Precision Rolling Bearings

TECHNICAL DATA CONTENTS

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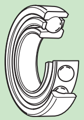
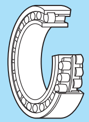
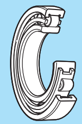
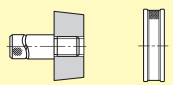
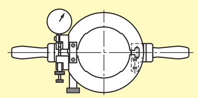
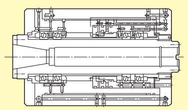
1. Classification of Precision Rolling Bearings for Machine Tools

1.1 Main spindle bearings

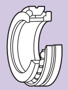
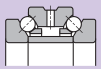
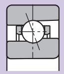
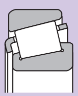
Table 1 Types of precision rolling bearings for machine tools

Bearing type	Cross section	Bearing type	Bearing bore mm	Contact angle	Remarks	Page
 Angular contact ball bearing	Standard, ULTAGE	78C	ϕ25–ϕ170	15°	● A bearing type code containing a suffix U means an ULTAGE series bearing. Optimized interior structure and resin cage help positively inhibit temperature rise (applicable to 79 and 70 types with bore diameter of 10 to 130 mm). ● Bearings with prefix 5S have ceramic balls.	126–153
		79 (U), 5S-79 (U)	ϕ10–ϕ170	15°, 25°, 30°		
		70 (U), 5S-70 (U)	ϕ10–ϕ200	15°, 25°, 30°		
		72C	ϕ10–ϕ130	15°		
	High speed, ULTAGE	2LA-HSE9U 5S-2LA-HSE9U	ϕ50–ϕ170	15°, 20°, 25°	● ULTAGE series ● Use of special material and introduction of surface modification contribute to much improved wear resistance and anti-seizure property. ● Optimized specifications for the interior structure lead to higher speed, rigidity and reliability. ● Bearings with prefix 5S have ceramic balls.	154–177
		2LA-HSE0 5S-2LA-HSE0				
	Ultra high speed, ULTAGE	5S-2LA-HSF0	ϕ50–ϕ100	25°	● ULTAGE series ● Maintaining the advantages of HSE type, this type has small diameter ceramic balls to achieve higher speed and limited heat buildup. ● Bearings with prefix 5S have ceramic balls.	178–179
	Eco-friendly, ULTAGE	5S-2LA-HSL9U	ϕ50–ϕ170	20°, 25°	● ULTAGE series ● These bearings are identical to the HSE and HSF types except in that they are air-oil lubrication designs that have an eco-friendly nozzle. ● Featuring lower noise, reduced air and oil consumption, they positively improve operating environments and reduce energy consumption. ● Bearings with prefix 5S have ceramic balls.	180–189
		5S-2LA-HSL0				
		5S-2LA-HSFL0	ϕ50–ϕ100	25°		
	HSE with Lubrication hole, ULTAGE	5S-2LA-HSEW9U	ϕ50–ϕ100	20°, 25°	● ULTAGE series ● High speed angular contact ball bearings with lubrication hole on outer ring, designed especially for air-oil lubrication based on HSE type. These bearings have an effect on compact design and high rigidity of spindle. Air flow rate and oil consumption can be reduced. ● Bearings with prefix 5S have ceramic balls.	190–197
		5S-2LA-HSEW0				
	Standard, ULTAGE, Non-contact sealed type	79 LLB 5S-79 LLB	ϕ10–ϕ50	15°, 25°	● ULTAGE series ● Featuring a two-side non-contact seal design and a special grease, these bearings are a dedicated grease lubricated type that has achieved limited heat buildup through optimization of the interior structure. ● Bearings with prefix 5S have ceramic balls.	198–213
		70 LLB 5S-70 LLB				
	High speed, ULTAGE, Non-contact sealed type	2LA-BNS9 LLB 5S-2LA-BNS9 LLB	ϕ45–ϕ100	15°, 20°, 25°	● ULTAGE series ● Maintaining the advantages of HSE type, this dedicated grease lubricated type has an improved interior design (grease reservoir, both -side non-contact seal and special grease) to extend grease life. ● Bearings with prefix 5S have ceramic balls.	214–237
		2LA-BNS0 LLB 5S-2LA-BNS0 LLB				
		BNT9 5S-BNT9	ϕ10–ϕ65	15°	● Angular contact ball bearings for grinding machines/motors. ● All variants are flush ground. ● Bearings with prefix 5S have ceramic balls.	238–249
		BNT0 5S-BNT0	ϕ10–ϕ70			
		BNT2 5S-BNT2	ϕ10–ϕ80			

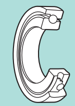
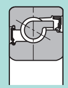
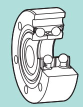
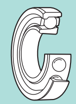
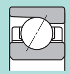
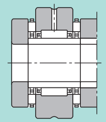
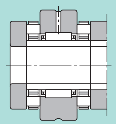
Bearing type	Cross section	Bearing type	Bearing bore mm	Contact angle	Remarks	Page
 Double-row cylindrical roller bearing	ULTAGE	NN49 (K)	ϕ100–ϕ320	—	● The bearing clearance can be either interchangeable radial internal clearance or non-interchangeable radial internal clearance. ● A variant (K) is available with a tapered bore to accommodate a tapered shaft. ● A bearing type code containing a suffix T6 means an ULTAGE series bearing. Optimized internal structure and resin cage help high speed and positively inhibit temperature rise (applicable to NN30 types with bore diameter of 65 to 130 mm).	270–275
		NN30 (K) NN30HS (K)	ϕ25–ϕ60 ϕ150–ϕ460			
		NN30HST6 (K) NN30HSRT6 (K)	ϕ65–ϕ140			
	ULTAGE	NNU49 (K)	ϕ100–ϕ500	—		
 Single-row cylindrical roller bearing	Standard	N10HS (K)	ϕ30–ϕ160	—	● The boundary dimensions of the N10HS(K) high speed single-row cylindrical roller bearing are the same as those of the N10(K). Only the bearing clearance is non-interchangeable. ● A ceramic-roller-type (5S-N10) is available on request.	276–279
	High speed, ULTAGE	N10HSRT6 (K)	ϕ55–ϕ100	—	● ULTAGE series ● Optimized internal design allows higher speed and results in lower temperature rise. ● The cage is made of a special resin to cope with a high speed operation. ● The allowable maximum speed is higher than that of the conventional high speed cylindrical roller bearing N10HS(K).	280–281
	Eco-friendly, ULTAGE	N10HSLT6 (K)	ϕ55–ϕ100	—	● ULTAGE series ● This is a dedicated air-oil lubricated type identical to the N10HSR(K) type except in that it incorporates an eco-friendly nozzle. ● Still maintaining the high speed performance of the N10HSR(K) type, this type boasts lower noise, reduced air and oil consumption, and positively improves operating environments and reduces energy consumption.	282–283
 Plug gauge, Ring gauge Taper gauge		Plug gauge TA	ϕ30–ϕ160	—	● Taper gauge for N10-HS(K) single-row cylindrical roller bearing and NN30(K) double-row cylindrical roller bearing.	284
		Ring gauge TB	ϕ30–ϕ160			
 Mounted internal clearance adjustment gauge		SB	ϕ35–ϕ160	—	● Mounted internal clearance adjustment gauge for N10-HSK(K) , N10-HSR(K) single-row cylindrical roller bearing and NN30(K), NN30HS(K) double-row cylindrical roller bearing.	285
 Adjustable preload bearing unit		Adjustable preload bearing unit	—	—	● Fixed position adjustable preload bearing unit. ● Incorporation of an adjustable preload sleeve and a duplex angular contact ball bearing allows the user to adjust the preload of an angular contact ball bearing in a wider range from a light preload to a heavy preload. ● Fixed position preload leads to a greater rigidity.	—

Classification of Precision Rolling Bearings for Machine Tools

NTN

Bearing type	Cross section	Bearing type	Bearing bore mm	Contact angle	Remarks	Page
 Double-direction angular contact thrust ball bearing		5629 (M)	Small-size $\phi100$–$\phi320$ Large-size (M) $\phi104$–$\phi330$	60°	●The small bearing is used on a cylinder bore or smaller-diameter side of a tapered bore of the NNU49, NN49 or NN30 double-row cylindrical roller bearing; the large bearing (suffix M) is used on the large hole side of a tapered bore.	298–301
		5620 (M)	Small-size $\phi25$–$\phi320$ Large-size (M) $\phi27$–$\phi330$			
 Angular contact ball bearing for axial load	ULTAGE 	HTA9U	$\phi100$–$\phi320$	30°, 40°	●ULTAGE series ●HTA9UDB type bearings are fully compatible with 5629 type bearings.	302–313
		HTA0U 5S-HTA0U	$\phi25$–$\phi320$ $\phi25$–$\phi130$		●ULTAGE series ●HTA0UDB type bearings are fully compatible with 5620 type bearings.	
 Tapered roller bearings		329	$\phi50$–$\phi190$	Nominal contact angle of 10° or greater, 17° or smaller	●Thin-wall type, ISO-compatible metric series.	318–321
		320	$\phi20$–$\phi170$			

1.2 Ball screw support bearings

Bearing type	Cross section	Bearing type	Bearing bore mm	Contact angle	Remarks	Page
 Angular contact thrust ball bearing for ball screw support	ULTAGE 	**BST 2A-BST** Open type	$\phi17$–$\phi55$	60°	●ULTAGE series ●Surface modification treatment on the bearing ring raceways has led to a longer bearing life and much improved fretting resistance. ●Owing to prelubrication with a special grease, the sealed type boasts a longer bearing life and simpler maintenance work. ●All variants are flush ground and are provided with a standard preload.	344–349
		BST LXL/L588 2A-BST LXL/ L588 Light-contact sealed type				
 Double-row thrust angular contact ball bearing unit for ball screw support	ULTAGE 	**BSTU LLX/L588** Light-contact sealed type	$\phi20$–$\phi100$	60°	●ULTAGE series. ●Greater high-load capacity with optimizations made to the internal bearing design. ●Use of newly developed light-contact seal to achieve both low torque and high dust resistance. ●Long operating life, and use of special grease with high fretting resistance. ●Outer ring mounting hole, and sealed grease lubrication groove for easier handling.	350–353
 Angular contact ball bearing for ball screw support		**HT**	$\phi6$–$\phi40$	30°	●The allowable axial load of this bearing type is greater owing to the improved interior design.	354–355
 Needle roller bearings with double-direction thrust needle roller bearing		**AXN**	$\phi20$–$\phi50$	—	●A clearance remains between the inner ring of radial bearing and the inner rings of both thrust bearings, allowing the user to determine the preload by, for example, tightening a nut etc. ●The targeted preload is attained based on the starting torque. ●The bearing clearance on certain preloaded bearings is controlled in advance so that an intended preload is attained by fully tightening the inner rings on both thrust bearing with nuts, or equivalent means.	356–357
 Needle roller bearings with double-direction thrust cylindrical roller bearing		**ARN**	$\phi20$–$\phi70$	—		358–359

2. Bearing Selection and Shaft & Housing Design

2.1 Bearing selection

Generally, the optimal bearing must be selected to suit the nature of the machine, the area within the machine, the spindle specification, bearing type, lubrication system and drive system of the intended machine through considerations of the design life, precision, rigidity and critical speed, etc. of the bearing. **Table 2.1** summarizes a typical bearing selection procedure, and **Table 2.2** gives an example flowchart according to which considerations are made to select an optimal main spindle bearing for a machine tool.

Table 2.1 Bearing selection procedure

Step	Items being considered	Items being confirmed
Confirm operating conditions of bearing and consider bearing type.	●Function and construction of components to house bearings ●Bearing mounting location ●Dimensional limitations ●Magnitude and direction of bearing load ●Magnitude of vibration and shock load ●Shaft speed ●Bearing arrangement (fixed side, floating side) ●Noise and torque of the bearing ●Bearing operating temperature range ●Bearing rigidity ●Installation/disassembly requirements ●Maintenance and inspection ●Cost-effectiveness ●Allowable misalignment of inner/outer rings	Determine bearing type and arrangement.
Select bearing dimensions.	●Design life of components to house bearings ●Dynamic/static equivalent load conditions ●Safety factor So ●Allowable speed ●Allowable axial load	Determine bearing dimensions.
Select bearing tolerances.	●Shaft runout tolerances ●Torque fluctuation ●High speed operation	Decide bearing grade.
Select bearing internal clearance.	●Material and shape of shaft and housing ●Fits ●Temperature difference between inner and outer rings ●Allowable misalignment of inner/outer rings ●Magnitude and nature of load ●Amount of preload	Decide bearing internal clearance.
Select cage.	●Rotational speed ●Noise level ●Vibration and shock load ●Lubrication	Determination of cage type
Select lubrication method.	●Operating temperature ●Rotational speed ●Lubrication method ●Sealing method ●Maintenance and inspection	Decide lubrication method, lubricant, and sealing method.
Consider special specifications.	●Operating conditions (special environments: high or low temperature, chemical) ●Requirement for high reliability	Decide special bearing specifications.
Select installation and disassembly procedures.	●Mounting dimensions ●Installation and disassembly procedures	Decide installation and disassembly procedures.

Bearing Selection and Shaft & Housing Design NTN

Table 2.2 Bearing selection flow chart for machine tool spindles

User's requirements

→ **Max. speed** / **Main spindle size** / **Lubrication system**

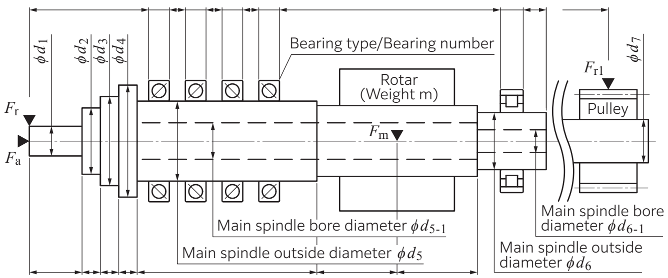
→ Bearing size. Consideration of allowable $d_m n$ value. → Consideration of fits. → Consideration of tightening force for inner ring spacer. Consideration of drive-up of front cover. → Estimation of bearing internal preload at maximum speed and consideration of bearing initial clearance according to **NTN**'s bearing selection guideline. → Finalization of bearing specifications. → Consideration of rigidity of bearing and main spindle. Consideration of bearing life.

NG (return to start) / **OK** →

Finalization of main spindle specifications

The articles necessary for basic considerations in selecting an optimal main spindle bearing for machine tool are summarized in **Table 2.3**.

Table 2.3 Selection procedure for bearings for main spindles of machine tools

(1) Type of Machine	NC Lathe, machining center, grinding machine, etc.
(2) Main spindle orientation	Vertical, horizontal, variable-direction, inclined, etc.
(3) Diameter and size of main spindle	#30, #40, #50, etc.
(4) Shape and mounting-related dimensions of main spindle	 Fig. 2.1 Main spindle shape and mounting-related dimensions (example)
(5) Intended bearing type, bearing size, and preloading method	Front (angular contact type, cylindrical roller type) or rear (angular contact type, cylindrical roller type) preloading system (fixed-position preloading, fixed-pressure preloading)
(6) Slide system free side	Cylindrical roller bearing, ball bushing (availability of cooling)
(7) Lubrication method	Grease, air-oil, oil mist (MicronLub)
(8) Drive system	Built-in motor, belt drive, coupling
(9) Presence/absence of jacket cooling arrangement on bearing area	w/, w/o
(10) Jacket cooling conditions	Synchronization with room temperature, machine-to-machine synchronization, oil feed rate (L/min)
(11) Operating speed range	Max. speed (min^{-1})
	Normal speed range (min^{-1})
	Operating speed range (min^{-1})
(12) Load conditions (machining conditions)	Load center
	Applied load Radial load F_r (N) Axial load F_a (N)
	Speed
	Machining frequency
	Intended bearing life

Technical Data

2.2 Bearing accuracy

Bearing accuracy

Accuracies of rolling bearings, that is, dimensional accuracy and running accuracy of rolling bearings are defined by applicable ISO standards and JIS B 1514 standard (Rolling bearings - Tolerances) (see **Table 2.4** and **Table 2.5**). The dimensional accuracy governs the tolerances that must be satisfied when mounting a bearing to a shaft or housing, while the running accuracy defines a permissible run-out occurring when rotating a bearing by one revolution. Methods for measuring the accuracy of rolling bearings (optional methods) are described in JIS B 1515 (Measuring methods for rolling bearings). **Table 2.6** summarizes some typical methods for measuring running accuracy of rolling bearings.

Table 2.4 Bearing types and applicable tolerance

Bearing type		Applicable standard	Tolerance class				
Angular contact ball bearings		JIS B 1514-1 (ISO 492)	Class 0	Class 6	Class 5	Class 4	Class 2
Cylindrical roller beargins			Class 0	Class 6	Class 5	Class 4	Class 2
Needle roller bearings			Class 0	Class 6	Class 5	Class 4	—
Tapered roller bearings	Metric	JIS B 1514	Class 0,6X	(Class 6)1)	Class 5	Class 4	—
	Inch	ANSI/ABMA Std.19	Class 4	Class 2	Class 3	Class 0	Class 00
	J series	ANSI/ABMA Std.19.1	Class K	Class N	Class C	Class B	Class A
Double-direction angular contact thrust ball bearings		**NTN** standard	—	—	Class 5	Class 4	—

1) The class is the **NTN** standard class.

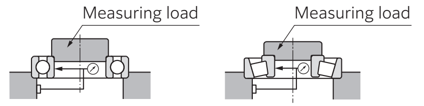
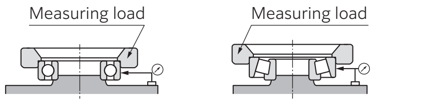
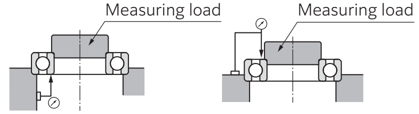
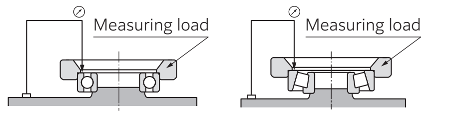
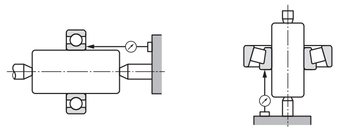
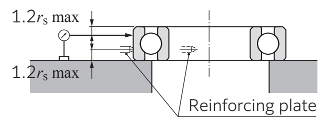
Table 2.5 Comparison of tolerance classifications of national standards

Standard	Applicable standard	Tolerance Class					Bearing Types
Japanese industrial standard (JIS)	JIS B 1514	Class 0,6X	Class 6	Class 5	Class 4	Class 2	All type
International Organization for Standardization (ISO)	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 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)	DIN 620	P0	P6	P5	P4	P2	All type
American National Standards Institute (ANSI) American Bearing Manufacturer's Association (ABMA)	ANSI/ABMA 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/ABMA Std.19.1	Class K	Class N	Class C	Class B	Class A	Tapered roller bearings (Metric series)
	ANSI/ABMA Std.19	Class 4	Class 2	Class 3	Class 0	Class 00	Tapered roller 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 slightly different from those of ABMA standards.

To attain a higher level of running accuracy required of a main spindle of machine tool, a high-precision bearing that satisfies the user's main spindle specifications must be chosen. Usually, a high-precision bearing per JIS accuracy Class 5, 4 or 2 is selected according to an intended application. In particular, the radial run-out, axial run-out and non-repetitive run-out of a main spindle bearing greatly affect the running accuracy of the main spindle and therefore have to be strictly controlled. With the recent super high-precision machine tools, the control of N.R.R.O. (Non-Repetitive Run-Out) has increasing importance, and the main spindle on a turning machine or machining center incorporates an N.R.R.O. accuracy controlled bearing. For further information about N.R.R.O., refer to the following section. Note that to attain a higher accuracy with a main spindle, careful considerations need to be exercised for the accuracies (circularity, cylindricity, coaxiality) of machine components other than a bearing (shaft, housing) as well as machining method and finish accuracy of the shaft and housing. For the information about the accuracies of shaft and housing, refer to a section given later.

Table 2.6 Measuring methods for running accuracies

Accuracy characteristics	Measurement methods	
Radial runout of inner ring of assembled bearing (K_{ia})		Radial runout of the inner ring is the difference between the maximum and minimum reading of the measuring device when the inner ring is turned one revolution.
Radial runout of outer ring of assembled bearing (K_{ea})		Radial runout of the outer ring is the difference between the maximum and minimum reading of the measuring device when the outer ring is turned one revolution.
Axial runout of inner ring of assembled bearing (S_{ia})		Axial runout of the inner ring is the difference between the maximum and minimum reading of the measuring device when the inner ring is turned one revolution.
Axial runout of outer ring of assembled bearing (S_{ea})		Axial runout of the outer ring is the difference between the maximum and minimum reading of the measuring device when the outer ring is turned one revolution.
Perpendicularity of inner ring face with respect to the bore (S_d)		The squareness of the inner ring side surface is the difference between the maximum and minimum readings of the measuring device when the inner ring is turned one revolution together with the tapered mandrel.
Perpendicularity of outer ring outside surface with respect to the face (S_D)		The squareness of the outer ring outside diameter surface is the difference between the maximum and minimum readings of the measuring device when the outside ring is turned one revolution along the reinforcing plate.

N.R.R.O. (Non-Repetitive Run-Out) of bearing

Accuracies of rolling bearings are defined by applicable ISO standards and a JIS (Japanese Industrial Standards) standard, wherein the accuracies are discussed under the descriptions of radial run-out (K_{ia}), axial run-out (S_{ia}), etc. According to the methods for measuring running accuracies in **Table 2.6**, run-out is read by turning a bearing by only one revolution (each reading is synchronized with the revolution of the bearing being analyzed).

In fact, however, a rolling bearing for machine tool is used in a continuous revolving motion that involves more than one revolution. As a result, the actual run-out accuracy with a rolling bearing includes elements that are not synchronous with the revolution of the bearing (for example, a difference in diameter among rolling elements involved, as well as roundness on the raceway surfaces of inner ring and outer ring), causing the trajectory of plotting with running accuracies to vary with each revolution.

The run-out of an element not in synchronization with the revolutions of bearing is known as N.R.R.O. (<u>N</u>on- <u>R</u>epetitive <u>R</u>un-<u>O</u>ut) and is equivalent to the amplitude in the Lissajous figure illustrated in **Fig. 2.3**.

The effect of N.R.R.O. on a rolling bearing onto the accuracies is illustrated in **Fig. 2.4** by taking a main spindle of turning machine as an example.

This diagram illustrates a machining process where the outside surface of a work piece mounted to the main spindle is shaved by a turning operation. If the outside surface is cut with a new trajectory with every revolution, the outside shape of work piece will be distorted. Furthermore, if the accuracies of shaft and housing are not high enough or bearings are assembled onto the shaft and/or housing improperly, the bearing ring can be deformed, possibly leading to a run-out that is not in synchronization with the revolutions of bearing.

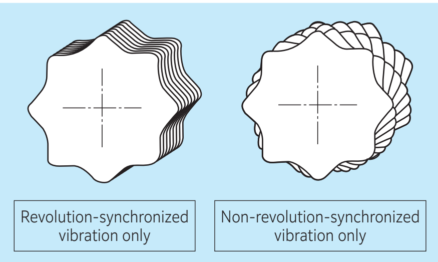


Fig. 2.2

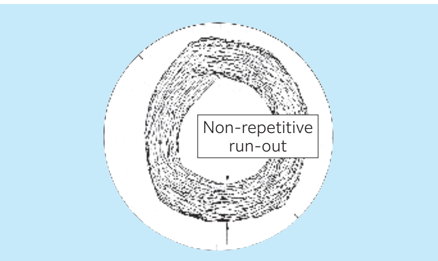


Fig. 2.3 Lissajous figure

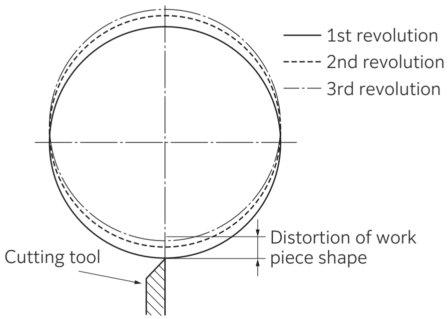


Fig. 2.4 Model of cutting operation

Accuracies of shaft and housing

Depending on the fit of a bearing to a shaft and a housing, the bearing internal clearance can vary. For this reason, an adequate bearing fit has to be attained so that the bearing can perform as designed (Refer to the recommended fits section).

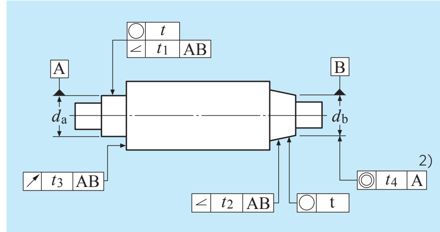
Also, the axial tightening torque on a bearing needs to be considered. To avoid deformation of bearing raceway surface owing to axial tightening of the bearing, it is necessary to carefully determine the dimensions of components associated with a tightening force the magnitude of tightening force and the number of tightening bolts.

The clearance on a tapered bore cylindrical roller bearing is adjusted by changing the drive-up to the taper. Because of this, the critical factors associated with an appropriate fit of a bearing to a shaft and/or a housing are the dimensional accuracies of the taper, contact surface on the taper, and the squareness of the end face of the inner ring relative to the shaft centerline during the drive-up process.

Typical accuracy values for a spindle and housing are summarized in **Table 2.7** and **Table 2.8**.

Typical accuracy for spindle

Table 2.7 Form accuracy of spindle 1)

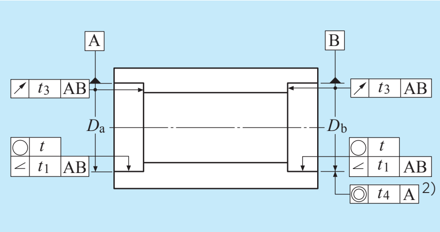


Accuracy	Symbol	Tolerance 3)	Fundamental permissible tolerance IT		
			P5	P4	P2
Deviation from circular form	○	t	$\frac{IT3}{2}$	$\frac{IT2}{2}$	$\frac{IT0^{4)}}{2}$
Angularity	∠	t_1	$\frac{IT3}{2}$	$\frac{IT2}{2}$	$\frac{IT0^{4)}}{2}$
	∠	t_2	—	$\frac{IT3}{2}$	$\frac{IT2}{2}$
Run out	↗	t_3	IT3	IT3	IT2
Eccentricity	◎	t_4	IT5	IT4	IT3

1) The form tolerance, symbol, and reference face of spindle are in accordance with ISO R1101.
2) The length of the bearing fit surface is often too small to measure concentricity. Therefore, this criterion applies only when the fit surface has a width sufficient as a reference face.
3) When determining a tolerance for permissible form accuracy, the reference dimensions used are shaft diameters d_a and d_b. For example, when using a JIS Class 5 bearing for a dia. 50 mm shaft, the tolerance of roundness is t = IT3/2 = 4/2=2 μm.
4) IT0 is preferred if the diameter tolerance of the bearing fit surface is IT3.

Typical accuracy for housing

Table 2.8 Form accuracy of housing 1)



Accuracy	Symbol	Tolerance 3)	Fundamental permissible tolerance IT		
			P5	P4	P2
Deviation from circular form	○	t	$\frac{IT3}{2}$	$\frac{IT2}{2}$	$\frac{IT1}{2}$
Angularity	∠	t_1	$\frac{IT3}{2}$	$\frac{IT2}{2}$	$\frac{IT1}{2}$
Run out	↗	t_3	IT3	IT3	IT2
Eccent ricity	◎	t_4	IT5	IT4	IT3

1) The form tolerance, symbol and reference face of the housing are in accordance with ISO R1101.
2) The length of the bearing fit surface is often too small to measure concentricity. Therefore, this criterion applies only when the fit surface has a width sufficient as a reference face.
3) Housing bore diameters D_a and D_b are the reference dimensions used when the tolerance for permissible form accuracy are determined.
For example, when a JIS Class 5 bearing is used for a housing with a 50 mm inside bore, the tolerance of roundness is t = IT3/2 = 5/2 = 2.5 μm.

Fundamental tolerance IT

Table 2.9 Fundamental tolerance IT

Classification of nominal dimension (mm)		Fundamental tolerance IT value (μm)					
over	incl.	IT0	IT1	IT2	IT3	IT4	IT5
6	10	0.6	1	1.5	2.5	4	6
10	18	0.8	1.2	2	3	5	8
18	30	1	1.5	2.5	4	6	9
30	50	1	1.5	2.5	4	7	11
50	80	1.2	2	3	5	8	13
80	120	1.5	2.5	4	6	10	15
120	180	2	3.5	5	8	12	18
180	250	3	4.5	7	10	14	20
250	315	4	6	8	12	16	23
315	400	5	7	9	13	18	25
400	500	6	8	10	15	20	27

Note) For machine tool spindles, the shaft hardness is recommended to be at least HRC 50 and the housing is recommended to be at least HRC 30 to assist bearing replacement during repairs.

Bearing Selection and Shaft & Housing Design NTN

Technical Data

2.3 Bearings and rigidity

The rigidity of the main spindle of a machine tool is associated with both bearing rigidity and shaft rigidity. Bearing rigidity is typically governed by the elastic deformation between the rolling elements and raceway surface under load. Usually, bearings are preloaded in order to increase the rigidity.

Under same loading conditions, a roller bearing has a higher rigidity than a ball bearing of the same size. However, having sliding portions, a roller bearing is disadvantageous in supporting a high speed shaft.

Shaft rigidity is greater with a larger shaft diameter. However, the supporting bearing must have a sufficient size and its $d_m n$ value [pitch center diameter across rolling elements d_m (mm) multiplied by speed n (min^{-1})] must be accordingly greater. Of course, a larger bearing is disadvantageous for high speed applications.

To sum up, the rigidity required of the shaft arrangement must be considered before the bearing rigidity (bearing type and preload) and shaft rigidity are determined.

■ Bearings rigidity

The rigidity of a bearing built into a spindle directly affects the rigidity of the spindle.

In particular, a high degree of rigidity is required of the main spindle of a machine tool to ensure adequate productivity and accurate finish of workpieces.

Bearing rigidity is governed by factors such as the following:

(1) Types of rolling elements
(2) Size and quantity of rolling elements
(3) Material of rolling elements
(4) Bearing contact angle
(5) Preload on bearing

■ Type of rolling elements (roller or ball)

The surface contact pattern of the rolling element and raceway is line contact with a roller bearing, while a ball bearing is point contact. As a result, the dynamic deformation of a bearing relative to a given load is smaller with a roller bearing.

■ Size and number of rolling elements

The size and number of rolling elements of a bearing are determined based on the targeted performance of the bearing.

Larger rolling elements lead to a greater bearing rigidity. However, a bearing having larger rolling elements tends to be affected by gyratory sliding centrifugal force, and, as a result, its high speed performance will be degraded. Incidentally, a greater number of rolling elements helps increase bearing rigidity, but at the same time creates an increased number of heat generation sources, possibly leading to greater temperature rise.

For this reason, smaller size of rolling elements are used for high speed applications.

To achieve both "high speed" and "high rigidity", each type of the **NTN** angular contact ball bearing for a machine tool is manufactured according to optimized specifications for interior structure (see **Fig. 2.5**).

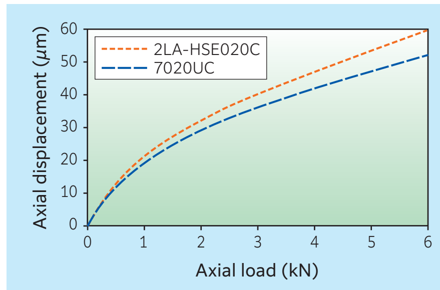


Fig. 2.5

■ **Material of rolling element (ceramic and steel)**

Certain **NTN** bearings incorporate ceramic rolling elements. As Young's modulus of silicon nitride (308 GPa) is greater than that of bearing steel (208 GPa), the rigidity with this type of bearing is accordingly greater (see **Fig. 2.6**).

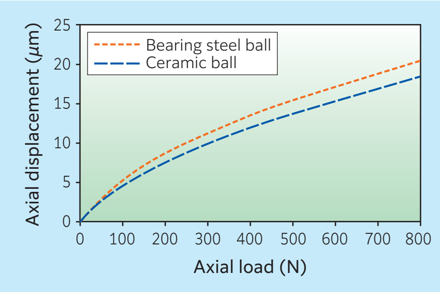


Fig. 2.6

■ **Bearing contact angle**

A smaller contact angle on an angular contact ball bearing results in greater radial rigidity. When used as a thrust bearing, this type of bearing should have a greater contact angle to enable greater axial rigidity (see **Fig. 2.7**).

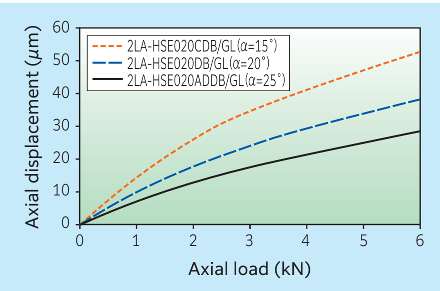


Fig. 2.7

■ **Preload on bearing**

A greater preload on a given bearing results in greater rigidity (see **Fig. 2.8**). However, too great of a preload on a bearing can lead to overheating, seizure, and/or early spalling (flaking) of the bearing. It is possible to use bearings in three- or four-row configurations in order to achieve increased axial rigidity (see **Fig. 2.9**).

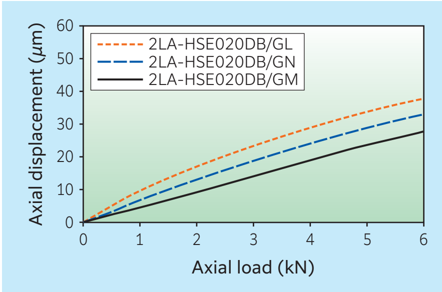


Fig. 2.8

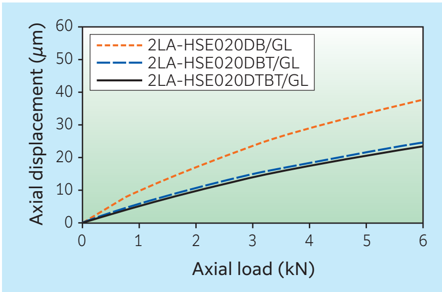


Fig. 2.9

■ **Preloading technique and preload**

Bearing preloading techniques can be categorized as fixed position preloading and constant pressure preloading (see **Fig. 2.10**).

Definite position preloading is useful in enhancing the rigidity of a bearing unit, as the positional relationship across individual bearings can be maintained. On the other hand, as preloading is achieved with spring force, the constant pressure preloading technique can maintain a preload constant even when the bearing-to-bearing distance varies due to heat generation on the spindle or a change in load.

The standard preload for a duplex bearing is given in the relevant section for each bearing.

If an angular contact ball bearing is to be used for a high speed application, such as for the main spindle of a machine tool, determine the optimal preload by considering the increase in contact stress between rolling elements and the raceway surface that results from gyratory sliding and centrifugal force. When considering such an application, consult **NTN** Engineering.

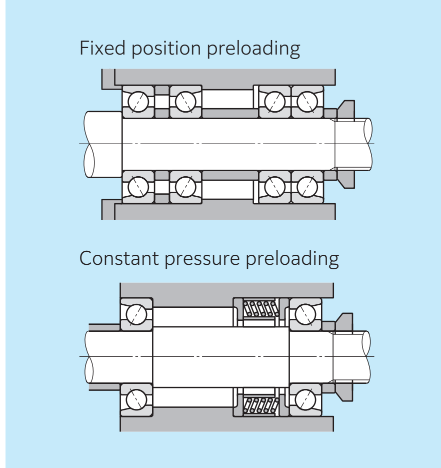


Fig. 2.10

■ Preload and rigidity

The effect of preloading for an increase in bearing rigidity is summarized in **Fig. 2.11**.

When the inner rings in the diagram are tightened to bring them together, bearings I and II are each axially displaced by dimension δ_o, thereby attaining a preload F_o. In this situation, if an axial load F_a is further exerted from outside, the displacement on bearing I increases by δ_a, while the displacement on bearing II decreases.

At this point, the loads on bearings I and II are F_{I} and F_{II}, respectively. When compared with δ_b (the displacement occurring when an axial load F_a is exerted onto a non-preloaded bearing I), displacement δ_a is small. Thus, a preloaded bearing has higher rigidity.

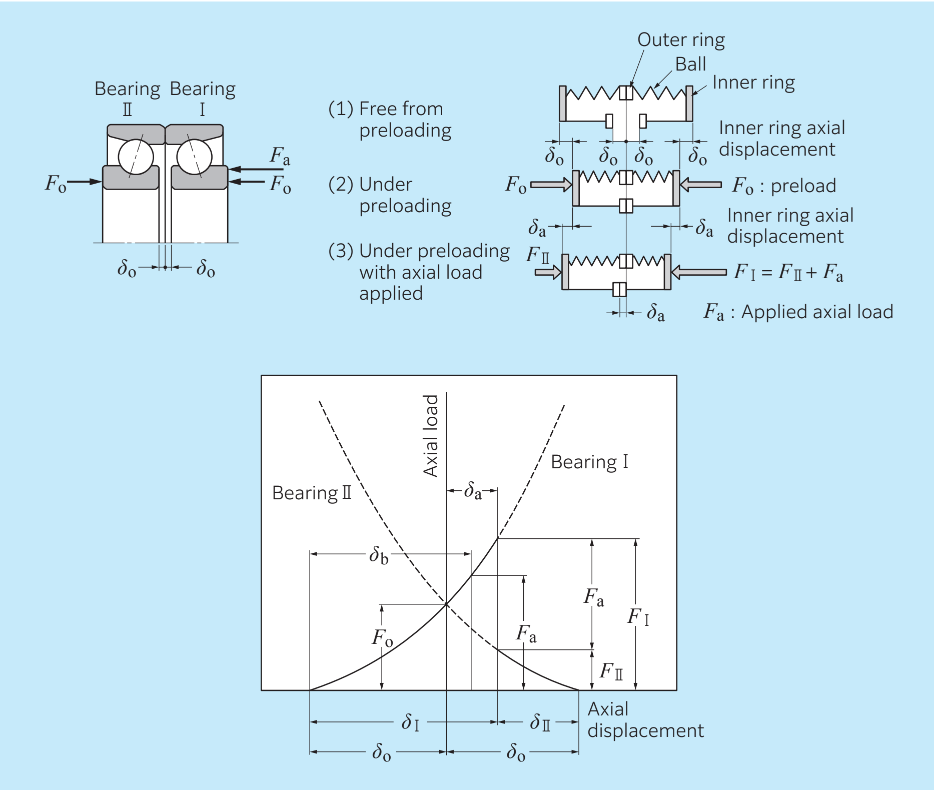


Fig. 2.11 Preload diagram

■ Gyratory sliding

Every rolling element (ball) in an angular contact ball bearing revolves on the axis of rotation A-A' as illustrated in **Fig. 2.12**. A revolving object tends to force the axis of rotation to a vertical or horizontal attitude. As a result, the rolling element develops a force to alter the orientation of the axis of rotation. This force is known as a gyratory moment (M).

When the force due to the gyratory moment is greater than the resistance force (rolling element load multiplied by the coefficient of friction between the raceway and rolling element) , gyratory sliding occurs on the raceway surface. This leads to heat generation, wear and seizure. Therefore, it is necessary to provide a sufficient resistance force to inhibit gyratory sliding. **NTN**'s recommended preload is based on this theory.

The gyratory moment that will occur can be calculated by the formula below.

$$M = k \times \omega_b \times \omega_c \times \sin\beta$$

$$k = \frac{1}{10} \times m \times d_w^2$$

$$= 0.05 \times \rho \times d_w^5$$

$$M \propto d_w^5 \times n^2 \times \sin\beta$$

M : Gyratory moment
ω_b : Autorotation angular velocity of rolling element
ω_c : Angular velocity of revolution
m : Mass of rolling element
ρ : Density of rolling element
d_w : Diameter of rolling element
β : Angle of axis of rotation of rolling element
n : Speed of inner ring

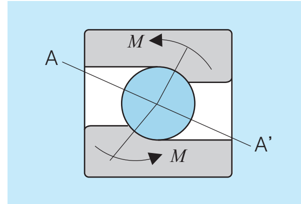


Fig. 2.12 Gyratory sliding

■ Spin sliding

Every rolling element (ball) in an angular contact ball bearing develops spin sliding that is unavoidable owing to the structure of the bearing, relative to the raceway surface of either the inner ring or outer ring (see **Fig. 2.13**).

Usually, at a lower speed range, pure rolling motion occurs between an inner ring raceway and rolling elements and spin sliding develops between an outer ring raceway and rolling elements (this state is known as inner ring control). At a higher speed range, pure rolling motion occurs between an outer ring raceway and rolling elements and spin sliding develops between an inner ring raceway and rolling elements (this state is known as outer ring control). A point where transfer from inner ring control to outer ring control occurs is known as control transfer point. An amount of spin sliding and control transfer point can vary depending on the bearing type and bearing data. Generally, the amount of spin sliding will be greater with an outer ring control state.

According to J. H. Rumbarger and J. D. Dunfee, when the amount of spin sliding exceeds 4.20×10^6 (N/mm^2 • mm/s), increase of heat generation and wear start.

The example of wear on a bearing owing to spin sliding is given in **Fig. 2.14**.

The magnitude of spin-derived wear is governed by a PV value (amount of spin sliding) during operation of the main spindle. Therefore, the optimum bearing for main spindle must be selected. The possibility of spin-derived wear occurrence varies depending on the bearing type, model number and specifications.

Also, the magnitude of spin-derived wear is significantly affected by how well the raceway surface is lubricated. Regardless of the type of sliding, even minor sliding can lead to wear if oil film is not formed well. For this reason, a reliable lubrication arrangement needs to be incorporated.

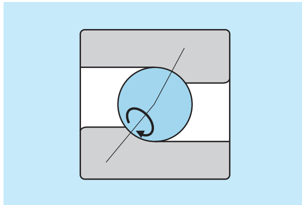
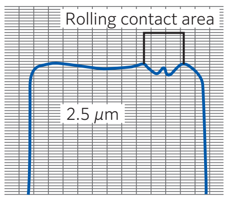


Fig. 2.13 Spin sliding

The form of wear on the bearing raceway derived from spin sliding appears as . The wear on the raceway surface on inner ring that resulted from spin sliding is given below.



Bearing: 7026T1
Axial load: 2 kN
Speed: 5 000 min^{-1}
Lubrication: Grease
Run time: 50 h

Possible causes for type wear

(1) Contact ellipse and direction of spin sliding

(2) Sliding velocity (V)

(3) Bearing pressure within ellipse (P)

(4) PV value owing to spin

(5) Wear on raceway surface

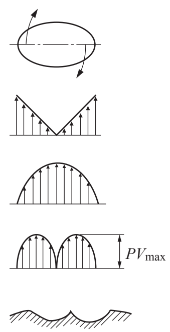


Fig. 2.14 Mechanism of wear on bearing owing to spin sliding

2.4 Designing shaft and housing

In designing a bearing and housing, it is very important to provide a sufficient shoulder height for the bearing and housing so as to maintain bearing and housing accuracies and to avoid interference with the bearing related corner radius.

The chamfer dimensions are shown in **Table 2.10** and the recommended shoulder height and corner radii on the shaft and housing are listed in **Table 2.11**.

■ Bearing corner radius dimensions

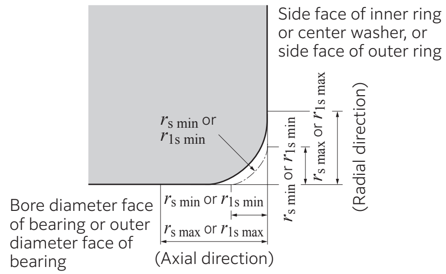


Fig. 2.15

Table 2.10 Allowable critical-value of bearing chamfer

(1) Radial bearings (Except tapered roller bearings)

Unit: mm

$r_{s\,min}$ 1) or $r_{1s\,min}$	Nominal bore diameter d		$r_{s\,max}$ or $r_{1s\,max}$	
	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	40 —	0.6 0.8	1 1
0.6	— 40	40 —	1 1.3	2 2
1	— 50	50 —	1.5 1.9	3 3
1.1	— 120	120 —	2 2.5	3.5 4
1.5	— 120	120 —	2.3 3	4 5
2	— 80 220	80 220 —	3 3.5 3.8	4.5 5 6
2.1	— 280	280 —	4 4.5	6.5 7
2.5	— 100 280	100 280 —	3.8 4.5 5	6 6 7
3	— 280	280 —	5 5.5	8 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" or "r_1" and are described in the dimensional table.

(2) Metric tapered roller bearings

Unit: mm

$r_{s\,min}$ 2) or $r_{1s\,min}$	Nominal bore 3) 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	40 —	0.7 0.9	1.4 1.6
0.6	— 40	40 —	1.1 1.3	1.7 2
1	— 50	50 —	1.6 1.9	2.5 3
1.5	— 120 250	120 250 —	2.3 2.8 3.5	3 3.5 4
2	— 120 250	120 250 —	2.8 3.5 4	4 4.5 5
2.5	— 120 250	120 250 —	3.5 4 4.5	5 5.5 6
3	— 120 250 400	120 250 400 –	4 4.5 5 5.5	5.5 6.5 7 7.5
4	— 120 250 400	120 250 400 —	5 5.5 6 6.5	7 7.5 8 8.5
5	— 180	180 —	6.5 7.5	8 9
6	— 180	180 —	7.5 9	10 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 bearings whose dimensional series (refer to the dimensional table) are specified in the standard of ISO 355 or JIS B 1512-3.
For further information concerning bearings outside of these standards or tapered roller bearings using US customary units, please contact **NTN** Engineering.

(3) Thrust bearings

Unit: mm

$r_{s\,min}$ or $r_{1s\,min}$ 4)	$r_{s\,max}$ or $r_{1s\,max}$ Radial and axial direcition
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.

■ **Abutment height and fillet radius**

The shaft and housing abutment height (h) should be larger than the bearing's 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 (r_a) must be smaller than the bearing's minimum allowable chamfer dimension ($r_{s\,min}$) so that it does not interfere with bearing seating. **Table 2.11** lists abutment height (h) and fillet radius (r_a).

For bearings that support very large axial loads, shaft abutments (h) should be higher than the values in the table.

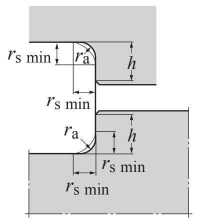


Table 2.11 Fillet radius and abutment height

Unit: mm

Chamfer length	Fillet radius (radius)	Shoulder height h (min)
$r_{s\,min}$ or $r_{1s\,min}$	$r_{as\,max}$	Normal use 1)
0.05	0.05	0.3
0.08	0.08	0.3
0.1	0.1	0.4
0.15	0.15	0.6
0.2	0.2	0.8
0.3	0.3	1.25
0.6	0.6	2.25
1	1	2.75
1.1	1	3.5
1.5	1.5	4.25
2	2	5
2.1	2	6
2.5	2	6
3	2.5	7
4	3	9
5	4	11
6	5	14
7.5	6	18
9.5	8	22
12	10	27
15	12	32
19	15	42

1) If bearing supports large axial load, the height of the shoulder must exceed the value given here.
Note: $r_{as\,max}$ maximum allowable fillet radius.

Where a fillet radius ($r_{a\,max}$) larger than the bearing chamfer dimension is required to strengthen the shaft or to relieve stress concentration [see **Fig. 2.16 (a)**], or where the shaft abutment height is too low to afford adequate contact surface with the bearing [see **Fig. 2.16 (b)**], spacers may be used effectively.

Relief dimensions for ground shaft and housing fitting surfaces are given in **Table 2.12**.

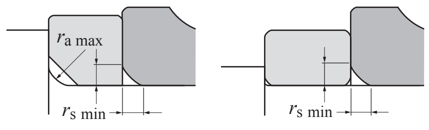


Fig. 2.16 Bearing mounting with spacer

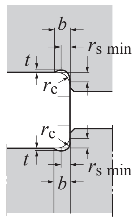


Table 2.12 Relief dimensions for grounding

Unit: mm

$r_{s\,min}$	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
2.5	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

3. Load Rating and Life

3.1 Bearing life

Even in bearings operating under normal conditions, the surfaces of the raceway and rolling elements are constantly being subjected to repeated compressive stresses which causes spalling (flaking, separation) of these surfaces to occur. This spalling 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 spalling of either the raceway surface or the rolling element surfaces occurs.

Other causes of bearing failure are often attributed to problems such as seizure, abrasions, cracking, chipping, scuffing, rust, etc. However, these so called "causes" of bearing failure are usually caused by improper installation, insufficient or improper lubrication, faulty sealing or improper 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 spalling aspect.

Usually, the load exerted on the main spindle of a machine tool is relatively small compared to the dynamic rated load on the bearing. Therefore, the fatigue life of a bearing seldom poses a problem.

The following operating conditions, rather than a bearing's rating life, can significantly affect the bearing functions (running accuracy, rigidity, heat generation, etc.) and require special consideration.

(1) High speed operation.
(2) Heavy preload.
(3) Large bending of the shaft.
(4) Large temperature difference between the inner and outer rings.

For further information, please consult **NTN** Engineering.

■ **Basic rating life and basic dynamic load rating**

A group of seemingly identical bearings when subjected to identical 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 rating life is defined as follows.

The basic rating 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 rating life (90 % reliability) is expressed in the total number of hours of operation.

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

The relationship between the basic rating life, the basic dynamic load rating and the bearing load is given in the formula below.

For ball bearings: $$L_{10} = \left(\frac{C}{P}\right)^3 \quad \cdots\cdots\cdots\cdots (3.1)$$

$$L_{10h} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^3 \quad \cdots\cdots (3.2)$$

For roller bearings: $L_{10} = \left(\frac{C}{P}\right)^{10/3}$ ··········· (3.3)

$$L_{10h} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^{10/3} \cdots (3.4)$$

Where:

L_{10} : Basic rating life, 10^6 revolutions

L_{10h}: Basic rating life, h

C : Basic dynamic load rating, N {kgf} (C_r: radial bearings, C_a: thrust bearings)

P : Equivalent dynamic load, N {kgf} (P_r: radial bearings, P_a: thrust bearings)

n : Rotational speed, min^{-1}

When several bearings are incorporated in machines or equipment as complete units, all the bearings in the unit are considered as a whole when computing bearing life (see formula 3.5).

$$L = \frac{1}{\left(\frac{1}{L_1{}^e} + \frac{1}{L_2{}^e} + \cdots \frac{1}{L_n{}^e}\right)^{1/e}} \cdots\cdots\cdots\cdots\cdots (3.5)$$

Where:

L : Total basic rating life of entire unit, h

L_1 , $L_2 \cdots L_n$: Basic rating life of individual bearings, 1, 2, ··· n, h

e = 10/9 ······ For ball bearings

e = 9/8 ········ For roller bearings

When the load conditions vary at regular intervals, the life can be given by formula (3.6).

$$L_m = \left(\frac{\phi_1}{L_1} + \frac{\phi_2}{L_2} + \cdots \frac{\phi_j}{L_j}\right)^{-1} \cdots\cdots\cdots\cdots\cdots (3.6)$$

Where:

L_m: Total life of bearing, h

ϕ_j : Frequency of individual load conditions ($\Sigma\phi_j = 1$)

L_j : Life under individual conditions, h

Adjusted rating life

The basic bearing rating life (90 % reliability factor) can be calculated by the formula (3.2) mentioned. 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 manufacturing process. Bearing life is also sometimes affected by operating conditions such as lubrication, temperature and rotational speed.

Basic rating life adjusted to compensate for this is called “adjusted rating life,” and is determined by using the formula (3.7).

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot L_{10} \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (3.7)$$

Where:

L_{na}: Adjusted rating life in millions of revolutions (10^6)

a_1 : Reliability factor

a_2 : Bearing characteristics factor

a_3 : Operating conditions factor

● Life adjustment factor for reliability a_1

The value of reliability factor a_1 is provided in **Table 3.1** for reliability of 90 % or greater.

Table 3.1 Reliability factor a_1

Reliability %	L_n	Reliability factor a_1
90	L_{10}	1.00
95	L_5	0.64
96	L_4	0.55
97	L_3	0.47
98	L_2	0.37
99	L_1	0.25
99.2	$L_{0.8}$	0.22
99.4	$L_{0.6}$	0.19
99.6	$L_{0.4}$	0.16
99.8	$L_{0.2}$	0.12
99.9	$L_{0.1}$	0.093
99.92	$L_{0.08}$	0.087
99.94	$L_{0.06}$	0.080
99.95	$L_{0.05}$	0.077

● Life adjustment factor for material a_2

Bearing characteristics concerning life vary according to bearing material, quality of material and use of special manufacturing processes. In this case, life is adjusted by the bearing characteristics factor a_2.

The basic dynamic load ratings listed in the catalog are based on **NTN**'s standard material and process, therefore, the adjustment factor a_2 = 1. a_2 > 1 may be used for specially enhanced materials and manufacturing methods. If this applies, consult **NTN** Engineering.

● Life adjustment factor for operating conditions a_3

Operating conditions factor a_3 is used to compensate for when the lubrication condition worsens due to rise in temperature or rotational speed, lubricant deteriorates, or becomes contaminated with foreign matters.

Generally speaking, when lubricating conditions are satisfactory, the a_3 factor has a value of one. And when lubricating conditions are exceptionally favorable and all other operating conditions are normal, a_3 can have a value greater than one. a_3 is however less than 1 in the following cases:

- Dynamic viscosity of lubricating oil is too low for bearing operating temperature (13 mm^2/s or less for ball bearings, 20 mm^2/s for roller bearings)
- Rotational speed is particularly low (pitch circle diameter across rolling elements d_m mm and rotational speed n min^{-1} is d_mn value < 10 000)
- Bearing operating temperature is too high
- Lubricant is contaminated with foreign matter or moisture

Life calculation for machine tool main spindle bearing

By dramatic improvement in bearing materials and bearing manufacturing techniques, bearings can offer a life several times as long as that calculated from the formula (3.7) as long as they are mounted with minimal mounting errors and are fully free from foreign matter and adequately lubricated. This finding was obtained by a series of experiments performed by **NTN**. The formula for calculating life of a machine tool main spindle bearing uses the life correction factor, a_{NTN}. This correction factor is based on a contact stress of 1.5 GPa at the fatigue limit specified in ISO 281: 1990/Amd. 2: 2000 under clean and well lubricated conditions.

Bearing life theory

(1) Conventional Lundberg-Palmgren (L-P) theory

According to this theory, a stress that governs rolling fatigue is considered, that is, a maximum dynamic shear stress τ_0 that is exerted, at a depth of Z_0 from the rolling contact surface, in a plane parallel with the rolling contact surface. Referring to a theory of Neuber, et. al. which claims that the durability of a material deteriorates as the volume being subjected to a stress application decreases, the L-P theory assumes that a fissure occurring at a weak point of material at around the depth Z_0 reaches the surface and leads to develop failure [spalling (flaking, separation)]. The probability of survival S of a volume V that is subjected to N times of stress application is determined by the formula below according to the Weibull theory.

$$\ell_n \frac{1}{S} \propto \frac{N^e \tau_0^c V}{z_0^h} \quad \cdots\cdots (3.8)$$

Where:

S : Probability of survival of stress volume V
N : Number of repeated stress applications
e : Weibull slope (index to represent variation in life)
τ_0 : Maximum shear stress
Z_0 : Depth from surface at which maximum shear stress occurs
c, h : Indexes

From the basic formula for the bearing life relative to rolling fatigue (3.8), a generic life formula below is obtained:

$$L_{10} = \left(\frac{C}{P}\right)^p \quad \cdots\cdots (3.9)$$

Where:

L_{10} : Basic rating life, 10^6 revolutions
C : Basic dynamic load rating, N {kgf}
P : Dynamic equivalent load, N {kgf}
p : $(c - h + 2)/3e$ (point contact)
$(c - h + 1)/2e$ (line contact)

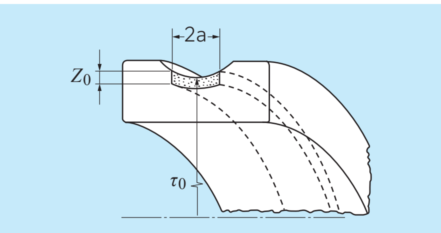


Fig. 3.1 Stress volume resulting from rolling contact according to L-P theory

(2) NTN's new bearing life theory

While the L-P theory intends to define internally occurring spalling owing to the shear stress within a material that results from hertzian contact, **NTN**'s new bearing life theory is designed not only to evaluate surface-initiated spalling but also to determine life of each small segment (ΔL_1) based on a local stress (σ_1). This is done by dividing an area from the interior to the contact surface of the material into small segments as illustrated in **Fig. 3.2**, and finally obtaining the overall bearing life L by the formula (3.12).

$$\ell_n \frac{1}{\Delta S_i} \propto \frac{\Delta N_i^e \sigma_i^c \Delta V_i}{z_i^h} \quad \cdots\cdots (3.10)$$

$$\Delta L_i = \Delta N_i \propto (\sigma_i^{-c} \Delta V_i^{-1} z_i^h)^{1/e} \quad \cdots\cdots (3.11)$$

$$L = \left\{ \sum_{i=1}^{n} \Delta L_i^{-e} \right\}^{-1/e} \quad \cdots\cdots (3.12)$$

Where:

ΔS_i : probability of survival of stress volume ΔV_i of divided segment
L : Overall bearing life
Z_i : Depth of divided small stress volume ΔV_i from the surface
n : Number of segments
σ_u : Fatigue limit stress

- A stress below which a bearing does not develop failure [spalling (flaking, separation)] under ideal lubrication conditions.
- ISO 281: 1990/Amd. 2: 2000 specifies 1.5 GPa as a the maximum contact stress at a fatigue limit. **NTN** uses it as a Von Mises stress equivalent to the maximum contact stress 1.5 GPa.
- When σ_i is smaller than σ_u (fatigue limit), the life of a region in question (ΔL_1) will be infinitely long.

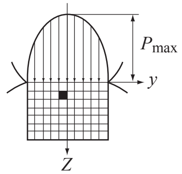


Fig. 3.2 Calculation model

■ NTN's new bearing life formula

The correlation between the **NTN**'s life correction factor a_{NTN} and corrected rating life L_{nm} is defined by the formula (3.13) below.

$$L_{nm} = a_1 \cdot a_{NTN} \cdot \left(\frac{C}{P}\right)^p \quad \cdots\cdots (3.13)$$

Where:

L_{nm} : Corrected rating life, 10^6 revolutions
a_1 : Reliability coefficient
a_{NTN} : Life correction factor that reflects material properties, fatigue limit stress, contamination with foreign matter and oil film parameter (Λ) ($0.1 \le a_{NTN} \le 50$)
C : Basic dynamic load rating, N {kgf}
P : Dynamic equivalent load, N {kgf}
p : Index 3 (ball bearing) 10/3 (roller bearing)

(1) Effect of fatigue limit

NTN's new bearing life formula introduces a concept of fatigue life according to which the bearing life is infinitely long at a particular contact stress as illustrated in **Fig. 3.3** assuming no foreign matter is trapped in the bearing and the bearing is reliably lubricated.

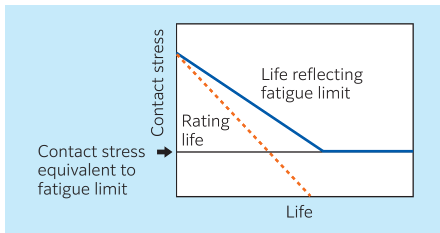


Fig. 3.3 Basic concept of fatigue limit

(2) Effect of foreign matter

The effect of foreign matter is treated as surface-initiated spalling that starts from a dent resulting from trapped foreign matter. **NTN** performs a bearing life calculation, assuming that the size of foreign matter and the stress concentration area in the middle portion (the size of this area corresponds with that of the foreign matter) in the surface layer as well as the amount of foreign matter significantly affect the bearing life.

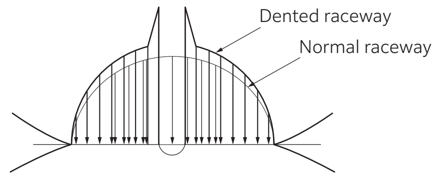


Fig. 3.4 Contact stress distribution resulting from dent

(3) Effect of oil film parameter (Λ)

The oil film parameter can be used to calculate bearing life. The oil film parameter, designated by Λ, is the ratio of the oil film thickness to the roughness of the surface. It can be used to calculate the average stress across the surface layer of two contacting surfaces, such as a rolling element and raceway. From this surface layer stress, the contact stress can be determined. Bearing life is then calculated from the contact stress.

[Conditions of two objects on surface layer]
Calculation model

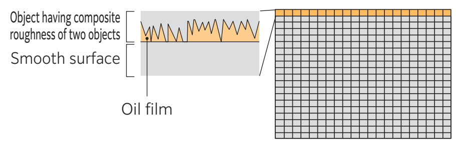


Fig. 3.5 Model of stress load onto the surface layer

■ **New life calculation formula chart**

Various statuses of contamination with foreign matter are defined in **Table 3.2**. The values of ISO codes and NAS classes are those for ball bearings that are subjected to more severe operating conditions.

Table 3.2 Status of contamination

Condition of contamination	Extremely clean	Clean	Normal	Lightly contaminated	Moderately contaminated	Highly contaminated	Severely contaminated
Contamination coefficient	1	0.8	0.5	0.4	0.3	0.2	0.1
Guideline for application	Filtered						No filter
	Less than 10 μm	10 to 30 μm	30 to 50 μm	50 to 70 μm	70 to 100 μm	100 μm or more	Ingress of much dust
ISO cleanliness code (ISO 4406)	13/10	15/12	17/14	19/16	21/18	23/20	25/22
NAS class	4	6	8	10	12	—	—

(1) Effect of foreign matter on correlation between load (P/C) and life correction factor a_{NTN}

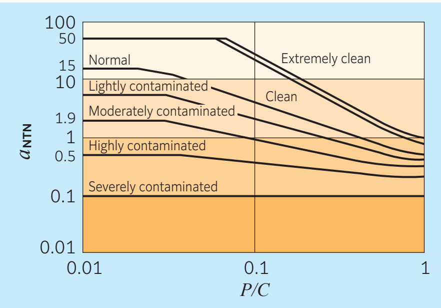


Fig. 3.6 Correlation between P/C and a_{NTN} (effect of foreign matter in ball bearing)

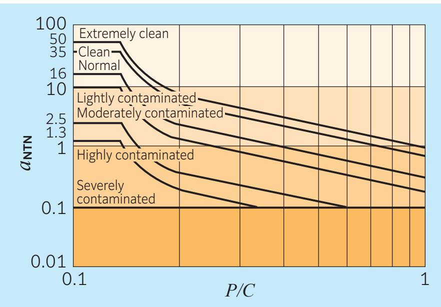


Fig. 3.7 Correlation between P/C and a_{NTN} (effect of foreign matter in roller bearing)

(2) Effect of oil film parameter (Λ) on correlation between load (P/C) and life correction factor a_{NTN}

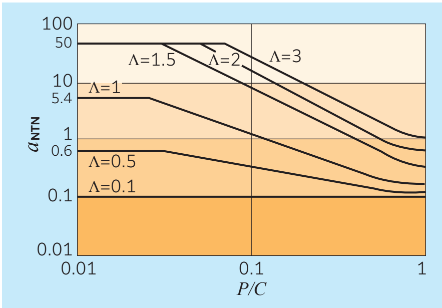


Fig. 3.8 Correlation between P/C and a_{NTN} (effect of Λ with ball bearing)

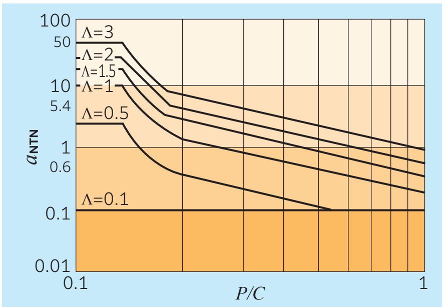


Fig. 3.9 Correlation between P/C and a_{NTN} (effect of Λ with roller bearing)

3.2 Static load rating and allowable axial load

■ **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, occurring 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 static load rating 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 thrust bearings. The maximum applied load values for contact stress occurring at the rolling element and raceway contact points are given below.

For ball bearings 4 200 MPa
For roller bearings 4 000 MPa

Referred to as “basic static radial load rating” for radial bearings and “basic static axial load rating” for thrust bearings, basic static load rating is expressed as C_{0r} or C_{0a} respectively and is provided in the bearing dimensions table.

■ **Allowable static equivalent load**

Generally the static equivalent load which can be permitted is limited by the basic static rating load as stated above. However, depending on requirements regarding friction and smooth operation, these limits may be greater or lesser than the basic static rating load.

This is generally determined by taking the safety factor S_0 given in **Table 3.3** and formula (3.14) into account.

$$S_0 = C_0 / P_0 \quad \cdots\cdots (3.14)$$

Where:

S_0 : Safety factor
C_0: Basic static load rating, N {kgf}
radial bearings : C_{0r}
thrust bearings: C_{0a}
P_0: Static equivalent load, N {kgf}
radial bearings : P_{0r}
thrust bearings: P_{0a}

Table 3.3 Minimum safety factor values S_0

Operating conditions	Ball bearings	Roller bearings
Applications that require quiet rotation	2	3
Applications subjected to impact loads	1.5	3
Normal rotation applications	1	1.5

Note: When vibration and/or shock loads are present, a load factor based on the shock load needs to be included in the P_0 max value.

Load Rating and Life

Allowable axial load

A greater axial load can be exerted on a main spindle bearing on a machine tool allowing for tool changes while the machine is stationary. When an angular contact ball bearing is subjected to a larger axial load, the contact ellipse between its rolling elements and raceway surface can overflow the raceway surface (see **Fig. 3.10**). Furthermore, even if the contact ellipse remains within the raceway surface, overstressing can cause problems such as denting.

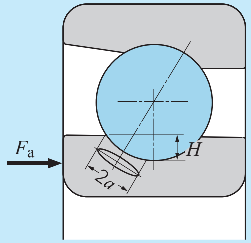
The limit of this load is known as the "allowable axial load."

- **The end of contact ellipse on the raceway surface reaches the shoulder of either an inner or outer ring.**
- **The contact stress on the raceway surface reaches 3 650 MPa in either the inner or outer ring raceway.**

NTN, the maximum allowable load that does not cause such problems is defined as the "allowable axial load."

Note that the contact stress of 3 650 MPa on the raceway surface is a value that leads to a permanent deformation of 0.00002 to 0.00005 times as much as the rolling element diameter and has been determined through many years of experience.

The allowable axial load for each bearing is found in the associated dimensions table.



H : Shoulder height of bearing
a : Semimajor of contact ellipse
F_a: Axial load

Fig. 3.10

Allowable Speed

4. Allowable Speed

High bearing speed leads to high temperature rise on the bearing owing to frictional heating within the bearing. When the temperature of the bearing exceeds a particular limit, the lubricant performance deteriorates significantly, possibly leading to bearing overheating or seizure.

The factors that can affect the maximum allowable bearing speed include:

(1) Bearing type
(2) Bearing size
(3) Lubrication system (grease lubrication, air-oil lubrication, jet lubrication, etc.)
(4) Internal clearance or preload on the bearing
(5) Bearing arrangement (2-row to 5-row)
(6) Bearing load
(7) Accuracies of shaft, housing, etc.

The maximum allowable speeds listed in the bearing dimensions tables are reference values and are applicable only to individual bearings that are adequately lubricated and correctly preloaded under a condition where the heat is reliably removed from the bearing arrangement.

In the case of grease lubrication, these speeds are attainable only when the bearing is filled with an adequate amount of high-quality grease as given in **Table 7.3**, the bearing is sufficiently run in, and heat is removed by an arrangement such as a cooling jacket. In the case of oil lubrication, these speeds are attained only by an air-oil lubrication system if an adequate amount of ISO VG22 to 32 spindle oil is supplied and the heat is removed by an arrangement such as a cooling jacket. When using a large amount of lubricant, a jet lubrication system excels in lubrication and cooling performance, and can permit operation at the maximum allowable speed. However, this lubrication system involves a high power loss and should be employed carefully.

Speed factor of fixed position preloading

The bearing arrangements (2-row to 5-row) and speed reduction ratios (speed factors) for maximum allowable speed due to post-assembly preloads are summarized in **Table 4.1**.

The maximum allowable speed of a particular bearing can vary depending on the relation between heat generation and heat dissipation in the bearing as well as how well the bearing is lubricated.

Furthermore, to continue operation at high speeds and with varying preloads, it is recommended that bearings at the upper limit are multiplied by a factor of 0.8.

Table 4.1 Speed factor by bearing arrangement and preload

Bearing arrangement	Matching	GL	GN	GM
	DB	0.85	0.8	0.65
	DBT	0.7	0.6	0.5
	DTBT	0.8	0.75	0.6
	DTBTT	0.7	0.6	0.5

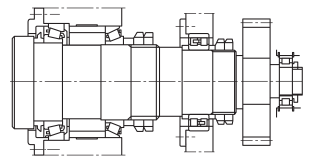
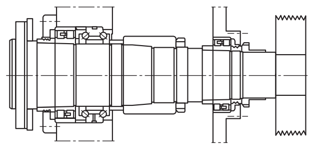
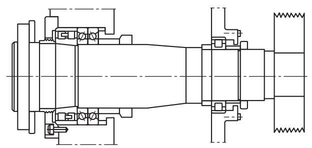
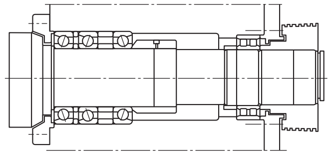
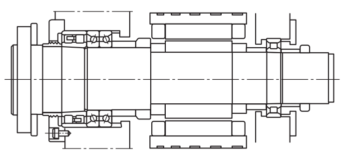
5. Bearing Arrangements and Structures of Bearings for Main Spindles

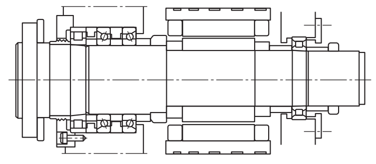
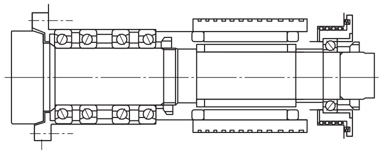
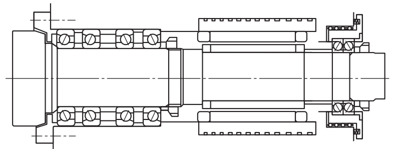
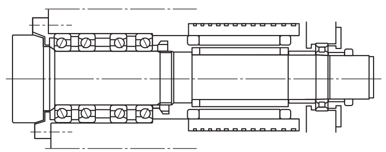
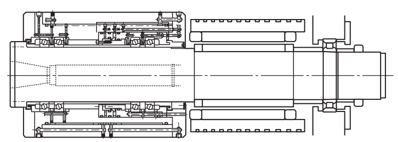
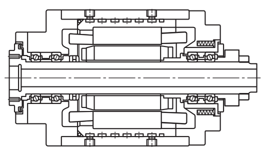
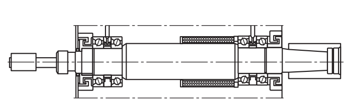
5.1 Bearing arrangement for main spindles

Typical examples of bearing arrangements for main spindles of machine tools are summarized in **Table 5.1**.

An optimal bearing arrangement must be determined through considerations about the properties required of the main spindle in question (maximum speed, radial and axial rigidities, main spindle size, required accuracies, lubrication system, etc.). And, machine tool models incorporate built-in motor type main spindles. However, heat generation on a built-in motor can affect the accuracy of the main spindle and performance of lubricant, so a main spindle bearing should be selected very carefully.

Table 5.1 Typical examples of bearing arrangements for main spindles

Bearing arrangement for main spindle	Bearing type	Typical applications
 Gear-driven configuration	[Type I] Tapered roller bearing + Tapered roller bearing + Double-row cylindrical roller bearing	Large turning machine Oil country lathe General-purpose turning machine Lubrication method ●Grease lubrication
 Belt-driven configuration	[Type II] Double-row cylindrical roller bearing + Double-direction angular contract thrust ball bearing + Double-row cylindrical roller bearing	CNC turning machine Machining center Boring machine Milling machine Lubrication method ●Grease lubrication
 Belt-driven configuration	[Type III] Double-row cylindrical roller bearing + High speed duplex angular contact ball bearing for axial load + Single-row cylindrical roller bearing NOTE: high speed variant of type II	CNC turning machine Machining center Milling machine Lubrication method ●Grease lubrication
 Belt-driven configuration	[Type IV] Duplex angular contact ball bearing (DBT arrangement) + Double-row cylindrical roller bearing NOTE: high speed variant of type II or III	CNC turning machine Machining center Milling machine Lubrication method ●Grease lubrication
 Built-in motor-driven configuration	[Type V] Double-row cylindrical roller bearing + High speed duplex angular contact ball bearing for axial load + Single-row cylindrical roller bearing NOTE: high speed variant of type III with built-in motor-driven configuration	CNC turning machine Machining center Milling machine Lubrication method ●Grease lubrication ●Air-oil lubrication

Bearing arrangement for main spindle	Bearing type	Typical applications
 Built-in motor-driven configuration	[Type VI] Single-row cylindrical roller bearing + High speed duplex angular contact ball bearing for axial load + Single-row cylindrical roller bearing NOTE: high speed variant of type V	CNC turning machine Machining center Lubrication method ●Grease lubrication ●Air-oil lubrication
 Built-in motor-driven configuration	[Type VII] Duplex angular contact ball bearing (DTBT arrangement) + Single-row angular contact roller bearing (w/ ball slide) NOTE: ultra high speed variant	Machining center <vertical> Lubrication method ●Grease lubrication ●Air-oil lubrication
 Built-in motor-driven configuration	[Type VIII] Duplex angular contact ball bearing (DTBT arrangement) + Duplex angular contact roller bearing (w/ ball slide) NOTE: ultra high speed variant	Machining center <vertical> Lubrication method ●Grease lubrication ●Air-oil lubrication
 Built-in motor-driven configuration	[Type IX] Duplex angular contact ball bearing (DTBT arrangement) + Single-row cylindrical roller bearing NOTE: ultra high speed variant	Machining center Lubrication method ●Grease lubrication ●Air-oil lubrication
 Built-in motor-driven configuration	[Type X] Adjustable preload bearing unit + Duplex angular contact ball bearing (DBT arrangement) + Single-row cylindrical roller bearing NOTE: high-rigidity/ultra high speed variant	Machining center Lubrication method ●Air-oil lubrication
 Built-in motor-driven configuration	[Type XI] Duplex angular contact ball bearing (DT arrangement) + Duplex angular contact ball bearing (DT arrangement)	Machining center Small turning machine Grinding machine Lubrication method ●Grease lubrication ●Air-oil lubrication
 Belt-driven configuration	[Type XII] Duplex angular contact ball bearing (DT arrangement) + Duplex angular contact ball bearing (DT arrangement)	Grinding machine Lubrication method ●Grease lubrication ●Air-oil lubrication ●Oil-mist lubrication

5.2 Bearing selection based on bearing arrangement for main spindle

An optimal bearing product that best suits the application is selected by referring to the bearing selection table in **Table 5.2**, which contains the possible bearing arrangements for main spindles.

- Designate the free side and fixed side.
- Select the bearing arrangement type (Ⅰ to Ⅻ) on the free or fixed side.
- Select a set of bearing specifications applicable to the selected arrangement type.
- Choose a lubrication system suitable for the selected bearing specifications.
- Select a product group that satisfies the above-mentioned considerations.

Table 5.2 Bearing selection table

Fix side	Free side	Bearing specifications	Lubrication system	Applicable product groups Steel balls/ceramic balls	Considerations for selection procedure
Duplex angular contact ball bearing or adjustable preload bearing mechanism + Duplex angular contact ball bearing Bearing arrangement [Type Ⅳ, Ⅶ, Ⅷ, Ⅸ, Ⅺ, or Ⅻ]	Single-row angular contact ball bearing or duplex angular contact ball bearing (w/ ball bush) Bearing arrangement [Type Ⅶ, Ⅷ, Ⅺ, or Ⅻ]	Angular contact ball bearing for radial load Contact angle 30° or smaller	Grease lubrication	Sealed 【15˚, 25˚】 79 LLB/5S-79 LLB 70 LLB/5S-70 LLB 【15˚, 20˚, 25˚】 2LA-BNS9 LLB/5S-2LA-BNS9 LLB 2LA-BNS0 LLB/5S-2LA-BNS0 LLB	Bearing selection ① High speed performance (general) High ⇔ Low Contact angle 15°, 20°, 25°, 30° ② Rigidity • Radial rigidity High ⇔ Low Contact angle 15°, 20°, 25°, 30° • Axial rigidity Low ⇔ High Contact angle 15°, 20°, 25°, 30°, 40°, 60° • Complex rigidity (radial and axial) High (4-row) Medium (3-row) Low (2-row) ③ Recommended arrangement 4-row (DTBT) or 2-row (DB) ④ Recommended lubrication specifications Standard main spindle: Grease High speed main spindle: Air-oil Low-noise: Grease or eco-friendly air-oil ⑤ Presence of cooling jacket around the bearing. In particular, grease lubrication is recommended.
			Air-oil lubrication / Grease lubrication	【15˚】78C 72C 【15˚, 25˚, 30˚】 79U/5S-79U, 70U/5S-70U 【15˚, 20˚, 25˚】 2LA-HSE9U/5S-2LA-HSE9U 2LA-HSE0/5S-2LA-HSE0	
			Air-oil lubrication / Grease lubrication	Bearings for grinding machines/ motors 【15˚】BNT9/5S-BNT9 BNT0/5S-BNT0 BNT2/5S-BNT2	
			Air-oil lubrication	Ultra high speed/dedicated air-oil lubrication series 【25˚】5S-2LA-HSF0	
			Air-oil lubrication	Eco-friendly type 【20˚, 25˚】 5S-2LA-HSL9U 5S-2LA-HSL0 5S-2LA-HSFL0 With re-lubricating hole on the outer ring 【20˚, 25˚】 5S-2LA-HSEW9U 5S-2LA-HSEW0	
Cylindrical roller bearing + Duplex angular contact ball bearing Bearing arrangement [Type Ⅱ, Ⅲ, Ⅴ or Ⅵ]	Double-row cylindrical roller bearing or single-row cylindrical roller bearing Bearing arrangement [Type Ⅰ, Ⅱ, Ⅲ, Ⅳ, Ⅴ, Ⅵ, Ⅸ or Ⅹ]	Cylindrical roller bearing	Air-oil lubrication / Grease lubrication	Double-row NN30/NN30K NN30HS/NN30HSK NN30HST6/NN30HST6K NN30HSRT6/NN30HSRT6K NN49/NN49K NNU49/NNU49K	
			Air-oil lubrication / Grease lubrication	Single-row N10HS/N10HSK N10HSRT6/N10HSRT6K	
			Air-oil lubrication	Single-row Eco-friendly type N10HSLT6/N10HSLT6K	
		Angular contact ball bearing for axial load Contact angle less than 60°	Air-oil lubrication / Grease lubrication	【30˚】HTA9UA HTA0UA/5S-HTA0UA 【40˚】HTA9U HTA0U/5S-HTA0U 【60˚】5629/5629M 5620/5620M	
		Thrust contract ball bearing	Grease lubrication		
Tapered roller bearing + Cylindrical roller bearing Bearing arrangement [Type Ⅰ]		Cylindrical roller bearing	Oil lubrication / Grease lubrication	329XU 4T-320X/320XU Inch series tapered roller bearing	

5.3 Adjustable preload bearing unit

A recent trend in the machine tool industry is a steady increase of operating speeds. The maximum $d_m n$ value [pitch circle diameter across rolling elements d_m (mm) multiplied by speed n (min^{-1})] reached by main spindles with air-oil lubricated lubrication can be as high as 2.5 to 3.8 × 10^6. At the same time, main spindles are requiring increased rigidity. Therefore, main spindle bearings must be capable of both high speed operation and high rigidity. This can be achieved through optimal preloading.

A fixed preload (spring preload) system is usually employed to satisfy both these high speed and high rigidity requirements. A spindle unit with fixed-position preload that is adjustable for different speed conditions is advantageous for optimizing the rigidity of the unit.

The **NTN** Adjustable Preload Bearing Unit is a high speed, high-rigidity unit that features fixed position preload that can be adjusted for different speed conditions.

The **NTN** Adjustable Preload Bearing Unit is illustrated in **Fig. 5.1**. Hydraulic pressure is used to shift the position of the adjustable preload sleeve situated in the rear bearing section of the unit. This changes the preload on the bearings.

A spindle incorporating a 3-step adjustable preload bearing unit is illustrated in **Fig. 5.2**. The sleeve in the adjustable preload section is comprised of two hydraulic pressure chambers, A and B, as well as a spiral groove for sliding motion. The preload can be adjusted to one of three settings by changing the hydraulic pressure in each of the chambers. To achieve instantaneous and reliable adjustment, high-pressure oil (at the same pressure as in the hydraulic chambers) is supplied to the spiral groove on the outside of the sleeve. This oil provides lubrication so that the sleeve can move smoothly.

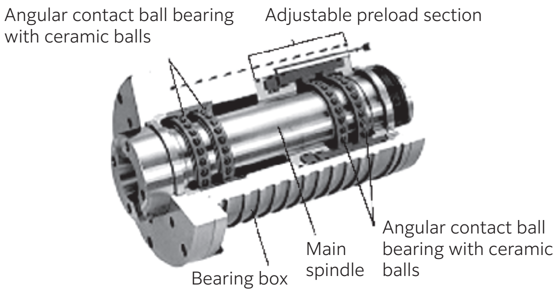


Fig. 5.1 Adjustable preload bearing unit

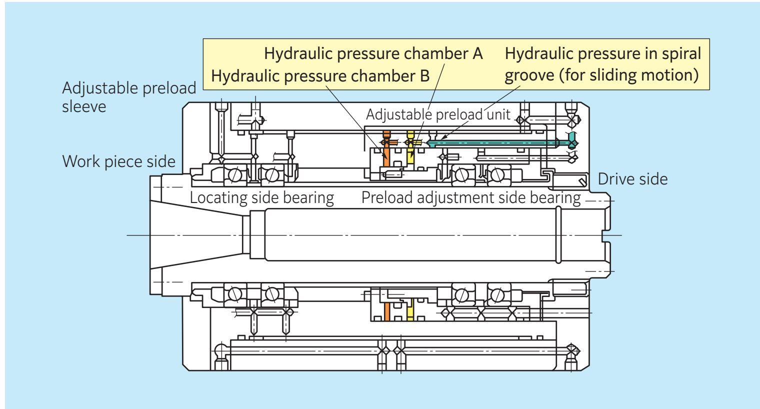


Fig. 5.2 Typical spindle configuration incorporating 3-step Adjustable Preload Type Bearing Unit

■ Operating mechanism

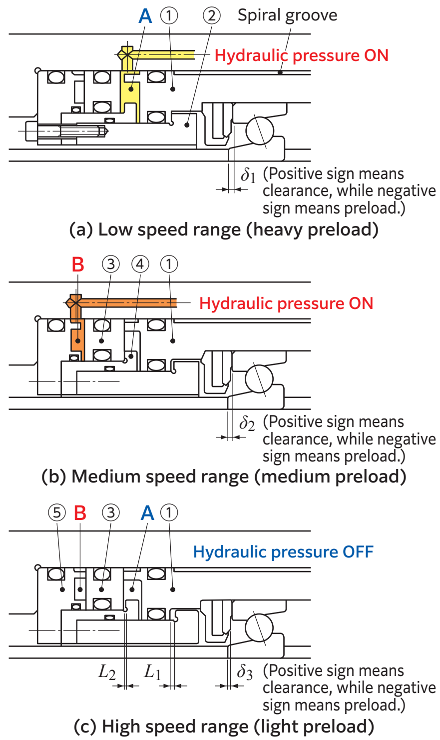
Fig. 5.3 shows the hydraulic operation of the unit for three preloading conditions as well as the associated motion of the adjustable preload sleeve.

• **Low speed operation (heavy preload): Chamber A is pressurized.**
Component ① moves to the right by a preset clearance L_1 and contacts Component ②. The axial clearance is δ_1 [see **Fig. 5.3 (a)**].

• **Medium speed operation (medium preload): Chamber B is pressurized.**
Components ① and ③ move to the right by a preset clearance L_2, causing Component ③ to contact Component ④. The axial clearance is δ_2 [see **Fig. 5.3 (b)**].

• **High speed operation (light preload): Chambers A and B are not pressurized.**
Components ① and ③ return [1] to the left due to the reaction force on the bearing. This causes Component ③ to contact Component ⑤, thereby returning the axial clearance to the initial setting of δ_3 [see **Fig. 5.3 (c)**].

1) The return motion of the components ① and ③ is achieved by the reaction force of bearing or a separately provided spring.



∗ $L_1 = 2(\delta_3 - \delta_1)$ $L_2 = 2(\delta_3 - \delta_2)$

Fig. 5.3 Operating mechanism of Adjustable preload

5.4 Bearing jacket cooling system

With a built-in motor drive system, the main spindle is directly driven by a motor and is therefore suitable for rapid acceleration or deceleration. However, this system can be adversely affected by temperature rise. A cooling jacket with a spiral groove around the housing allows cooling oil to flow through the unit.

If heat generated by the motor affects the bearing, overheating of the bearing as well as degradation of the grease can occur.

■ Considerations about cooling of jacket

A typical bearing arrangement is shown in **Fig. 5.4** and **Fig. 5.5**, comprising a double-row cylindrical roller bearing and an angular contact ball bearing set. The cooling groove on the jacket in **Fig. 5.4** starts at around an area above the angular contact ball bearings and does not cool the double-row cylindrical roller bearing effectively (The fit of the angular contact ball bearings with the bore of the housing is a loose fit, the bearings are not in direct contact with the housing). In the configuration in **Fig. 5.5**, the cooling groove extends to the region above the double-row cylindrical roller bearing, and cools both the angular contact ball bearings and the double-row cylindrical roller bearing effectively.

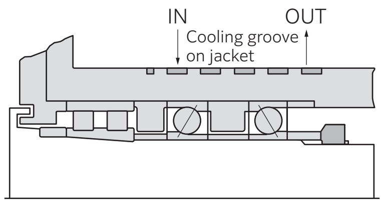


Fig. 5.4 Inadequate cooling groove on jacket

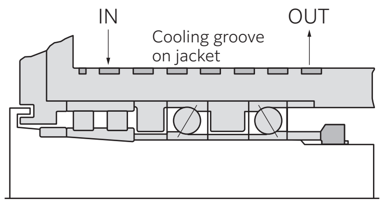


Fig. 5.5 Adequate cooling groove on jacket

[Test conditions]

Bearing	5S-2LA-HSE014DB
Preload after assembled	100 N (fixed position preloading)
Lubrication	Air-oil lubrication

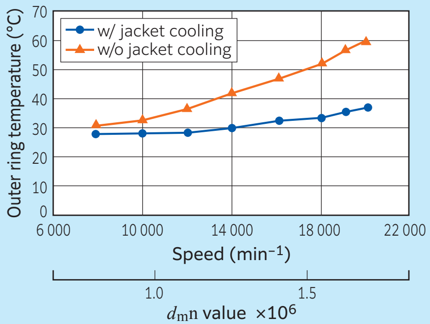


Fig. 5.6 Variation in bearing temperature depending on presence/absence of jacket cooling (angular contact ball bearing)

[Test conditions]

Bearing	NN3028HST6
Mounted radial clearance	−5 μm
Lubrication	Grease lubrication

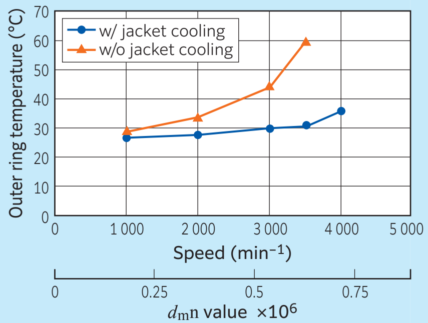


Fig. 5.7 Variation in bearing temperature depending on presence/absence of jacket cooling (cylindrical roller bearing)

6. Handling of Bearings

6.1 Cleaning and filling with grease

To achieve maximum speed and limited temperature rise with a precision rolling bearing, it is vital to handle the bearing correctly.

The handling of bearings involves cleaning, drying, filling with grease (if necessary), and the running-in operation. For each step, follow the precautions and instructions.

A sealed bearing contains prefilled grease. Do not clean (rinse) and dry this type of bearing. Only wipe away rust-preventive oil with a clean cloth before assembling the bearing.

■ Cleaning (removal of rust-preventive oil)

- Immerse the bearing in kerosene or a highly volatile solvent such as naphthesol and wash it turn the by hand. Then remove the kerosene using benzene or alcohol. Use clean compressed air to blow away the rinsing fluid.
  (For air-oil lubrication, it is recommended that after cleaning, the bearing should either be coated with the application specific lubricant or a less viscous oil)

■ Drying

If the bearing is to be used with grease lubrication, it is necessary to thoroughly dry the bearing to avoid leakage of grease. After drying, be sure to immediately fill the bearing with grease.

Drying can be performed by blowing hot air onto the bearing or placing the bearing in a chamber at constant temperature. When drying by hot air, be sure to consider the cleanliness of the air.

Filling with grease

The procedures for greasing ball and roller bearings can be found below.

After filling with grease, turn the bearing by hand to uniformly distribute the grease to the whole rolling surface.

<Ball bearings> See Table 6.1

- By using an injector or small plastic bag, fill grease between balls in equal amounts, aiming at the inner ring rolling surface.
- For a bearing with a ring-guided cage, also apply grease to the guide surface of the cage using a spatula or similar tool.

Table 6.1 Filling grease into angular contact ball bearing

	By using an injector, fill grease between balls in equal amounts, aiming at the rolling surface of the inner ring. Apply grease to the guide surface as well for outer ring guide cages.
	[After completion of filling]
	Turn the bearing by hand while applying an appropriate load in the contact angle direction so that the any area in the interior of bearing is sufficiently lubricated with grease. When doing so, check that the grease adheres to the surface of the balls.

- If grease cannot be filled into the inner ring rolling surface because of a small gap between the cage and the inner ring add grease to the outer ring rolling surface. In this case, carefully turn the bearing so that the grease is fully spread on the inner ring side.

<Roller bearings> See Table 6.2

- Apply grease to the outer (inner) side of rollers, and while turning the rollers with fingers, spread the grease to the inner ring side.

Table 6.2 Filling grease into cylindrical roller bearing

	Apply grease to the outer circumference surface of cage.
	Spread the grease over the roller outer diameter surface. Apply grease to the bearing rollers and rotate by hand to coat the inner ring surface. Ensure that grease will coat the pocket, mating, and roller end surfaces of the cage.
	[After applying grease] If a lump of grease remains on the cage rib outside diameter surface, the running-in operation can take a longer time. Apply the grease to the outer diameter surface of the cage ribs. Use fingers to coat the surface towards the roller ends (see photo on left).

Running-in operation

(1) Air-oil or oil-mist lubrication

The running-in operation is relatively simple with oil lubrication because no peak temperature occurs and the bearing temperature stabilizes within a relatively short time. **NTN** recommends that the speed of bearing is to be increased in steps of 2 000 to 3 000 min^{-1} until the maximum speed is reached.

Every speed setting should be maintained for about 30 minutes. However, for the speed range where the $d_m n$ value (pitch circle diameter across rolling elements multiplied by speed) exceeds 1.0 × 10^6, increase the bearing speed in steps of 1 000 to 2 000 min^{-1} to ensure the stable running.

(2) Grease lubrication

For a grease-lubricated bearing, a running-in operation is very important in attaining stable temperature rise. During a running-in operation, a large temperature rise (peak) occurs while the bearing speed is increased, and then the bearing temperature eventually stabilizes. Refer to the section “6.12 Running in operation for main spindle bearing.”

<Ball bearings>

NTN recommends that the bearing speed be increased in steps of 1 000 to 2 000 min^{-1} and be further increased only after the temperature has stabilized at the current speed setting.

However, for the speed range where the $d_m n$ value exceeds 0.4 × 10^6, increase the bearing speed in steps of 500 to 1 000 min^{-1} to ensure the stable running.

<Roller bearings>

Compared with contact ball bearings, the time to peak temperature or saturation in running-in operation of roller bearings tends to be longer. Also, there will be temperature rise due to whipping of the grease and the temperature rise may be unstable. To cope with this problem, run the roller bearing in the maximum speed range for a prolonged period.

Increase the bearing speed in steps of 500 to 1 000 min^{-1} only after the bearing temperature has stabilized at the current speed setting.

For the speed range where the $d_m n$ value exceeds 0.3 × 10^6, increase the bearing speed in steps of 500 min^{-1} to ensure safety.

6.2 Mounting

When mounting a bearing to a main spindle, follow either of the mounting techniques described below:

(1) Press-fitting with hydraulic press
(2) Mounting by heating bearings

With either technique, it is important to minimize the adverse effects of the mounting process to maintain bearing accuracy.

(1) Press-fitting with hydraulic press
Before press-fitting a bearing with a hydraulic press or hand press, the press-fitting force due to the interference between the shaft and inner ring must be calculated. A hydraulic press having a capacity greater than the required press-fitting force must used. Next, using an inner ring press-fitting jig, the inner ring is correctly press-fitted to the shoulder of shaft. Please be careful not to exert a force on the outer ring (see **Fig. 6.1**).

After the press-fitting operation, it is important to measure the accuracies of various portions of the bearing to verify that the bearing has been correctly mounted to the shaft. When using a multi row bearings, measure the runout after assembly and correct misalignment across the outer rings as necessary (see **Fig. 6.2** and **Fig. 6.3**).

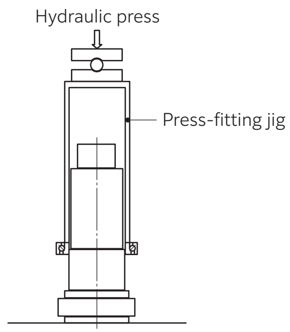


Fig. 6.1 Press-fitting pressure

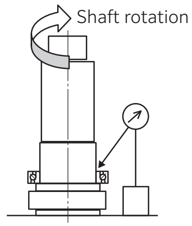


Fig. 6.2 Checking for face runout of inner ring

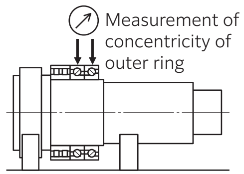


Fig. 6.3 Checking for concentricity of outer ring

Handling of Bearings NTN
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■ **Calculation of press-fitting force**
The press-fitting force occurring from the interference between the shaft and inner ring can be determined by the formula given below.

According to the calculated press-fitting force, a hydraulic press having a sufficiently large capacity must be used to mount the bearing. The variations in dimensional errors among the bearings should be considered. The force needed to press the inner ring to the shaft can be obtained with the following formula (6.1).

Force to press-fitting inner ring to shaft

$$K_d = \mu \cdot P \cdot \pi \cdot d \cdot B \quad \cdots\cdots \quad (6.1)$$

Where:

- K_d : Force for press-fitting or extracting an inner ring, N
- P : Surface pressure on fitting surface, MPa (see **Table 6.3**)
- d : Shaft diameter, inner ring bore diameter, mm
- D : Outer ring outside diameter, mm
- B : Inner ring, width
- μ : Sliding friction coefficient (when press-fitting inner ring over cylindrical shaft: 0.12)

Table 6.3

	Fitting conditions and calculation formulas	Symbol (Unit: mm)
Fitting surface pressure MPa	Fits between solid steel shaft and inner ring $P = \frac{E}{2}\frac{\Delta_{deff}}{d}\left[1-\left(\frac{d}{D_i}\right)^2\right] \quad \cdots\cdots \quad (6.2)$	d : Shaft diameter, inner ring bore diameter d_0 : Hollow shaft bore diameter D_i : Inner ring average raceway diameter
	Fits between hollow steel shaft and inner ring $P = \frac{E}{2}\frac{\Delta_{deff}}{d}\frac{[1-(d/D_i)^2][1-(d_0/d)^2]}{[1-(d_0/D_i)^2]} \quad \cdots\cdots \quad (6.3)$	Δ_{deff} : Effective interference E : Modulus of longitudinal elasticity = 208 000 MPa

$$\Delta_{deff} = \frac{d}{d+2}\Delta d \quad \cdots\cdots \quad (6.4)$$

(In the case of a ground shaft)
Δd : Theoretical interference fitting, μm

$$D_i = 1.05\frac{4d+D}{5} \quad \cdots\cdots \quad (6.5)$$

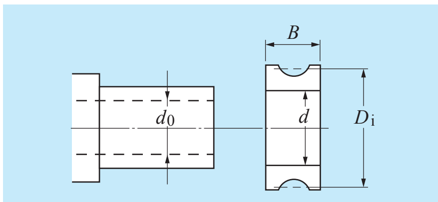

Fig. 6.4

<Example of calculation for press-fitting force>
The calculation for press-fitting force for tight fit of 2 μm interference between the shaft and inner ring for the standard angular contact ball bearing is as summarized below:

- 7020UC (ϕ100 × ϕ150 × 24)
- Interference fit of 2 μm (solid shaft)

$$\Delta_{deff} = \frac{100}{102} \times 0.002 = 0.00196$$

$$D_i = 1.05 \times \frac{4 \times 100 + 150}{5} = 115.5$$

$$P = \frac{208\,000}{2} \times \frac{0.00196}{100}\left[1-\left(\frac{100}{115.5}\right)^2\right] = 0.51 \text{ MPa}$$

$$K_d = 0.12 \times 0.51 \times \pi \times 100 \times 24 = 460 \text{ N}$$

To accommodate for variation in the friction, incorporate a safety factor of 2 to 3. As a result, the required press-fitting force is:
460 × (2 to 3) = 920 to 1 380 N

(2) Mounting by heating bearings

When mounting a bearing to a shaft using a constant temperature chamber, bearing heater or the like, follow the instructions below.

Heat the bearing at a temperature that reflects the interference between the shaft and inner ring (see **Fig. 6.5**).

Assuming linear expansion coefficient 12.5×10^{-6}, heating temperature ΔT, inner ring bore diameter ϕd, and interference fit

$\delta = 12.5 \times 10^{-6} \times d \times \Delta T$

Ex.) If ϕd = 100 mm, and δ = 0.030 (30 μm, tight fit), then the required heating temperature ΔT = 24 °C.

Therefore, the bearing temperature is heated to approximately room temperature +30 °C to allow for cooling during assembly.

NOTE

- If a resin material is used for the cage of angular contact ball bearing, do not excessively heat the bearing (approx. 80 °C max.).
- As a result of heating bearings after cooling, the inner ring will axially shrink, and there will be clearance between the bearing side face and shaft shoulder (see **Fig. 6.6**). For this reason, keep the bearing and shaft forced together with a press or the like after the unit returns to normal temperature. After cooling, check that the bearing is mounted to the shaft correctly.
- When using a bearing heater, be sure to avoid overheating. To prevent bearing from being magnetized, use equipment that has a demagnetizing feature.

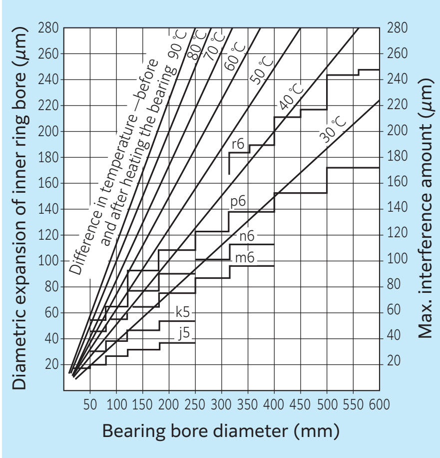

Fig. 6.5 Required heating temperature for mounting by heating inner ring

Remarks: The maximum interference amounts are interference values associated with Class 0 bearings.

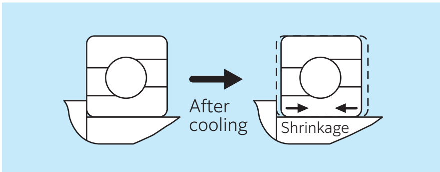

Fig. 6.6 Cooling after mounting by heating bearings

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6.3 Tightening of inner ring

When mounting and securing a bearing to a main spindle, the inner ring side face is usually clamped with a stepped sleeve or precision bearing nut, and the front cover situated on the outer ring side face is bolted down. When utilizing a stepped sleeve or precision bearing nut to clamp the inner ring, the following precautions must be followed.

■ Tightening with stepped sleeve

The stepped sleeve is designed that the hydraulically expanded sleeve is inserted over the shaft, and a predetermined drive-up force (tightening force) is applied to the shaft. Then the hydraulic pressure is released in order to secure the sleeve onto shaft and provide a tightening force to the bearing. This technique is a relatively simple locking method (see **Fig. 6.7**).

Note however after being locked in position by interference with the shaft, the sleeve can come loose because of deflection of the shaft or a moment load applied to the shaft.

For this reason, in many cases, a stepped sleeve is used together with a bearing nut as illustrated in **Fig. 6.8**.

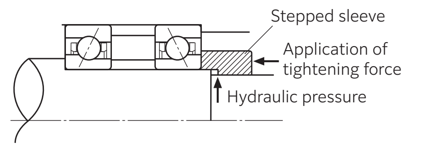


Fig. 6.7 Tightening with stepped sleeve

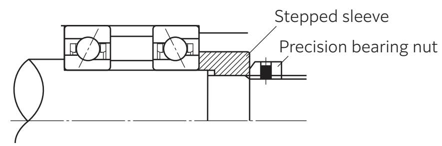


Fig. 6.8 Tightening with stepped sleeve + precision bearing nut

■ Tightening with precision bearing nut

Required tightening force is achieved with the precision bearing nut (precision locknut) by correctly controlling the tightening torque.

Note that when a bearing has been locked with a precision bearing nut (lock nut), the nut can develop inclination owing to the clearance on the threaded portions. If this problem occurs, fine adjustment will be necessary to obtain necessary running accuracy for the shaft.

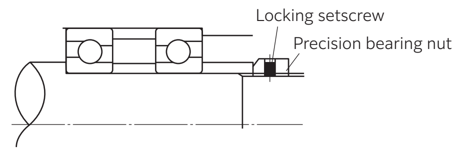


Fig. 6.9 Tightening with precision bearing nut

■ Correlation between tightening torque and tightening force with precision bearing nut

The correlation between tightening torque and tightening force with a precision bearing nut can be defined with the formula given below.

Because the thread face of the precision bearing nut, the thread face of the shaft and the bearing surface and nut constitute sliding surfaces, the correlation between tightening torque and tightening force will vary depending on the friction coefficient. Therefore, the nut needs to be thoroughly run on the shaft thread in advance to ensure smooth and uniform tightening.

It is also necessary to determine the correlation between tightening torque and tightening force by using a load washer or the like in advance.

$$F = \frac{M}{(d/2)\tan(\beta + \rho) + r_n \mu_n} \quad \cdots\cdots\cdots\cdots (6.6)$$

F : Precision bearing nut tightening force, N
M : Precision bearing nut tightening torque, N • mm
d : Effective diameter of thread, mm
ρ : Friction angle of thread face

$$\tan\rho = \frac{\mu}{\cos\alpha} \quad \cdots\cdots\cdots\cdots (6.7)$$

β : Lead angle of thread, °
$\tan\beta$ = number of threads × pitch/πd ……………… (6.8)
r_n : Average radius of nut surface, mm
μ_n : Friction coefficient of nut surface
$\mu_n \cong 0.15$
μ : Friction coefficient of thread face
$\mu \cong 0.15$
α : Half angle of thread, °

<Example calculation>

- Precision bearing nut
  AN20 (see **Fig. 6.10**)
- Thread data
  M100 × 2 (Class 2 thread)
  Effective diameter
  $d = \phi 98.701$ mm
  Half angle of thread $\alpha = 30°$

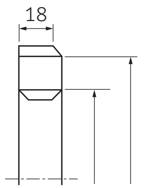


Fig. 6.10

The correlation between a tightening torque and tightening force with the precision bearing nut can be calculated as follows:

$$\tan\rho = \frac{0.15}{\cos 30°} \qquad \rho = 9.826°$$

$$\tan\beta = \frac{1 \times 2}{\pi \times 98.701} \qquad \beta = 0.370°$$

$$r_n = \frac{(101 + 120)/2}{2} = 55.25$$

$$F = \frac{M}{\frac{98.701}{2}\tan(0.370 + 9.826) + 55.25 \times 0.15} = \frac{M}{17.163}$$

6.4 Elastic deformation of spacer by tightening force

When incorporating a bearing into a main spindle, the bearing must be correctly forced into a predetermined position and maintained with a predetermined bearing pressure in order to maintain appropriate accuracies, clearances and rigidities of the bearing and main spindle.

When axially locating a duplex angular contact ball bearing by using a bearing spacer the cross-sectional area of spacer as well as (depending on the tightening force) the bearing pressure and elastic deformation by tightening of the spacer must be considered.

■ Correlation between inner ring spacer tightening force and amount of elastic deformation

When securing an angular contact ball bearing onto a main spindle, the bearing inner ring is tightened and locked by the shoulder of main spindle and a precision bearing nut and/or stepped sleeve. This inner ring tightening force causes the spacer to develop elastic deformation in the axial direction, varying the axial clearance on the bearing. In the case of a back-to-back duplex bearing (DB, DTBT or DBT) for a main spindle in particular, the inner ring tightening force will decrease the bearing clearance, estimated leading to an increased post-assembly preload and operating preload. A possible inner ring tightening force-derived axial deformation can develop in the form of deformation of both the inner ring and inner ring spacer. **NTN**'s experience has shown that only the elastic deformation on inner ring spacers needs to be considered.

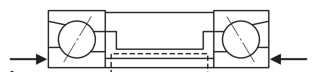


Fig. 6.11 Elastic deformation of inner ring spacer

The amount of deformation of a spacer is calculated using the following formula:

$$\delta = \frac{P \times L}{A \times E} \quad \cdots\cdots\cdots\cdots (6.9)$$

δ : Elastic deformation, mm
P : Inner ring tightening force, N
L : Inner ring spacer width, mm
A : Inner ring cross-sectional area, mm^2
E : Young's modulus 208 000, MPa

The require tightening force exerted onto inner ring spacers varies depending on the bearing manufacturer. From its experience, **NTN** adopts the typical values listed in **Table 6.4** (refer to next page).

Table 6.4 Nut tightening force

Bearing bore diameter (mm)	Nut tightening force (N)	Nut tightening torque <reference value> (N • m)	Front cover drive-up (mm)
6	1 470	2	0.01–0.02
8		2	
10	2 200	4	
12		5	
15	2 900	8	
17		9	
20	2 940–4 900	10–17	
25		13–22	
30		15–26	
35		18–30	
40	4 900–9 800	34–68	
45		38–75	
50		42–83	
55	9 800–14 700	92–138	
60		100–150	
65		108–162	
70		116–174	
75		124–186	
80	14 700–24 500	199–331	
85		211–351	
90		223–372	
95		235–392	
100		247–412	
105		259–432	
110		271–452	
120		295–492	
130		319–532	
140	24 500–34 300	572–800	0.02–0.03
150		613–858	
160		655–917	
170		695–973	
180		736–1 031	
190		779–1 090	
200		818–1 145	
220	<reference value> (34 300–44 100)	—	
240		—	
260		—	
280		—	
300		—	

Note 1) **NTN** has specified the nut tightening forces in this table based on experiences. However, **NTN** has no production record for bore diameter of 220 mm or larger. The nut tightening forces listed are only to be used for reference.
2) The nut tightening torque is calculated with a friction coefficient of 0.15 between the nut seating face and screw thread surface.
3) When tightening nuts, it is recommended to tighten them to twice the set value, then loosen them, and finally re-tighten them to the recommended set value.
4) For ball screw support bearings (BST), a tightening force approximately 2 to 3 times as large as the preload is recommended. The values shown in **Table 6.4** are also recommended for front arrangement bearings (DF, DTFT).

● Handling of Bearings NTN

6.5 Front cover drive-up

When mounting and securing a bearing onto a main spindle, the inner ring is usually tightened with a stepped sleeve or precision bearing nut and the outer ring side is bolted down. When locking the outer ring with a front cover, the following points need to be considered.

■ Front cover pressing amount

The bearing outer ring is tightened and locked between the shoulder of the housing and front cover at the main spindle front section. The front cover is installed by utilizing bolt holes (6 to 8 positions) on its flange. The usual pressing allowance on the outer ring and the front cover, which **NTN** has adopted through experience, falls in a range of 0.01 to 0.02 mm. Too large a pressing amount on the outer ring or a smaller number of fastening bolts may lead to poor roundness of the bearing ring.

Typical fit and deterioration in roundness of a raceway surface resulting from a pressing amount of 0.05 mm on the outer ring are shown in **Fig. 6.14**. Also, typical outer ring pressing amount and deterioration of a raceway surface with a fit of 5 μm loose are provided in **Fig. 6.15**.

To avoid deformation of the outer ring raceway surface, **NTN** recommends that the outer ring be installed to a highly accurate housing in transition fit with a large number of bolts.

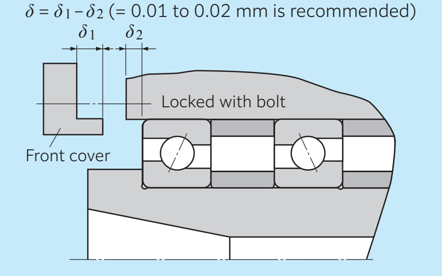


Fig. 6.12 Front cover pressing allowance

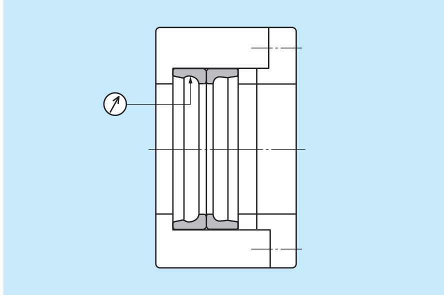


Fig. 6.13 Measuring position for roundness on outer ring raceway surface

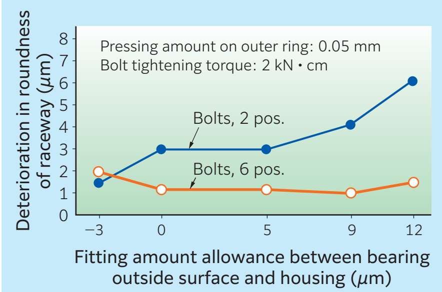


Fig. 6.14 Effect of fit of outer ring on roundness of raceway surface

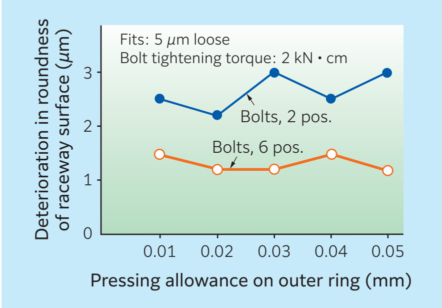


Fig. 6.15 Pressing allowance on outer ring vs. deterioration in roundness of raceway surface

Technical Data

6.6 Checking axial rigidity

In the typical method for checking for the axial rigidity of a bearing installed to a machine tool, the main spindle itself is pushed with a push-pull gauge to measure the resultant axial displacement. A method using a dial gauge is described below.

Two dial gauge are placed on two locations (axisymmetric locations separated by 180°) at the leading end of the main spindle. Use magnetic stands to secure the dial gauge to the end face of housing. Then, apply the load onto the main spindle and the resultant axial displacement is measured.

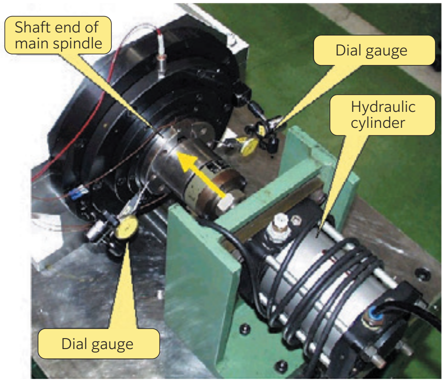


Photo 6.1

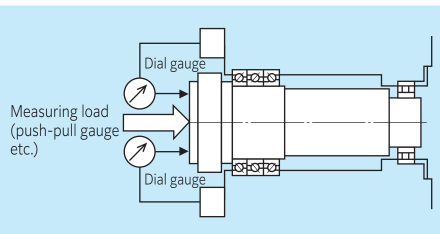


Fig. 6.16 Checking for axial rigidity

6.7 Clearance adjustment for cylindrical roller bearing

When incorporating a cylindrical roller bearing into a main spindle of a machine tool such as an NC turning machine or machining center, and setting the internal clearance to zero or to a negative clearance, the inner ring of the bearing usually has a tapered bore.

The internal clearance is adjusted by fitting the tapered bore bearing onto the tapered portion of the main spindle and driving the bearing in the axial direction to expand the inner ring.

For adjusting the internal clearance, two methods are available: a method consisting of clearance measurement for each bearing and adjustment with a spacer (s), and a mounted internal clearance adjustment gauge.

■ Method with clearance measurement and adjustment with spacer (s)

Adjust the bearing internal clearance by following the procedure described below:

(1) Calculation of outer ring shrinkage (see Fig. 6.17)

- Calculate the interference at the fitting area Δ_{deff} between the outer ring and housing.

  Measure the housing bore diameter first, and then calculate the interference Δ_{deff} from the outer ring outside diameter listed on the bearing inspection sheet.

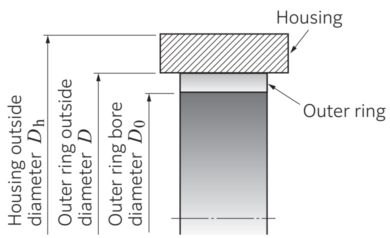


Fig. 6.17 Fits of outer ring and housing

EX. 1

Bearing outer ring outside diameter
ϕ150 mm (Inspection sheet = –0.005)
Housing bore diameter D
ϕ150 mm (measurement value = –0.007)
Interference at fitting area
Δ_{deff} = 0.002 (2 μm tight)

- Calculate the outer ring shrinkage ΔG with the formula (6.10).

$$\Delta G = \Delta_{deff} \cdot \frac{D_0}{D} \cdot \frac{1-(D/D_h)^2}{1-(D_0/D)^2 \cdot (D/D_h)^2} \quad \cdots\cdots (6.10)$$

EX. 2

Housing outside diameter $D_h = \phi200$, outer ring outside diameter $D = \phi150$, outer ring bore diameter $D_0 = \phi137$

$$\Delta G = 0.002 \cdot \frac{137}{150} \cdot \frac{1-(150/200)^2}{1-(137/150)^2 \cdot (150/200)^2}$$
$$= 0.0015 \quad \cdots\cdots (6.11)$$

(2) Measurement of bearing position and bearing radial clearance on a temporarily mounted bearing

- Mount the bearing inner ring with the cage and rollers onto the tapered shaft (see **Fig. 6.18**).

  In this process, force the inner ring until its tapered bore face is fully seated, and then measure the distance between the shaft shoulder and inner ring side face (L_1).

NOTE: After mounting the inner ring, check that the bearing side face is square to the main spindle centerline.

- At this point, mount the outer ring, move the outer ring up and down by hand and then measure the internal clearance after mounting (Δr_1) (see **Fig. 6.19**).
- Calculate the estimated bearing clearance Δ_1 after press-fitting the outer ring into the housing with the formula (6.12). The result of the calculation reflects the outer ring shrinkage ΔG.

$$\Delta_1 = \Delta r_1 - \Delta G \quad \cdots\cdots (6.12)$$

EX. 3

Internal clearance after mounting $\Delta r_1 = 0.030$
Outer ring shrinkage $\Delta G = 0.0015$
Estimated bearing clearance
$\Delta_1 = 0.030 - 0.0015 = 0.0285$

(3) Adjustment of spacer width between shaft shoulder and inner ring

To adjust the bearing clearance to a predetermined target value (δ) after mounting, determine the spacer width L_n with the formula (6.13) (see **Fig. 6.20** and **Fig. 6.21**).

$$L_n = L_1 + f(\delta - \Delta_1) \quad \cdots\cdots (6.13)$$
$$(n = 2, 3, 4 \cdots)$$

The value f in the formula (6.13) is found in the **Table 6.5** (refer to next page).

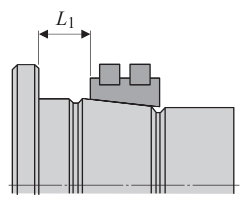


Fig. 6.18 Measurement of bearing position

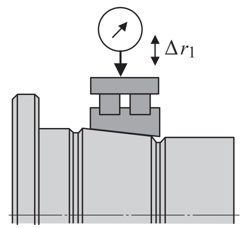


Fig. 6.19 Measurement of bearing radial clearance

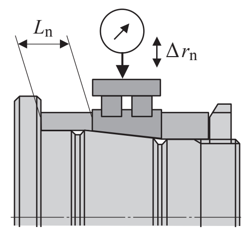


Fig. 6.20 Clearance measurement after insertion of spacer

Table 6.5 Value f

Value d_m/d_i	Value f
0 –0.2	13
0.2–0.3	14
0.3–0.4	15
0.4–0.5	16
0.5–0.6	17
0.6–0.7	18

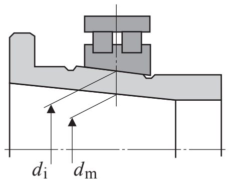


Fig. 6.21 Explanation of d_m / d_i

EX. 4

In the case of NN3020K, if bearing bore diameter $d = \phi 100$, width B = 37, and $d_i = d + 1/12 \cdot B/2$,

then d_i = dia. $\phi 101.5417$.

If the targeted post-mounting clearance value $\delta = 0.015$, $L_1 = 15$, $d_m = \phi 60$, $\Delta_1 = 0.0285$, then $d_m/d_i = 60/101.5417 = 0.5909$, and, therefore, $f = 17$.

Thus, the spacer width L_n between the shoulder and inner ring equivalent to $\delta = 0.015$ will be the value shown by the formula below:

$$L_n = 15 + 17 \times (0.015 - 0.0285) = 14.7705$$

(4) Bearing clearance measurement after insertion of spacer (see Fig. 6.20)

Insert a spacer that satisfies the spacer width L_n between the shoulder and inner ring determined in the previous step, and tighten the inner ring until the spacer does not move. Next, move the bearing outer ring up and down by hand and measure the internal clearance after mounting (post-mounting internal clearance) Δr_n. The estimated bearing clearance Δ_n after press-fitting of the outer ring into the housing is determined with the formula below:

$$\Delta_n = \Delta r_n - \Delta G \quad \cdots\cdots (6.14)$$

(n = 2, 3, 4 • • •)

(5) Final adjustment for spacer width

- Repeat the steps (3) and (4) above to gradually decrease the spacer width L_n so as to adjust the post-mounting bearing clearance to the targeted clearance.
- By plotting the correlation between the spacer width and post-mounting clearance as illustrated in **Fig. 6.22**, the spacer width for the final targeted clearance will be more readily obtained.

  Positive clearance:

  All rollers are sliding rather than rolling.

  Clearance = 0:

  About half of the rollers are rolling but the rest are sliding.

  Negative clearance:

  All rollers are rolling.

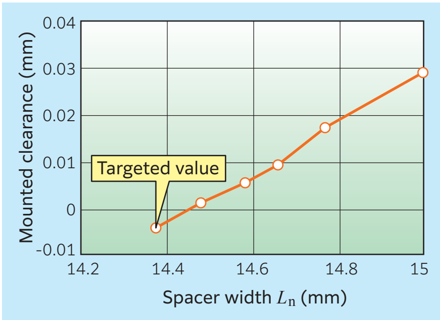


Fig. 6.22 Correlation between spacer width L_n and mounted clearance Δ_n

■ Method using mounted internal clearance adjustment gauge

The mounted internal clearance adjustment gauge has a cylindrical ring, which has a cut-out so that the ring can be opened and closed. The bore surface of the ring is used as a location for measurement. The clearance at the location for measurement is proportional to the reading on the dial gauge. As illustrated in **Fig. 6.23**, the mounted internal clearance adjustment gauge consists of a ring gauge, dial gauge, and attachment components. Its fixture protects the interference gauge against possible deformation when not in use. For the measuring operation, detach the fixture.

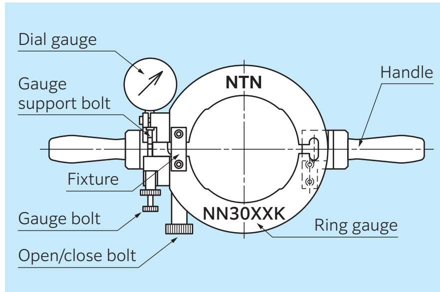


Fig. 6.23 Descriptions of various components on mounted internal clearance adjustment gauge

● Usage of mounted internal clearance adjustment gauge

(1) Measurement of outer ring raceway diameter (bore diameter)

- Mount the outer ring into the housing. (For easy mounting, heat the housing.)
- Wait until the temperature of the outer ring is same as that of the inner ring, and then measure the outer ring raceway diameter (bore diameter). Take measurements at several points and calculate the average, and then zero the gauge at this average value (see **Photo 6.2**).

Photo 6.2

(2) Setup of mounted internal clearance adjustment gauge

- Place the cylinder gauge, onto the bore surface of clearance adjustment gauge as shown in **Photo 6.3**, and adjust it with the open/close bolt so that its Dial 1 is set to zero.
- When the reading of Dial 1 of the cylinder gauge is zero, adjust the gauge bolt so that the pointer of Dial 2 points at the red mark (correction amount of the gauge) (see **Photo 6.4**).
  With the gauge bolt, adjust the gauge so that the short pointer is situated at the scale 2 position (With the large size, insert the pin into the hole of the open/close bolt and make fine-adjustment).

NOTE 1) **Photo 6.4** shows the inner ring and rollers. When the correction amount of the gauge is adjusted, adjust it only with the thickness gauge.

NOTE 2) The pointer of Dial 2 is directed to the red mark. The purpose of this is to compensate clearance error caused due to the structure of mounted internal clearance adjustment gauge. The correction amount can vary from gauge to gauge.

NOTE 3) When the pointer of Dial 2 is in line with the red mark, the zero reading on Dial 2 coincides with the zero bearing clearance.

(3) Setting up the mounted internal clearance adjustment gauge on the main spindle

- Mount the cage and roller with inner ring onto the main spindle, and lightly tighten the precision bearing nut.
- Tightening the open/close bolt (see **Fig. 6.23**) on the clearance adjustment gauge will cause the gauge bore to expand.
  With the gauge bore expanded by about 0.15 mm, insert the gauge into the outside diameter portion of the roller set in the inner ring. Be careful not to damage the rollers (see **Photo 6.5**).
- Loosening the open/close bolt will cause the gauge bore to shrink.
  Loosen the open/close bolt to bring the gauge bore into contact with the outside diameter of the roller set in the inner ring.
- Lightly swing the clearance adjustment gauge in the circumferential direction to stabilize the pointer on the dial gauge.

(4) Setup of inner ring clearance

- Tighten the precision bearing nut of the main spindle. This should be done gradually to prevent shock loading.
- Tightening the precision bearing nut further until the reading on the dial of the clearance adjustment gauge becomes zero in case the clearance is aimed at 0 μm.
- Once the reading on gauge gets zero, carefully swing the adjustment gauge again to check that the measurement value is correct.
- Loosen the open/close bolt on the clearance adjustment gauge to expand the gauge bore and remove the gauge from the inner ring.

(5) Determination of spacer width

- The inner ring should now be in the position where the reading on the dial of clearance adjustment gauge was zero in step (4). By using a block gauge, measure the distance between the inner ring side face and shaft shoulder (dimension ℓ in **Fig. 6.24**).
- Measure this dimension in at least three locations, and finally adjust the spacer width ℓ to the average of three measurements.
- Loosen and remove the precision bearing nut, inner ring spacer and inner ring from the main spindle.

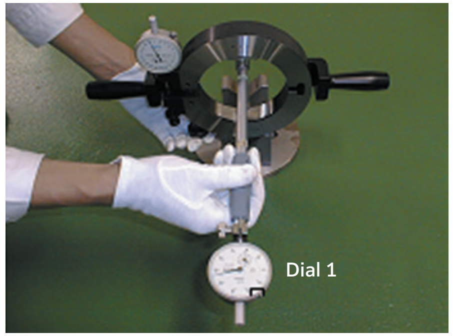


Photo 6.3

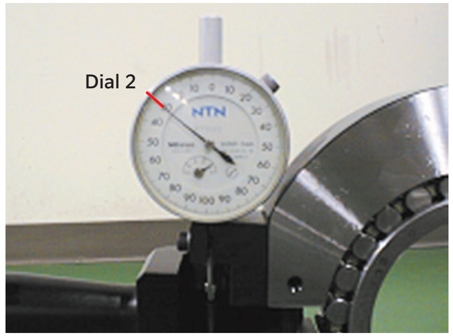


Photo 6.4

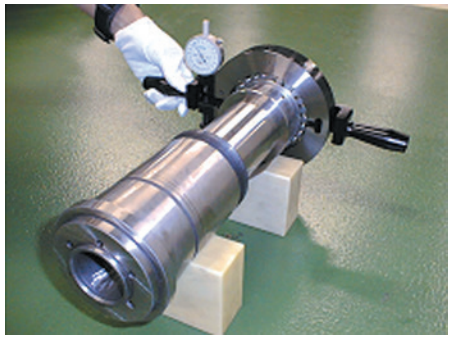


Photo 6.5

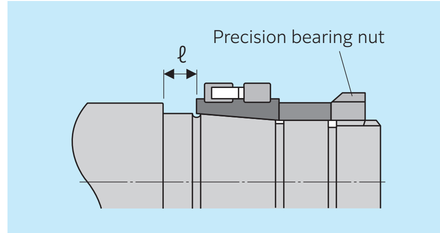


Fig. 6.24 Spacer width dimension

(6) Assembly and check of the mounted roller outside diameter

- Insert a spacer of width ℓ. Then insert the inner ring and mounting spacer and tighten the precision bearing nut.
- According to a procedure similar to that in steps (3) "Setting up the mounted internal clearance adjustment gauge on the main spindle" and (4) "Setup of inner ring clearance", check the mounted roller outside diameter and the clearance setting. Note this process is only a re-check procedure, and may be omitted once the clearance measurements fall in a smaller range.

● **Replacement of mounted internal clearance by clearance correction factor**

(1) Clearance correction factor

Because of the structure of the **NTN** mounted internal clearance adjustment gauge, the ratio of the clearance reading on location for measurement to the reading on dial gauge is 1:2.5 (clearance correction factor), (The clearance reading on the dial gauge is 2.5 times as large as the mounted internal clearance). For reference, a clearance reading conversion table is given in **Table 6.6**.

NOTE: Note that the clearance correction factor of certain bearing numbers is not 1:2.5. Clearance correction factor is given on the table of inspection results.

Table 6.6 Clearance reading conversion table (when clearance correction factor 2.5)

Reading on dial gauge (μm)	Mounted internal clearance on location for measurement (μm)	Reading on dial gauge (μm)	Mounted internal clearance on location for measurement (μm)
0.5	0.2	5.5	2.2
1.0	0.4	6.0	2.4
1.5	0.6	6.5	2.6
2.0	0.8	7.0	2.8
2.5	1.0	7.5	3.0
3.0	1.2	8.0	3.2
3.5	1.4	8.5	3.4
4.0	1.6	9.0	3.6
4.5	1.8	9.5	3.8
5.0	2.0	10.0	4.0

(2) Mounted internal clearance (when clearance indication value 1:2.5)

The reading on the dial gauge is converted into a mounted internal clearance in the following manner:

• CASE 1

The reading relative to the zero point is in the clockwise direction (CW) (see **Fig. 6.25**). The value of the mounted internal clearance (+) is 1/2.5 times as large as the reading on dial gauge.

Reading on dial gauge in **Fig. 6.25** = 2.5
Mounted internal clearance = 2.5/2.5 = (+) 1 μm

• CASE 2

The reading relative to the zero point is in the counterclockwise direction (CCW) (see **Fig. 6.26**). The value of the mounted internal clearance (–) is 1/2.5 times as large as the reading on dial gauge.

Reading on dial gauge in **Fig. 6.26** = 5.0
Mounted internal clearance = 5.0/2.5 = (–) 2 μm

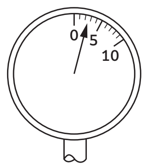


Fig.6.25
Reading on dial gauge: +2.5 μm (mounted internal clearance: +1 μm)

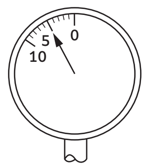


Fig.6.26
Reading on dial gauge: –5 μm (mounted internal clearance: –2 μm)

● **Example for setting mounted internal clearance**

When setting the mounted internal clearance, adjust the dial gauge by shifting from the zero point to the "targeted clearance × clearance correction factor".

Examples for when clearance correction factor 2.5 are shown in **Fig. 6.27** and **Fig. 6.28**.

NOTE: Note the direction when adjusting the dial gauge by shifting from the zero point.

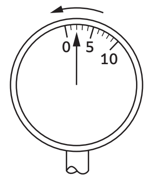


Fig. 6.27 Adjustment for negative clearance (mounted internal clearance: –0.8 μm)

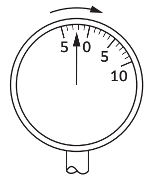


Fig. 6.28 Adjustment for positive clearance (mounted internal clearance: +1.0 μm)

<Precautions for using and storing the mounted internal clearance adjustment gauge>

When using the mounted internal clearance adjustment gauge, follow the precautions described below:

- When transferring the outer ring raceway diameter measured with the cylinder gauge to the mounted internal clearance adjustment gauge, use the adjustment gauge in a vertical attitude (see **Photo 6.6**).
- When not using the mounted internal clearance adjustment gauge, place it in a horizontal attitude (see **Photo 6.7**). Also, after completion of clearance measuring operation, apply rust-preventive oil to the mounted internal clearance adjustment gauge and store in a dry location.

Photo 6.6 Vertical storage attitude

Photo 6.7 Horizontal storage attitude

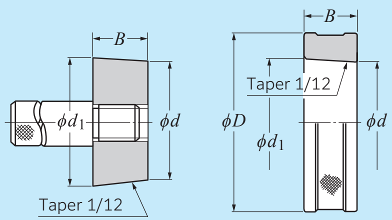
6.8 Tapered bore cylindrical roller bearing and main spindle taper angle

In order for a precision bearing to perform as designed, it must be correctly mounted to a shaft and housing. In particular, when employing a tapered bore cylindrical roller bearing, accurate finish for the tapered main spindle and appropriate fit between the bearing bore and the main spindle are very important to ensure high accuracy of the main spindle. **NTN** recommends that the customer use the **NTN** tapered shaft ring gauge, which that is finished to same accuracies as the bearing, so that the customer can achieve higher precision. **NTN** also offers a plug gauge so that the customer can check the accuracy of the ring gauge.

■ Taper gauge for precision roller bearings

Each **NTN** precision cylindrical roller bearing taper gauge consists of a female gauge and a male gauge (plug gauge) (see **Fig. 6.29**).

Using blue paste or an equivalent as well as a ring gauge, check the fit of the bearing bore with the main spindle taper. The correct fit between the main spindle and the bearing leads to higher accuracy of the main spindle. The plug gauge is intended to check the accuracy of the associated ring gauge. Use the plug gauge to verify the taper accuracies of the associated ring gauge (see **Fig. 6.30**).



Plug gauge (TA) Ring gauge (TB)

* Some have integrated plug gauge and handle.

Fig. 6.29 Taper gauge

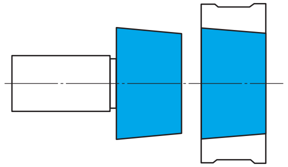


Fig. 6.30 Blue paste on taper gauge

■ Taper angle

NTN machines the tapered bore of its cylindrical roller bearings and the taper angle of its taper gauges according to the tolerances below:

- Nominal taper angle 1/12 (4° 46′ 18.8″)
- Tolerance for precision roller bearing with 1/12 taper angle is +12″ ± 12″ (JIS Class 4 and 2)
- Targeted tolerance for taper gauge 1/12 is +9″.

Usually, Using blue paste between the tapered bore of a cylindrical roller bearing and a plug gauge exhibits a strong contact mark on the small diameter side as show in **Fig. 6.31**. This is because **NTN** has slightly adjusted the taper angle of the bearing bore to accommodate for the difference in thickness of the inner ring below each row of rollers.

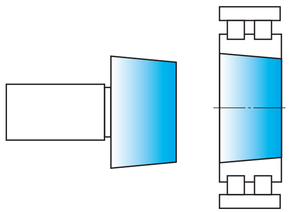


Fig. 6.31

■ Checking main spindle taper with ring gauge

When checking the main spindle taper angle with a ring gauge, perform the following steps.

- Thoroughly clean the surface of the ring gauge, and apply a thin layer of blue paste to four equally-spaced points.
- Clean the tapered surface of the shaft, and gently insert into the ring gauge.
- The ring gauge to be lightly turning it.
- Check the patterns of blue paste deposited on the shaft surface.
- At this point, attach a strip of clear adhesive tape onto each blue paste spot, and peel off each strip.

Attach strips of adhesive tape onto white paper and check how much blue paste was deposited onto each point. Check that more than 80 % of the applied blue paste was deposited on the tapered surface.

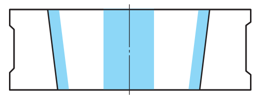


Fig. 6.32 Application of blue paste to ring gauge

Table 6.7 Examples of blue paste records

Region A	Small	Large
Region B	Small	Large
Region C	Small	Large
Region D	Small	Large

Small: small diameter side
Large: large diameter side

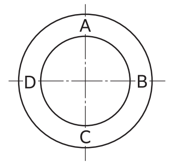


Fig. 6.33 Regions subjected to measurement with blue paste

6.9 Handling precautions

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.

Bearings are vulnerable to impact. Do not hit them with a hammer directly or drop them on the floor (see **Fig. 6.34**).

In addition, bearings are sensitive to foreign particle contamination. When foreign particles enters the bearing during rotation, denting and/or scratches may occur, resulting in objectionable noise and vibration levels and rough bearing rotation (see **Fig. 6.35**). Therefore, when handling bearings, it is necessary to keep the periphery clean.

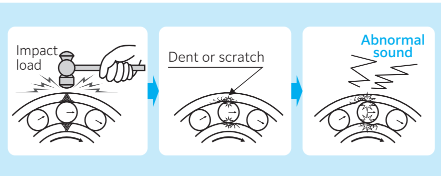


Fig. 6.34 Damage caused by impact

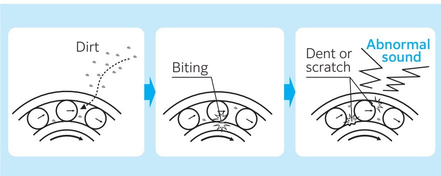


Fig. 6.35 Damage caused by foreign particle contamination

For optimal bearing performance, proper bearing handling methods must be used. The handling methods described herein are general guidelines. Depending on the type and size of bearing needed, special handling "methods" may be necessary. For more detailed information, please consult **NTN** Engineering.

Using proper protective equipment and tools are also essential when installing or removing bearings, to avoid damage to the machinery and ensure the safety of the technician. Further information on proper installation and removal procedures is detailed in the following sections.

6.10 Bearing storage

Most rolling bearings are coated with a rust preventive oil before being packed and shipped. Please observe the following guidelines when storing bearings.

1. Ideally, bearings should be stored indoors at room temperature with a relative humidity of less than 60 %. Avoid places in direct sunlight or in contact with outer walls because excessive temperature fluctuation or humidity rise may cause condensation.
2. Bearings should not be stored directly on the ground. Instead, they should be placed on a shelf or pallet at least 20 cm above the ground. The maximum number of shipping boxes to be stacked for storage should limited to four whenever possible (see **Fig. 6.36**).
3. Precision rolling bearings, large rolling bearings and thin ring or race rolling bearings must be laid down horizontally for storage (see **Fig. 6.37**). Storing them standing vertically may cause raceway deformation.

   To avoid damage during transportation such as fretting or false brinelling, ensure that the individual bearing boxes are packed laying down horizontally within the shipping box. Fill remaining space with dunnage (see **Fig. 6.38**).

   Some products have a ↑ symbol on the shipping box to prevent improper storage placement. Follow the indication on the box in this case (see **Fig. 6.39**).

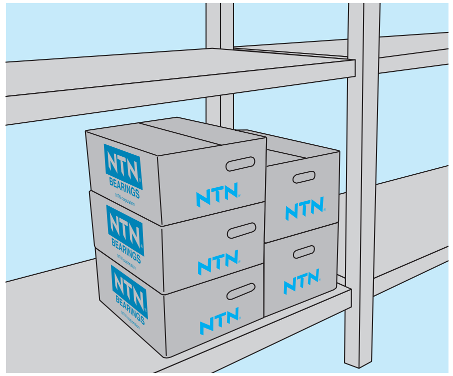


Fig. 6.36 Storing bearings on a shelf

Fig. 6.38 Transportation and storage by shipping box

Fig. 6.37 Storing one-bearing boxes on a shelf

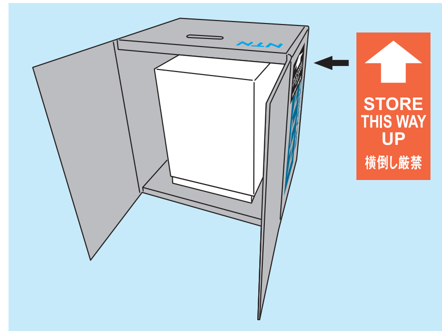


Fig. 6.39 Horizontally placing box prohibited

6.11 Bearing installation

A jig, a measuring instrument, a lubricant, and a clean and dry workshop will be needed for bearing installation. Further, if possible, it is desirable to install miniature/small ball bearings and precision rolling bearings in a clean room because intrusion of dirt and foreign matter significantly affects bearing performance.

Improper installation of bearings may cause marks from the rolling elements on the raceways, adversely affecting the bearing life. For details, on machining accuracy and mounting accuracy of bearings, shafts, and housings, for details please refer to the see tecnical data "2. Bearing selection and shaft & housing design."

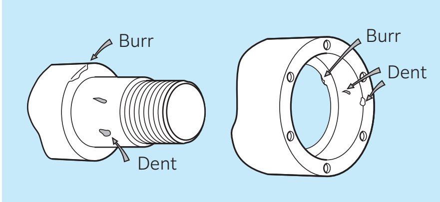


Fig. 6.40 Burrs and dents

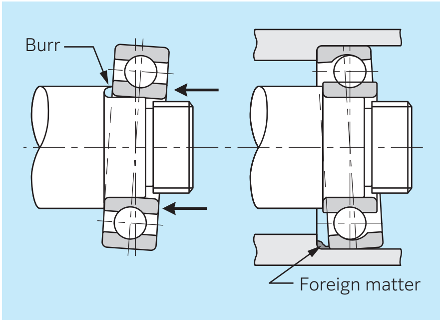


Fig. 6.41 Example of improper bearing installation

■ Installation preparations

● Fitting surface of shafts and housings

When a bearing is installed on a shaft or in a housing with surfaces containing burrs or dents, the bearing may not seat properly, causing vibration and noise during operation (see **Fig. 6.40** and **Fig. 6.41**).

6.12 Running-in operation for main spindle bearings

Run-in is important for ensuring smooth operation of grease-lubricated main spindle bearings.

The following two modes of running-in are recommended:

(1) The bearing speed is gradually increased in steps. After the temperature is saturated at each speed setting, the speed is increased to the next step (see **Fig. 6.42**).

(2) The bearing is run for one minute at around the maximum operating speed of the spindel. This cycle is repeated two or three times (see **Fig. 6.43**) as needed.

(1) is the ordinary method used, however it takes slightly longer to reach the maximum operating speed of the spindel. In contrast, (2) can shorten the running-in time, however higher risk of sudden bearing temperature rise is considerable, so that running speed and its holding time must be set carefully.

Generally, the temperature of a main spindle bearing is measured on the front cover. The temperature difference across the bearing outer ring and front cover reaches 2 to 3 °C, and at the same time, the temperature difference between the hottest rolling element and the inner ring raceway surface seems to reach 50 to 10 °C. For this reason, **NTN** recommends that the machine is stopped if the temperature on front cover reaches approximately 60 °C. It is recommended that the temperature falls to 30 °C or less before restarting the running-in operation.

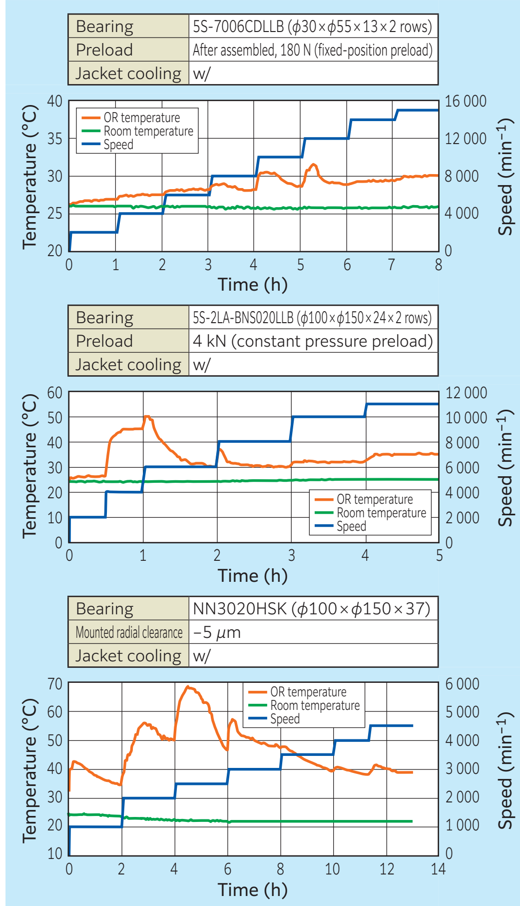


Fig. 6.42

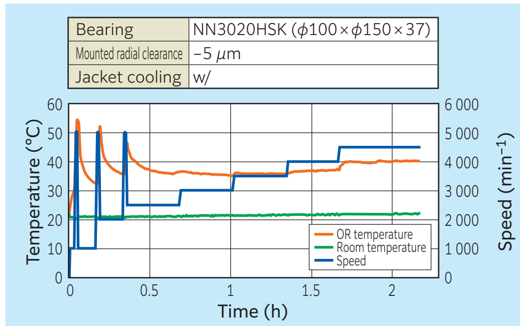


Fig. 6.43

7. Lubrication of Bearings

The purpose of rolling 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. Lubricant is necessary for operating rolling bearings. For rolling bearings, lubrication has the following advantages:

(1) **Reduction** of **friction** and **wear**
It prevents direct metallic contact between the rolling and sliding elements of bearing components and reduces friction and wear.

(2) **Prolonged** bearing **life**
The rolling fatigue life is prolonged by forming an oil film on the rolling contact surface part.

(3) **Friction heat dissipation** and cooling
Circulating lubrication can dissipate heat generated from friction or conducted from the outside.

(4) Others
It **prevents foreign materials from entering** inside the bearing and suppresses corrosion (rust) by covering the bearing surface with oil.

In order to exhibit these effects, a lubrication method that matches service conditions is required. In addition to this, a quality lubricant must be selected, the proper amount of lubricant must be used and the bearing must be designed to prevent foreign matter from getting in or lubricant from leaking out.

The main spindle of a machine tool usually uses an extremely low volume of lubricant so heat generation from stirring of the lubricant is minimal.

Fig. 7.1 summarizes the relationships between oil volume, friction loss, and bearing temperature.

The lubrication methods available for bearings in a machine tool include grease lubrication, oil mist lubrication, air-oil lubrication, and jet lubrication. Each method has unique advantages. Therefore, the lubricating system that best suits the lubrication requirements should be used.

Table 7.1 and **Table 7.2** summarize the features of various lubrication methods.

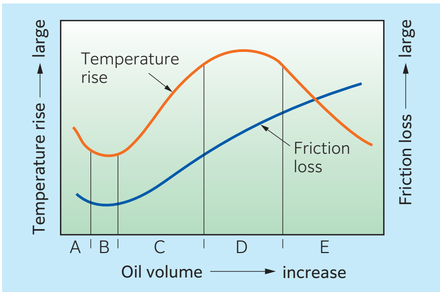


Fig. 7.1

Table 7.1 Oil volume, friction loss and bearing temperature (see Fig. 7.1)

Range	Characteristics	Lubrication method
A	When oil volume is extremely low, direct metallic contact occurs in places between the rolling elements and raceway surfaces. Bearing abrasion and seizing may occur.	—
B	A thin oil film develops over all surfaces, friction is minimal and bearing temperature is low.	Grease lubrication Oil mist lubrication Air-oil lubrication
C	As oil volume increases, heat buildup is balanced by cooling.	Circulating lubrication
D	Regardless of oil volume, temperature rises at a fixed rate.	Circulating lubrication
E	As oil volume increases, cooling dominates and bearing temperature decreases.	Forced circulating lubrication Jet lubrication

Table 7.2 Evaluation of various lubricating systems

Criterion \ Lubrication method	Grease lubrication	Oil mist lubrication	Air-oil lubrication	Jet lubrication
Handling	◎	○	○	△
Reliability	○	△	○	◎
Temperature rise	△	△	○	◎
Cooling effect	×	△	○	◎
Sealing structure	△	○	○	×
Power loss	○	○	○	×
Environmental contamination	○	×	△	○
Allowable $d_m n$ value [1]	1.4×10^6	2.2×10^6	2.5×10^6	4.0×10^6

Legend ◎ : Excellent ○ : Good △ : Fair × : Poor
1) The permissible $d_m n$ values are approximate values:
$d_m n$: pitch circle diameter across rolling elements (mm) multiplied by speed (min^{-1})

7.1 Grease lubrication

Grease is popular amongst other lubricants because of its simpler maintenance. With an adequate amount of quality grease prefilled, this system can be used over a wide range of speed. The allowable maximum speed varies with the type and size of bearing: for a high speed angular contact ball bearing, the $d_m n$ value should be 1.4×10^6 as a guideline. For applications exceeding this range, consult **NTN** Engineering.

■ Grease types

When grease temperature rises during high speed operation, which can be seen for machine tool spindles, a grease with a consistency of NLGI2 or NLGI3 is recommended. For the base oil, ester oil and synthetic oil are used in addition to mineral oil. Urea, which has excellent high temperature properties, is used as a thickener in addition to lithium soap and barium complex soap.

Table 7.3 lists technical data for greases commonly used for machine tool main spindles.

■ Amount of grease required

Usually, a bearing for the main spindle of a machine tool requires that grease volume be low so heat generated by the stirring of the grease during high speed operation is minimal. A guideline for the amount of grease used for a main spindle bearing is given below.

- Angular contact ball bearing
  ($d_m n$ value $\leq 0.65 \times 10^6$);
  15 to 20 % of bearing free space
  ($d_m n$ value $> 0.65 \times 10^6$);
  12 to 17 % of bearing free space
- Cylindrical roller bearing;
  10 to 15 % of bearing free space
- Tapered roller bearing;
  15 to 20 % of bearing free space

The above is a guideline to determine the amount of grease required based on bearing free space listed in the bearing dimensions table. It is recommended to aim for the lower limit to reduce the running-in operation time.

Before filling a bearing with grease, remove the rustproof coating from the bearing with clean wash oil and allow the bearing to dry completely. Then fill and uniformly distribute an appropriate amount of grease in the bearing with an syringe, plastic bag, etc.

Lubrication of Bearings

NTN

Table 7.3 Typical greases for machine tool main spindle bearings

Grease brand	SE-1	MP-1	ISOFLEX NBU 15	STABURAGS NBU 8 EP	Multemp LRL No.3	Multemp PS No.2
Thickener	Urea		Ba complex soap		Li soap	
Base oil	PAO + ester	Synthetic oil	Diester + mineral oil	Mineral oil	Synthetic oil	Ester + PAO
Base oil viscosity (40 °C) mm^2/s	22	40.6	23	105	37.3	15.9
Blend consistency NLGI No.	2	3	2	2	3	2
Dropping point °C	> 220	> 250	> 220	> 220	208	190
Operating temperature range °C	−50 to 120	−40 to 150	−40 to 130	−20 to 140	−40 to 150	−50 to 130
Application	Applied to ULTAGE Series grease-lubricated sealed angular contact ball bearings	Applied to ULTAGE Series grease-lubricated sealed angular contact ball bearings	Most commonly used for main spindles	Suitable for roller bearings subject to large loads	Wider operating temperature range	For low temperature and low torque
NTN grease code	L749	L448	15K	L135	12K	1K

Note: 1. Representative values are shown for the base oil viscosity, consistency, and dropping point.
2. The upper and lower limits of the operating temperature range differ depending on the usage environment and requirement specifications. Please consult with **NTN** Engineering.

Technical Data

7.2 Air-oil lubrication

Air-oil lubrication (also known as oil-air lubrication or oil and air lubrication) is widely adopted for main spindle bearings in order to cope with the higher speed and precision of machine tools and to ensure more reliable lubrication.

Air-oil lubrication employs a method by which compressed air is used to provide lubricating oil in precisely controlled amounts. Generally, an air-oil lubrication unit a volumetric piston-type distributor that accurately meters the required minimum amount of lubricating oil and provides it at optimal intervals controlled by a timer.

■ Features of air-oil lubrication

Air-oil lubrication has the following features over:

- Accurately supplies a minimal amount of oil.
- Can be adjusted to provide the proper amount of lubricant for individual bearings.
- It is easy to control the amount of oil depending on the viscosity of the lubricant.
- Compressed air helps cool the bearing.
- It reduces the amount of oil mist.
- Low oil consumption.
- Use of compressed air can prevent contamination of the bearing by other coolants.

■ Example of an air-oil lubrication system

Fig. 7.2 shows the configuration and example of an air-oil lubrication system.

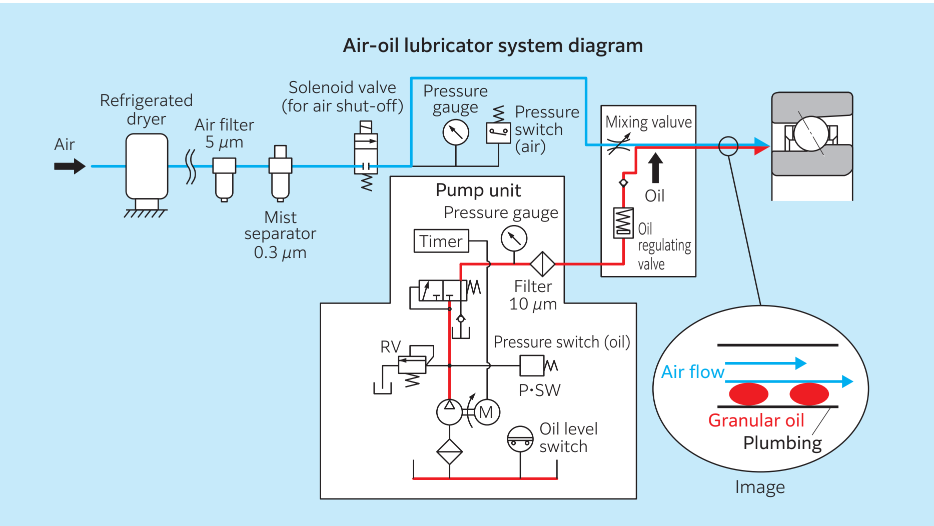


Fig. 7.2 Example of air-oil lubricating system

■ Air-oil lubrication nozzle spacer

Air-oil lubrication requires a specialized nozzle because it supplies the lubricating oil to the inside of the bearing by means of compressed air (see **Fig. 7.3**).

A nozzle with a hole diameter of 1.0 to 1.5 mm and a length 4 to 6 times the hole diameter is recommended.

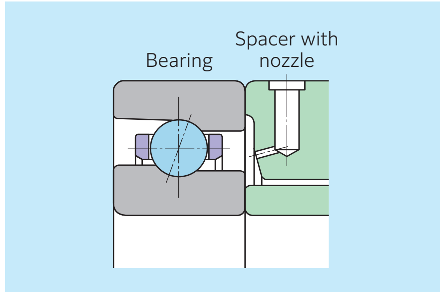


Fig. 7.3 Feed system for air-oil lubrication

■ Exhaust method for air-oil lubrication

Air-oil lubrication uses a large volume of air to feed lubricating oil to the bearing. Therefore, it is essential that the air fed into the bearing be allowed to escape. If the air is not smoothly exhausted, the lubricating oil will remain in the bearing and possibly contribute to bearing seizure. In the design stage, remember to allow ample space on the exhaust side of the bearing in order to increase exhaust efficiency and provide a larger oil drain hole to ensure smooth airflow. In addition, for types that allow for repositioning of the spindle, it is recommended that the shoulder dimensions of all parts is designed to prevent lubricating oil from flowing back into the bearing after a change in the attitude of the main spindle. Unnecessary dimensional differences can also contribute to stagnancy of the lubricating oil.

■ Recommended targeted position with nozzle

(1) Angular contact ball bearings

Table 7.4 Air-oil/oil mist nozzle spacer dimensions

Unit: mm

Bearing No.	θ	A	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	D	E
7900U	15°	14.6	12.4	13.4	18.5	1
7901U	15°	16.6	14.4	15.4	20.5	1
7902U	15°	19.5	17.2	18.2	25	1
7903U	15°	21.5	19.2	20.2	27	1
7904U	15°	26.3	24	25	32.5	1
7905U	15°	31.3	29	30	37.5	1
7906U	15°	36.3	34	35	42.5	1
7907U	15°	41.5	39.2	40.2	50.5	1
7908U	15°	48.1	45.8	46.8	56.5	1
7909U	15°	52.8	50.5	51.5	63	1
7910U	15°	57.3	54.3	55.8	67.5	1.5
7911U	15°	64.1	61.1	62.6	73.5	1.5
7912U	15°	69.1	66.1	67.6	78.5	1.5
7913U	15°	74.1	71.1	72.6	84	1.5
7914U	15°	80.9	77.9	79.4	93	1.5
7915U	15°	85.9	82.9	84.4	97.5	1.5
7916U	15°	91.4	88.4	89.9	103	1.5
7917U	15°	97.4	94.4	95.9	112	1.5
7918U	15°	102.4	99.4	100.9	117	1.5
7919U	15°	107.4	104.4	105.9	122	1.5
7920U	15°	113.9	110	112	131	1.5
7921U	15°	118.9	115	117	136	1.5
7922U	15°	123.9	120	122	141	1.5
7924U	15°	135.4	132	134	155	1.5
7926U	15°	146.9	143	145	169	1.5
7000U	15°	15.4	13.1	14.1	22	1
7001U	15°	18.1	15.8	16.8	24.5	1
7002U	15°	21.3	19	20	27.5	1
7003U	15°	23.3	21	22	31	1
7004U	15°	28.6	25.8	26.8	37.5	1
7005U	15°	33.1	30.5	31.5	41.5	1
7006U	15°	39.6	36.5	37.5	49.5	1
7007U	15°	44.6	41	42	56	1
7008U	15°	50.4	47	48	61.5	1
7009U	15°	55.9	52	54	67.5	1
7010U	15°	60.9	57	59	72.5	1.5
7011U	15°	67.4	63	65	82	1.5
7012U	15°	72.4	68	70	87	1.5
7013U	15°	77.4	73	75	92	1.5
7014U	15°	83.9	78	80	101	1.5
7015U	15°	88.9	83	85	106	1.5
7016U	15°	95.4	90	92	115	1.5
7017U	15°	100.4	95	97	120	1.5
7018U	15°	106.9	101	103	129	1.5
7019U	15°	111.9	106	108	134	1.5
7020U	15°	116.9	112	114	139	1.5
7021U	15°	123.4	117	120	148	1.5
7022U	15°	129.9	122	125	157	1.5
7024U	15°	139.9	133	136	167	1.5
7026U	15°	153.9	143	146	184	1.5

Note) Spacer dimensions are the same for all contact angles (15°, 25°and 30°).

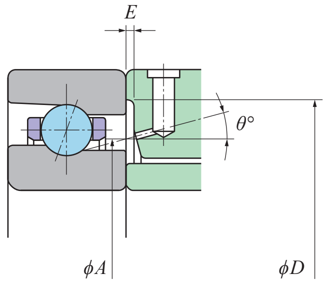


Fig. 7.4 79U, 70U and HSE types

Table 7.5 Air-oil/oil mist nozzle spacer dimensions

Unit: mm

Bearing No.	θ	A	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	D	E
HSE910U	15°	58.9	55	56	67	1.5
HSE911U	15°	64.8	61	62	74	1.5
HSE912U	15°	69.8	66	67	79	1.5
HSE913U	15°	74.8	71	72	84	1.5
HSE914U	15°	81.6	77	79	93	1.5
HSE915U	15°	86.6	82	84	98	1.5
HSE916U	15°	91.6	87	89	103	1.5
HSE917U	15°	98.1	93	95	112	1.5
HSE918U	15°	103.1	98	100	117	1.5
HSE919U	15°	108.1	103	105	122	1.5
HSE920U	15°	115.3	109	111	131	1.5
HSE921U	15°	120.3	114	116	136	1.5
HSE922U	15°	125.3	119	121	141	1.5
HSE924U	15°	136.9	130	132	155	1.5
HSE926U	15°	148.4	141	143	169	1.5
HSE928U	15°	158.4	151	153	179	1.5
HSE930U	15°	172.1	164	166	196	1.5
HSE932U	15°	182.1	174	176	206	1.5
HSE934U	15°	192.1	184	186	216	1.5
HSE010	15°	61.6	57	59	73	1.5
HSE011	15°	69.7	63	65	82	1.5
HSE012	15°	74.7	68	70	87	1.5
HSE013	15°	79.7	73	75	92	1.5
HSE014	15°	86.9	78	80	101	1.5
HSE015	15°	91.9	83	85	106	1.5
HSE016	15°	99.2	90	92	115	1.5
HSE017	15°	104.2	95	97	120	1.5
HSE018	15°	111.4	101	103	129	1.5
HSE019	15°	116.4	106	108	134	1.5
HSE020	15°	121.4	112	114	138	1.5
HSE021	15°	128.7	117	119	148	1.5
HSE022	15°	135.2	122	126	158	1.5
HSE024	15°	145.2	133	136	167	1.5
HSE026	15°	158.5	143	149	187	1.5
HSE028	15°	170.8	153	160	197	1.5
HSE030	15°	181.5	165	171	210	1.5
HSE032	15°	193.2	175	183	225	1.5
HSE034	15°	207.8	185	197	245	1.5

Note) Spacer dimensions are the same for all contact angles (15°, 20° and 25°).

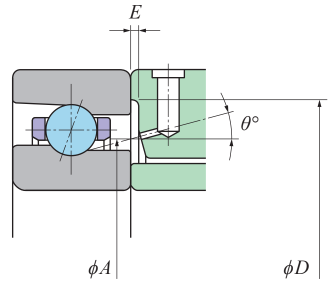


Fig. 7.5 BNT and HTA types

Table 7.6 Air-oil/oil mist nozzle spacer dimensions

Unit: mm

Bearing No.	θ	A	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	D	E
BNT900	12°	14.3	12.2	13.2	18.5	1
BNT901	12°	16.3	14.2	15.2	20.5	1
BNT902	12°	19.2	17.1	18.1	24	1
BNT903	12°	21.2	19.1	20.1	26	1
BNT904	12°	26	23.5	24.5	32.5	1
BNT905	12°	31	28.5	29.5	37.5	1
BNT906	12°	35.8	33.5	34.5	42.5	1
BNT907	12°	41.1	38.5	39.5	50	1
BNT908	12°	47.1	44.4	45.4	56	1
BNT909	12°	52.3	49	50	61.5	1
BNT000	15°	15.1	13	14	22	1
BNT001	15°	17.7	15.6	16.6	24	1
BNT002	15°	21	18.6	19.6	28	1
BNT003	15°	22.9	20.6	21.6	30	1
BNT004	15°	28.1	25	26	37	1
BNT005	15°	32.6	30.5	31.5	41.5	1
BNT006	15°	39.1	35.5	36.5	49.5	1
BNT007	15°	44	41	42	56	1
BNT008	15°	49.8	47	48	61	1
BNT009	15°	55.2	52	53	68	1
BNT200	15°	17.5	15.4	16.4	24.5	1
BNT201	15°	18.9	16.8	17.8	26.5	1
BNT202	15°	21.4	19.3	20.3	29	1
BNT203	15°	24.6	22	23	34	1
BNT204	15°	30	26.5	27.5	40.5	1
BNT205	15°	34.8	32	33	45.5	1
BNT206	15°	40.9	37.5	38.5	54.5	1
BNT207	15°	46.6	43.5	44.5	64	1
BNT208	15°	52.5	49	50	71.5	1
BNT209	15°	56.9	54.5	55.5	76.5	1

Table 7.7 Air-oil/oil mist nozzle spacer dimensions

Unit: mm

Bearing No.	θ	A	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	D	E
HTA920	15°	116.4	110	112	130	1.5
HTA921	15°	121.4	115	117	135	1.5
HTA922	15°	126.4	120	122	140	1.5
HTA924	15°	138.7	132	134	153	1.5
HTA926	15°	151	143	145	167	1.5
HTA928	15°	161	153	155	177	1.5
HTA930	15°	174.9	165	167	195	1.5
HTA932	15°	184.9	175	177	205	1.5
HTA934	15°	194.9	185	187	215	1.5
HTA936	15°	208.1	197	199	233	1.5
HTA938	15°	218.1	208	210	242	1.5
HTA940	15°	232.5	220	222	260	1.5
HTA006	15°	39.5	35.5	36.5	49.5	1
HTA007	15°	44.3	41	42	56	1
HTA008	15°	49.9	47	48	61	1
HTA009	15°	56.1	52	53	68	1
HTA010	15°	61.1	57	59	73	1.5
HTA011	15°	69.3	63	65	82	1.5
HTA012	15°	74.3	68	70	87	1.5
HTA013	15°	79.3	73	75	92	1.5
HTA014	15°	86.4	78	80	101	1.5
HTA015	15°	91.4	83	85	106	1.5
HTA016	15°	98.7	90	92	115	1.5
HTA017	15°	103.7	95	97	120	1.5
HTA018	15°	111	101	103	129	1.5
HTA019	15°	116	106	108	134	1.5
HTA020	15°	121	112	114	138	1.5
HTA021	15°	128.4	117	119	148	1.5
HTA022	15°	134.9	122	126	158	1.5
HTA024	15°	144.9	133	136	167	1.5
HTA026	15°	158.1	143	149	187	1.5
HTA028	15°	170.4	153	160	197	1.5
HTA030	15°	181.2	165	171	210	1.5
HTA032	15°	192.7	175	183	225	1.5
HTA034	15°	207.4	185	197	245	1.5

Note) Spacer dimensions are the same for all contact angles (30°and 40°).

Lubrication of Bearings

When lubricant is supplied between the cage and outer ring

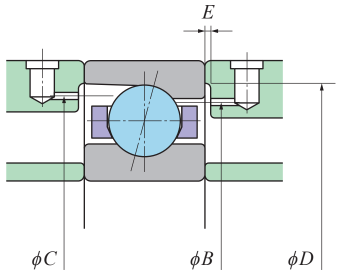


Fig. 7.6 78C and 79C types

Table 7.8 Air-oil/oil mist nozzle spacer dimensions Unit: mm

Bearing No.	When lubricant is supplied between the cage and outer ring					
	B	*C*	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	*D*	*E*
7805C	32.6	33.3	28	29	34	1
7806C	37.6	38.2	33	34	39	1
7807C	42.6	43.1	38	39	44	1
7808C	47.8	48.4	43	44	49	1
7809C	53.2	54.3	48.5	49.5	54	1
7810C	59.5	60.2	54	55	60.5	1
7811C	66.2	67.4	59	61	68	1
7812C	71.7	72.8	64.5	66.5	73.5	1
7813C	77.7	78.7	70.5	72.5	79.5	1
7814C	82.4	83.6	75.5	77.5	84.5	1
7815C	87.8	88.8	80.5	82.5	89.5	1
7816C	92.5	93.6	85.5	87.5	94.5	1
7817C	101	102.5	91.5	93.5	103.5	1
7818C	106	107.3	96.5	98.5	108.5	1
7819C	111	112.4	101.5	104	113.5	1
7820C	115.6	117	106.5	110	118.5	1
7821C	120.7	122	111.5	115	123.5	1
7822C	129.2	131.1	117.5	122	132.5	1
7824C	139.2	141.1	127.5	132	142.5	1
7826CT1	152.3	154.5	139	144	156.5	1.5
7828CT1	162.3	164.5	149	155	166.5	1.5
7830CT1	175.3	177.8	160.5	167.5	180.5	1.5
7832CT1	185.5	188	170.5	177.5	190.5	1.5
7834CT1	198.7	201.5	181	188	204.5	1.5
7928CT1B	171.3	176.9	153	163	179	1.5
7930CT1B	187.2	193.8	165	179	197	1.5
7932CT1B	198.3	201.9	175	190	205	1.5
7934CT1B	208.2	211.9	185	200	215	1.5

Note) 7805C to 7834CT1, 7928CT1B to 7934CT1B ··· *B* is recommended.
If targeting at *B* is impossible, targeting of *C* is acceptable.

(a) When lubricant is supplied between the cage and inner ring

(b) When lubricant is supplied between the cage and outer ring

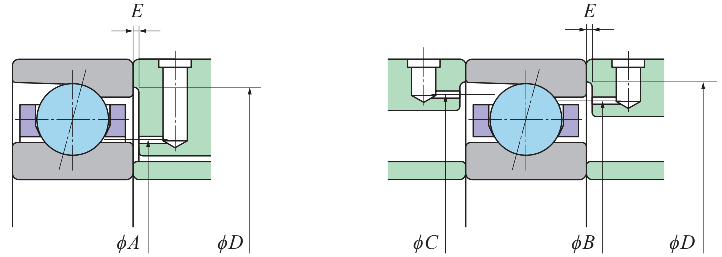


Fig. 7.7 70C and 72C types

Table 7.9 Air-oil/oil mist nozzle spacer dimensions Unit: mm

Bearing No.	(a) When lubricant is supplied between the cage and inner ring			(b) When lubricant is supplied between the cage and outer ring				Common to (a) & (b)	
	A	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	*B*	*C*	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	*D*	*E*
7200C	—	—	—	23	23.8	15.5	17.5	25	1
7201C	—	—	—	24.9	25.8	17.5	19.5	27	1
7202C	—	—	—	28.3	29.4	20.5	22.5	30	1
7203C	—	—	—	32.4	33.7	23.5	26.5	35	1
7204C	—	—	—	38.4	40.2	26.5	31	41.5	1
7205C	—	—	—	43.3	44.7	32	36	46.5	1
7206C	—	—	—	51.1	53	37.5	44	54.5	1
7207C	—	—	—	59.1	61.2	43.5	52	64	1
7208C	—	—	—	65.9	68.3	49	58	71.5	1
7209C	—	—	—	71.3	73.8	54.5	63	76.5	1
7210C	—	—	—	76.4	78.8	59.5	68	81	1.5
7211C	—	—	—	84.6	87.4	66	76	90	1.5
7212C	—	—	—	94.4	97.5	72	85	99.5	1.5
7213C	—	—	—	100.8	104.1	77.5	92	108.5	1.5
7214C	—	—	—	106.2	109.6	83	96	114	1.5
7215C	—	—	—	112.2	115.6	88.5	102	118	1.5
7216C	—	—	—	119.5	123.2	94	109	127	1.5
7217C	—	—	—	128	131.8	100	117	136	1.5
7218C	—	—	—	136.2	140.4	106	125	146	1.5
7219C	119.4	111.5	113.5	144.4	149	111.5	132	155	1.5
7220C	126.1	117.5	120	152.7	157.7	117.5	141	164	1.5
7221C	131.6	122.5	125	159.9	165.1	122.5	148	173.5	1.5
7222C	138.3	129	131	168.5	174.1	129	157	182	1.5
7224C	149.3	141	143	181.5	187.2	141	169	196	1.5
7226C	161.3	152.5	155	193	199.2	152.5	181	210	1.5
7028CT1B	162.9	153	157	183.5	187.4	153	172	197	1.5
7030CT1B	174.4	165	169	196.6	200.9	165	185	210	1.5
7032CT1B	185.7	175	180	209.8	214.2	175	198	225	1.5
7034CT1B	199.2	185	193	226	231.3	185	214	245	1.5
7036CT1B	212.2	197	206	242	248	197	230	263	1.5
7038CT1B	222.2	210	216	252	258	210	240	270	1.5
7040CT1B	235.2	220	229	268	275	220	255	290	1.5

Note) 7200C to 7218C ·········· *B* is recommended.
7219C to 7226C, 7028CT1B to 7040CT1B ·········· *A* is recommended.
If targeting at *A* is impossible, targeting at *B* is acceptable. If both *A* and *B* are impossible, targeting of *C* is acceptable.

Lubrication of Bearings

(2) Cylindrical roller bearings

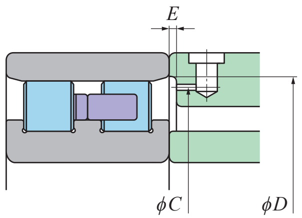


Fig. 7.8 NN30 and NN30T6 types

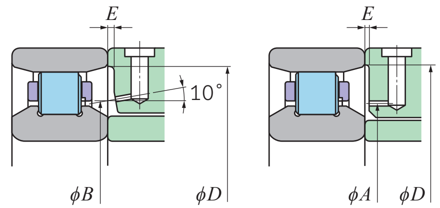


Fig. 7.9 N10HS types

Table 7.10 Unit: mm

Bearing No.	*C*	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	*D*	*E*
NN3005	40.3	31	33.8	42	1
NN3006	47	38	40.5	50	1
NN3007	53.5	43	47.0	57	1
NN3008	59.5	48	53.0	63	1
NN3009	66	54	59.5	69	1
NN3010	71	59	64.5	74	1.5
NN3011	79	65	72.5	83	1.5
NN3012	84	70	77.5	88	1.5
NN3013	90 (89)	75	82.5	93	1.5
NN3014	98	82	90	102	1.5
NN3015	103	87	95	107	1.5
NN3016	111	93	103	115	1.5
NN3017	116	98	108	120	1.5
NN3018	125	105	117	130	1.5
NN3019	130	110	122	135	1.5
NN3020	135	115	127	140	1.5
NN3021	144 (143)	120	135	149	1.5
NN3022	153 (152)	127	144	158	1.5
NN3024	163 (162)	137	154	168	1.5
NN3026	179	150	171	185	1.5
NN3028	189	160	181	195	1.5
NN3030	202	172	194	210	1.5
NN3032	215.5	183	208	223	1.5
NN3034	232	196	224	240	1.5
NN3036	251	209	243	259	1.5
NN3038	261	219	253	269	1.5

NOTE) With certain products, the dimension C of L1 cage differs from that of T6 cage. The values in parentheses ( ) are dimensions C of L1 cages. Other dimensions of L1 cages are same as those of T6 cages.

Table 7.11 Unit: mm

Bearing No.	*A*	*B*	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	*D*	*E*
N1006HS	—	40.4	37	38	50	1
N1007HS	—	46.5	42	43	57	1
N1008HS	—	51.7	47	48	63	1
N1009HS	—	57.7	52	53	69	1
N1010HS	—	62.7	57	58	74	1.5
N1011HS	—	69.7	63.5	64.5	83	1.5
N1012HS	—	74.8	68.5	69.5	88	1.5
N1013HS	—	79.7	73.5	74.5	93	1.5
N1014HS	86	—	78.5	80.5	102	1.5
N1015HS	91	—	83.5	85.5	107	1.5
N1016HS	97.5	—	88.5	90.5	115	1.5
N1017HS	102.5	—	93.5	95.5	120	1.5
N1018HS	110	—	102	104	130	1.5
N1019HS	115	—	107	109	135	1.5
N1020HS	120	—	112	114	140	1.5
N1021HS	125.9	—	118	120	149	1.5
N1022HS	133.1	—	123	125	158	1.5
N1024HS	143.3	—	133	135	168	1.5
N1026HS	157.2	—	143	145	185	1.5
N1028HS	167.2	—	153	155	195	1.5
N1030HS	179.6	—	165	167	210	1.5
N1032HS	191.1	—	175	177	223	1.5

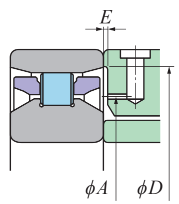


Fig. 7.10 N10HSR types

Table 7.12 Unit: mm

Bearing No.	*A*	Outside diameter of inner ring spacer	Bore diameter of outer ring spacer	*D*	*E*
N1009 HSRT6	58.3	52	53	69	1.0
N1011 HSRT6	71.5	63.5	64.5	83	1.5
N1012 HSRT6	76.6	68.5	69.5	88	1.5
N1013 HSRT6	81.5	73.5	74.5	93	1.5
N1014 HSRT6	89.7	78.5	80.5	102	1.5
N1016 HSRT6	101.3	88.5	90.5	115	1.5
N1018 HSRT6	113.8	102	104	130	1.5
N1020 HSRT6	123.8	112	114	140	1.5

Lubrication of Bearings

NTN

7.3 Jet lubrication

With this lubricating system, a high speed jet of lubricant is injected into the bearing from the side. This is the most reliable lubricating technique and is typically used on the main spindle bearings of jet engines and gas turbines. It is currently capable of a $d_m n$ value of up to approximately 4.0×10^6.

When used as a lubricating system for the main spindle of a machine tool, it can minimize the temperature rise of the bearing. However, the resultant torque loss is great, as a large amount of oil is supplied to each bearing. Therefore, this arrangement requires a powerful motor to drive the main spindle. Low viscosity oil (2 to 3 mm^2/s) is used.

Fig. 7.11 shows examples of the temperature rise with air-oil lubrication and jet lubrication, while **Fig. 7.12** graphically plots test results of power loss.

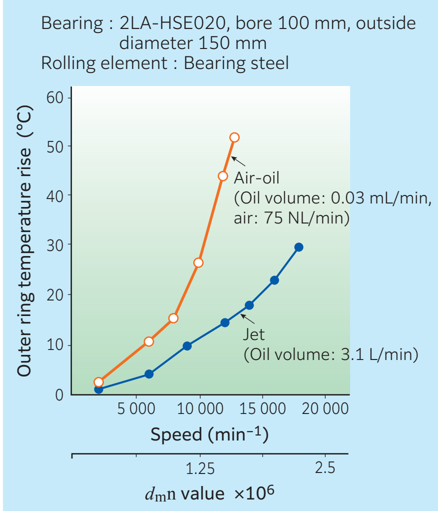


Fig. 7.11 Comparison of temperature rise of outer ring with air-oil lubrication and jet lubrication

(The temperature rise with air-oil lubrication is relative to room temperature; the temperature with jet lubrication is relative to lubricant temperature.)

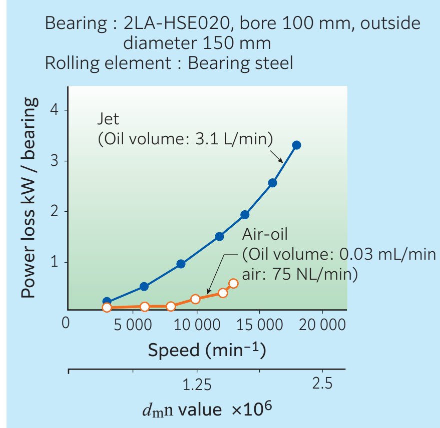


Fig. 7.12 Comparison of power loss with air-oil lubrication and with jet lubrication

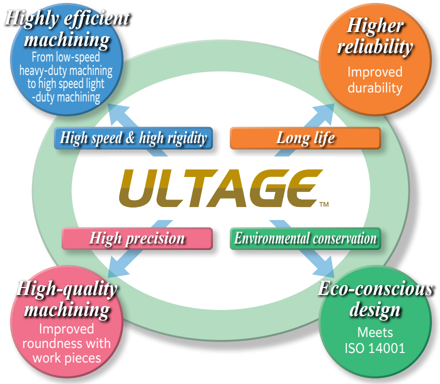
Technical Data

8. Precision Bearing Technologies

8.1 ULTAGE™ series precision bearings for machine tool main spindles

NTN has responded to need for improved efficiency, reliability, quality and environmental responsibility for machine tools by developing the ULTAGE series of precision bearings. ULTAGE series of bearings demonstrates excellent performance thanks to the optimal internal design; a new approach to surface quality; and the use of special materials, special grease, and seals on both sides.

ULTAGE is the name for **NTN**'s goal of achieving the ultimate performance with precision bearings, and expresses the "ULTIMATE" performance on any type of "STAGE."



■ Concept

Our ideal is to offer a ultra high speed precision bearing that offers excellent reliability while remaining eco-friendly.

[Design]

The internal bearing design has been optimized to cope with varying applications and operating conditions in order to realize high speed and high rigidity, limited temperature rise, high precision, energy saving and low noise emission. It performs optimally in a variety of situations.

[Material]

Adoption of special material and a special surface modification technique has resulted in greatly enhanced reliability.

[Lubrication]

Use of unique eco-conscious technology and special grease contributes to decreased pollution and enhanced energy savings.

[Precision]

Our ultra high precision technology, in conjunction with our proven precision bearing technology, will help attain further improved precision.

8.2 Material and surface modification

The ULTAGE series high speed and ultra high speed precision bearings for machine tool main spindle employs a special material that boasts excellent anti-seizure properties and wear resistance, as well as a unique surface modification technique.

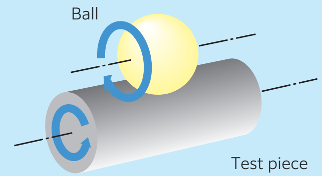
■ Life under normal temperatures

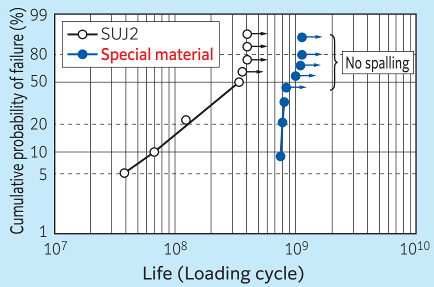
The test results obtained from point contact test pieces under greater loading are graphically plotted in **Fig. 8.1**.

[Test conditions]

Test piece	ϕ12×22 mm cylindrical roller
Ball	ϕ19.05 (3/4")
Max. contact stress	5.88 GPa
Loading frequency	46 240 cycles/min
Lubricant	Turbine oil ISO VG56 (oil bath)

[Schematic of test rig]





	L_{10} (×10⁷cycles)	Life ratio
SUJ2	6.3	1
Special material	79.8	12.7

The rolling fatigue life of the special material is approximately 13 times as long as that of SUJ2.

Fig. 8.1 Life test results with point contact test pieces

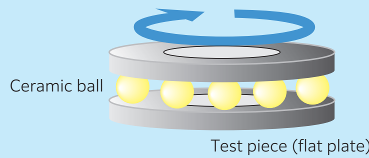
■ Life under high temperature

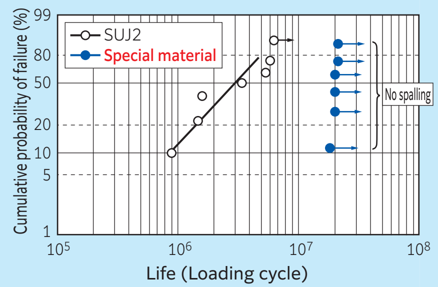
The test results obtained from thrust-type test pieces at 200 °C are graphically plotted in **Fig. 8.2**.

[Test conditions]

Test piece	ϕ47×ϕ29×t7 mm flat plate
Ball	ϕ6.35 (1/4") Ceramic ball
Max. contact stress	5.5 GPa
Loading frequency	3 000 cycles/min
Oil temperature	200 °C
Lubricant	Ether-based oil

[Schematic of test rig]





	L_{10} (×10⁵cycles)	Life ratio
SUJ2	9.1	1
Special material	No spalling	>30

The rolling fatigue life of the special material at a high temperature of 200 °C is more than 30 times as long as that of SUJ2.

Fig. 8.2 High temperature life test results with thrust-type test pieces

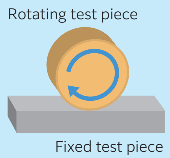
■ **Improved wear resistance**

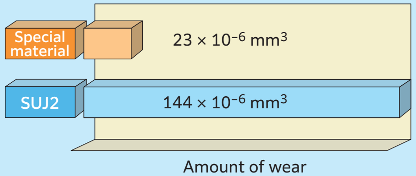
Test results with a Sawin type friction and wear test machine are illustrated in **Fig. 8.3**.

[Test conditions]

Speed	4 775 min^{-1}
Peripheral speed	10 m/s
Test duration	10 min
Max. contact stress	2.0 GPa
Lubricant	ISO VG2
Rotating test piece	Si_3N_4
Fixed test piece	Material being tested

[Schematic of test rig]





The wear resistance of the special material is 6 times better than that with SUJ2.

Fig. 8.3 Test results with Sawin type friction and wear test machine

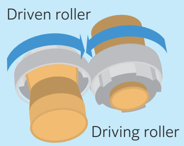
■ **Improved anti-seizure property**

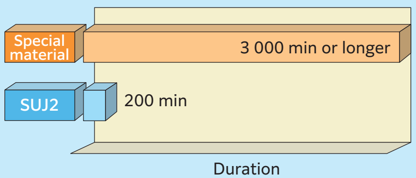
Test results with a two roller testing machine are illustrated in **Fig. 8.4**.

[Test conditions]

Driving side speed	200 min^{-1}
Driven side speed	120 min^{-1}
Sliding velocity	10.1 m/min
Max. contact stress	2.9 GPa
Lubricant	ISO VG2
Driving roller	Material being tested
Driven roller	Material being tested

[Schematic of test rig]





The anti-seizure property of the special material is more than 15 times better than as that with SUJ2.

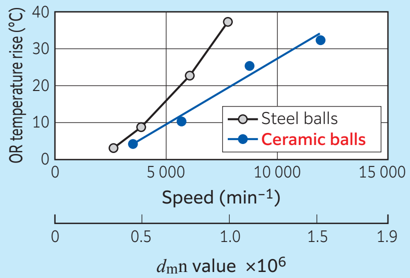
Fig. 8.4 Test results with a two roller testing machine

■ **Adoption of ceramic balls**

A comparison of temperature rise, which can vary depending on the material of rolling element, is illustrated in **Fig. 8.5**.

[Test conditions]

Bearing	Basic bearing number HSE020 (ϕ100 × ϕ150 × 24 × 2 rows)
Preload	2.5 kN (constant pressure preloading)
Lubrication	Grease lubrication
Jacket cooling	w/o

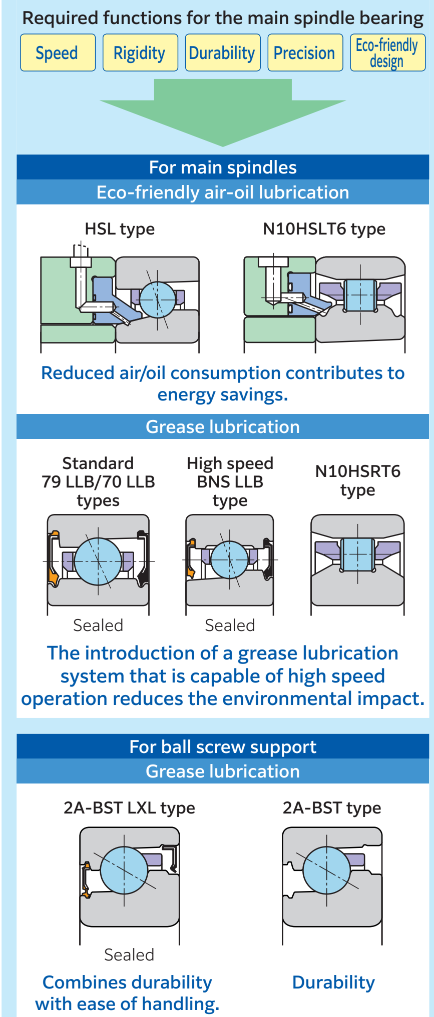


The ULTAGE series bearings employ well-proven ceramic balls to efficiently limit temperature rise.

Fig. 8.5 Comparison of temperature rise with steel and ceramic rolling elements

8.3 Environmentally conscious technology

The eco-friendly ULTAGE series is available in two specifications: an eco-friendly air-oil lubrication design that offers energy savings by reducing air and oil consumptions; and a grease-lubricated, sealed design that reduces environmental impact by employing a grease lubrication system that is capable of higher-speed operation.

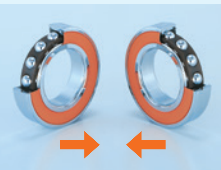
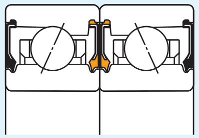
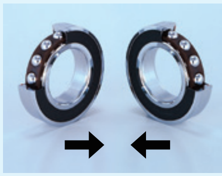
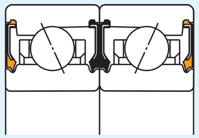


■ **Grease-lubricated sealed angular contact ball bearings**

(1) Ease of handling

ULTAGE series sealed angular contact ball bearings are grease-prefilled bearings. No grease filling is necessary; you need only wipe off the rust-preventive oil before assembly. Seals of different colors are employed to differentiate the front and back. The black front face and orange back face are easily identified, which also makes it easy to orient the bearings in combinations (see **Table 8.1**).

Table 8.1 Bearing combinations and seal colors

DB set (back-to-back)		DF set (face-to-face)	
Orange seal + Orange seal		Black seal + Black seal	
			

(2) Suggestions for simplified spindle structure

The ULTAGE series sealed angular contact ball bearing makes possible high speed operation with grease lubrication thanks to optimized internal design. Grease lubrication with minimal mist splash simplifies main spindle structure and contributes to lower environmental impact as well as cost reduction (see **Fig. 8.6**).

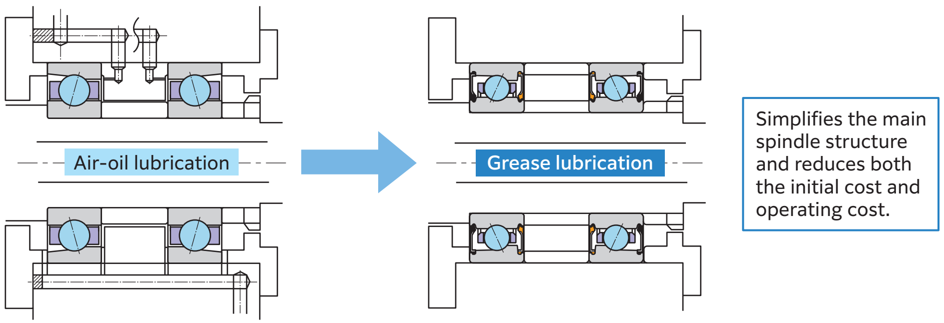


Fig. 8.6 Alteration to lubrication system (air-oil lubrication to grease lubrication)

■ **Eco-friendly air-oil lubricated angular contact ball bearings and cylindrical roller bearings**

When combined with the eco-friendly nozzle, the eco-friendly air-oil lubricated angular contact ball bearing (HSL/HSFL types) or cylindrical roller bearing [N10HSL (K) type] can reduce the emissions of oil mist and noise.

(1) Reduction of oil mist

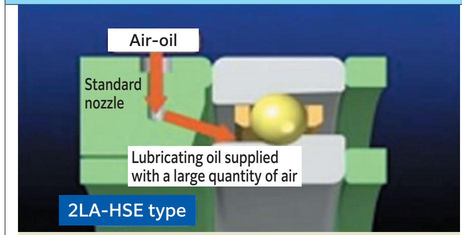
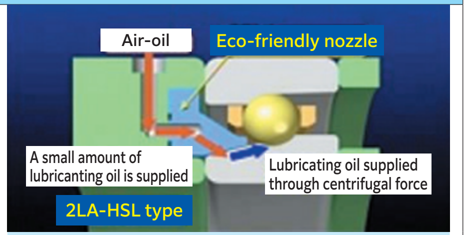
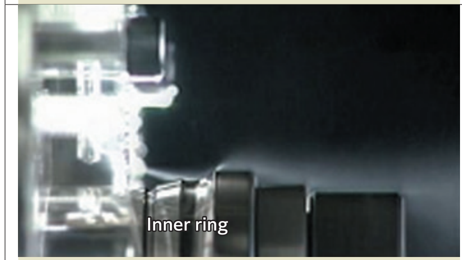
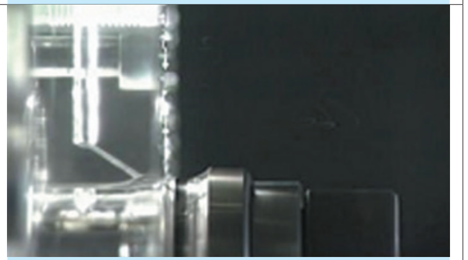
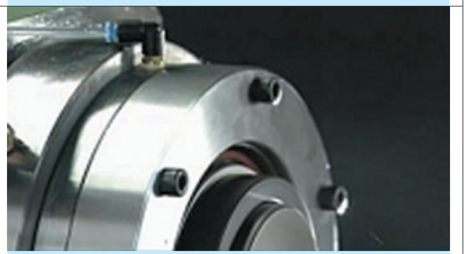
The eco-friendly air-oil lubricated bearing does not spray compressed air from the nozzle; instead, it uses the centrifugal force of the rotating inner ring to supply lubricating oil into the bearing. For this reason, this type of bearing conserves both air and oil. In addition, it reduces the amount of oil mist emitted from the labyrinth seal of the spindle. The **Table 8.2** reveal the difference between the amount of oil mist emitted from the standard bearing and that emitted from the eco-friendly bearing.

The lubricating oil discharged with air passes through the inside of the bearing and is then exhausted as a large volume of mist.

The lubricating oil exhausted from the bearing in the mist state is collected through the discharge port of the main spindle housing, but some of the oil mist leaks from the main spindle labyrinth seal and contaminates the immediate environment around the machine.

Adoption of the eco-friendly bearing therefore improves the working environment.

Table 8.2 Comparison of oil mist emissions between standard bearing and eco-friendly bearing

Standard bearing	Eco-friendly bearing
 Standard bearings consume a great deal of air when supplying lubricating oil to the bearing.	 The eco-friendly type uses centrifugal force to supply lubricating oil into the bearing.
 The oil emitted from the nozzle is in a mist state.	 The oil emitted from the nozzle is in a liquid state.
 A large amount of oil mist, contaminating the working environment.	 It reduces the amount of oil mist discharged and improves the working environment.

(2) Noise Reduction

The standard air-oil lubrication method uses air to supply a slight amount of oil. It also uses a special nozzle spacer, as shown in **Fig. 8.7** and **Fig. 8.9**.

In addition, this method uses a nozzle measuring 1 to 1.5 mm in diameter to supply oil to the raceway surface of the bearing at the rate of 30 to 40 NL/min/bearing. To supply this oil, the nozzle emits compressed air as a jet to break the air barrier of the bearing, which is created when running at high speed. In this way, the air is used as a tool for supplying oil. The eco-friendly bearing developed by **NTN** reduces the amount of air consumed, thus reducing the whistling noise of the flowing air. The mechanism used in this type of bearing is as follows: the centrifugal force of the bearing inner ring feeds a small amount of oil from the nozzle to the raceway surface of the bearing along the tapered surface (see **Fig. 8.8** and **Fig. 8.10**).

Since the function of the compressed air is only to deliver lubricating oil to the cavity of the inner ring, a large quantity of air is not required. In addition, since the air used to supply the oil is released between the tapered surfaces, the whistling noise of air is also reduced.

When the eco-friendly bearing is employed, the noise is reduced by 6 to 8 dBA.

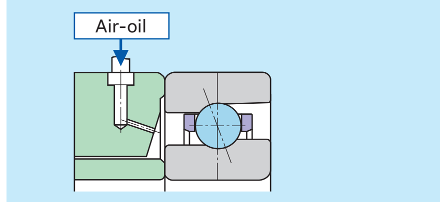


Fig. 8.7 Standard nozzle

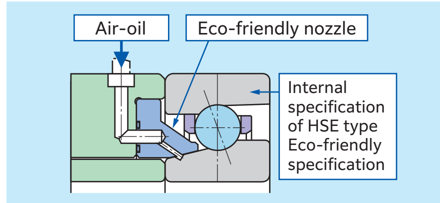


Fig. 8.8 Eco-friendly type nozzle

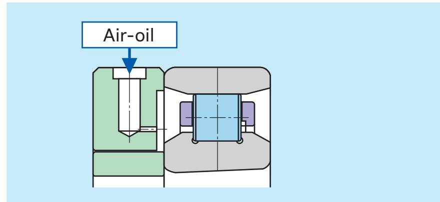


Fig. 8.9 N10HS type

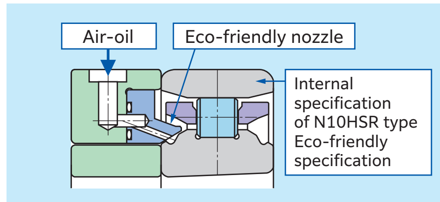
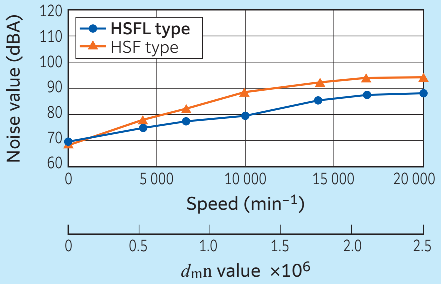


Fig. 8.10 N10HSL type

Example:

In the high speed region in excess of 10 000 min^{-1}, noise is reduced by 6 to 8 dBA (see **Fig. 8.11**).

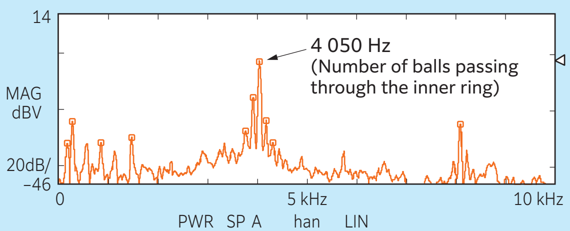


[Test conditions]

Bearing	5S-2LA-HSFL020 (eco-friendly bearing) 5S-2LA-HSF020 (ϕ100×ϕ150×24×2 rows)
Speed	0 to 20 000 min^{-1}
Preload	2.5 kN (constant pressure preloading)

Fig. 8.11 Comparison of noise values

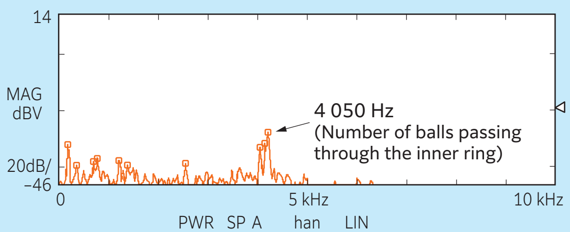
The eco-friendly bearing is particularly good for reducing “screeching” noise. The high-frequency component of the noise generated at high speeds is well attenuated. The reason for this is as follows: when the air jet emitted from the standard nozzle hits the rolling elements, a high-pitched noise is generated; in contrast, the eco-friendly nozzle does not emit air on the rolling elements, which reduces screeching noise (see **Fig. 8.12** and **Fig. 8.13**).



[Measurement conditions]

Bearing	5S-2LA-HSF020
Number of balls	37
Speed	14 000 min^{-1}

Fig. 8.12 Bearing noise frequency analysis results (Standard nozzle)



[Measurement conditions]

Bearing	5S-2LA-HSFL020
Number of balls	37
Speed	14 000 min^{-1}

Fig. 8.13 Bearing noise frequency analysis results (Eco-friendly nozzle)