124
100	120	10	31	25	56	50	83	75	114	105	145
120	140	10	38	30	68	60	100	90	135	125	175
140	160	15	44	35	80	70	120	110	161	150	210

Table 8.7 (Cont.) Radial internal clearance of cylindrical roller bearings, needle roller bearings (Non-interchangeable) Unit μm

Bearing with tapered bore												Nominal bore diameter d mm	
C9NA		C0NA[2]		C1NA[2]		C2NA		NA[1]		C3NA			
min	max	min	max	min	max	min	max	min	max	min	max	over	incl.
5	5	7	17	10	20	20	30	35	45	45	55	—	10
5	10	7	17	10	20	20	30	35	45	45	55	10	18
5	10	7	17	10	20	20	30	35	45	45	55	18	24
5	10	10	20	10	25	25	35	40	50	50	60	24	30
5	12	10	20	12	25	25	40	45	55	55	70	30	40
5	15	10	20	15	30	30	45	50	65	65	80	40	50
5	15	10	20	15	35	35	50	55	75	75	90	50	65
10	20	15	30	20	40	40	60	70	90	90	110	65	80
10	25	20	35	25	45	45	70	80	105	105	125	80	100
10	25	20	35	25	50	50	80	95	120	120	145	100	120
15	30	25	40	30	60	60	90	105	135	135	160	120	140
15	35	30	45	35	65	65	100	115	150	150	180	140	160
15	35	30	45	35	75	75	110	125	165	165	200	160	180
20	40	30	50	40	80	80	120	140	180	180	220	180	220
20	45	35	55	45	90	90	135	155	200	200	240	200	225
25	50	40	65	50	100	100	150	170	215	215	265	225	250
25	55	40	65	55	110	110	165	185	240	240	295	250	280
30	60	45	75	60	120	120	180	205	265	265	325	280	315
30	65	45	75	65	135	135	200	225	295	295	360	315	355
35	75	50	90	75	150	150	225	255	330	330	405	355	400
45	85	60	100	85	170	170	255	285	370	370	455	400	450
50	95	70	115	95	190	190	285	315	410	410	505	450	500

2) C9NA, C0NA and C1NA are applied only to precision bearings of class 5 and higher.

Unit μm

Bearing with tapered bore										Nominal bore diameter d mm	
C2		Normal		C3		C4		C5			
min	max	min	max	min	max	min	max	min	max	over	incl.
—	—	—	—	—	—	—	—	—	—	2.5	6
—	—	—	—	—	—	—	—	—	—	6	10
—	—	—	—	—	—	—	—	—	—	10	14
—	—	—	—	—	—	—	—	—	—	14	18
7	17	13	26	20	33	28	42	37	55	18	24
9	20	15	28	23	39	33	50	44	62	24	30
12	24	19	35	29	46	40	59	52	72	30	40
14	27	22	39	33	52	45	65	58	79	40	50
18	32	27	47	41	61	56	80	73	99	50	65
23	39	35	57	50	75	69	98	91	123	65	80
29	47	42	68	62	90	84	116	109	144	80	100
35	56	50	81	75	108	100	139	130	170	100	120
40	68	60	98	90	130	120	165	155	205	120	140
45	74	65	110	100	150	140	191	180	240	140	160

Table 8.8 Axial internal clearance of metric double row and duplex tapered roller bearings

(Except series 329X, 322C, 323C)

Nominal bore diameter d mm		Contact angle $\alpha \leq 27°$ ($e \leq 0.76$)							
		C2		Normal		C3		C4	
over	incl.	min	max	min	max	min	max	min	max
18	24	25	75	75	125	125	170	170	220
24	30	25	75	75	125	145	195	195	245
30	40	25	95	95	165	165	235	210	280
40	50	20	85	85	150	175	240	240	305
50	65	20	85	110	175	195	260	280	350
65	80	20	110	130	220	240	325	325	410
80	100	45	150	150	260	280	390	390	500
100	120	45	175	175	305	350	480	455	585
120	140	45	175	175	305	390	520	500	630
140	160	60	200	200	340	400	540	520	660
160	180	80	220	240	380	440	580	600	740
180	200	100	260	260	420	500	660	660	820
200	225	120	300	300	480	560	740	720	900
225	250	160	360	360	560	620	820	820	1020
250	280	180	400	400	620	700	920	920	1140
280	315	200	440	440	680	780	1020	1020	1260
315	355	220	480	500	760	860	1120	1120	1380
355	400	260	560	560	860	980	1280	1280	1580
400	500	300	600	620	920	1100	1400	1440	1740

Note: Radial internal clearance is approximately obtained from:

$$\Delta r = \frac{e}{1.5}\Delta_a$$

where Δr =radial internal clearance, μm
Δ_a =axial internal clearance, μm
e =constant, see bearing tables

Table 8.8 (Cont.) Axial internal clearance of metric double row and duplex tapered roller bearings

(Except series 329X, 322C, 323C) Unit μm

Contact angle α>27° (e>0.76)								Nominal bore diameter d mm	
C2		Normal		C3		C4			
min	max	min	max	min	max	min	max	over	incl.
10	30	30	50	50	70	70	90	18	24
10	30	30	50	60	80	80	100	24	30
10	40	40	70	70	100	90	120	30	40
10	40	40	70	80	110	110	140	40	50
10	40	50	80	90	120	130	160	50	65
10	50	60	100	110	150	150	190	65	80
20	70	70	120	130	180	180	230	80	100
20	70	70	120	150	200	210	260	100	120
20	70	70	120	160	210	210	260	120	140
30	100	100	160	180	240	240	300	140	160
—	—	—	—	—	—	—	—	160	180
—	—	—	—	—	—	—	—	180	200
—	—	—	—	—	—	—	—	200	225
—	—	—	—	—	—	—	—	225	250
—	—	—	—	—	—	—	—	250	280
—	—	—	—	—	—	—	—	280	315
—	—	—	—	—	—	—	—	315	355
—	—	—	—	—	—	—	—	355	400
—	—	—	—	—	—	—	—	400	500

Table 8.10 Radial internal clearance of bearings for electric motor

Unit μm

Nominal bore diameter d mm		Radial internal clearance CM[1)]			
		Deep groove ball bearings		Cylindrical[2)] roller bearings	
over	incl.	min	max	min	max
10[3)]	18	4	11	—	—
18	24	5	12	—	—
24	30	5	12	15	30
30	40	9	17	15	30
40	50	9	17	20	35
50	65	12	22	25	40
65	80	12	22	30	45
80	100	18	30	35	55
100	120	18	30	35	60
120	140	24	38	40	65
140	160	24	38	50	80
160	180	—	—	60	90
180	200	—	—	65	100

1) Suffix CM is added to bearing numbers.
2) Non-interchangeable clearance.
3) This diameter is included in the group.

Table 8.9 Radial internal clearance of spherical roller bearings

Nominal bore diameter d mm		Bearing with cylindrical bore									
		C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
14	18	10	20	20	35	35	45	45	60	60	75
18	24	10	20	20	35	35	45	45	60	60	75
24	30	15	25	25	40	40	55	55	75	75	95
30	40	15	30	30	45	45	60	60	80	80	100
40	50	20	35	35	55	55	75	75	100	100	125
50	65	20	40	40	65	65	90	90	120	120	150
65	80	30	50	50	80	80	110	110	145	145	180
80	100	35	60	60	100	100	135	135	180	180	225
100	120	40	75	75	120	120	160	160	210	210	260
120	140	50	95	95	145	145	190	190	240	240	300
140	160	60	110	110	170	170	220	220	280	280	350
160	180	65	120	120	180	180	240	240	310	310	390
180	200	70	130	130	200	200	260	260	340	340	430
200	225	80	140	140	220	220	290	290	380	380	470
225	250	90	150	150	240	240	320	320	420	420	520
250	280	100	170	170	260	260	350	350	460	460	570
280	315	110	190	190	280	280	370	370	500	500	630
315	355	120	200	200	310	310	410	410	550	550	690
355	400	130	220	220	340	340	450	450	600	600	750
400	450	140	240	240	370	370	500	500	660	660	820
450	500	140	260	260	410	410	550	550	720	720	900
500	560	150	280	280	440	440	600	600	780	780	1000
560	630	170	310	310	480	480	650	650	850	850	1100
630	710	190	350	350	530	530	700	700	920	920	1190
710	800	210	390	390	580	580	770	770	1010	1010	1300
800	900	230	430	430	650	650	860	860	1120	1120	1440
900	1000	260	480	480	710	710	930	930	1220	1220	1570
1000	1120	290	530	530	780	780	1020	1020	1330	1330	1720
1120	1250	320	580	280	860	860	1120	1120	1460	1460	1870
1250	1400	350	640	640	950	950	1240	1240	1620	1620	2080

Table 8.9 (Cont.) Radial internal clearance of spherical roller bearings

Unit μm

Bearing with tapered bore										Nominal bore diameter d mm	
C2		Normal		C3		C4		C5			
min	max	min	max	min	max	min	max	min	max	over	incl.
—	—	—	—	—	—	—	—	—	—	14	18
15	25	25	35	35	45	45	60	60	75	18	24
20	30	30	40	40	55	55	75	75	95	24	30
25	35	35	50	50	65	65	85	85	105	30	40
30	45	45	60	60	80	80	100	100	130	40	50
40	55	55	75	75	95	95	120	120	160	50	65
50	70	70	95	95	120	120	150	150	200	65	80
55	80	80	110	110	140	140	180	180	230	80	100
65	100	100	135	135	170	170	220	220	280	100	120
80	120	120	160	160	200	200	260	260	330	120	140
90	130	130	180	180	230	230	300	300	380	140	160
100	140	140	200	200	260	260	340	340	430	160	180
110	160	160	220	220	290	290	370	370	470	180	200
120	180	180	250	250	320	320	410	410	520	200	225
140	200	200	270	270	350	350	450	450	570	225	250
150	220	220	300	300	390	390	490	490	620	250	280
170	240	240	330	330	430	430	540	540	680	280	315
190	270	270	360	360	470	470	590	590	740	315	355
210	300	300	400	400	520	520	650	650	820	355	400
230	330	330	440	440	570	570	720	720	910	400	450
260	370	370	490	490	630	630	790	790	1000	450	500
290	410	410	540	540	680	680	870	870	1100	500	560
320	460	460	600	600	760	760	980	980	1230	560	630
350	510	510	670	670	850	850	1090	1090	1360	630	710
390	570	570	750	750	960	960	1220	1220	1500	710	800
440	640	640	840	840	1070	1070	1370	1370	1690	800	900
490	710	710	930	930	1190	1190	1520	1520	1860	900	1000
530	770	770	1030	1030	1300	1300	1670	1670	2050	1000	1120
570	830	830	1120	1120	1420	1420	1830	1830	2250	1120	1250
620	910	910	1230	1230	1560	1560	2000	2000	2470	1250	1400

8.4 Preload

Normally, bearings are used with a slight internal clearance under operating conditions. However, in some applications, bearings are given an initial load; this means that the bearings' internal clearance is negative before operation. This is called "preload" and is commonly applied to angular ball bearings and tapered roller bearings.

8.4.1 Purpose of preload

Giving preload to a bearing results in the rolling element and raceway surfaces being under constant elastic compressive forces at their contact points. This has the effect of making the bearing extremely rigid so that even when load is applied to the bearing, radial or axial shaft displacement does not occur. Thus, the natural frequency of the shaft is increased, which is suitable for high speeds.

Preload is also used to prevent or suppress shaft runout, vibration, and noise; improve running accuracy and locating accuracy; reduce smearing, and regulate rolling element rotation. Also, for thrust ball and roller bearings mounted on horizontal shafts, preloading keeps the rolling elements in proper alignment.

The most common method of preloading is to apply an axial load to two duplex bearings so that the inner and outer rings are displaced axially in relation to each other. This preloading method is divided into fixed position preload and constant pressure preload.

8.4.2 Preloading methods and amounts

The basic pattern, purpose and characteristics of bearing preloads are shown in Table 8.11. The definite position preload is effective for positioning the two bearings and also for increasing the rigidity. Due to the use of a spring for the constant pressure preload, the preload force can be kept constant, even when the distance between the two bearings fluctuates under the influence of operating heat and load.

Also, the standard preload for the paired angular contact ball bearings is shown in Table 8.12. Light and normal preload is applied to prevent general vibration, and medium and heavy preload is applied especially when rigidity is required.

8.4.3 Preload and rigidity

The increased rigidity effect preloading has on bearings is shown in Fig. 8.5. When the offset inner rings of the two paired angular contact ball bearings are pressed together, each inner ring is displaced axially by the amount δ_o and is thus given a preload, F_o, in the direction shown. Under this condition, when external axial load F_a is applied, bearing I will have an increased displacement by the amount δa and bearing II's displacement will decrease. At this time the loads applied to bearing I and II are F_i and F_{ii}, respectively.

Under the condition of no preload, bearing I will be displaced by the amount δ_b when axial load F_a is applied. Since the amount of displacement, δ_a, is less than δ_b, it indicates a higher rigidity for δ_a.

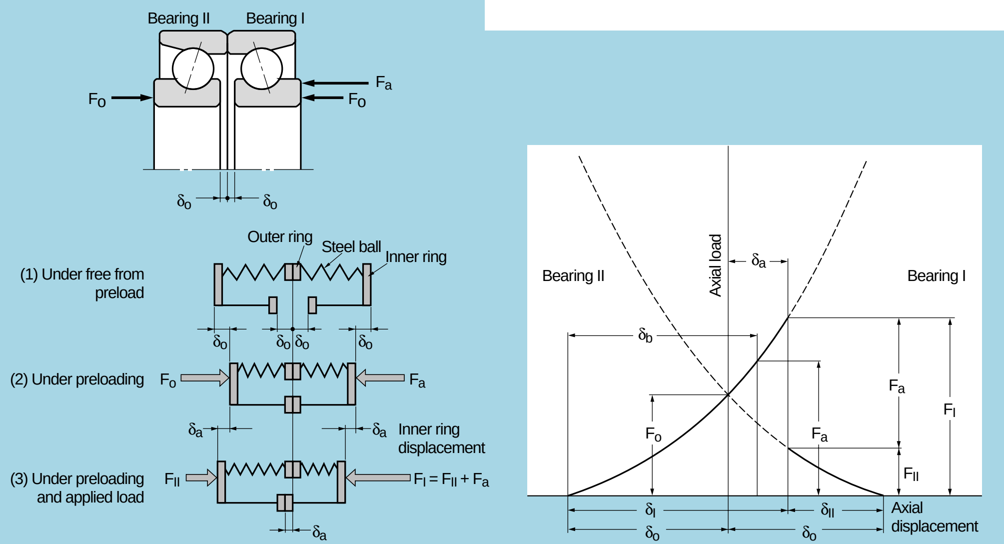
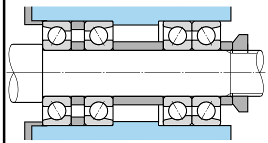
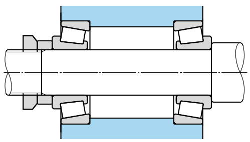
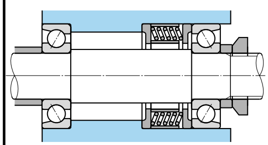
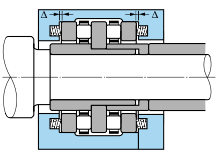


Fig. 8.5 Fixed position preload versus axial displacement

Table 8.11 Preloading methods and characteristics

Method	Basic pattern	Applicable bearings	Object	Characteristics	Applications
Fixed position preload		Precision angular contact ball bearings	Maintaining accuracy of rotating shaft, preventing vibration increasing rigidity	Preload is accomplished by a predetermined offset of the rings or by using spacers. For the standard preload see Table 8.12	Grinding machines, lathes, milling machines, measuring instruments
		Tapered roller bearings, thrust ball bearings angular contact ball bearings	Increasing bearing rigidity	Preload is accomplished by adjusting a threaded screw. The amount of preload is set by measuring the starting torque or axial displacement. Relationship between the starting torque M and preload T is approximately given by the following formulas: for duplex angular contact ball bearings: $M = \dfrac{d_p \bullet T}{330 \sim 430} \quad N \bullet mm^{1)}$ for duplex tapered roller bearings: $M = \dfrac{d_p^{0.8} \bullet T}{54 \sim 107} \quad N \bullet mm^{1)}$	Lathes, milling machines, differential gears of automotives, printing machines, wheel axles
Constant pressure preload		Angular contact ball bearings, deep groove ball bearings, precision tapered roller bearings	Maintaining accuracy and preventing vibration and noise with a constant amount of preload without being affected by loads or temperature	Preloading is accomplished by using coil or belleville springs. Recommended preloads are as follows: for deep groove ball bearings: (4 to 8) *d* N for angular contact ball bearings: see Table 8.12	Internal grinding machines, electric motors, high speed shafts in small machines, tension reels
		Tapered roller bearings with steep angle, spherical roller thrust bearings, thrust ball bearings	Preventing smearing on raceway of non-loaded side under axial loads	Preload is accomplished by using coil or belleville springs. Recommended preloads are as follows: for thrust ball bearings: $T = 0.42(n \bullet C_{oa})^{1.9} \times 10^{-13} \quad N^{2)}$ $T = 0.00083\, C_{oa} \quad N^{2)}$ whichever is greater for spherical roller thrust bearings: $T = 0.025\, C_{oa}^{0.8} \quad N^{2)}$	Rolling mills, extruding machines

Note: In the above formulas, d_p=pitch diameter of bearing, mm; d_p=(Bore+Outside dia)/2; T=preload, N; d=bearing bore, mm; n=number of revolutions, r/min; C_{oa}=basic static axial load rating, N

The starting torque *M* however, is greatly influenced by lubricants and a period of run-in time.

Table 8.12 Standard preloads for angular contact ball bearings Units: (N)

Nominal bore diameter *d* mm		Bearing series															
		78C				79C, HSB9C				70C, BNT0, HSBOC				72C, BNT2			
over	incl.	Light	Normal	Medium	Heavy	Light	Normal	Medium	Heavy	Light	Normal	Medium	Heavy	Light	Normal	Medium	Heavy
—	12	—	—	—	—	—	—	—	—	20	30	100	150	20	50	100	200
12	18	—	—	—	—	—	—	—	—	20	30	100	200	20	50	150	300
18	32	10	30	80	150	20	50	100	200	30	80	150	300	50	100	300	500
32	40	10	30	80	150	30	80	200	300	50	150	300	600	80	200	500	800
40	50	20	50	100	200	40	100	250	500	50	150	300	700	100	300	600	1000
50	65	30	100	200	400	50	120	300	600	100	200	500	1000	150	400	800	1500
65	80	30	100	200	400	80	200	400	800	100	300	700	1500	200	500	1000	2000
80	90	50	150	300	600	100	250	500	1000	150	400	1000	2000	300	700	1500	3000
90	95	50	150	300	600	100	250	500	1000	150	400	1000	2000	300	700	2000	4000
95	100	50	150	300	600	120	300	700	1500	150	400	1000	2000	300	700	2000	4000
100	105	50	150	300	600	120	300	700	1500	200	600	1500	2500	400	1000	2500	5000
105	110	80	200	500	1000	120	300	700	1500	200	600	1500	2500	400	1000	2500	5000
110	120	80	200	500	1000	150	400	900	2000	200	600	1500	2500	400	1000	2500	5000
120	140	100	300	600	1300	200	500	1000	2500	300	800	2000	4000	500	1500	3000	6000
140	150	150	400	800	1500	250	700	1500	3000	300	800	2000	4000	500	1500	3000	6000
150	160	150	400	800	1500	250	700	1500	3000	500	1000	2500	6000	700	2000	4500	8000
160	170	150	500	1000	2000	250	700	1500	3000	500	1000	2500	6000	700	2000	4500	8000
170	180	150	500	1000	2000	300	900	2000	4000	500	1000	2500	6000	700	2000	4500	8000
180	190	200	600	1300	2500	300	900	2000	4000	600	1500	3500	7000	800	2500	5000	10000
190	200	200	600	1300	2500	500	1300	3000	6000	600	1500	3500	7000	800	2500	5000	10000

Note: Symbols /GL, /GN, /GM and GH are suffixes on NTN bearing part numbers indicating Light, Normal, Medium and Heavy preloads, respectively.

9. Limiting Speed

As bearing speed increases, the temperature of the bearing also increases due to friction heat generated in the bearing interior. If the temperature continues to rise and exceeds certain limits, the efficiency of the lubricant drastically decreases, and the bearing can no longer continue to operate in a stable manner. Therefore, the maximum speed at which it is possible for the bearing to continuously operate without the generation of excessive heat beyond specified limits, is called the limiting speed (r/min).

The limiting speed of a bearing depends on the type of bearing, bearing dimensions, type of cage, load, lubricating conditions, and cooling conditions.

The limiting speeds listed in the bearing tables for grease and oil lubrication are for standard NTN bearings under normal operating conditions, correctly installed, using the suitable lubricants with adequate supply and proper maintenance. Moreover, these values are based on normal load conditions ($P \leq 0.09C$, $F_a/F_r \leq 0.3$). For ball bearings with contact seals (LLU type), the limiting speed is determined by the peripheral lip speed of the seal.

For bearings to be used under heavier than normal load conditions, the limiting speed values listed in the bearing tables must be multiplied by an adjustment factor. The adjustment factors f_L and f_c are given in Figs. 9.1 and 9.2.

Also when radial bearings are mounted on vertical shafts, lubricant retentions and cage guidance are not favorable compare to horizontal shaft mounting. Therefore, the limiting speed should be reduced to approximately 80% of the listed speed.

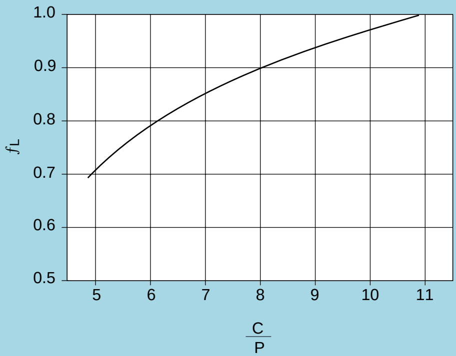


Fig. 9.1 Value of adjustment factor f_L depends on bearing load

For speeds other than those mentioned above, and for which data is incomplete, please consult NTN.

It is possible to operate precision bearings with high speed specification cages at speeds higher than those listed in the bearing tables, if special precautions are taken. These precautions should include the use of forced oil circulation methods such as oil jet or oil mist lubrication.

Under such high speed operating conditions, when special care is taken, the standard limiting speeds given in the bearing tables can be adjusted upward. The maximum speed adjustment values, f_B, by which the bearing table speed can be multiplied, are shown in Table 9.1. However, for any application requiring speeds greater than the standard limiting speed, please consult NTN.

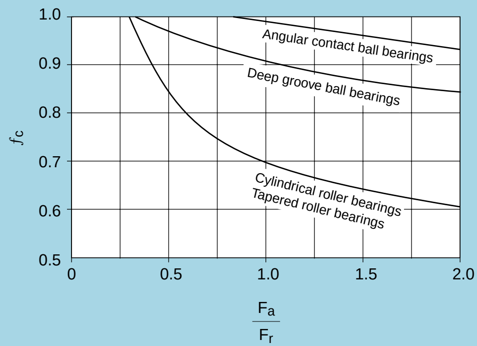


Fig. 9.2 Value of adjustment factor f_c depends on combined load

Table 9.1 Adjustment factor, f_B, for Limiting Speeds

Type of bearing	Adjustment factor f_B
Deep groove ball bearings	3.0
Angular contact ball bearings	2.0
Cylindrical roller bearings	2.5
Tapered roller bearings	2.0

10. Friction and Temperature Rise

10.1 Friction

One of the main functions required of a bearing is that it must have low friction. Under normal operating conditions rolling bearings have a much smaller friction coefficient than the slide bearings, especially starting friction.

The friction coefficient for rolling bearings is calculated on the basis of the bearing bore diameters and is expressed by formula (10.1).

$$\mu = \frac{2M}{Pd} \cdots\cdots\cdots\cdots\cdots\cdots (10.1)$$

where,

μ : Friction coefficient
M : Friction moment, N•mm
P : Load, N
d : Bearing bore diameter, mm

Although the dynamic friction coefficient for rolling bearings varies with the type of bearings, load, lubrication, speed and other factors; for normal operating conditions, the approximate friction coefficients for various bearing types are listed in Table 10.1.

Table 10.1 Friction coefficient for bearings

Bearing type	Coefficient $\times 10^{-3}$
Deep groove ball bearings	1.0~1.5
Angular contact ball bearings	1.2~1.8
Self-aligning ball bearings	0.8~1.2
Cylindrical roller bearings	1.0~1.5
Needle roller bearings	2.0~3.0
Tapered roller bearings	1.7~2.5
Spherical roller bearings	2.0~2.5
Thrust ball bearings	1.0~1.5
Trust roller bearings	2.0~3.0

10.2 Temperature rise

Almost all friction loss in a bearing is transformed into heat within the bearing itself and causes the temperature of the bearing to rise. The amount of thermal generation caused by friction moment can be calculated using formula (10.2).

$$Q = 0.105 \times 10^{-6} M \bullet n \cdots\cdots\cdots\cdots\cdots (10.2)$$

where,

Q : Thermal value kW
M : Friction moment N•mm
n : Rotational speed r/min

Bearing operating temperature is determined by the equilibrium or balance between the amount of heat generated by the bearing and the amount of heat conducted away from the bearing. In most cases the temperature rises sharply during initial operation, then increases slowly until it reaches a stable condition and then remains constant. The time it takes to reach this stable state will vary according to the amount of heat generated, the heat absorbing capacity of the housing and surrounding parts, the amount of cooling surface, amount of lubricating oil, and the surrounding ambient temperature. If the temperature continues to rise and does not become constant, it must be assumed that there is some improper function.

Excessive bearing heat can be caused by: moment load, insufficient internal clearance, excessive preload, too little or too much lubricant, foreign matter in the bearing, or by heat generated at the sealing device.

11. Lubrication

11.1 Lubrication of rolling bearings

The purpose of bearing lubrication is to prevent direct metallic contact between the various rolling and sliding elements. This is accomplished through the formation of a thin oil (or grease) film on the contact surfaces. However, for rolling bearings, lubrication has the following advantages.

(1) Friction and wear reduction
(2) Friction heat dissipation
(3) Prolonged bearing life
(4) Prevention of rust
(5) Protection against harmful elements

In order to achieve the above effects, the most effective lubrication method for the operating conditions must be selected. Also, a good quality, reliable lubricant must be selected. In addition, an effectively designed sealing system prevents the intrusion of damaging elements (dust, water, etc.) into the bearing interior, removes dust and other impurities from the lubricant, and prevents the lubricant from leaking from the bearing.

Almost all rolling bearings use either grease or oil lubrication methods, but in some special applications, a solid lubricant such as molybdenum disulfide or graphite may be used.

11.2 Grease lubrication

Grease type lubricants are relatively easy to handle and require only the simplest sealing devices—for these reasons, grease is the most widely used lubricant for rolling bearings.

11.2.1 Type and characteristics of grease

Lubricating grease are composed of either a mineral oil base or a synthetic oil base. To this base a thickener and other additives are added. The properties of all greases are mainly determined by the kind of base oil used by the combination of thickening agent and various additives.

Standard greases and their characteristics are listed in Table 11.1. As performance characteristics of even the same type of grease will vary widely from brand to brand, it is best to check the manufacturers' data when selecting a grease.

Table 11.1 Types and characteristics of greases

Name of grease	Lithium grease			Sodium grease (Fiber grease)	Calcium grease (Cup grease)
Thickener	Li soap			Na soap	Ca soap
Base oil	Mineral oil	Diester oil	Silicone oil	Mineral oil	Minera oil
Dropping point °C	170~190	170~190	200~250	150~180	80~90
Applicable Tempe-rature range °C	–30~+130	–50~+130	–50~+160	–20~+130	–20~+70
Mechanical properties	Excellent	Good	Good	Excellent or Good	Good or Impossible
Pressure resistance	Good	Good	Impossible	Good	Good or Impossible
Water resistance	Good	Good	Good	Good or Impossible	Good
Applications	The widest range of application Grease generally used in roller bearings	Excellent in low temperature and wear characterist-stics	Suitable for high and low tempera-tures Unsuitable for heavy load use because of low oil film strength	Some of the grease is emulsified when mixed in water Relatively excellent high temperature resistance	Excellent in water resistance, but in-ferior in heat resis-tance Low speed and heavy load use

11.2.2 Base oil

Natural mineral oil or synthetic oils such as diester oil, silicone oil and fluorocarbon oil are used as grease base oils.

Mainly, the properties of any grease is determined by the properties of the base oil. Generally, greases with a low viscosity base oil are best suited for low temperatures and high speeds; while greases made from high viscosity base oils are best suited for heavy loads.

11.2.3 Thickening agents

Thickening agents are compounded with base oils to maintain the semi-solid state of the grease. Thickening agents consist of two types of bases, metallic soaps and non-soaps. Metallic soap thickeners include: lithium, sodium, calcium, etc.

Non-soap base thickeners are divided into two groups; inorganic (silica gel, bentonite, etc.) and organic (poly-urea, fluorocarbon, etc.)

The various special characteristics of a grease, such as limiting temperature range, mechanical stability, water resistance, etc. depend largely on the type of thickening agent is used. For example, a sodium based grease is generally poor in water resistance properties, while greases with bentone, poly-urea and other non-metallic soaps as the thickening agent are generally superior in high temperature properties.

11.2.4 Additives

Various additives are added to greases to improve various properties and efficiency. For example, there are anti-oxidents, high-pressure additives (EP additives), rust preventives, and anti-corrosives.

For bearing subject to heavy loads and/or shock loads, a grease containing high-pressure additives should be used. For comparatively high operating temperatures or in applications where the grease cannot be replenished for long periods, a grease with an oxidation stabilizer is best to use.

11.2.5 Consistency

The consistency of a grease, i.e. the stiffness and liquidity, is expressed by a numerical index.

Calcium compound grease (Complex grease)	Sodium grease	Aluminum grease	Non-soap based grease (Non-soap grease)	
Ca compound soap	Ca+Na soap Ca+Li soap	Al soap	Bentone, Silica gel, Urea, Carbon Black	
Mineral oil	Mineral oil	Mineral oil	Mineral oil	Synthetic oil
200~280	150~180	70~90	250 or more	250 or more
–20~+150	–20~+120	–10~+80	–10~+130	–50~+200
Good	Excellent or Good	Good or Impossible	Good	Good
Good	Excellent or Good	Good	Good	Good
Good	Good or Impossible	Good	Good	Good
Some of the grease containing extreme pressures additives are suitable for heavy load use For general roller bearings	Excellent in pressure resistance and mechanical stability Suitable for bearings which receive vibrations	Excellent in stickiness (adhesiveness) Suitable for bearings which receive vibrations	These can be applied to the range from low to high temperatures. Excellent characteristics are obtained in heat and by suitably arranging the thickening agents and base oils Grease for general roller bearings.	

The NLGI values for this index indicate the relative softness of the grease; the larger the number, the stiffer the grease. The consistency of a grease is determined by the amount of thickening agent used and the viscosity of the base oil. For the lubrication of rolling bearings, greases with the NLGI consistency numbers of 1,2, and 3 are used.

General relationships between consistency and application of grease are shown in Table 11.2.

Table 11.2 Consistency of grease

NLGI Consis-tency No.	JIS (ASTM) Worked penetration	Applications
0	355 ~ 385	For centralized greasing use
1	310 ~ 340	For centralized greasing use
2	265 ~ 295	For general use and sealed bearing use
3	220 ~ 250	For general and high temperature use
4	175 ~ 205	For special use

11.2.6 Mixing of greases

When greases of different kinds are mixed together, the consistency of the greases will change (usually softer), the operating temperature range will be lowered, and other changes in characteristics will occur. As a general rule, greases with different bases oil, and greases with different thickener agents should never be mixed.

Also, greases of different brands should not be mixed because of the different additives they contain.

However, if different greases must be mixed, at least greases with the same base oil and thickening agent should be selected. But even when greases of the same base oil and thickening agent are mixed, the quality of the grease may still change due to the difference in additives.

For this reason, changes in consistency and other qualities should be checked before being applied.

11.2.7 Amount of grease

The amount of grease used in any given situation will depend on many factors relating to the size and shape of the housing, space limitations, bearing's rotating speed and type of grease used.

As a general rule, housings and bearings should be only filled from 30% to 60% of their capacities.

Where speeds are high and temperature rises need to be kept to a minimum, a reduced amount of grease should be used. Excessive amount of grease cause temperature rise which in turn causes the grease to soften and may allow leakage. With excessive grease fills oxidation and deterioration may cause lubricating efficiency to be lowered.

11.2.8 Replenishment

As the lubricating efficiency of grease declines with the passage of time, fresh grease must be re-supplied at proper intervals. The replenishment time interval depends on the type of bearing, dimensions, bearing's rotating speed, bearing temperature, and type of grease.

An easy reference chart for calculating grease replenishment intervals is shown in Fig. 11.1

This chart indicates the replenishment interval for standard rolling bearing grease when used under normal operating conditions.

As operating temperatures increase, the grease re-supply interval should be shortened accordingly.

Generally, for every 10°C increase in bearing temperature above 80°C, the relubrication period is reduced by exponent "1/1.5".

(Example)

Find the grease relubrication time limit for deep groove ball bearing 6206, with a radial load of 2.0 kN operating at 3,600 r/min.

C_r/P_r=19.5/2.0 kN=9.8, from Fig. 9.1 the adjusted load, f_L, is 0.96.

From the bearing tables, the allowable speed for bearing 6206 is 11,000 r/min and the numbers of revolutions permissible at a radial load of 2.0 kN are

$$n_o = 0.96 \times 11000 = 10560 \text{ r/min} \cdots\cdots\cdots\cdots\cdots \text{A}$$

therefore,

$$\frac{n_o}{n} = \frac{10560}{3600} = 2.93 \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \text{B}$$

Using the chart in Fig. 11.1, find the point corresponding to bore diameter d=30 (from bearing table) on the vertical line for radial ball bearings. Draw a straight horizontal line to vertical line I. Then, draw a straight line from that point (A in example) to the point on line II which corresponds to the n_o/n value (2.93 in example). The point, C, where this line intersects vertical line III indicates the relubrication interval h. In this case the life of the grease is approximately 5,500 hours.

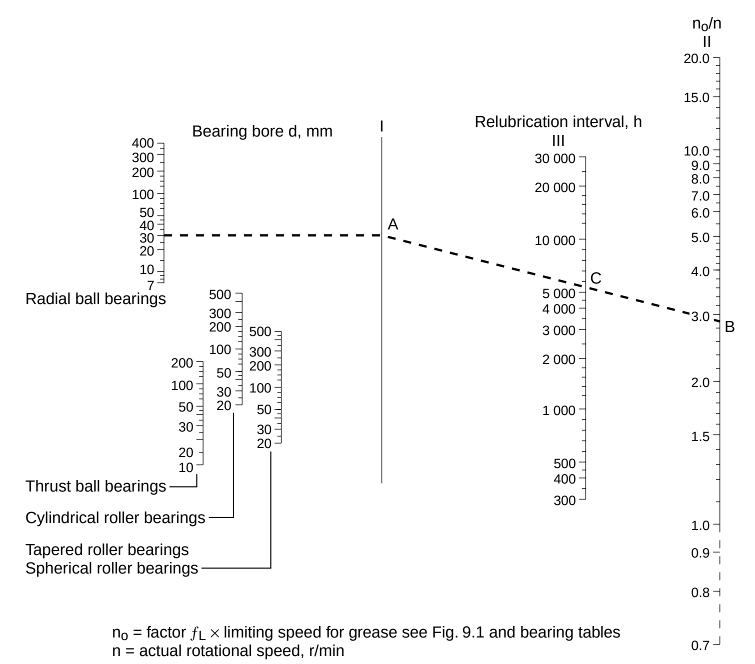


Fig. 11.1 Diagram for relubrication interval of greasing

11.3 Oil lubrication

Generally, oil lubrication is better suited for high speed and high temperature applications than grease lubrication. Oil lubrication is especially effective for those application requiring the bearing generated heat (or heat applied to the bearing from other sources) to be carried away from the bearing and dissipated to the outside.

11.3.1 Oil lubrication methods

1) Oil bath
Oil lubrication is the most commonly used method for low to moderate speed applications. However, the most important aspect of this lubrication method is oil quantity control.
For most horizontal shaft applications, the oil level is normally maintained at approximately the center of the lowest rolling elements when the bearing is at rest.
With this method, it is important that the housing design does not permit wide fluctuations in the oil level, and that an oil gauge be fitted to allow easy inspection of the oil level with the bearing at rest or in motion (Fig. 11.2).

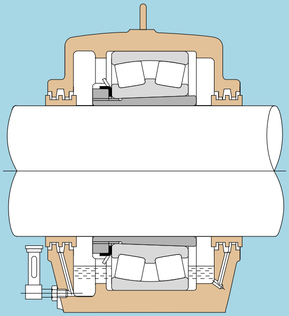


Fig. 11.2 Oil bath lubrication

For vertical shafts at low speeds, the oil level should be up to 50% to 80% submergence of the rolling elements. However, for high speeds or for bearings used in pairs or multiple rows, other lubrication methods, such as drip lubrication or circulation lubrication, should be used (see below).

2) Oil splash
In this method the bearing is not directly submerged in the oil, but instead, an impeller or similar device is mounted on the shaft and the impeller picks up the oil and sprays it onto the bearing. This splash method of lubrication can be utilized for considerably high speeds.
As shown in the vertical shaft example in Fig. 11.3, a tapered rotor is attached to the shaft just below the bearing. The lower end of this rotor is submerged in the oil, and as the rotor rotates, the oil climbs up the surface of the rotor and is thrown as spray onto the bearing.

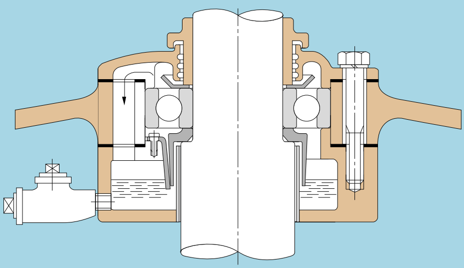


Fig. 11.3 Oil spray lubrication

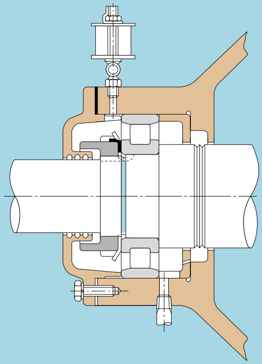


Fig. 11.4 Drip lubrication

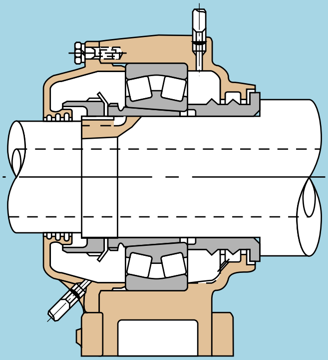


Fig. 11.5 Circulating lubrication (Horizontal shaft)

3) Drip lubrication
Used for comparatively high speeds and for light to medium load applications. an oiler is mounted on the housing above the bearing and allows oil to drip down on the bearing, striking the rotating parts, turning the oil to mist (Fig. 11.4). Another method allows only small amounts of oil to pass through the bearing at a time. The amount of oil used varies with the type of bearing and its dimensions, but, in most cases, the rate is a few drops per minute.

4) Circulating lubrication
Used for bearing cooling applications or for automatic oil supply systems in which the oil supply is centrally located.
The principal advantage of this method is that oil cooling devices and filters to maintain oil purity can be installed within the system.
With this method however, it is important that the circulating oil definitely be evacuated from the bearing chamber after it has passed through the bearing. For this reason, the oil inlets and outlets must be provided on opposite sides of the bearing, the drain port must be as large as possible, or the oil must be forcibly evacuated from the chamber (Fig. 11.5). Fig. 11.6 illustrates a circulating lubrication method for vertical shafts using screw threads.

5) Disc lubrication
In this method, a partially submerged disc rotates at high speed pulling the oil up by centrifugal force to an oil reservoir located in the upper part of the housing. The oil then drains down through the bearing. Disc lubrication is only effective for high speed operations, such as supercharger or blower bearing lubrication (Fig. 11.7).

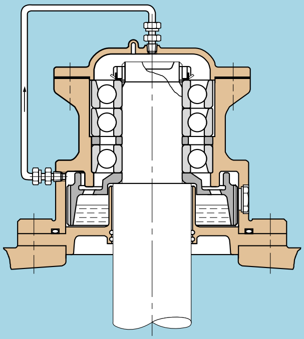


Fig. 11.6 Circulating lubrication (Vertical shaft)

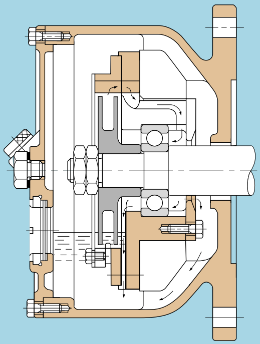


Fig. 11.7 Disc lubrication

6) Oil mist lubrication
Using pressurized air, the lubrication oil is atomized before it passes through the bearing. This method is especially suited for high speed lubrication due to the very low lubricant resistance. As shown in Fig. 11.8, one lubricating device can lubricate several bearings at one time. Also, oil consumption is very low.

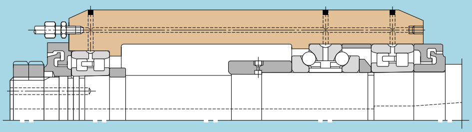


Fig. 11.8 Oil mist lubrication

7) Air-oil lubrication
With the air-oil lubrication system, an exact measured minimum required amount of lubricating oil is fed to each bearing at correct intervals. As shown in Fig. 11.9, this measured amount of oil is continuously sent under pressure to the nozzle.
A fresh lubricating oil is constantly being sent to the bearing, there is no oil deterioration, and with the cooling effect of the compressed air, bearing temperature rise can be kept to a minimum. The quantity of oil required to lubricate the bearing is also very small, and this infinitesimal amount of oil fed to the bearing does not pollute the surrounding environment.
Note: This air-oil lubrication unit is now available from NTN.

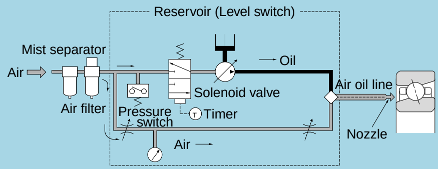


Fig. 11.9 Air-Oil lubrication supply system

8) Oil jet lubrication
This method lubricates the bearing by injecting the lubricating oil under pressure directly into the side of the bearing. This is the most reliable lubricating system for severe (high temperature, high speed, etc.) operating conditions.
This is used for lubricating the main bearings of jet engines and gas turbines, and all types of high speed equipment. This system can be used in practice for *dn* values up to approximately 2.5×10^6.
Usually the oil lubricant is injected into the bearing by a nozzle adjacent to the bearing, however in some applications, oil holes are provided in the shaft, and the oil is injected into the bearing by centrifugal force as the shaft rotates.

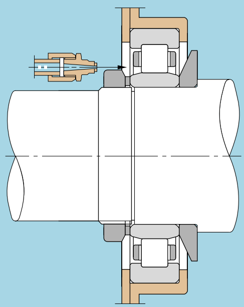


Fig. 11.10 Oil jet lubrication

11.3.2 Lubricating oil

Under normal operating conditions, spindle oil, machine oil, turbine oil and other minerals are widely used for the lubrication of rolling bearings. However, for temperatures above 150°C or below –30°C, synthetic oils such as diester, silicone and fluorosilicone are used.

For lubricating oils, viscosity of the oil is one of the most important properties and determines the oil's lubricating efficiency. If the viscosity is too low, the oil film will not be sufficiently formed, and it will damage the load carrying surface of the bearing. On the other hand, if the viscosity is too high, the viscosity resistance will also be high and cause temperature increases and friction loss. In general, for higher speed, a lower viscosity oil should be used, and for heavy loads, a higher viscosity oil should be used.

In regard to operating temperature and bearing lubrication, Table 11.3 lists the minimum required viscosity for various bearings. Fig. 11.11 is a lubricating oil viscosity-temperature comparison chart is used in the selection of lubricating oil. It shows which oil would have the appropriate viscosity at a given temperature. For lubricating oil viscosity selection standards relating to bearing operating conditions, see Table 11.4.

Table 11.3 Minimum viscosity of lubricating oil for bearings

Bearing type	Dynamic viscosity mm^2/s
Ball bearings, cylindrical roller bearings, needle roller bearings	13
Spherical roller bearings, tapered roller bearings, thrust needle roller bearings	20
Spherical roller thrust bearings	30

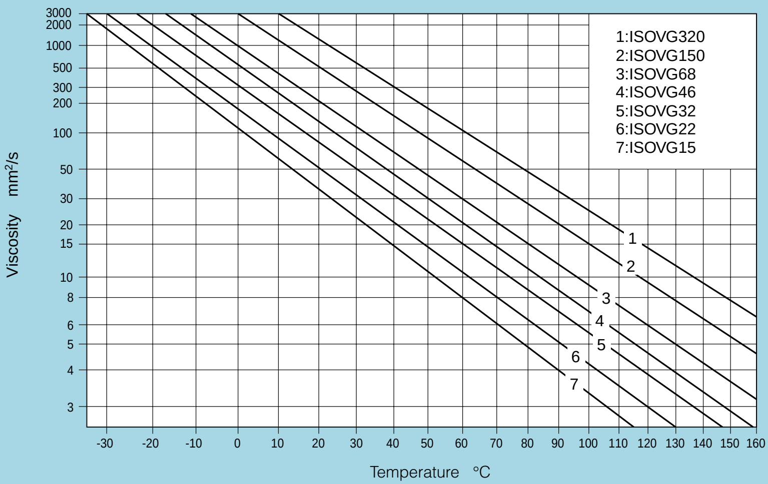


Fig. 11.11 Relation between viscosity and temperature

Table 11.4 Selection standards for lubricating oils

Operating temperature of bearings °C	*dn*–value	Viscosity grade of lubricating oil		Bearing type
		Ordinary load	Heavy or Impact load	
–30 to 0	Up to the allowable revolution	22 32	46	All type
0 to 60	Up to 15,000	46 68	100	All type
	15,000 to 80,000	32 46	68	All type
	80,000 to 150,000	22 32	32	Except thrust ball bearings
	150,000 to 500,000	10	22 32	Single row radial ball bearings, cylindrical roller bearings
60 to 100	Up to 15,000	150	220	All type
	15,000 to 80,000	100	150	All type
	80,000 to 150,000	68	100 150	Except thrust ball bearings
	150,000 to 500,000	32	68	Single row radial ball bearings, cylindrical roller bearings
100 to 150	Up to the allowable revolution	320		All type
0 to 60		46 68		Spherical roller bearings
60 to 100		150		

Notes: 1. In case of oil drip or circulating lubrication
2. In case the usage conditions' range is not listed in this table, please refer to NTN.

11.3.3 Oil quality

In forced oil lubrication systems, the heat radiated away by housing and surrounding parts plus the heat carried away by the lubricating oil is approximately equal to the amount of heat generated by the bearing and other sources.

For standard housing applications, the quantity of oil required can be found by formula (11.1).

$$Q = K \bullet q \cdots\cdots\cdots\cdots\cdots\cdots (11.1)$$

where,

Q : Quantity of oil for one bearing cm3/min
K : Allowable oil temperature rise factor (Table 11.5)
q : Minimum oil quantity cm^3/min (From chart)

Because the amount of heat radiated will vary according to the shape of the housing, for actual operation it is advisable that the quantity of oil calculated by formula (11.1) be multiplied by a factor of 1.5 to 2.0. Then, the amount of oil can be adjusted to correspond to the actual machine operating conditions. If it is assumed for calculation purposes that no heat is radiated by the housing and that all bearing heat is carried away by the oil, then the value for shaft diameter, d, (second vertical line from right in Fig. 11.12) becomes zero, regardless of the actual shaft diameter.

Table 11.5 Factor *K*

Temperature rise, °C	*K*
10	1.5
15	1
20	0.75
25	0.6

(Example)

For tapered roller bearing 30220U mounted on a flywheel shaft with a radial load of 9.5 kN, operating at 1,800 rpm; what is the amount of lubricating oil required to keep the bearing temperature rise below 15°C?

d=100 mm, dn=100×1,800=18×10^4 mm r/min

from Fig. 11.12, q=180 cm^3/min.

Assume the bearing temperature is approximately equal to the outlet oil temperature, from Table 11.5, since K=1, Q=1×180=180 cm3/min.

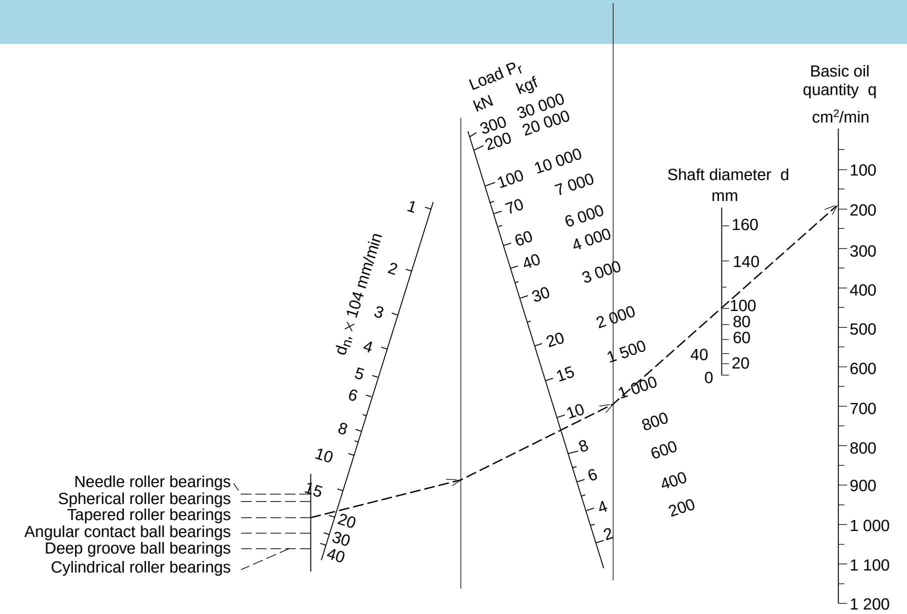


Fig. 11.12 Guidance for oil quantity

11.3.4 Relubrication interval

The interval of oil change depend on operating conditions, oil quantity, and type of oil used. A general standard for oil bath lubrication is that if the operating temperature is below 50°C, the oil should be replaced once a year. For higher operating temperatures, 80°C to 100°C for example, the oil should be replaced at least every three months.

In critical applications, it is advisable that the lubricating efficiency and oil deterioration be checked at regular intervals in order to determine when the oil should be replaced.

12. Sealing Devices

Bearing seals have two main functions: 1) to prevent lubricant from leaking out and 2) to prevent dust, water and other contaminants from entering the bearing. When selecting a seal the following factors need to be taken into consideration: the type of lubricant (oil or grease), seal sliding speed, shaft fitting errors, space limitations, seal friction and resultant heat, and cost.

Sealing devices for rolling bearings fall into two main classifications: contact and non-contact types.

12.1 Non-contact seals

Non-contact seals utilize a small clearance between the seal and the sealing surface; therefore, there is no wear, and friction is negligible.

Consequently, very little frictional heat is generated making non-contact seals very suitable for high speed applications.

As shown in Fig. 12.1, non-contact seals can have the simplest of designs. With its small radial clearance, this type of seal is best suited for grease lubrication, and for use in dry, relatively dust free environments.

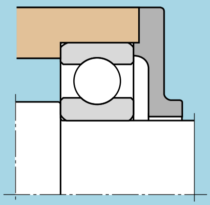


Fig. 12.1 Clearance seal

When several concentric oil grooves (Fig. 12.2) are provided on the shaft or housing, the sealing effect can be greatly improved. If grease is filled in the grooves, the intrusion of dust, etc. can be prevented.

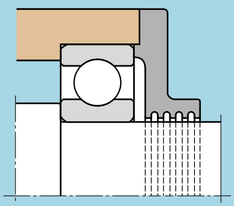


Fig. 12.2 Oil groove seal

For oil lubrication, if helical concentric oil grooves are provided in the direction opposite to the shaft rotation (horizontal shafts only), lubricating oil that flows out along the shaft can be returned to the inside of the housing (see Fig. 12.3). The same sealing effect can be achieved by providing helical grooves on the circumference of the shaft.

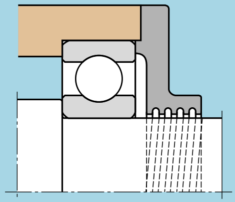


Fig. 12.3 Helical oil groove seal

Labyrinth seals employ a multistage labyrinth design which elongates the passage, thus improving the sealing effectiveness. Labyrinth seals are used mainly for grease lubrication, and if grease is filled in the labyrinth, protection efficiency (or capacity) against the entrance of dust and water into the bearing can be enhanced.

The axial labyrinth passage seal shown in Fig. 12.4 is used on one-piece housings and the radial seal shown in Fig. 12.5 is for use with split housings.

In applications where the shaft is set inclined, the labyrinth passage is slanted so as to prevent contact between the shaft and housing projections of the seal (Fig. 12.6).

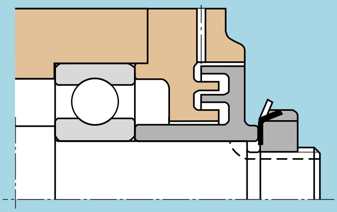


Fig. 12.4 Axial labyrinth seal

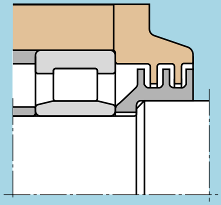


Fig. 12.5 Radial labyrinth seal

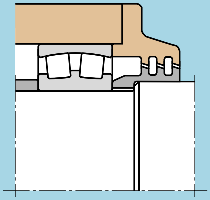


Fig. 12.6 Aligning labyrinth seal

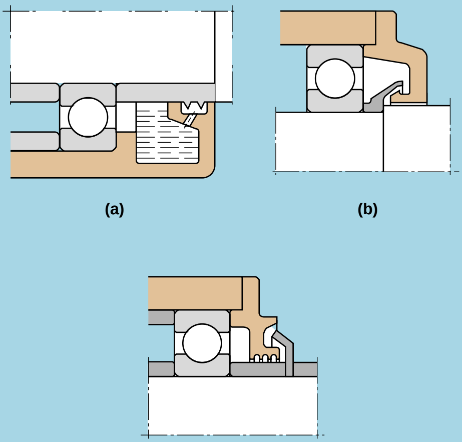
Axial and radial clearance values for labyrinth seals are given in Table 12.1.

Table 12.1 Clearance for labyrinth seals

Shaft diameter mm	Radial clearance on diameter mm	Axial clearance mm
~50 50~200	0.20~0.40 0.50~1.00	1~2 3~5

For oil lubrication, if projections are provided on the sleeve as shown in Fig. 12.7 (a), oil that flows out along the sleeve will be thrown off by centrifugal force and returned through ducts. In the example shown in Fig. 12.7 (b) oil leakage is prevented by the centrifugal force of the slinger.

Also, in Fig. 12.7 (c), a slinger can be mounted on the outside to prevent dust and other solid contaminants from entering.



(a) (b) (c)

Fig. 12.7 Slinger

12.2 Contact seals

Contact seals accomplish their sealing action through the constant pressure of a resilient part of the seal on the sealing surface. Contact seals are generally far superior to non-contact seals in sealing efficiency, although their friction torque and temperature rise coefficients are somewhat higher.

The simplest of all contact seals are felt seals. Used primarily for grease lubrication (Fig. 12.8), felt seals work very well for keeping out fine dust, but are subject to oil permeation and leakage to some extent. Therefore, the Z type rubber seal shown in Fig. 12.9 and GS type shown in Fig. 12.10, have been used more widely.

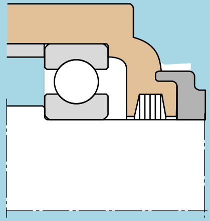


Fig. 12.8 Felt seal

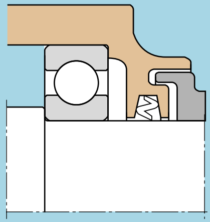


Fig. 12.9 Z Grease seal

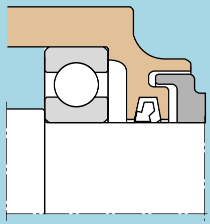


Fig. 12.10 GS Grease seal

Oil seals are used very widely and commonly, so their shapes and dimensions are standardized under JIS B2402. Using a ring shaped coil spring in the lip to exert optimum contact pressure and also to allow the seal lip to follow the shaft runout, gives this type of seal excellent sealing efficiency.

The direction of the sealing action changes depending on which direction the lip faces. If the lip faces outward (Fig. 12.11 (a)), it will protect against dust, water and other contaminants entering the bearing. If the lip faces inward (Fig. 12.11 (b)), it can prevent lubricant leakage from the housing.

For needle roller bearings, NTN's special seals are now available (see page E-82). Depending upon usage conditions, the seal lip may be made of nitrile rubber, silicone rubber, fluorinated rubber or PTFE resin etc.

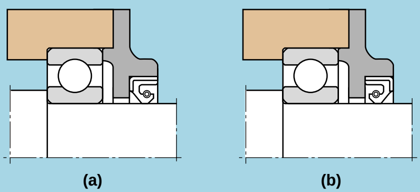


Fig. 12.11 Oil seal

V-ring seals shown in Fig. 12.12 are used for either oil or grease lubrication. As only the edge of the V-ring makes contact with the comparatively large seal lip, it is able to follow any side runout.

V-ring seals are very suitable for high speeds as the V-ring contacts the seal lip with only light contact pressure. For lip sliding speeds in excess of 12 m/s, the fit of the seal ring is lost and it needs to be held in place with a clamping band.

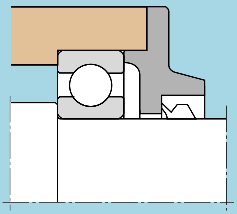


Fig. 12.12 V-Ring seal

These seals are made of elastic, high polymer material, and, depending on the type of material, they can be used for wide range of operational temperatures. The limiting operating temperature ranges for various materials are shown in Table 12.2.

Table 12.2 Permissible temperature of seals

Seal material		Permissible operating temperature range °C
Synthetic rubber	nitrile	–25 to 100
	acrylic	–15 to 160
	silicone	–70 to 230
	fluorinated	–30 to 220
PTFE synthetic resin		–50 to 220
Felt		–40 to 120

Allowable speeds for contact seals vary with the type of lubrication, operating temperature, roughness of the sealing contact surface, etc. A general reference chart showing allowable speeds for seal types is shown in Table 12.3.

Table 12.3 Allowable rubbing speed for seals

Type	Allowable speed, m/s
Felt	4
Grease seal	6
Oil seal, nitrile rubber	15
Oil seal, fluorinated rubber	32
V-ring	40

The general relationship between the shaft contact sealing surface roughness (R_a) and seal lip speed is shown in Table 12.4. In order to increase water resistance of the shaft, it should be heat treated or hard chrome plated, etc. The surface hardness of the shaft should be at least HRC40 or above, and if possible over HRC55.

Table 12.4 Surface roughness of shafts

Circumferential speed m/s		Surface roughness R_a
over	incl.	
	5	0.8a
5	10	0.4a
10		0.2a

12.3 Combination seals

Where operating conditions are especially severe (large amounts of water, dust, etc.), or in places where pollution caused by lubricant leakage cannot be tolerated; seals may be used in combination. Fig. 12.13 shows a combined labyrinth and oil groove slinger seal, and Fig. 12.14 shows a contact and non-contact seal combination.

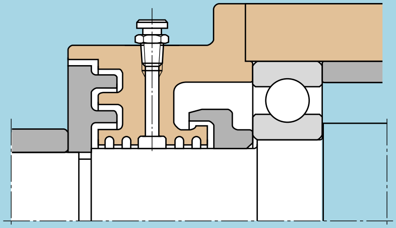


Fig. 12.13 Non-contact combination seal

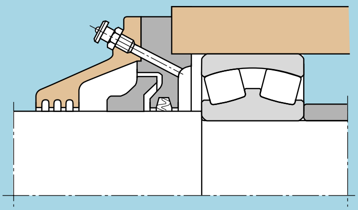


Fig. 12.14 Combined seal

13. Bearing Materials

13.1 Ring and rolling element materials

While the contact surfaces of the bearing rings and rolling elements are subjected to repeated heavy stress, they still must maintain high precision and rotational accuracy. To accomplish this, the rings and rolling elements must be made of a material that has high hardness, is resistant to rolling fatigue, is wear resistant and has good dimensional stability.

High carbon chromium bearing steel, which can be deep hardened by the so-called through hardening method, and case hardening steel with a hardened carburized outer layer are used for the rings and rolling elements of standard bearings. The hardness of the rings and rolling elements is usually on the order of HRC 58 to HRC 65.

The most widely used and most suitable materials for rolling bearings are high carbon steels. The chemical composition for JIS G 4805 standard high carbon chromium steels is shown in Table 13.1. The most commonly used of these steels, SUJ2, is equivalent to such steels as AISI 52100 (U.S.A.), DIN 100 C_r6 (West Germany), and GS 534A 99 (U.K.). For bearings with large cross section dimensions, SUJ3 or SUJ5 having good hardening properties are used.

For case hardening steel; chrome steel (SC_r), chrome molybdenum steel (SCM) and nickel chrome molybdenum steel (SNCM) are used; their chemical compositions for are shown in Table 13.2. Because of its combination of a hard surface layer which has been carburized and hardened to an appropriate depth, and a relatively pliable inner core, case hardening steel has excellent efficiency against shock load. NTN uses case hardening steel for almost all of tapered roller bearings.

The most common cause of fatigue cracking in bearings is the inclusion of non-metallic impurities in the material. By using clean materials, low in these non-metallic impurities, the rolling fatigue life of the bearing is lengthened. For all its bearings, NTN uses steel low in oxygen content and non-metallic impurities, and refined by a vacuum degassing process as well as outside hearth smelting.

For bearings requiring high reliability and long life, vacuum melted steel (CEVM) and electro-slag melted steel (ESR) which are even higher in purity are used. For information about bearings constructed of these materials, please consult NTN.

Table 13.1 High carbon chromium bearing steel

Specification	Symbol	Chemical composition %						
		C	Si	Mn	P	S	Cr	Mo
JIS G 4805	SUJ 2	0.95~1.10	0.15~0.35	0.50max.	0.025 max.	0.025 max.	1.30~1.60	—
	SUJ 3	0.95~1.10	0.40~0.70	0.90~1.15	0.025max.	0.025max.	0.90~1.20	—
	SUJ 4	0.95~1.10	0.15~0.35	0.50 max.	0.025max.	0.025max.	1.30~1.60	0.10~0.25
	SUJ 5	0.95~1.10	0.40~0.70	0.90~1.15	0.025max.	0.025max.	0.90~1.20	0.10~0.25

Table 13.2 Case hardening steel

Specification	Symbol	Chemical composition %							
		C	Si	Mn	P	S	Ni	Cr	Mo
JIS G 4104	SC_r420	0.18~0.23	0.15~0.35	0.60~0.85	0.030 max.	0.030max.	—	0.90~1.20	—
JIS G 4105	SCM420	0.18~0.23	0.15~0.35	0.60~0.85	0.030max.	0.030max.	—	0.90~1.20	0.15~0.30
JIS G 4103	SNCM420	0.17~0.23	0.15~0.35	0.40~0.70	0.030max.	0.030max.	1.60~2.00	0.40~0.65	0.15~0.30
	SNCM815	0.12~0.18	0.15~0.35	0.30~0.60	0.030max.	0.030max.	4.00~4.50	0.70~1.00	0.15~0.30

Table 13.3 High speed steel

Specification	Symbol	Chemical composition %								
		C	Si	Mn	P	S	Ni	Cr	Mo	V
AMS 6490	M50	0.77~0.85	0.25max.	0.35max.	0.015max.	0.015max.	0.15max.	3.75~4.25	4.00~4.50	0.90~1.10

For bearings operated in high temperatures, high speed steel (M50), is used. For applications requiring high corrosion resistance, stainless steel (SUS 440C) is used. The chemical composition for these steels is shown in Tables 13.3 and 13.4. For bearings whose raceway surfaces are induction hardened, machine structural carbon steel (S48C to S50C), and chrome molybdenum steel (SCM440) which has a relatively high carbon content are used (for chemical composition, see Table 13.5).

Table 13.4 Stainless steel

Specification	Symbol	Chemical composition %							
		C	Si	Mn	P	S	Cr	Ni	Mo
JIS G 4303	SUS440C	0.95~1.20	1.00max.	1.00max.	0.04max.	0.030max.	16.00~18.00	0.6max.	0.75max.

Table 13.5 Induction hardening steel

Specification	Symbol	Chemical composition %						
		C	Si	Mn	P	S	Cr	Mo
JIS G 4051	S48C	0.45~0.51	0.15~0.35	0.60~0.90	0.030max.	0.035max.	—	—
	S50C	0.47~0.51	0.15~0.35	0.60~0.90	0.030max.	0.035max.	—	—
JIS G 4105	SCM440	0.38~0.43	0.15~0.35	0.60~0.85	0.030max.	0.030max.	0.90~1.20	0.15~0.30

13.2 Cage materials

Bearing cage materials must have the strength to withstand rotational vibrations and shock loads. These materials must also have a low friction coefficient, be light weight, and be able to withstand bearing operating temperatures.

For small and medium sized bearings, pressed cages of cold or hot rolled sheet steel are used. However, depending on the application, brass sheet or stainless steel is also available. The chemical compositions are shown in Table 13.6.

For large bearings, machined cages of machine structural carbon steel (S30C) (Table 13.7) or high tensile cast brass (HBsCI) (Table 13.8) are widely used. However, spheroidal graphite cast iron or aluminum alloy cages are also used.

Injection molded plastic cages are now also widely used, and most are made from fiberglass reinforced heat resistant polyamide resin. Plastic cages are light in weight, corrosion resistant, and have excellent damping and sliding properties.

Table 13.6 Materials for pressed cage

Specification	Symbol	Chemical composition %				
		C	Si	Mn	P	S
BAS361	SPB2	0.13~0.20	0.04max.	0.25~0.60	0.030max.	0.030max.
JIS G 3141	SPCC	0.12max.	—	0.50max.	0.040max.	0.045max.
JIS G 3131	SPHC	0.15max.	—	0.60max.	0.050max.	0.050max.

Table 13.7 Materials for machined cage

Specification	Symbol	Chemical composition %				
		C	Si	Mn	P	S
JIS G 4051	S30C	0.27~0.33	0.15~0.35	0.60~0.90	0.030max.	0.035max.

Table 13.8 Materials for machined cage

Specification	Symbol	Chemical composition %								
		C	Si	Mn	P	S	Ni	Cr	Mo	V
JIS H 5102	HBsCI	55.0max.	Remains	1.5max.	0.5~1.5	0.5~1.5	1.0max.	1.0max.	0.4max.	0.1max.

14. Shaft and Housing Design

14.1 Fixing of bearings

When fixing a bearing in position on a shaft or in a housing, there are many instances where the interference fit alone is not enough to hold the bearing in place. Bearing ring must be fixed in place by various methods so they do not axially move when placed under load.

The most common method of fixing bearings in place is to hold the ring end face against the shaft or housing abutment by means of bolts or screws. Fig. 14.1 illustrates inner ring clamping methods, and Fig. 14.2 outer ring clamping methods. Fig. 14.3 and 14.4 show the use of snap ring methods which also make construction extremely simple.

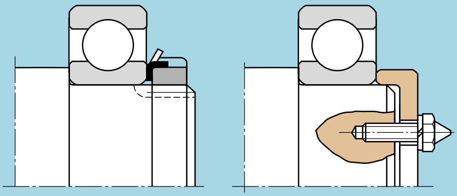


Fig. 14.1 Inner ring clamping

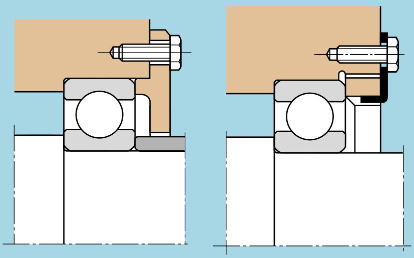


Fig. 14.2 Outer ring clamping

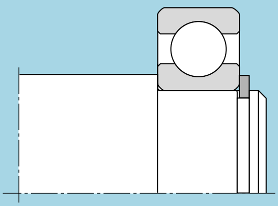


Fig. 14.3 Snap ring (Shaft)

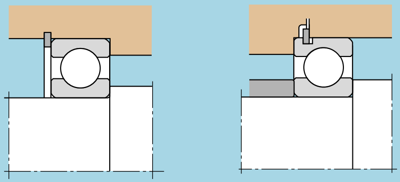


Fig. 14.4 Snap ring (Housing)

For bearings with tapered bores, examples of the use of adapters are shown in Fig. 14.5. When fitting bearings on non-stepped shafts, fixing the bearing axially depends on the friction between the sleeve and the shaft. Fig. 14.6 shows the use of withdrawal sleeves and clamping with nuts or end-plates on shaft ends. For installing tapered bore bearings directly on tapered shafts, the bearing is held in place by a split ring inserted in groove provided in the shaft, and tightened on the shaft by the split ring nut (Fig. 14.7).

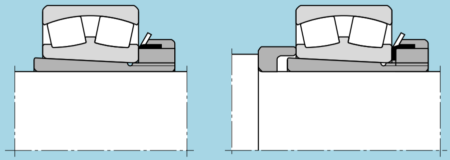


Fig. 14.5 Adapter sleeve mounting

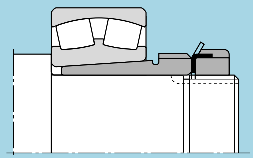


Fig. 14.6 Withdrawal sleeve mounting

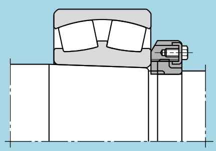


Fig. 14.7 Split ring mounting

14.2 Bearing fitting dimensions

The shaft and housing abutment height (*h*) should be larger than the bearings' maximum allowable chamfer dimensions ($r_{s\,max}$), and the abutment should be designed so that it directly contacts the flat part of the bearing end face. The fillet radius must be smaller than the bearing's minimum allowable chamfer dimension ($r_{s\,min}$) so that it does not interfere with bearing seating. Table 14.1 lists abutment height (*h*) and fillet radius (r_a). For bearings subjected to heavy axial loads, shaft abutments (*h*) should be higher than the values in the table.

In cases where a fillet radius (r_a) larger than the bearings' chamfer dimension is required to maximize shaft strength or to minimize stress concentration (Fig. 14.8a); or where the shaft abutment height is too low to afford adequate contact surface with the bearing (Fig. 14.8b), spacers may be used effectively.

Relief dimensions for ground shaft and housing fitting surfaces are given in Table 14.2.

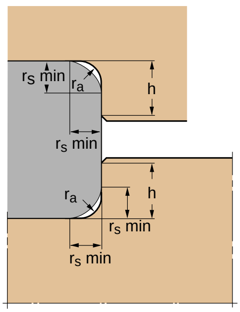


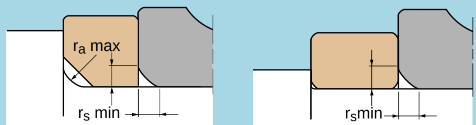
Table 14.1 Fillet radius *r*a and abutment height *h*

Unit mm

Chamfer dimension *r*s min	Fillet radius *r*as max	Minimum shoulder height *h*	
		Normal use1)	Special use2)
0.1	0.1	0.4	
0.15	0.15	0.6	
0.2	0.2	0.8	
0.3	0.3	1.25	1
0.6	0.6	2.25	2
1	1	2.75	2.5
1.1	1	3.5	3.25
1.5	1.5	4.25	4
2	2	5	4.5
2.1	2	6	5.5
2.5	2	6	5.5
3	2.5	7	6.5
4	3	9	8
5	4	11	10
6	5	14	12
7.5	6	18	16
9.5	8	22	20

1) For bearings subjected to heavy axial loads, shaft adjustments (h) should be higher than the values listed in the table.

2) The values in the "Special Case" column should be adopted in cases where thrust loading is extremely small;with the exception of tapered roller bearings, angular contact bearings, or spherical roller bearings.



(a) **(b)**

Fig. 14.8 Bearing mouting with spacer

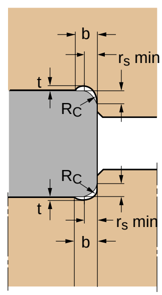


Table 14.2 Relief dimensions for ground shaft

Unit mm

Chamfer dimension *r*s min	Relief dimensions *b*	*t*	*R*c
1	2	0.2	1.3
1.1	2.4	0.3	1.5
1.5	3.2	0.4	2
2	4	0.5	2.5
2.1	4	0.5	2.5
3	4.7	0.5	3
4	5.9	0.5	4
5	7.4	0.6	5
6	8.6	0.6	6
7.5	10	0.6	7

14.3 Shaft and housing accuracy

For normal use, the accuracies for shaft and housing fitting surface dimensions and configurations, as well as fitting surface roughness and abutment squareness, are given in Table 14.3.

Table 14.3 Accuracy of shaft and housing Units (μm)

Characteristics		Shaft	Housing
Circularity Cylindricity (max.)		IT3	IT4
Sqareness of step (max.)		IT5	IT5
Surface roughness	Small size bearings	0.8a	1.6a
	Large size bearings	1.6a	3.2a

15. Bearing Handling

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

15.1 Storage

Most all rolling bearings are coated with a rust preventative before being packed and shipped, and they should be stored at room temperature with a relatively humidity of less than 60%. Under optimum storage conditions, and if the package remains intact, bearings can be stored for many years.

15.2 Fitting

When bearings are being mounted on shafts or in housings, the bearing rings should never be struck directly with a hammer or drift as damage to the bearing may result. Any force applied to the bearing should always be evenly distributed over the entire bearing ring face. Also, when fitting both rings simultaneously, applying pressure to one ring only should be avoided as indentations in the raceway surface may be caused by the rolling elements, or other internal damage may result.

15.2.1 Fitting preparation

Bearings should be fitted in a clean, dry work area. Especially for small and miniature bearings, a “clean room” should be provided as any dust in the bearing will greatly affect bearing efficiency.

Before installation, all fitting tools, shaft, housings and related parts should be cleaned and any burrs or cutting chips removed if necessary.

Shaft and housing fitting surfaces should also be checked for roughness, dimensional and design accuracy, and ensure that they are within allowable tolerance limits.

Bearings should be unwrapped just prior to installation. Normally, bearings to be used with grease lubrication can be installed as is, without removing the rust preventative. However, for bearings to be oil lubricated, or in cases where mixing the grease and rust preventative would result in loss of lubrication efficiency, the rust preventative should be removed by washing with benzene or petroleum solvent and drying before installation. Bearings should also be washed and dried before installation if the package has been damaged or there are other chances that the bearing has been contaminated. Double shielded bearings and sealed bearings should never be washed.

15.2.2 Fitting cylindrical bore bearings

Bearings with relatively small interference fits can be press fit at room temperature by using a sleeve against the ring face as shown in Fig. 15.1. By applying the fitting pressure to the center of the bearing, even pressure on the entire ring circumference can be attained. Usually, bearings are mounted by striking the sleeve with a hammer; however, when installing a large number of bearings, a mechanical or hydraulic press should be used.

When mounting a non-separable bearing on a shaft and in a housing at the same time, a pad which distributes the fitting pressure evenly over the inner and outer rings is used as shown in Fig. 15.2.

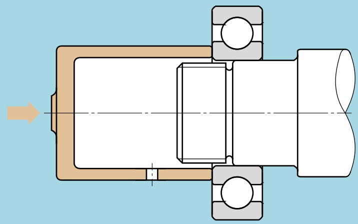


Fig. 15.1 Press mounting of inner ring

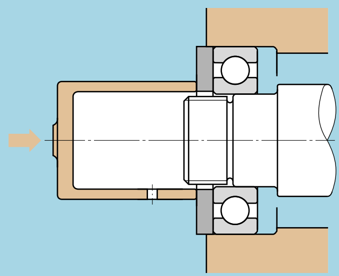


Fig. 15.2 Press mounting of inner and outer rings simultaneously

When fitting bearings having a large inner ring interference fit, or when fitting bearings on shafts that have a large diameter, a considerable amount of force is required to mount the bearing at room temperature. Mounting can be facilitated by heating and expanding the inner ring before hand. The required relative temperature difference between the inner ring and the fitting surface depends on the amount of interference and the shaft fitting surface diameter. Fig. 15.3 shows the relation between the bearing inner ring bore diameter temperature differential and the amount of thermal expansion. In any event, bearings should never be heated above 120°C.

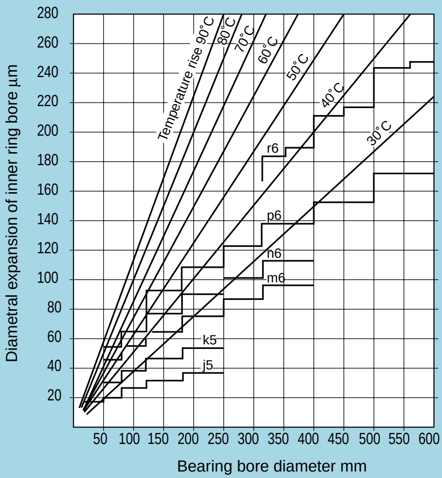


Fig. 15.3 Temperature differential required for shrinkage fit of inner ring

The most commonly used method of heating bearings is to immerse them in hot oil. However, to avoid overheating parts of the bearings, they should never be brought in direct contact with the heating element or bottom of the oil tank.

Bearings should be suspended inside the heating tank or placed on a wire grid.

If the bearings are dry heated with a heating cabinet or hot plate, they can be mounted without drying.

This method can also be used for prelubricated shielded and sealed bearings.

For heating the inner rings of NU, NJ or NUP cylindrical roller and similar type bearings without any ribs or with only a single rib, an induction heater can be used as shown in Fig. 15.4. With this method, bearings can be quickly installed while in a dry state.

When heated bearings are installed on shafts, the inner ring must be held against the shaft abutment until the bearings has been cooled in order to prevent gaps from occurring between the ring and the abutment face.

The same induction heating method described above can also be used for dismounting the inner ring with the use of a pawl.

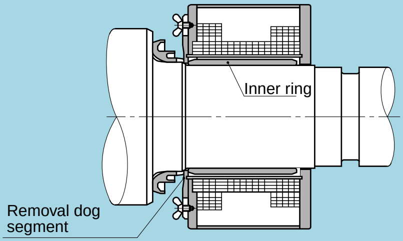


Fig. 15.4 Removal of inner ring using an induction heater

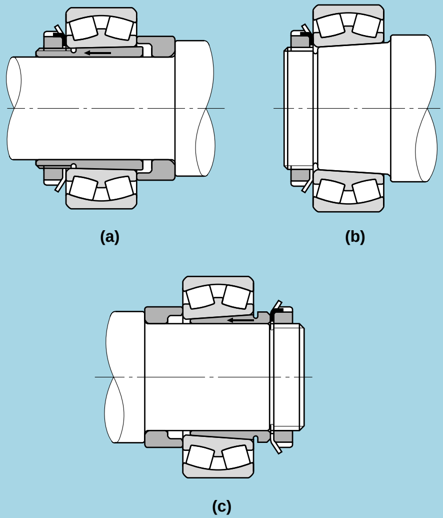


Fig. 15.5 Mounting with nuts

15.2.3 Mounting bearings with tapered bore

Small bearings with a tapered bore are driven on to the tapered seating; i.e. tapered shaft, withdrawal sleeves or adapter sleeves; with tightening a nut. This nut is tightened by using a hammer or impact wrench (Fig. 15.5).

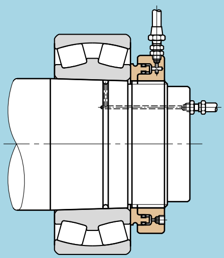
Large bearings are mounted by hydraulic method because the fitting force is considerably large.

In Fig. 15.6 the fitting surface friction and nut tightening torque needed to mount bearings with tapered bore directly on tapered shafts are lessened by injecting high pressure oil between the fitting surfaces.

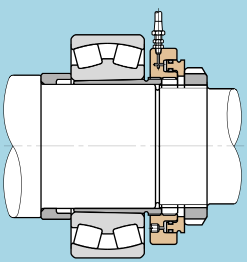
In Fig. 15.7 (a) a hydraulic nut method of pressing the bearing on a tapered shaft is shown.

Use of a hydraulic nut with adapters and withdrawal sleeves is shown in Fig. 15.7 (b) and (c).

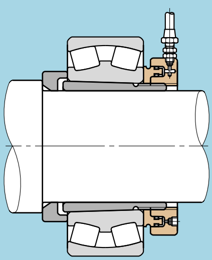
A mounting method using a hydraulic withdrawal sleeve is shown in Fig. 15.8.



(a)



(b)



(c)

Fig. 15.7 Mounting using hydraulic nut

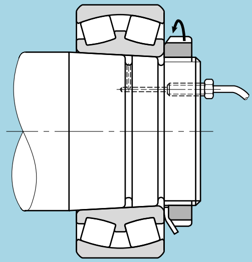


Fig. 15.6 Mounting using oil injection

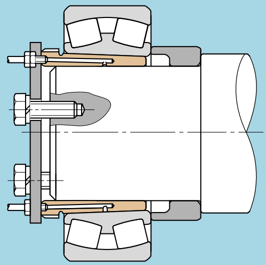


Fig. 15.8 Hydraulic sleeve mounting

In tapered bore bearings, as the inner ring is pressed axially onto the shaft or adapter or withdrawal sleeve, the interference will increase and the bearing internal radial clearance will decrease. The amount of interference can be estimated by measuring the amount of radial clearance decrease.

The internal radial clearance between the rollers and the outer ring should be measured with a thickness gauge under no load and the rollers held in the correct position. The measured clearance should be the same at both rows. In place of using the decrease in the amount of internal radial clearance to estimate the interference, it is possible to estimate it by measuring the distance the bearing has been driven onto the shaft.

Table 15.1 indicates the interference which will be given as a result of the internal radial clearance decrease, or the distance the bearing has been driven onto the shaft for tapered bore spherical roller bearings.

For conditions such as heavy loads, high speeds, and large temperature differentials between the inner and outer rings, etc. which require large interference fits, bearings which have a minimum internal radial clearance of C3 or greater should be used.

When using Table 15.1, the maximum values for clearance and axial displacement driven up should be used. For these applications, the remaining clearance must be greater than the minimum remaining clearance listed in Table 15.1.

15.2.4 Installation of Outer ring

For tight interference fits, the outer rings of small type bearings can be installed by pressing into housings at room temperature. For large interference fits, the housing can be heated before installing the bearing, or the bearing outer ring can be cooled with dry ice, etc. before installation.

15.3 Clearance adjustment

As shown in Fig. 15.9, for angular contact ball bearings or tapered roller bearings, the desired amount of axial internal clearance can be set at the time of mounting by tightening or loosening the adjusting nut. These bearings can also be preloaded by turning the adjusting nut until a minus axial internal clearance is reached.

There are three basic methods to ascertain if negative clearance is adjusted. One method is to actually measure the

Table 15.1 Mounting spherical roller bearings with tapered bore bearings　Unit mm

Bearing bore diameter *d*		Reduction in radial internal clearance		Axial displacement drive up				Minimum permisible residual clearance		
				Taper:1:12		Taper:1:30				
over	incl.	min	max	min	max	min	max	Normal	C3	C4
30	40	0.020	0.025	0.35	0.4	—	—	0.015	0.025	0.040
40	50	0.025	0.030	0.4	0.45	—	—	0.020	0.030	0.050
50	65	0.030	0.040	0.45	0.6	—	—	0.025	0.035	0.055
65	80	0.040	0.050	0.6	0.75	—	—	0.025	0.040	0.070
80	100	0.045	0.060	0.7	0.9	1.75	2.25	0.035	0.050	0.080
100	120	0.050	0.070	0.75	1.1	1.9	2.75	0.050	0.065	0.100
120	140	0.065	0.090	1.1	1.4	2.75	3.5	0.055	0.080	0.110
140	160	0.075	0.100	1.2	1.6	3.0	4.0	0.055	0.090	0.130
160	180	0.080	0.110	1.3	1.7	3.25	4.25	0.060	0.100	0.150
180	200	0.090	0.130	1.4	2.0	3.5	5.0	0.070	0.100	0.160
200	225	0.100	0.140	1.6	2.2	4.0	5.5	0.080	0.120	0.180
225	250	0.110	0.150	1.7	2.4	4.25	6.0	0.090	0.130	0.200
250	280	0.120	0.170	1.9	2.7	4.75	6.75	0.100	0.140	0.220
280	315	0.130	0.190	2.0	3.0	5.0	7.5	0.110	0.150	0.240
315	355	0.150	0.210	2.4	3.3	6.0	8.25	0.120	0.170	0.260
355	400	0.170	0.230	2.6	3.6	6.5	9.0	0.130	0.190	0.290
400	450	0.200	0.260	3.1	4.0	7.75	10	0.130	0.200	0.310
450	500	0.210	0.280	3.3	4.4	8.25	11	0.160	0.230	0.350
500	560	0.240	0.320	3.7	5.0	9.25	12.5	0.170	0.250	0.360
560	630	0.260	0.350	4.0	5.4	10	13.5	0.200	0.290	0.410
630	710	0.300	0.400	4.6	6.2	11.5	15.5	0.210	0.310	0.450
710	800	0.340	0.450	5.3	7.0	13.3	17.5	0.230	0.350	0.510
800	900	0.370	0.500	5.7	7.8	14.3	19.5	0.270	0.390	0.570
900	1000	0.410	0.550	6.3	8.5	15.8	21	0.300	0.430	0.640
1000	1120	0.450	0.600	6.8	9.0	17	23	0.320	0.480	0.700
1120	1250	0.490	0.650	7.4	9.8	18.5	25	0.340	0.540	0.770

axial internal clearance while tightening the adjusting nut (Fig. 15.10). Another method is to check rotation torque by rotating the shaft or housing while adjusting the nut. Still another method (Fig. 15.11) is to insert shims of the proper thickness.

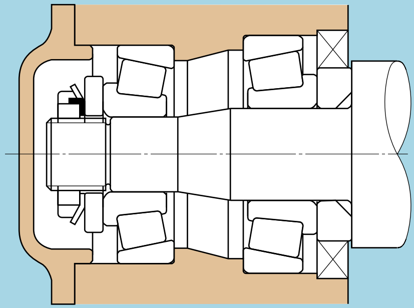


Fig. 15.9 Axial internal clearance adjustment

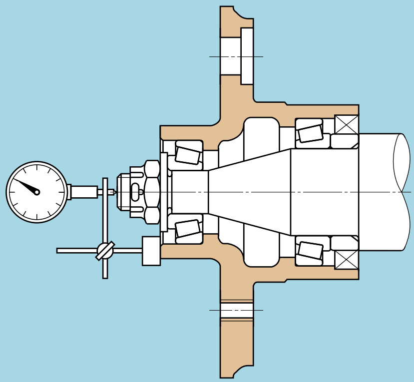


Fig. 15.10 Axial internal clearance measuring

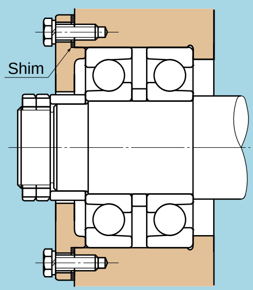


Fig. 15.11 Clearance adjustment with shims

15.4 Running test

To ensure that the bearing has been properly installed, a running test is performed after mounting. The shaft or housing is first rotated by hand, and if no problems are observed, a low speed, no load power test is performed. If no abnormalities are observed, the load and speed are gradually increased to operating conditions. During a test, if any unusual noise, vibration or temperature rise is observed, the test should be stopped and the equipment examined. If necessary, the bearing should be dismounted for inspection.

To check bearing running noise, the sound can be amplified and the type of noise ascertained with a listening instrument placed against the housing. A clear, smooth, continuous running sound is normal.

A high metallic or irregular sound indicates some error on function. Vibration can be accurately checked with a vibration measuring instrument, and the amplitude and frequency characteristics measured against a fixed standard.

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

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

15.5 Dismounting

Bearings are often removed as part of periodic inspection procedures or during the replacement of other parts. However, the shaft and housing are almost always reinstalled, and in some cases the bearings themselves are reused. These bearings, shafts, housings and other related parts must be designed to prevent damage during the dismounting procedures, and the proper dismounting tools must be employed. When removing inner or outer rings which have been installed with interference fits, the dismounting force should be applied to that ring only and not applied to other parts of the bearing, as this may cause internal damage to the bearings' raceway or rolling elements.

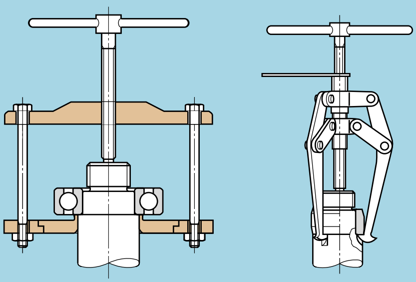
15.5.1 Dismounting of bearing with cylindrical bore

For small bearings, the pullers shown in Fig. 15.12 or the press method shown in Fig. 15.13 can be used for dismounting. When used properly these methods can improve dismounting efficiency and can prevent damage from occurring to the bearings. To facilitate dismounting procedures, care should be given to planning design of shafts and housings, such as providing extraction grooves on the shaft and housing for puller claws as shown in Figs. 15.14 and 15.15.

Threaded bolt holes should also be provided in housings for pressing out outer rings (Fig. 15.16).

Large bearings, having been in for a long service period and installed with shrink fits, require considerable dismounting force, and fretting corrosion is likely to have occurred on the seating surface. In these instances, the dismounting friction can be relieved by injecting oil under high pressure between the shaft and inner ring surfaces (Fig. 15.17).

For NU, NJ and NUP type cylindrical roller bearings, the induction heating method shown in Fig. 15.4 can also be used for easier dismounting of the inner ring. This method is highly efficient for dismounting the same dimension bearings frequently.



(a) (b)

Fig. 15.12 Puller dismounting

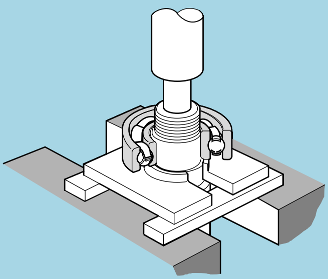


Fig. 15.13 Press dismounting

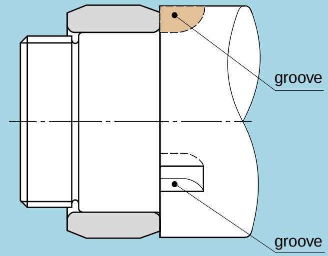


Fig. 15.14 Extracting grooves

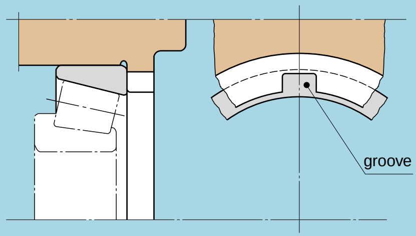


Fig. 15.15 Extraction groove for outer ring removal

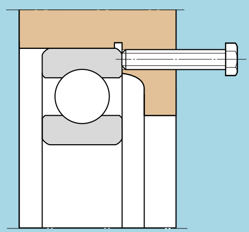


Fig. 15.16 Outer ring dismounting bolt

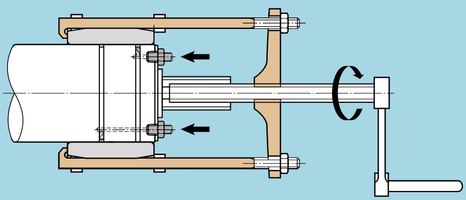


Fig. 15.17 Hydraulic dismounting

15.5.2 Dismounting of bearings with tapered bore

Small bearings with adapters can be easily dismounted by loosening the lock-nut and driving the inner ring off with a metal block (Fig. 15.18). Those bearings which have been installed with withdrawal sleeves can be extracted by tightening the nut (Fig. 15.19).

For large bearings on tapered shafts, adapters, or withdrawal sleeves; dismounting is greatly facilitated with hydraulic methods. Fig. 15.20 shows a hydraulic injection dismounting method. High pressure oil is injected between the fitting surface of the conical shaft and bearing.

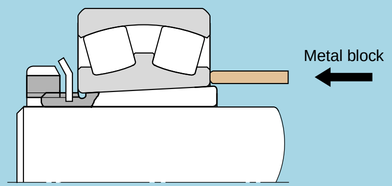


Fig. 15.18 Adapter dismounting

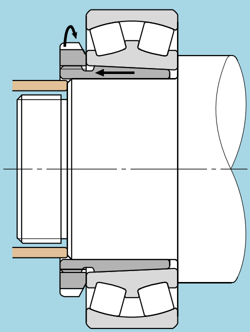


Fig. 15.19 Withdrawal sleeve extraction

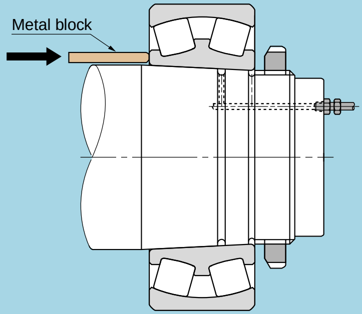
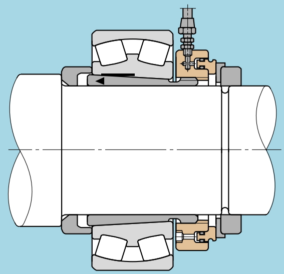
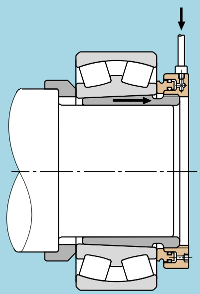


Fig. 15.20 Hydraulic injection dismounting

The metal block is used for protection of sudden movement of the bearing which occurs during injection. In Fig. 15.21 hydraulic nuts are used with adapters or withdrawal sleeves for dismounting, and a hydraulic withdrawal sleeve extraction method is shown in Fig. 15.22.



(a) Adapter sleeve



(b) Withdrawal sleeve

Fig. 15.21 Hydraulic nut dismounting

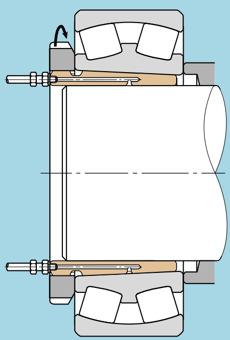


Fig. 15.22 Hydraulic withdrawal sleeve

16. Bearing Damage and Corrective Measures

While it is of course impossible to directly observe bearings in operation, one can get a good idea of how they are operating by monitoring noise, vibration, temperature and lubricant conditions. Types of damage typically encountered are present in Table 16.1.

Table 16.1 Bearing damage and corrective measures

Damage	Description	Causes	Correction
Flaking	The surface of the race-way begins wearing away. Conspicuous hills and valleys form soon afterward.	• Excessive loads or improper handling. • Improper mounting. • Improper precision in the shaft or housing. • Insufficient clearance. • Contamination • Rust. • Drop in hardness due to abnormally high temperatures.	• Review application conditions. • Select a different type of bearing. • Reevaluate the clearance. • Improve the precision of the shaft and housing. • Reevaluate the layout (deign) of the area around the bearing. • Review assembly procedures. • Review lubricant type and lubrication methods.
Seizure	The bearing heats up and becomes discolored. Eventually the bearing will seize up.	• Insufficient clearance (including clearances made smaller by local deformation). • Insufficient lubrication or improper lubricant. • Excessive loads (excessive pressure). • Skewed rollers.	• Check for proper clearance. (Increase clearances) • Review lubricant type and quantity. • Review application conditions. • Take steps to prevent misalignment. • Reevaluate the design of the area around the bearing (including fitting of the bearing). • Improve assembly procedures.
Cracking and notching	Localized flaking occurs. Little cracks or notches appear.	• Excessive shock loads. • Excessive interference. • Large flaking. • Friction cracking. • Inadequate abutment or chamfer. • Improper handling. (gouges from large foreign objects.)	• Review application conditions. • Select proper interference and review materials. • Improve assembly procedures and take more care in handling. • Take measures to prevent friction cracking. (Review lubricant type.) • Reevaluate the design of the area around the bearing.
Retainer damage	Rivets break or become loose resulting in retainer damage.	• Excessive moment loading. • High speed or excessive speed fluctuations. • Inadequate lubrication. • Impact with foreign objects. • Excessive vibration. •Improper mounting. (Mounted misaligned) • Abnormal temperature rise. (Plastic retainers)	• Review of application conditions. • Reevaluation of lubrication conditions. • Review of retainer type selection. • Take more care in handling. • Investigate shaft and housing rigidity.

Damage	Description	Causes	Correction
Smearing and scuffing	The surface becomes rough and some small deposits form. Scuffing generally refers to roughness on the race collar and the ends of the rollers.	• Inadequate lubrication. • Entrapped foreign particles. • Roller skewing due to a misaligned bearing. • Bare spots in the collar oil film due to large axial loading. • Surface roughness. • Excessive slippage of the rolling elements.	• Reevaluation of the lubricant type and lubrication method. • Review of operating conditions. • Setting of a suitable pre-load. • Improve sealing performance. • Take care to handle the bearing properly.
Rust and corrosion	The surface becomes either partially or fully rusted, occasional rust could occur along the rolling element pitch lines.	• Poor storage conditions. • Poor packaging. • Insufficient rust inhibitor. • Penetration by water, acid, etc. • Handling with bare hands.	• Take measures to prevent rusting while in storage. • Improve sealing performance. • Periodically inspect the lubricating oil. • Take care when handling the bearing.
Fretting	There are two types of fretting. In one, a rusty wear powder forms on the mating surfaces. In the other, brinelling indentations form on the raceway at the rolling element pitch.	• Insufficient interference. • Small bearing osscillation angle. • Insufficient lubrication. • Fluctuating loads. • Vibration during transport.	• Review the interference and apply a coat of lubricant. • Pack the inner and outer rings separately for transport. When the two cannot be separated, apply a pre-load. • Select a different kind of lubricant. • Select a different type of bearing.
Wear	The surfaces wear and dimensional deformation results. Wear is often accompanied by roughness and scratches.	• Entrapment of foreign particles in the lubricant. • Inadequate lubrication. • Skewed rollers.	• Review lubricant type and lubrication methods. • Improve sealing performance. • Take steps to prevent misalignment.
Electrical pitting	Pits form on the raceway. The pits gradually grow into ripples.	• Electric current flowing through the rollers.	• Creates a bypass circuit for the current. • Insulate the bearing so that current does not pass through it.
Dent and scratches	Scoring during assembly, gouges due to hard foreign objects, and surface denting due to mechanical shock.	• Entrapment of foreign objects. • Dropping or other mechanical shocks due to careless handling. • Assembled misaligned.	• Improve handling and assembly methods. • Take measures to prevent the entrapment of foreign objects. • Should the damage have been caused by foreign particles, thoroughly check all other bearing locations.
Slipping or creeping	Slipping is accompanied by mirrorlike or discolored surfaces on the ID and OD. Suffing may also occur.	• Insufficient interference in the mating section • Sleeve not fastened down properly. • Abnormal temperature rise. • Excessive loads.	• Reevaluate the interference. • Reevaluate operating conditions. • Review the precision of the shaft and housing.