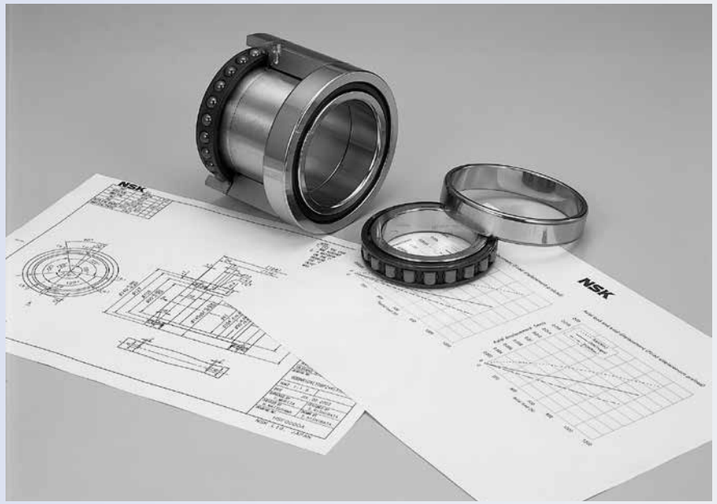


Part 5

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Rolling Fatigue Life and Basic Dynamic Load Rating

Bearing Life

The functions required of rolling bearings vary according to the bearing application, and these functions must be performed for a prolonged period. Even if bearings are properly mounted and correctly operated, they will eventually fail to perform satisfactorily due to an increase in noise and vibration, loss of running accuracy, deterioration of grease, or fatigue flaking of the rolling surfaces. In the broad sense of the term, bearing life refers to the period during which bearings continue to operate and satisfy their required functions. Depending on the cause of bearing failure, this bearing life may be defined as:

- noise life
- grease life
- rolling fatigue life

Besides natural deterioration, bearings may fail due to:

- heat seizure
- fracture or cracks
- wear
- rings becoming scored
- damaged seals

If the bearing fails for such reasons, this should not be interpreted as normal bearing life because these failures often occur as a result of selecting the wrong bearings, improper design or manufacture of the bearing surroundings, incorrect mounting, or insufficient maintenance.

Rolling Fatigue Life and Basic Rating Life

When rolling bearings are operated under load, the raceways of their inner and outer rings and rolling elements are subjected to repeated cyclic stress. The rolling contact surfaces of the raceways and rolling elements experience metal fatigue, and scaly particles may separate from the bearing material in a phenomenon called “flaking”. Rolling fatigue life is represented by the total number of revolutions at which the bearing surface will start flaking due to stress. Rolling fatigue life is often also called “bearing life” when the term is used in a more narrow sense.

Even for seemingly identical bearings of the same type, size, and material that receive the same heat treatment and other processing, the rolling fatigue life varies greatly, even under identical operating conditions. This is because the flaking of materials due to fatigue is subject to many other variables. Consequently, the concept of “basic rating life”, in which rolling fatigue life is treated as a statistical phenomenon, is used in preference to actual rolling fatigue life.

Suppose a number of bearings of the same type are operated individually under the same conditions. After a certain period, 10% of them will fail as a result of flaking caused by rolling fatigue. The total number of revolutions at this point is defined as the basic rating life. If the speed is constant, the basic rating life is often expressed by the total number of operating hours completed when 10% of the bearings become inoperable due to flaking.

In determining bearing life, basic rating life is often the only factor considered. However, other factors must also be taken into account. For example, it is possible to approximate the grease life of bearings prelubricated with grease. Since noise life and abrasion life are determined according to individual standards for different applications, specific values for noise or abrasion life have to be determined empirically.

Basic Dynamic Load Rating

The basic dynamic load rating applies to bearings with stationary outer rings and rotating inner rings. It is defined as the constant load that the bearing can endure for a rating life of one million revolutions (10^6 rev). For radial bearings, the basic dynamic load is given as a central radial load in a constant direction and magnitude, while the basic dynamic load rating of thrust bearings is given as an axial load of constant magnitude in the same direction as the central axis. The basic dynamic load ratings are listed in the bearing tables under C_r for radial bearings and C_a for thrust bearings.

The basic dynamic load rating of multi-row bearing arrangements can be calculated using the following formulas:

For ball bearings: $C_j = j^{0.7} \times C_1$

For roller bearings: $C_j = j^{7/9} \times C_1$

where j : Number of rows ($j \geqq 2$)
C_1 : Basic dynamic load rating for single-row bearings
C_j : Basic dynamic load rating for multi-row bearings (where j is the number of rows)

Special care must be taken when calculating the basic dynamic load rating C_a for thrust bearings, as j refers to the number of rows sustaining axial load only. For example, only two rows sustain the axial load in a four-row DBB arrangement; thus $j = 2$.

Steel ball bearings are specified by ISO 281 and hybrid bearings (ceramic ball bearings) by ISO 20056-1.

Basic Rating Life

The following relationships exist between basic dynamic load rating, dynamic equivalent load, and basic rating life:

For ball bearings: $L_{10} = \left(\frac{C}{P}\right)^3$ (10^6 rev)

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

For roller bearings: $L_{10} = \left(\frac{C}{P}\right)^{10/3}$ (10^6 rev)

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

P : Dynamic equivalent load (N)
C : Basic dynamic load rating (N)
n : Rotational speed (min^{-1})

For bearings that run at a constant speed, it is convenient to express the fatigue life in terms of hours.

Dynamic Equivalent Load

In some cases, the loads applied on bearings are purely radial or axial loads; however, in most cases, the loads are a combination of both. In addition, such loads usually fluctuate in both magnitude and direction.

In such cases, the loads actually applied to bearings cannot be used for bearing life calculations; therefore, a hypothetical load should be estimated that has a constant magnitude, passes through the center of the bearing, and achieves the same bearing life as the bearing would attain under actual conditions of load and rotation. Such a hypothetical load is called the dynamic equivalent load.

Assuming the equivalent radial load as P_r, the radial load as F_r, the axial load as F_a, and the contact angle as α, the relationship between the equivalent radial load and bearing load can be approximated as follows:

$$P_r = XF_r + YF_a$$

where X : Radial load factor
Y : Axial load factor
} see Table 1.1

The axial load factor varies depending on the contact angle. For roller bearings, the contact angle remains the same regardless of the magnitude of the axial load. In the case of single-row deep groove ball bearings and angular contact ball bearings, the contact angle increases when the axial load is increased. Such change in the contact angle can be expressed by the ratio of the basic static load rating C_{0r} and the axial load F_a. Table 1.1 shows the axial load factor at the contact angle corresponding to this ratio. Regarding angular contact ball bearings, if the contact angle is 25°, 30°, or 40°, the effect of contact angle change on the load factor may be ignored under normal operating conditions.

For thrust bearings with contact angles of $\alpha \neq 90°$ that receive both radial and axial loads simultaneously, the equivalent axial load P_a becomes as follows:

$$P_a = XF_r + YF_a$$

Relationship Between Bearing Life and Contact Angle

When a load is applied to an angular contact ball bearing, the amount of stress at the contact point changes as a result of the varying load conditions of the balls, inner ring, and outer ring relative to the contact angle of the bearing.

Figure 1.1 illustrates loads acting on two rolling elements at a 30° contact angle and a 15° contact angle.

The relationship between an axial load applied to the bearing and the resulting load acting on the rolling element can be formulated as:

Load on rolling element = F_a/(number of balls × sin α).

Therefore, the larger the contact angle, the smaller the load acting on the rolling element. The load at the contact point and its consequential deformation are reduced, thus resulting in longer life.

When a radial load is applied, the smaller the contact angle, the smaller the load acting on the rolling element. This results in reduced load at the contact point.

For these reasons, bearing life varies according to the contact angle under the same load conditions.

Fig. 1.1

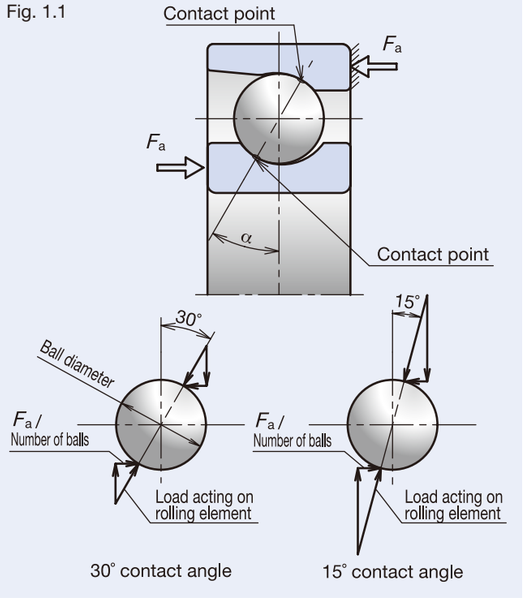


Table 1.1 Value of Factors X and Y for Calculation of Equivalent Load $P = XF_r + YF_a$

Bearing Type		if_oF_a/C_{or}	e	Single row (i=1) / DT (i=1) / DTD (i=1) / DTT (i=1)				DB (i=2) / DF (i=2) / DBB (i=2) / DFF (i=2)			
				$F_a/F_r \leqq e$		$F_a/F_r > e$		$F_a/F_r \leqq e$		$F_a/F_r > e$	
				X	Y	X	Y	X	Y	X	Y
Angular contact ball bearings ($C=C_r$)	15°	0.178	0.38	1	0	0.44	1.47	1	1.65	0.72	2.39
		0.357	0.40				1.40		1.57		2.28
		0.714	0.43				1.30		1.46		2.11
		1.07	0.46				1.23		1.38		2.00
		1.43	0.47				1.19		1.34		1.93
		2.14	0.50				1.12		1.26		1.82
		3.57	0.55				1.02		1.14		1.66
		5.35	0.56				1.00		1.12		1.63
	18°	0.181	0.44	1	0	0.43	1.30	1	1.42	0.70	2.11
		0.363	0.46				1.25		1.37		2.03
		0.726	0.48				1.18		1.30		1.92
		1.09	0.50				1.14		1.25		1.85
		1.45	0.52				1.10		1.21		1.79
		2.18	0.54				1.05		1.15		1.70
		3.63	0.57				1.00		1.10		1.63
	25°	—	0.68	1	0	0.41	0.87	1	0.92	0.67	1.41
	30°	—	0.80	1	0	0.39	0.76	1	0.78	0.63	1.24
	40°	—	1.14	1	0	0.35	0.57	1	0.55	0.57	0.93
Angular contact thrust ball bearings ($C=C_a$)	30°	—	0.72	—	—	0.48	1	0.74	0.55	0.48	1
	40°	—	1.05	—	—	0.60	1	1.01	0.64	0.60	1
	55°	—	1.79	—	—	0.81	1	1.60	0.60	0.81	1
	60°	—	2.17	—	—	0.92	1	1.90	0.56	0.92	1
Tapered roller bearings (1)		—	1.5tan α	1	0	0.4	0.4cot α	1	0.45cot α	0.67	0.67cot α

Bearing Type		e	DBD / DFD								DBT / DFT							
			Axial load supported by 2 rows					Axial load supported by 1 row				Axial load supported by 3 rows					Axial load supported by 1 row			
			i	$F_a/F_r \leqq e$		$F_a/F_r > e$		i	$F_a/F_r \leqq e$		$F_a/F_r > e$		i	$F_a/F_r \leqq e$		$F_a/F_r > e$		i	$F_a/F_r \leqq e$		$F_a/F_r > e$	
				X	Y	X	Y		X	Y	X	Y		X	Y	X	Y		X	Y	X	Y
Angular contact ball bearings ($C=C_r$)	15°	0.178	3/2	1	0.87	0.59	1.97	3	1	3.07	0.95	3.19	4/3	1	0.59	0.54	1.81	4	1	4.35	1.17	3.90
		0.357			0.83		1.86			2.91		3.02			0.56		1.71			4.12		3.70
		0.714			0.77		1.73			2.71		2.82			0.52		1.60			3.84		3.45
		1.07			0.73		1.65			2.58		2.68			0.50		1.52			3.65		3.28
		1.43			0.70		1.58			2.48		2.57			0.48		1.46			3.51		3.15
		2.14			0.66		1.49			2.33		2.42			0.45		1.37			3.30		2.96
		3.57			0.61		1.37			2.14		2.22			0.41		1.26			3.02		2.71
		5.35			0.56		1.27			1.98		2.06			0.40		1.22			2.94		2.64
	18°	0.181	3/2	1	0.75	0.57	1.72	3	1	2.63	0.93	2.80	4/3	1	0.51	0.53	1.57	4	1	3.73	1.15	3.38
		0.363			0.72		1.66			2.53		2.69			0.49		1.51			3.59		3.26
		0.726			0.68		1.57			2.40		2.55			0.46		1.43			3.41		3.09
		1.09			0.65		1.51			2.31		2.45			0.45		1.38			3.28		2.97
		1.45			0.63		1.46			2.24		2.38			0.43		1.34			3.18		2.88
		2.18			0.60		1.39			2.12		2.26			0.41		1.27			3.02		2.74
		3.63			0.56		1.29			1.97		2.10			0.40		1.22			2.91		2.64
	25°	—	—	1	0.48	0.54	1.16	—	1	1.71	0.89	1.88	—	1	0.33	0.50	1.08	—	1	2.45	1.08	2.33
	30°	—	—	1	0.41	0.52	1.01	—	1	1.45	0.84	1.64	—	1	0.28	0.48	0.93	—	1	2.05	1.03	2.01
	40°	—	—	1	0.29	0.46	0.76	—	1	1.02	0.76	1.23	—	1	0.20	0.42	0.71	—	1	1.46	0.92	1.54
Angular contact thrust ball bearings ($C=C_a$)	30°	—	—	0.91	0.41	0.48	1	—	0.56	0.89	0.48	1	—	0.98	0.30	0.48	1	—	0.46	1.04	0.48	1
	40°	—	—	1.24	0.39	0.60	1	—	0.76	0.84	0.60	1	—	1.35	0.29	0.60	1	—	0.62	0.98	0.60	1
	55°	—	—	1.95	0.36	0.81	1	—	1.20	0.78	0.81	1	—	2.12	0.27	0.81	1	—	0.98	0.90	0.81	1
	60°	—	—	2.32	0.35	0.92	1	—	1.43	0.76	0.92	1	—	2.52	0.26	0.92	1	—	1.17	0.88	0.92	1
Tapered roller bearings (1)		—	—					—					—					—				

Bearing Type		e	DBBD / DFFD								DBTD / DFTD								DBTT / DFTT							
			Axial load supported by 3 rows				Axial load supported by 2 rows				Axial load supported by 4 rows				Axial load supported by 1 row				Axial load supported by 5 rows				Axial load supported by 1 row			
			$F_a/F_r \leqq e$		$F_a/F_r > e$		$F_a/F_r \leqq e$		$F_a/F_r > e$		$F_a/F_r \leqq e$		$F_a/F_r > e$		$F_a/F_r \leqq e$		$F_a/F_r > e$		$F_a/F_r \leqq e$		$F_a/F_r > e$		$F_a/F_r \leqq e$		$F_a/F_r > e$	
			X	Y	X	Y	X	Y	X	Y	X	Y	X	Y	X	Y	X	Y	X	Y	X	Y	X	Y	X	Y
Angular contact thrust ball bearings ($C=C_a$)	30°	0.72	0.84	0.64	0.48	1	0.63	0.64	0.48	1	1.03	0.64	0.48	1	0.39	0.64	0.48	1	1.06	0.64	0.48	1	0.34	0.64	0.48	1
	40°	1.05	1.15	0.60	0.60	1	0.87	0.60	0.60	1	1.41	0.60	0.60	1	0.53	0.60	0.60	1	1.45	0.60	0.60	1	0.47	0.60	0.60	1
	55°	1.79	1.82	0.56	0.81	1	1.37	0.56	0.81	1	2.22	0.56	0.81	1	0.84	0.56	0.81	1	2.28	0.56	0.81	1	0.74	0.56	0.81	1
	60°	2.17	2.15	0.55	0.92	1	1.62	0.55	0.92	1	2.63	0.55	0.92	1	1.00	0.55	0.92	1	2.71	0.55	0.92	1	0.88	0.55	0.92	1

(1) For tapered roller bearings, please also refer to the values listed in the bearing tables on pages 160 and 161.

New life calculations

The concept of rolling bearing life was standardized in the 1960s with the basic rating life L_{10}, and this metric has been widely used for evaluating bearing performance since. The basic rating life refers to the statistical lifespan that 90% of identical bearings with a standard internal design, materials, and manufacturing quality will achieve under normal operating conditions. This guideline has served effectively over the years as a basic benchmark for selecting and comparing bearings.

However, in real-world applications, differences in bearing steel quality, lubrication conditions, operating environment (temperature, contamination, etc.) and load conditions often lead to significant discrepancies between the basic rating life and actual service life. For example, in a clean environment with good lubrication, contact stress between the rolling elements and raceway stays below the fatigue limit, resulting in an actual life that far exceeds the basic rating life. Conversely, under adverse conditions such as poor lubrication or where foreign matter has entered the bearing, actual life may become much shorter than the basic rating life due to localized stress concentrations and increasing damage. Therefore, bearing life cannot be determined conclusively from load conditions alone, but rather requires an assessment of interactions between multiple factors.

To account for such factors in fatigue life calculations, the basic rating life can be adjusted using the following adjustment factors.

$$L_{na} = a_1\, a_2\, a_3\, L_{10} \quad \cdots\cdots(1)$$

where L_{na} : Adjusted rating life in which reliability, material improvements, lubricating conditions, etc. are considered
L_{10} : Basic rating life with a reliability of 90%
a_1 : Life adjustment factor for reliability
a_2 : Life adjustment factor for special bearing properties
a_3 : Life adjustment factor for operating conditions

The life adjustment factor for reliability a_1 can be changed for reliabilities higher than 90% as per Table 1.2.

The life adjustment factor for special bearing properties a_2 is used to reflect longer fatigue life from improvements in materials.

Table 1.2 Reliability Factor a_1

Reliability (%)	90	95	96	97	98	99
a_1	1.00	0.64	0.55	0.47	0.37	0.25

Furthermore, since these factors are closely interrelated, they may be considered collectively. Taking the special bearing properties factor a_2 and the operating conditions factor a_3 as a single combined factor ($a_2 \times a_3$) allows a more systematic and practical approach to evaluating life.

With higher purity in bearing materials and more advanced manufacturing technologies, lubrication conditions and surface damage have become major factors influencing life beyond conventional fatigue-based assessments. As such, there has been increasing need to evaluate these combined factors more systematically. In response, the International Organization for Standardization (ISO) further developed the combined factor ($a_2 \times a_3$) concept and introduced life modification factor a_{ISO}. This factor comprehensively accounts for material fatigue limits, lubrication conditions as expressed by a lubrication parameter, and environmental effects such as the level of contamination. Rather than relying on empirical factors alone, the life modification factor a_{ISO} comprises a systematized approach that incorporates the interdependencies between contact stress, fatigue limit, and lubrication conditions. This positions a_{ISO} as a metric for better grounding bearing life on physical criteria. This theoretical framework was formally standardized in ISO 281:2007 as a systems approach for evaluating bearing life that accounts for the interplay between relevant factors and has been since integrated into JIS B 1518:2013. Life obtained using the life modification factor a_{ISO} is called modified rating life L_{nm} and is calculated as shown in Equation (2), with term definitions detailed in subsequent sections.

$$L_{nm} = a_1 \cdot a_{ISO} \cdot L_{10} \quad \cdots\cdots(2)$$

Life modification factor a_{ISO} is calculated using Equation (3). For practical purposes, keep $a_{ISO} \leq 50$.

$$a_{ISO} = f\left[\frac{e_c\, C_u}{P},\ \kappa\right] \quad \cdots\cdots(3)$$

C_u: Fatigue limit load

Bearings are thought to have a fatigue limit load C_u under which fatigue will not occur. This is the load applied to the bearing that results in fatigue limit stress at the point of maximum load contact on the raceway and is affected by the type of bearing, internal specifications, materials, etc. The bearing tables in this catalog list the fatigue limit load C_u values for each bearing designation.

e_c: Contamination factor

Indentations occur when solid particles mixed in the lubricant become trapped between the raceway surface and the rolling elements, causing localized increases in stress that shorten bearing life. The effect of such contaminants is evaluated using contamination factor e_c. Table 1.3 shows guideline values for e_c based on particle type, size, hardness, and quantity of the particles, as well as the thickness of the lubricant oil film (viscosity ratio κ) and bearing size (rolling element pitch diameter D_{pw}). The contamination factor should be set in consideration of operating conditions such as oil cleanliness, filtration conditions, and seal design. Note that average value for bearing bore/outside diameter (mm) may be used as the rolling element pitch circle diameter D_{pw}.

κ: Viscosity ratio

Lubricants form an oil film on rolling contact surfaces that separates the rolling elements from the raceway. Oil film conditions are expressed by viscosity ratio κ that compares actual kinematic viscosity during operation ν to the reference kinematic viscosity ν_1 as calculated using Equation (4).

If viscosity ratio $\kappa > 4$, treat $\kappa = 4$, and if $\kappa < 0.1$, the equation does not apply.

$$\kappa = \frac{\nu}{\nu_1} \quad \cdots\cdots(4)$$

ν : Kinematic viscosity during operation/viscosity of the lubricant at operating temperature
ν_1 : Reference kinematic viscosity based on the bearing speed and rolling element pitch diameter D_{pw} for the balls or roller set

Table 1.3 Contamination Factor e_C

Contamination Level	e_C	
	$D_{pw} < 100$mm	$D_{pw} \geqq 100$mm
Extremely high cleanliness Particle size within the oil film thickness; laboratory-level conditions	1	1
High cleanliness Oil processed through a very fine filter; standard grease-packed bearings and sealed bearings	0.8–0.6	0.9–0.8
Normal cleanliness Oil processed through a filter; standard grease-packed bearings and sealed bearings	0.6–0.5	0.8–0.6
Light contamination Some contamination of the lubricant	0.5–0.3	0.6–0.4
Typical contamination No seals, coarse filtration, or an environment where wear particles and foreign matter enter the bearing	0.3–0.1	0.4–0.2
Heavy contamination Very contaminated enviornment and insufficient bearing sealing	0.1–0	0.1–0
Extreme contamination	0	0

(Source: JIS B 1518:2013)

NSK life calculation tools can be accessed from product pages on the NSK Bearing Search/Online Catalog site.

https://www.nsk.com/engineering/

Tap the calculator icon next to supported designations to see detailed life calculations and more.

Features of Micro-UT™
NSK's Exclusive Material Evaluation Technology

NSK super precision bearings feature high-reliability steels with excellent cleanliness (see p. 14 for full details.) We developed exclusive technology to evaluate material cleanliness using methods such as Statistics of Extreme Valueand have pushed to advance the quality control of rolling bearing steel since the 1990s. However, it remains difficult to predict life by numerically when flaking originates from non-metallic inclusions. In addition, there had been no commercially available method to detect microscopic inclusions that could lead to damage. Therefore, NSK developed Micro-UT: a unique ultrasonic technology to detect material flaws. As shown in Fig. 1.2 (a), Micro-UT allows us to quickly grasp the size and quantity of microscopic inclusions under 100 μm and obtain reliable statistical data about their distribution. NSK uses a proprietary method to process and assess Micro-UT data to maintain quality, thereby ensuring the high reliability of NSK bearings produced worldwide. Additionally, Micro-UT allows us to better quantify the high reliability of NSK bearings with more accurate life prediction.

Highly Accurate Life Prediction

Although using steel with high cleanliness can extend the flaking life of rolling bearings, there has been no way to accurately quantify this benefit. Therefore, we developed methods alongside Micro-UT to better predict flaking life using statistical data on the size and quantity of non-metallic inclusions in steel.

Three main factors influence flaking life: stress, material cleanliness, and material strength. NSK has developed the theoretical life equation below using parameters that account for these factors, establishing a method that accurately predicts the flaking life of NSK bearings supported by cleanliness data obtained with Micro-UT:

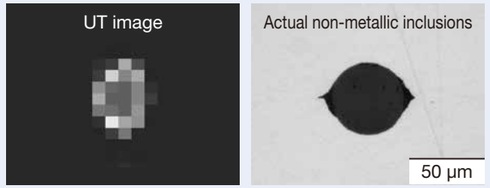
$$L = A\left[\frac{\tau}{\Delta K_{II} - \Delta K_{th}}\right]^{B}\left(\frac{C}{P}\right)^{p} \quad \cdots\cdots(5)$$

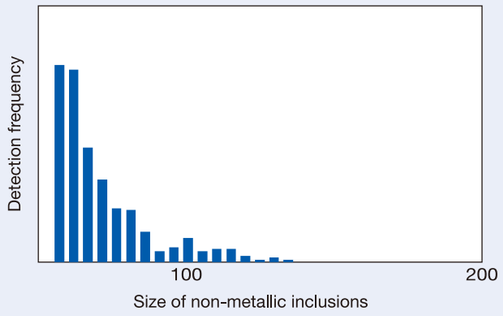
where A and B are empirical constants, τ is internal stress, and a parameter based on stress and defect size that is obtained from values given for the size of non-metallic inclusions and shear stress.

Figure 1.2 (b) compares predicted life from Equation (5) and data on the distribution of inclusions as obtained by Micro-UT with actual life from endurance tests. We can see that the life of NSK bearings far exceeds ISO predicted values, with our technology offering much more accurate predictions.

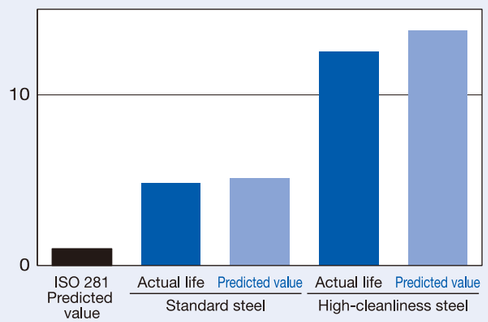
The basic dynamic load rating C of an NSK bearings is determined In accordance with in ISO 281:2007, but factors must be corrected to fully reflect the high reliability assured by our advanced technologies. With carefully selected, high-quality materials and processes, NSK bearings are your solution for reliability. NSK dynamic load ratings backed by our predictive technology can be found in Part 4 of this catalog and on our website.

Fig. 1.2 Example Micro-UT Data and Life Prediction Results





(a) Inclusions Detected by Micro-UT



(b) Calculations Using Highly Accurate Life Prediction Technology

Life of High-Speed Bearings

When bearings are operated at high speed, the effect of centrifugal force acting on the rolling elements must be considered (Fig. 1.3). In the basic rating life equation on page 190, a ratio of basic dynamic load rating and external load was used, and the internal load generated by centrifugal force was not taken into account.

NSK employs specific software to perform convergence calculations that obtain a balance of the forces acting on rolling elements, including centrifugal force and various other elements. An example result of such a calculation regarding bearing life in high-speed operation is given in Fig. 1.4.

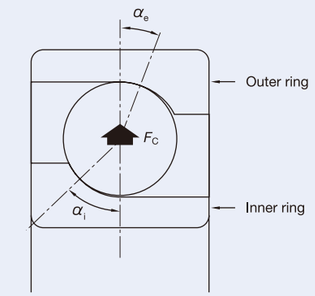
Please contact NSK for life calculations regarding bearings used in high-speed applications.

Note that bearing life as mentioned above indicates rolling fatigue life, which does not include bearing failure caused by seizure, wear, or the like. It is difficult to calculate how long a bearing will perform until a bearing failure takes place. Please refer to page 190 for details on the definition of bearing life.

Life of Ceramic Ball Bearings

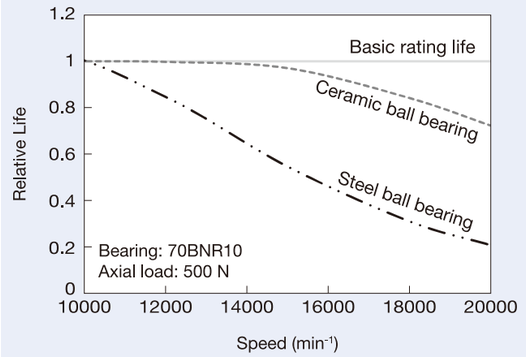
As ceramic balls have a lower density than steel balls, the increase in internal load due to centrifugal force in high-speed operation is less than with steel balls. Thus, the life calculation will show a longer life.

Fig. 1.3 Change in Contact Angle Due to Centrifugal Force



F_C : Centrifugal force acting on the rolling elements

Fig. 1.4 Example Calculated Life Considering Centrifugal Force



Life Calculation for Multiple Bearings as a Group

When multiple rolling bearings are used in one machine, the fatigue life of individual bearings can be determined if the load acting on individual bearings is known. In general however, the machine becomes inoperative if a bearing in any part fails. It may therefore be necessary in certain cases to know the fatigue life of a group of bearings used in one machine.

The fatigue life of bearings varies greatly, and the fatigue life calculation equation $L_{10} = \left(\frac{C}{P}\right)^{p}$ (where p=3 for ball bearings and $p=\frac{10}{3}$ for roller bearings) applies 90% life, or "basic rating life". This refers to either the number of revolutions or hours that 90% of bearings of the same type operated under the same conditions can reach.

In other words, the calculated fatigue life for one bearing has a probability of 90%. Since the endurance probability of a group of multiple bearings for a certain period is a product of the endurance probability of individual bearings for the same period, the basic rating life of a group of multiple bearings is not determined solely from the shortest basic rating life among the individual bearings. In fact, the group life is much shorter than the life of the bearing with the shortest fatigue life.

Defining the basic rating life of individual bearings as L_1, L_2, L_3 ... and the basic rating life of the entire group of bearings as L, the relationship may be expressed by the following equation:

$$L = \frac{1}{\left(\frac{1}{L_1^{1.1}} + \frac{1}{L_2^{1.1}} + \frac{1}{L_3^{1.1}} + \cdots\right)^{\frac{1}{1.1}}}$$

2. Static Load Ratings and Static Equivalent Loads

Basic Static Load Rating

When subjected to an excessive load or a strong shock load, rolling bearings may incur a local, permanent deformation of the rolling elements and raceway surface. This deformation increases in area and depth as the load increases, and when the load exceeds a certain limit, the smooth running of the bearing is impeded.

The basic static load rating produces the following calculated contact stress at the center of the contact area between the raceway surface and the rolling element subjected to the maximum stress:

Steel ball bearings : 4 200 Mpa
Ceramic ball bearings : 4 600 Mpa
Roller bearings : 4 000 Mpa

The basic static load rating value C_0 is listed in the bearing tables as C_{0r} for radial bearings and C_{0a} for thrust bearings.

The basic static load rating of multi-row bearing arrangements may be calculated using the following formula:

$$C_j = j \times C_1$$

where j : Number of rows
C_1 : Basic static load rating for a single-row bearing
C_j : Basic static load rating for a multi-row bearing (where j refers to the number of rows)

However, when calculating the basic static load rating for thrust bearings C_{0a}, j refers only to the number of rows that sustain an axial load.

Steel ball bearings are specified by ISO 281 and hybrid bearings (ceramic ball bearings) by ISO 20056-2.

Static Equivalent Load

The static equivalent load is a hypothetical load that produces a contact stress equal to the maximum contact stress the bearing is subjected to under actual load conditions while stationary in the area of contact between the most heavily stressed rolling element and the bearing raceway.

For radial bearings, the radial load passing through the bearing center is used as the static equivalent load. For thrust bearings, the axial load in the direction coinciding with the central axis is used as the static equivalent load.

(a) Static equivalent loads for radial bearings

The greater of the two values calculated from the following equations should be adopted as the static equivalent load for radial bearings:

$$P_0 = X_0 F_r + Y_0 F_a \quad \cdots\cdots(1)$$

$$P_0 = F_r \quad \cdots\cdots(2)$$

where P_0 : Static equivalent load (N)
F_r : Radial load (N)
F_a : Axial load (N)
X_0 : Static radial load factor
Y_0 : Static axial load factor

(b) Static equivalent loads for thrust bearings

$$P_0 = X_0 F_r + F_a \quad \alpha \neq 90° \quad \cdots\cdots(3)$$

where P_0 : Static equivalent load (N)
α : Nominal contact angle

Note that the accuracy of this equation decreases when $F_a < X_0 F_r$.

The values X_0 and Y_0 of equations (1) and (3) are shown in Table 2.1.

Note that $P_0 = F_a$ for thrust bearings with $\alpha = 90°$.

Table 2.1 Static Equivalent Loads

Bearing Type		Single Row		Double Row	
		X_0	Y_0	X_0	Y_0
Angular contact ball bearings	$\alpha=15°$	0.5	0.46	1	0.92
	$\alpha=18°$	0.5	0.46	1	0.92
	$\alpha=25°$	0.5	0.38	1	0.76
	$\alpha=30°$	0.5	0.33	1	0.66
Tapered roller bearings	$\alpha\neq0$	0.5	0.22cot α	1	0.44cot α
Cylindrical roller bearings	$\alpha=0$	$P_0 = F_r$			
Thrust ball bearings	$\alpha=90°$	$P_{0a} = F_a$			
Thrust roller bearings	$\alpha=90°$				
Thrust ball bearings	$\alpha\neq90°$	$P_{0a} = F_a + 2.3F_r \tan\alpha$ (where $F_a > 2.3 F_r \tan\alpha$)			
Thrust roller bearings	$\alpha\neq90°$				

Permissible Static Load Factor

The permissible static equivalent load on bearings varies depending on their basic static load rating, their application, and operating conditions. The permissible static load factor f_S is a safety factor that is applied to the basic static load rating, and it is defined by the ratio below:

$$f_S = (C_0/P_0)$$

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

Table 2.2 lists the generally recommended values for the permissible static load factor f_S.

Table 2.2 Permissible Static Load Factor f_s

Operating conditions	Lower limit of f_s	
	Ball bearings	Roller bearings
Low-noise applications	2.0	3.0
Bearings subjected to vibration and shock loads	1.5	2.0
Standard operating conditions	1.0	1.5

Permissible Axial Load for Angular Contact Ball Bearings

NSK has defined the limiting static axial load as the smaller of the two values listed below:

1. Limiting axial load that produces shoulder override
   The limiting load at which the contact ellipse generated between the ball and the raceway overrides the shoulder of the raceway groove (Fig. 2.1)

2. Limiting axial load in terms of surface pressure
   The limiting load at which the contact stress at the center of the contact area between the ball and the raceway groove reaches a level that leaves an indentation as defined in the basic static load rating (Fig. 2.2)

To maintain optimal bearing performance, NSK has defined permissible static axial load values by applying a safety factor to the limiting axial load based on many years of experience.

The formula for calculating the basic static axial load rating C_{0a} does not take the shoulder height of the raceway groove into account. Therefore, in some cases the C_{0a} value may exceed the limiting axial load that produces shoulder override. In such cases, the maximum load that the bearing can sustain is lower than the C_{0a} value, making the C_{0a} value unsuitable (Fig. 2.3). Therefore, instead of C_{0a} values, we have listed limiting axial load values in the bearing tables where necessary, particularly for angular contact thrust ball bearings as they are usually used to support heavy axial loads.

Fig. 2.1 Limiting Axial Load Producing Shoulder Override

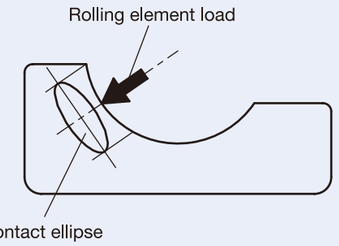


Fig. 2.2 Limiting Axial Load in Terms of Surface Pressure

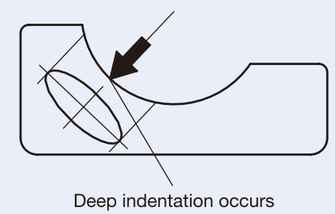
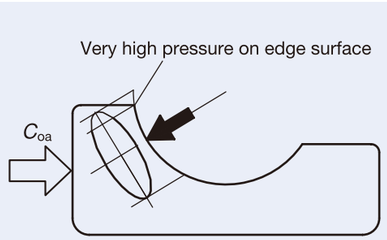


Fig. 2.3 C_{0a} and Limiting Axial Load



3. Angular Contact Ball Bearing Arrangements

Common arrangements of angular contact ball bearings for the fixed end of spindles use 2 row (DB), 3 row (DBD), and 4 row (DBB) matched sets. However, in 3-row combinations the preload is unevenly distributed between the bearings, resulting in a very limited range of preload unsuitable for high-speed applications.

Matched bearings are manufactured as sets, so when they are mounted adjacent to each other, the specified preload is automatically achieved. The variation of bore and outside diameter within each set of matched bearings is adjusted to less than 1/3 of the permissible tolerance.

Table 3.1 Arrangement Features

◎Excellent ◎Very good ○Good △Fair →One direction only ⇔Two directions

	DB	DF	DT	DBD	DBB
Load direction	⇔	⇔	→	⇔	⇔
Moment stiffness	◎	○	△	◎	◎
Speed capability	◎	◎	◎	△	○
Low heat generation	◎	◎	◎	△	○
Stiffness	○	○	△	◎	◎

Differences Between Arrangements

• Back-to-Back Arrangement DB

Axial loads in both directions and radial loads can be sustained. Since the distance between the effective load centers is large, this type is suitable if moments are applied. However, if housing accuracy is poor or mounting errors occur (misalignment), internal load may increase due to the high moment rigidity, potentially resulting in damage such as premature flaking.

• Face-to-Face Arrangement DF

Compared with the DB type, the distance between the effective load centers is small, so the capacity to sustain moments is inferior to the DB type. However, as this type can suppress increases in internal load from misalignment, it may be used when housing accuracy is poor, the shaft has low rigidly or significant deflection under load, etc.

• Tandem Arrangement DT

Axial loads in one direction and combined loads can be sustained. Since the internal load is shared between two bearings, this type is often used when load in one direction is heavy, with the drawback that preload must be applied externally by a spring or similar.

• 3-Row Arrangement DBD

Axial loads in both directions and radial loads can be sustained. However, as preload is not distributed evenly among the bearings, the internal load on the single-row side is twice that of the double-row side, which could cause damage during high-speed operation.

• 4-Row Arrangement DBB

Axial loads in both directions and radial loads can be sustained. DBB types provide approximately twice the preload and rigidity of DB arrangements at the same internal clearance and with higher permissible axial load.

Fig. 3.1 Distance Between Effective Load Centers of Back-to-Back and Face-to-Face Arrangements

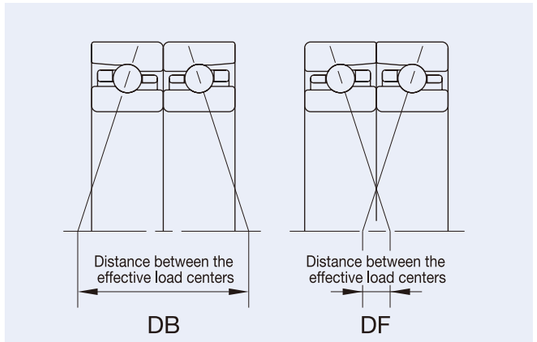


Fig. 3.2 Load Application Direction in Back-to-Back and Tandem Arrangements

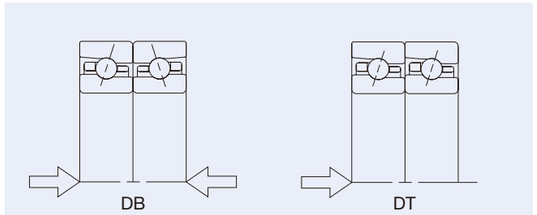
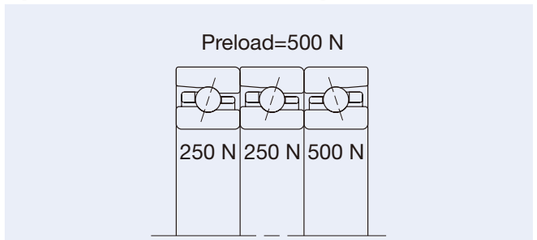


Fig. 3.3 Internal Preload in DBD Arrangement



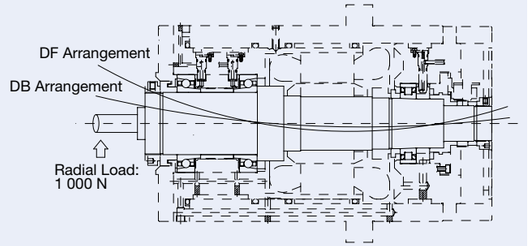
Shaft Bending in Arrangements

Moment rigidity is different between back-to-back and face-to-face arrangements. We have calculated typical shaft deflection for the spindle structure shown in Fig. 3.4. In this example, angular contact ball bearings (75BNR10XET) are used at the front end and shaft deflections are shown for both DB and DF configurations. A radial load of 1 000 N applied to the spindle nose will cause the following radial deflection of the spindle nose:

$$\sigma_{DB} = 2.4079 \times 10^{-2} \text{ (mm)}$$
$$\sigma_{DF} = 2.9853 \times 10^{-2} \text{ (mm)}$$

This demonstrates the effect of the distance between effective load centers on spindle bending.

Fig. 3.4 Spindle Deflection Curve



Mounting Instructions

Matching Direction

For matched bearings, the mounting order and load application direction are very important. As shown in Fig. 3.6, marks on the outer ring surfaces of the bearings form a “V” when the bearings are properly matched and aligned.

Radial Runout Mark

On the side surface or chamfered part of the inner rings, a “○” is marked to indicate the position of maximum radial run-out. Optimum accuracy is achieved when the bearing is mounted so the “○” is placed directly opposite the point of maximum shaft eccentricity.

Fig. 3.5 Mark at Position of Maximum Radial Runout of the Inner Ring

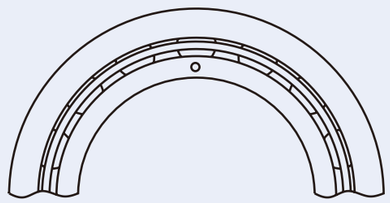
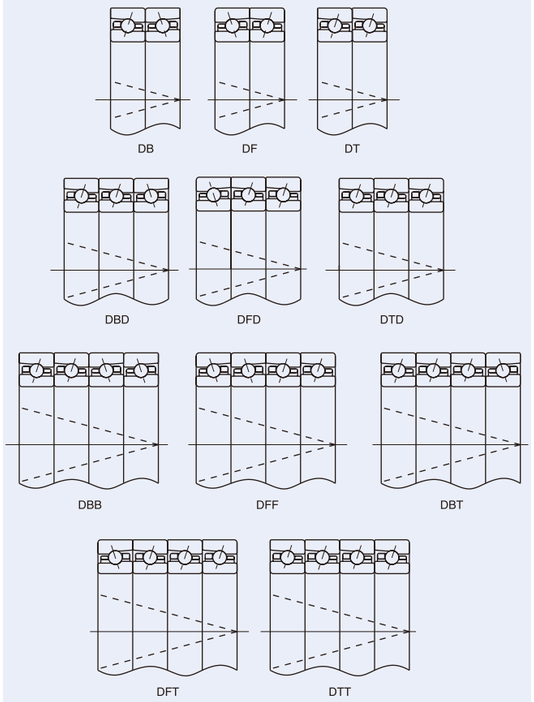


Fig. 3.6 Arrangements of Angular Contact Ball Bearings



3. Angular Contact Ball Bearing Arrangements

Important Points to Consider When Using Matched Bearings

The axial clearance (standout) of each bearing in a set of matched bearings (DB, DT, DFD etc.) has been adjusted and controlled to achieve a specified preload when bearings are correctly arranged per the markings on their outer ring surface. When positioned properly, these marks will form a "V" with the lines extending outwards continuously. For this reason, matched bearing sets can be removed and re-used, provided they are assembled together in the right orientation and sequence.

If the order or orientation of matched bearings is changed, the clearance between mating surfaces will not be adjusted properly, making the resulting clearance unknown and potentially causing excessive/insufficient preload or uneven load distribution. Matched bearing sets can be used in a different orientation or order only after measuring the standout of each bearing individually and using spacers to adjust the axial clearance. We do not recommend machining the bearings to achieve the correct standout, as grinding debris may enter the bearings.

To use the same bearings on various machines, we recommend stocking universal matching bearings rather than sets of matched bearings prepared for specific arrangements. For more information on universal matching bearings, please refer to page 204.

Fig. 3.7 DT Combination

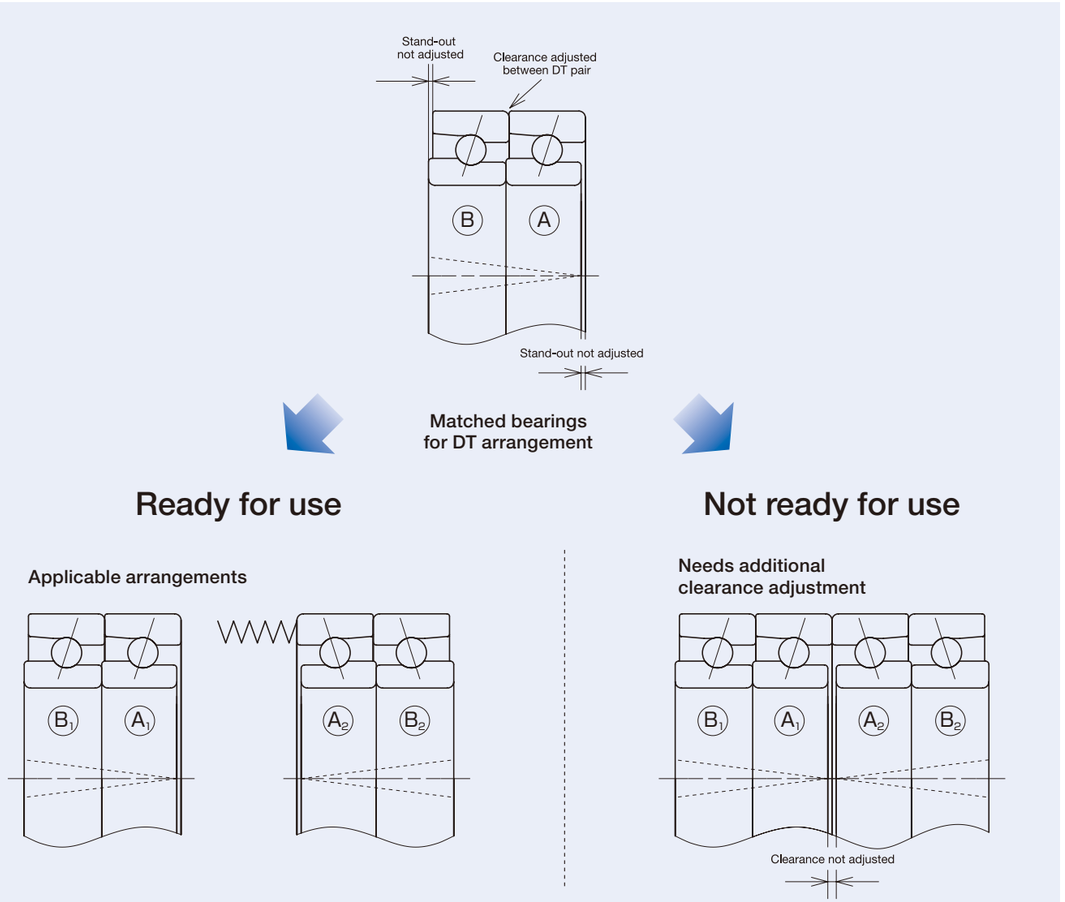
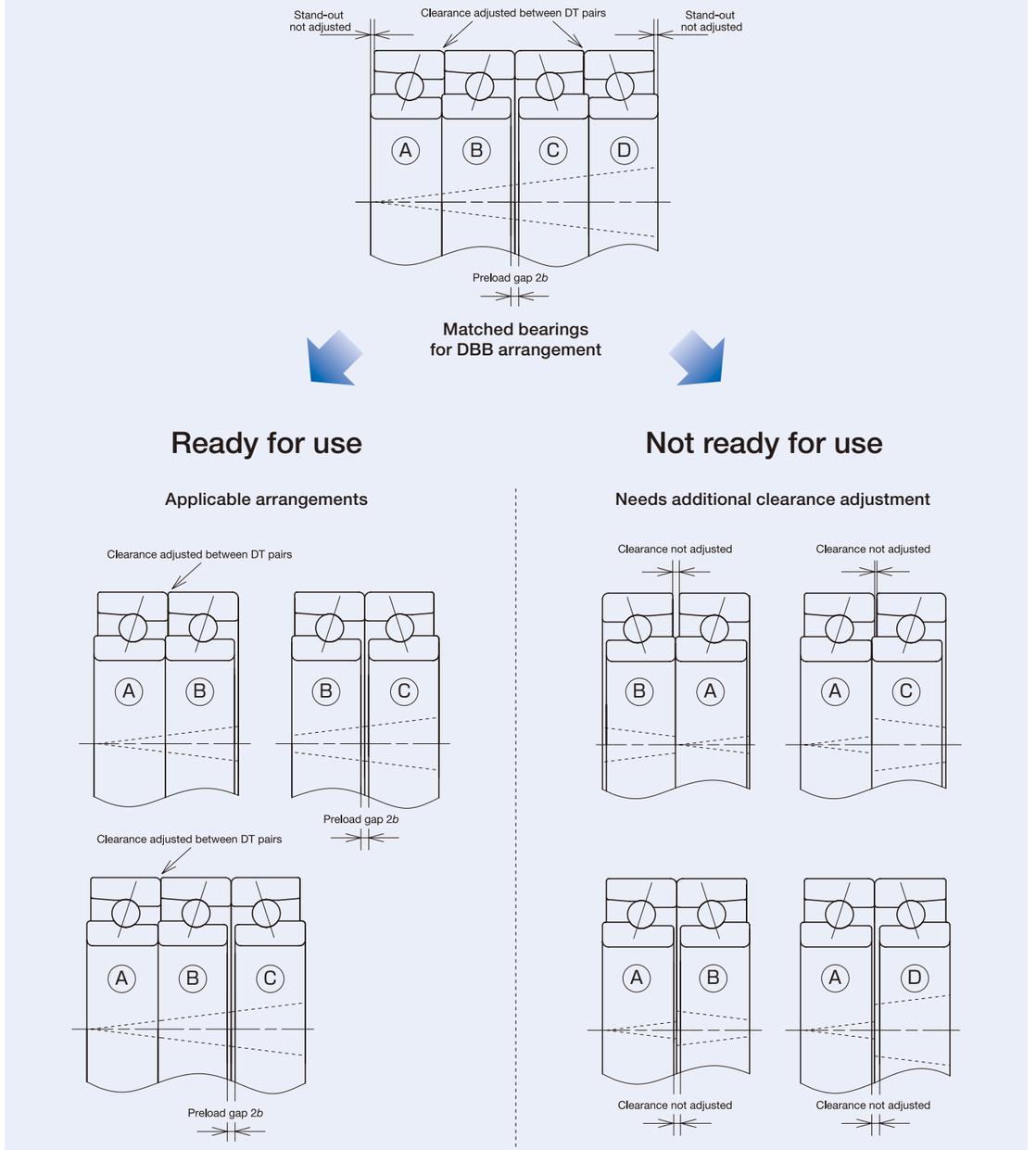


Fig. 3.8 DBB Combination

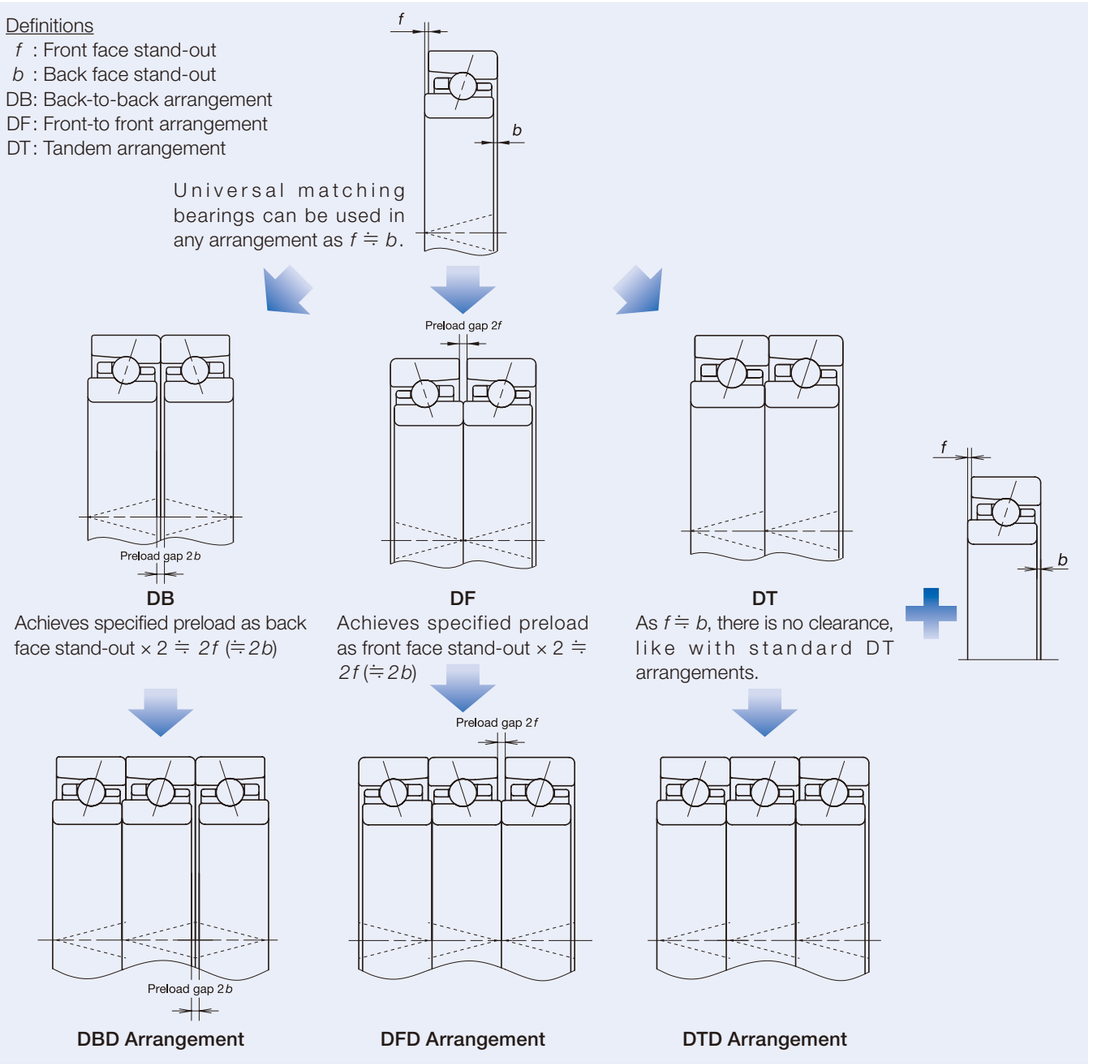


3. Angular Contact Ball Bearing Arrangements

Universal Matching Bearings

NSK manufactures universal matching (universal arrangement) bearings controlled to have the same amount of stand-out (offset) on their front and back faces for the same designation, regardless of how they are combined. Each universal matching bearing comes with a V-shaped mark on the surface of the outer ring to simplify identification of the correct direction when mounting and to ensure that the correct combination is achieved. The V-shaped mark points to the direction of the axial load that the inner ring supports (contact angle).

Fig. 3.9 Universal Arrangements



Types of Universal Matching Bearings

NSK supplies four types of universal matching bearings, as shown in Table 3.2. Variation of bore and outside diameter are controlled for multiple-row universal arrangement bearings.

Table 3.2 Types of Universal Matching Bearings and Features

	SU	DU	DUD	QU
Number of rows	1	2	3	4
Variation of bore and outside diameter	—	Controlled to be within 1/3 of tolerance		

Note: Variation control for P4Y class bearings is the same as for P4 class bearings.

Notes on Using Single Universal (SU) Bearings

- In the case of bearing sets (DB, DU etc.), the variations of bore and outside diameter within each set are controlled. However, when using single-row universal matching bearings (SU), users should check the actual bore/outside diameter variation printed on the box of each bearing and make sure variation of these dimensions is low. Otherwise, variations in housing or shaft fit may occur that result in imbalance of internal load, which may cause damage to the bearings.
- One way of keeping the variation of bore and outside diameter of SU bearings low is to choose the P4Y accuracy class that NSK provides. P4Y-rated bearings have a running accuracy of ISO Class 4. As bore diameter and outside diameter are strictly controlled, these bearings are suitable for custom matching.

Fig. 3.10 Tolerances of P4 and P4Y Accuracy

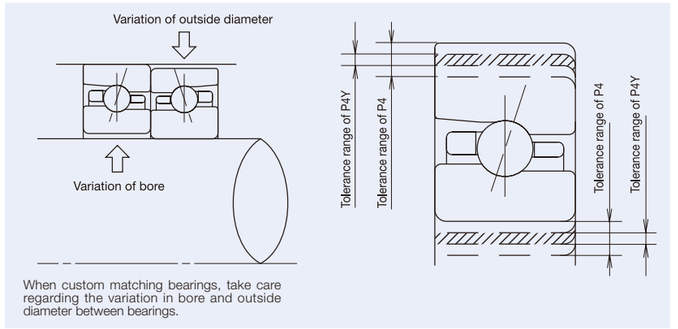
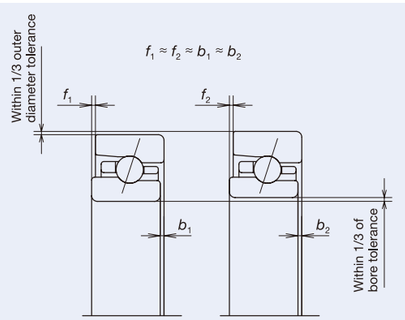
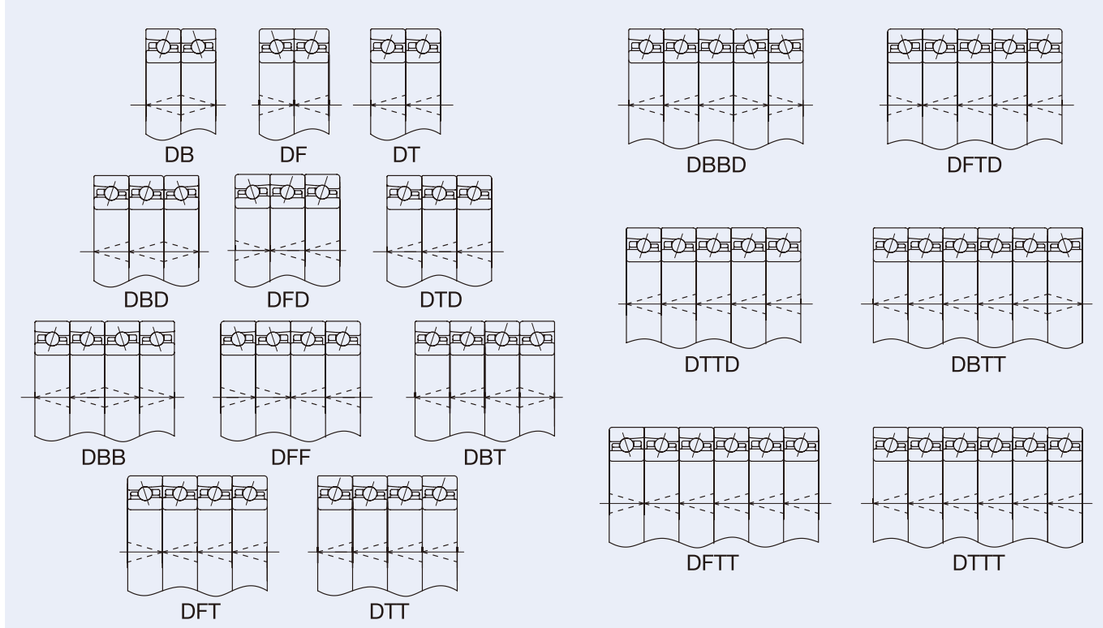


Fig. 3.11 DU Arrangement



Combination Marks and Matching Method for Universal Matching Bearings

Fig. 3.12 Arrangements of Universal Matching Bearings



4. Preload and Rigidity

When considering the rigidity of machine tool spindles, it is possible to think of the bearings as springs. When an axial load is applied to the spindle, axial displacement of the spindle is determined by the axial rigidity of the fixed end bearings.

Radial loads applied to the spindle end will cause radial deflection. While shaft deformation will account for most of this deflection, bearing deflection is also a factor that may not be ignored.

When high radial rigidity is required, cylindrical roller bearings are generally used. In this case, axial loads are usually sustained by angular contact thrust ball bearings. The bigger the contact angle of an angular contact ball bearing, the higher its axial rigidity.

Normally, preload is applied to bearings in order to increase the rigidity of machine tool spindles. However, excessively high preload may cause seizure and flaking. Many users increase rigidity by using a combination of two or more angular contact ball bearings. This is especially true for ball screw support bearings, where high rigidity is required, contact angle is as large as possible, and preload is higher than that for a spindle.

Purpose

The main purposes of applying preload to bearings in machine tool spindles are as follows:

- To improve the running accuracy of the spindle.
- To increase spindle rigidity.
- To minimize noise due to spindle vibration.
- To prevent abnormal wear due to oscillation and vibration
- To prevent sliding between the rolling elements and raceways

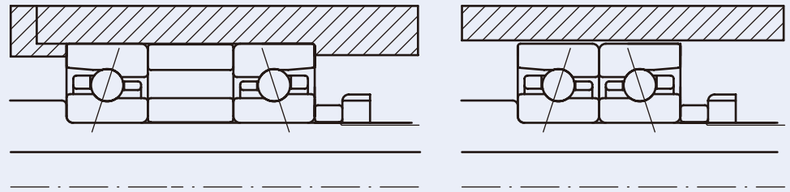
Usually, preload is applied to bearings using two or more bearings in an arrangement, such as angular contact ball bearings or tapered roller bearings. Cylindrical roller bearings can be preloaded by making the radial internal clearance negative.

(1) Position Preload

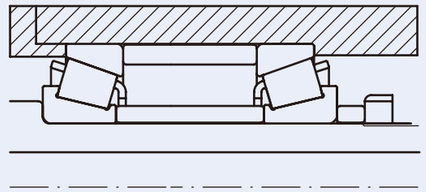
A position preload is achieved by fixing two axially opposed bearings so that their position relative to one another remains unchanged during operation. In practice, the following three methods are generally used to obtain position preload.

1. By using a spacer or shim of proper size to obtain the required spacing and preload, as shown in the left of Fig 4.1 (a) and in Fig. 4.1 (b) and (c).
2. By installing a bearing set with previously adjusted stand-out dimensions and axial clearance, as shown in the right of Fig. 4.1 (b).
3. By utilizing bolts or nuts to allow adjustment of axial preload, as shown in Fig. 4.1(d). In this case, starting torque should be measured to verify proper preload. However, this method cannot be recommended for high precision machine tool spindles due to difficulty in verifying the proper preload, thus risking vertical displacement (tilting) of the bearing.

Fig 4.1 Position Preload Examples

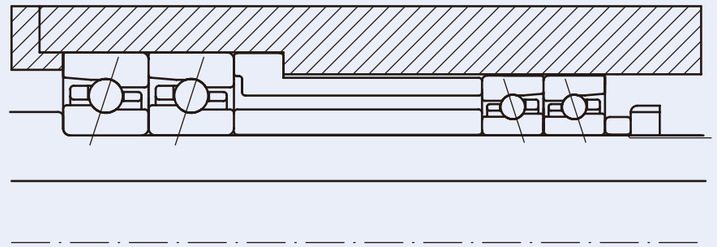


(a) Angular contact ball bearings in position preload

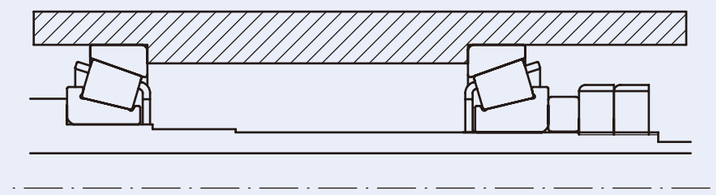


(b) Tapered roller bearings in position preload

Note: The cages of tapered roller bearings usually protrude from the bearing's end faces. If looking to reduce the bearing span, please bear in mind the required minimum spacer width.



(c) Bearing combinations of differently sized bearings in position preload



(d) Position preload without spacers

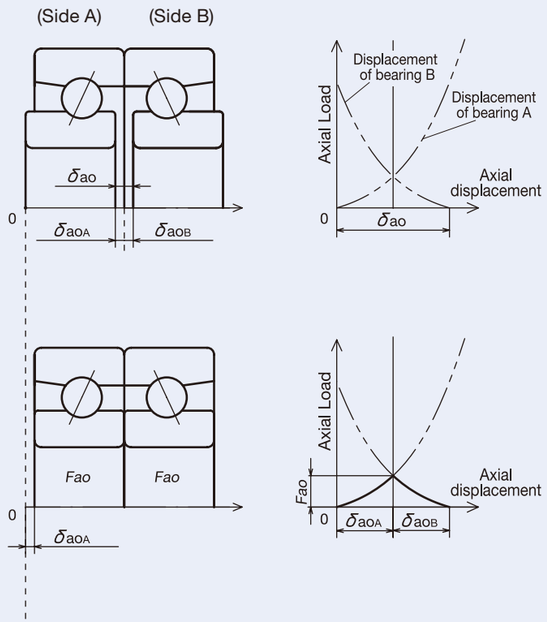
4. Preload and Rigidity

Axial Load and Displacement in Position Preload (DB Arrangement)

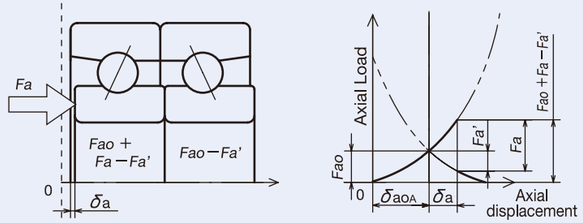
Fig 4.2 shows how preload and axial load interact in a back-to-back (DB) bearing arrangement.

Fig 4.2 Axial Displacement in Position Preload (DB Arrangement)

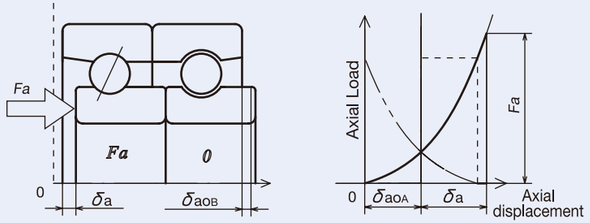
(1) When the inner rings are fixed axially, bearings A and B are displaced by δ_{a0A} and δ_{a0B}, respectively. Thus, the axial clearance δ_{a0} between the inner rings is eliminated. In this condition, preload F_{a0} is imposed on each bearing.



(2) When axial load F_a is applied to the preloaded bearing arrangement as shown, the inner ring of bearing A will be displaced by δ_a. A balance between the loads acting on each bearing will be achieved where the amount of displacement of bearing A equals that of bearing B.



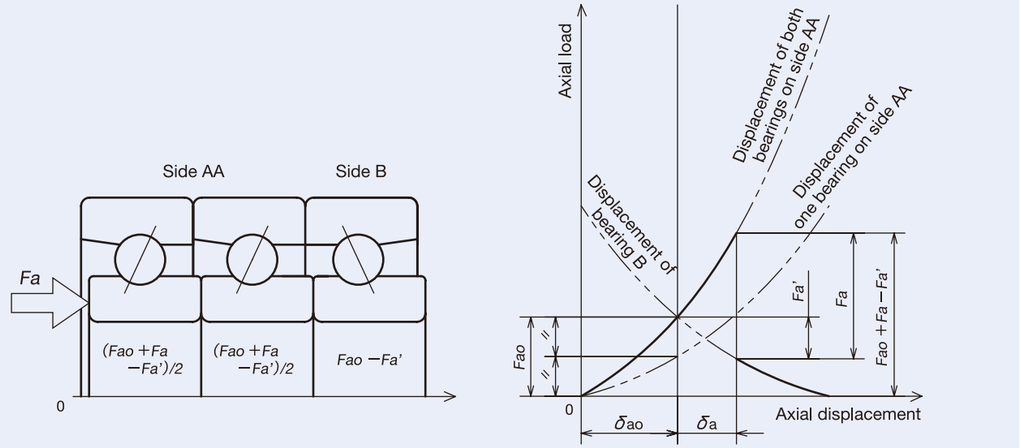
(3) When a large axial load is applied so that the displacement of the inner ring of bearing B exceeds δ_{a0B}, bearing B will no longer sustain axial load. Axial load will be sustained by bearing A only. Therefore, axial rigidity will decrease.



Axial Load and Displacement in Position Preload (DBD Arrangement)

In DBD arrangements, one direction of axial load is divided equally between two bearings. Apart from that, the basic concept is the same as with DB arrangements (shown in Fig 4.2.)

Fig 4.3 Axial Displacement in Position Preload (DBD Arrangement)



Preload Loss

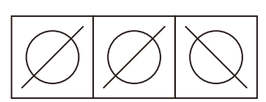
As Fig 4.2 (3) shows, when a preloaded bearing arrangement receives a large axial load, the bearing on the opposite side becomes unloaded. This can cause sliding between the rolling elements and raceways, and may cause excessive wear or bearing damage under high-speed rotation.

When angular contact ball bearings of the same type are used in an arrangement, preload loss will occur when a load 3 times the preload of a single DB or DF arrangement is applied. For tapered roller bearings, this loss-causing load is 2 times the preload of a single DB or DF arrangement. Preload loss factors for angular contact ball bearings and example calculations are shown below.

Table 4.1 Approximate Values for Load Causing Preload Loss

Combination	Arrangement Code	Direction	Preload Loss (multiply by preload of DB/DF arrangements)
	DB	→	3
		←	3
	DBD	→	6
		←	3
	DBB	→	6
		←	6
	DBT	→	9
		←	3
	DBBD	→	9
		←	6

Example calculations of preload loss for 7014A5TYNDBD with L preload:
Preload for DB/DF arrangements: 500 N
(See Page 79.)



Direction	Load Causing Preload Loss
→	500×6 = 3 000N
←	500×3 = 1 500N

4. Preload and Rigidity

Standard Clearance

NSK supplies matched angular contact ball bearings in four standard preload classes:

EL: Extra light preload L: Light preload
M: Medium preload H: Heavy preload

In addition, two special clearances are available for matched angular contact ball bearings and for tapered roller bearings:

CA: Special Measured (+) Axial Clearance
CP: Special Measured (–) Axial Clearance

Preload Adjustment with Spacers

The measured axial clearance of each bearing is listed in the bearing tables in Part 4 of this catalog. The listed values refer to DB and DF arrangements and do not include the influence of shaft and housing fit or nut clamping force.

When using spacers to adjust preload (for example: L→ EL, L→ M etc.), adjust the difference between measured axial clearance and target clearance with a spacer. With DB arrangements, reduce the width of the inner ring spacer to increase preload, and reduce the width of the outer ring spacer to decrease preload.

When using universal matching bearings, the axial clearance between the bearings equals the sum of the back face stand-out for each bearing in DB arrangements or the sum of front face stand-out for DF arrangements. Adjust to the required axial clearance using spacers.

Table 4.2 shows measuring loads for measuring axial clearance. The values in the table indicate the amount of load necessary to obtain stable measurements.

Fig 4.4 Special Clearances

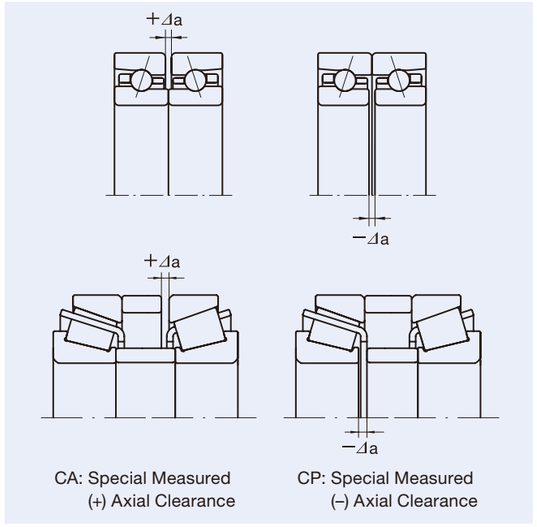
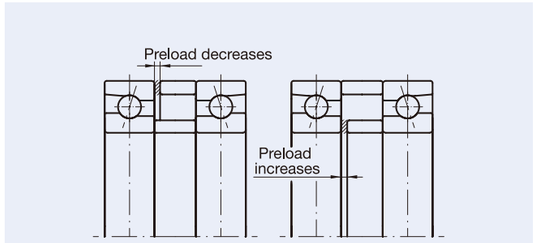


Fig 4.5 Adjustment of Axial Clearance with Spacers



Example adjustment by spacers:
100BER10ETDBELP4, change preload from EL → L
EL: 0
L: -12
Adjust so that the inner ring spacer is 12 um shorter than the outer ring spacer

Table 4.2 Measuring Loads for Axial Clearance Measurements

Nominal Bearing Outside Diameter *D* (mm)		Measuring Load (N)
over	incl.	
10 (1)	50	24.5
50	120	49
120	200	98
200	400	196

(1) Includes 10 mm.

Measuring Axial Clearance of Angular Contact Ball Bearing Arrangements

In the case of a DB arrangement, place the bearing without the inner ring spacer on the cradle as described in the figure below; then apply the measuring load to the inner ring. After the bearing is sufficiently stabilized, set the dial gauge to zero.

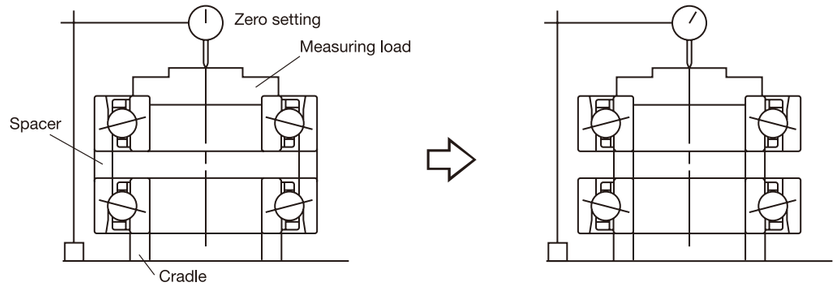
Next, after removing the outer ring spacer, place the bearing with the inner ring spacer on the cradle and apply measuring load in the same way as described above. The reading on the dial gauge now indicates the axial clearance. A positive reading signifies a CA clearance (positive clearance in axial direction). A negative reading signifies a CP clearance, which generates preload.

The axial clearance of a DF arrangement is measured in the same manner as with a DB arrangement, although the positions of spacers and cradles are different.

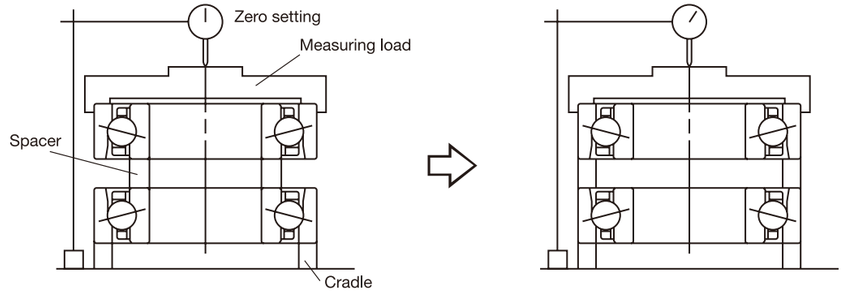
With a DT arrangement, the sum of front face stand-out and back face stand-out of the combined bearings at the mating surfaces should be zero.

For single-row bearings, axial clearance of a double-row arrangement can be obtained by adding the measured values of each bearing.

DB Arrangement

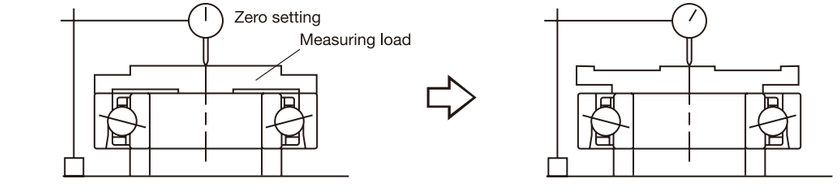


DF Arrangement

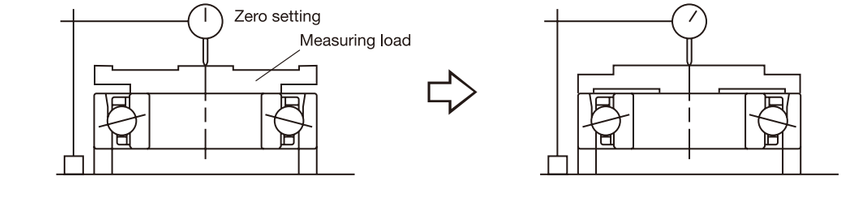


Single Row

Back face stand-out



Front face stand-out



4. Preload and Rigidity

Measuring Axial Clearance of Tapered Roller Bearing Arrangements (Stand-Out Measuring Method)

In the case of a DB arrangement, place a single-row bearing on the cradle with the back face of the inner ring facing down. Turn the outer ring (at least 10 turns) to stabilize the rollers. Then, measure inner ring width and assembled bearing width. Next, measure the inner ring width and assembled bearing width of the second bearing in the same way. Finally, measure widths K and L of the spacers.

Inserting these measurements into the equation below will obtain axial clearance Δ_a.

$$\Delta_a = (L\text{-}K)\text{-}(f_A+f_B) = (L\text{-}K)\text{-}[(T_A\text{-}B_A)+(T_B\text{-}B_B)]$$

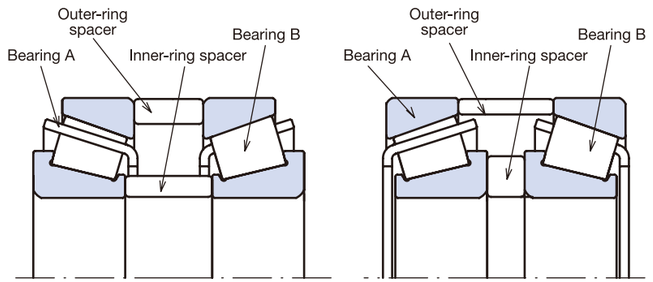
$\Delta_a < 0$ signifies a preload.

In the case of a DF arrangement, measure the outer ring instead of the inner ring. This gives the following equation for axial clearance:

$$\Delta_a = (K\text{-}L)\text{-}(b_A+b_B) = (K\text{-}L)\text{-}[(T_A\text{-}C_A)+(T_B\text{-}C_B)]$$

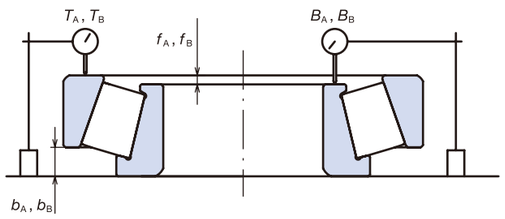
Tapered rolling bearings in DF arrangements are often operated without an inner-ring spacer, so that the inner rings of both bearings touch directly. In this case, $L = 0$.

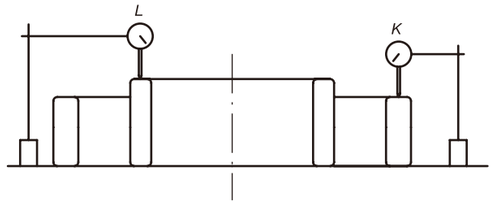
Since the cages protrude, make sure to place the bearings with the correct side onto the cradle and take care to avoid mechanical interaction with other parts so as to prevent cage deformation.

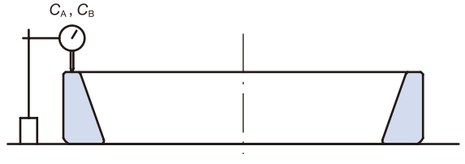


DB Arrangement

DF Arrangement







f : Front face stand-out
b : Back face stand-out
T : Assembled bearing width
C : Outer ring width
B : Inner ring width
K : Outer ring spacer width
L : Inner ring spacer width

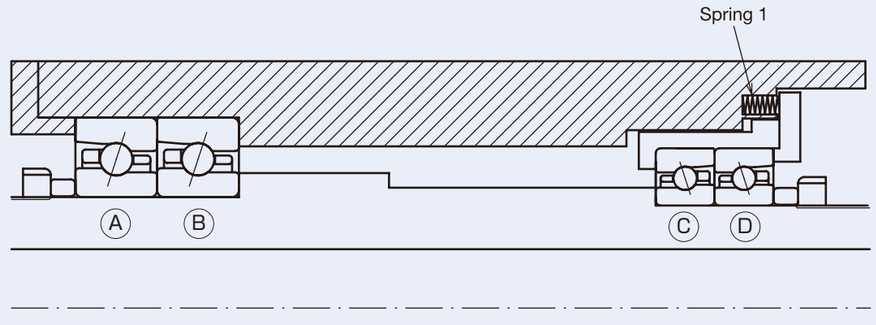
(2) Constant-Pressure Preload

Constant-pressure preload is achieved using a coil or disc spring. Even if the relative position of the bearings changes during operation, the magnitude of the preload remains relatively constant throughout various operation conditions. Therefore, constant-pressure preload is used for combinations of angular contact ball bearings in high-speed applications.

One example is shown in Fig. 4.6. Since spring rigidity is, as a rule, small compared with bearing rigidity, the change in spring preload due to bearing displacement is negligible. Thus, the preload applied to bearings C, D, and E in the figure below will be approximately the same in both vertical shaft and horizontal shaft configurations.

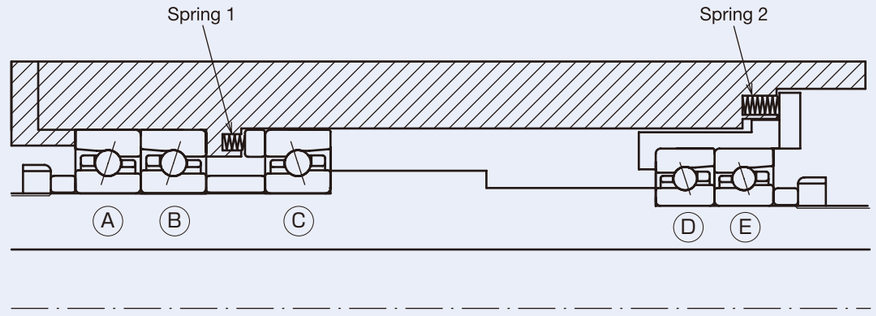
However, in addition to the forces of each spring, bearings A and B will also be affected by the shaft's own weight in vertical operations. Axial loads exerted from the free end towards the fixed end (from right to left in the figure below) will reduce preload of bearings A and B. It is necessary to consider these effects when setting preload for bearings A and B so that they will not become unloaded.

Fig. 4.6 Distribution of Preload in Constant-Pressure Preload



	Preload (N)			
	Bearing A	Bearing B	Bearing C	Bearing D
Horizontal shaft	250	250	250	250
Vertical shaft	200	200	250	250

(a) Shaft weight = 100 N, Spring 1 = 500 N



	Preload (N)				
	Bearing A	Bearing B	Bearing C	Bearing D	Bearing E
Horizontal shaft	250	250	300	100	100
Vertical shaft	200	200	300	100	100

(b) Shaft weight = 100 N, Spring 1 = 300 N, Spring 2 = 200 N

4. Preload and Rigidity

Preload and Axial Rigidity

Generally, when an axial load F_a is applied to a bearing, it is possible to calculate the axial displacement δ_a of the bearing using the following formulas:

Angular contact ball bearings: $\delta_a = cF_a^{2/3}$

Tapered roller bearings: $\delta_a = cF_a^{0.9}$

where c is a constant that depends on the bearing's internal design.

As such, the effect of preload to reduce displacement is stronger with ball bearings.

The formulas above to elastic displacement between the rolling elements (balls, tapered rollers) and the inner and outer rings. Actual axial displacement will also depend on the material and thickness of the shaft and housing as well as shaft fit and housing fit. For axial displacement that takes these mounting factors into account, please contact NSK.

Fig. 4.7 Axial Load and Axial Displacement in Single-Row Bearings

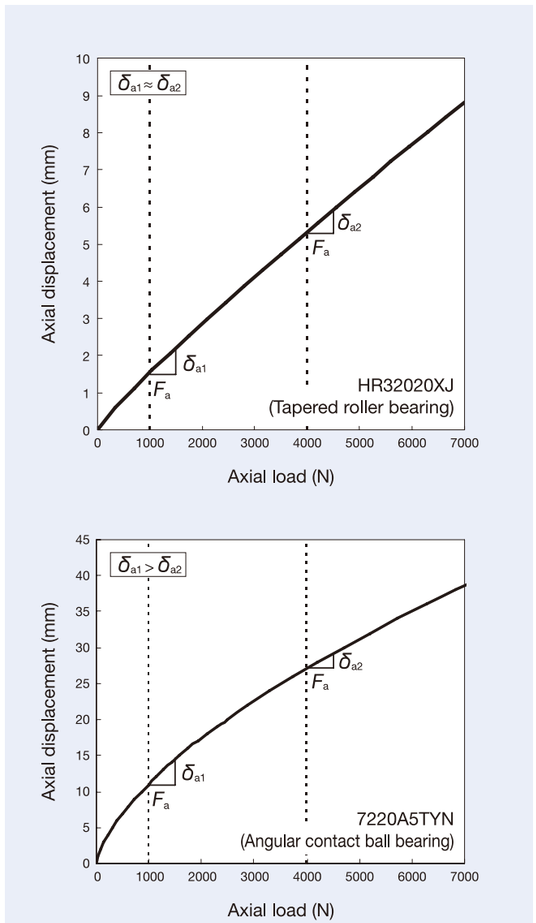
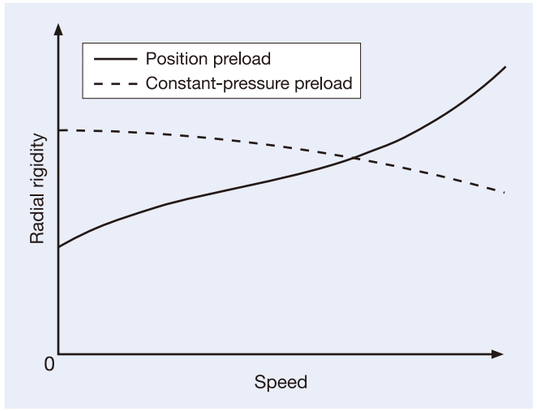


Fig. 4.8 Comparison of Rigidity by Preload Method



Comparison of Preload Methods

Position preload and constant-pressure preload can be compared as follows:

(1) Position preload provides greater bearing rigidity than constant-pressure preload at the same amount of preload. In other words, deflection due to external loads is less for bearings with position preload.

(2) Under position preload, the preload varies during operation depending on such factors as differences in axial expansion due to temperature differences between the shaft and the housing, differences in radial expansion due to temperature differences between the inner and outer rings, and the effect of centrifugal force acting on the rolling elements. Under constant-pressure preload, the variation of spring load due to shaft expansion and contraction, etc. is minimal, so its effect on preload is negligible.

With position preload, bearing rigidity will generally increase with spindle speed because preload increases during operation. In contrast, the preload level with constant-pressure preload will remain constant. However, the contact angle will change due to centrifugal force acting on the balls (Fig. 1.7 on page 197) and result in a decrease of rigidity (Fig. 4.8).

Consequently, position preloads are generally preferred where increasing rigidity is desired, while constant-pressure preloads are more suitable for high-speed applications, applications where axial vibration needs to be prevented, use with thrust bearings on horizontal shafts, etc.

Preload Amount

A larger preload results in higher rigidity. However, increasing preload reduces fatigue life and increases heat generation. In extreme cases, excessive preload may result in abnormal wear or even seizure.

Therefore, the amount of preload needs to be carefully selected taking into consideration the type of application and operating conditions, to avoid more preload than necessary.

High-Speed Spindles and Preload

When bearings are operated at high speed, internal load increases as a result of the expansion of the inner ring due to centrifugal force, the effect centrifugal force has on balls, and the temperature difference between inner and outer rings. Then, surface contact pressure between the balls and the raceways of the inner and outer ring will increase. For bearings with a contact angle, such as angular contact ball bearings, pure rolling motion will be accompanied by sliding as a result of spin moments and gyroscopic moments on the balls. Sliding increases as bearing speed increases. As a result, the intensity of heat generated in contact areas increases and the viscosity of the lubricating oil decreases. In some cases, a breakdown of the oil film occurs, which results in bearing seizure. This means that even if contact surface pressure were to remain unchanged from low-speed operation to high-speed operation, heat generation would still increase as speed increases due to sliding.

At NSK, we calculate contact surface pressure and slip rate generated during high-speed operation. Based on our own abundant empirical test data as well as market results, we have established limiting factors according to the lubricating method and rotating speed; this is the foundation on which we determine preload. For applications exceeding 800,000 $d_m n$, please contact NSK.

For high-speed bearings in constant-pressure preload, a medium preload (M) per row is commonly chosen as a guideline value and then adjusted according to the required rigidity and temperature rise in high-speed operation.

Preload for Thrust Bearings

Thrust bearings generally require an axial load in order to prevent slipping between the rolling elements and the washers.

The minimum axial load required to prevent rolling element slippage can be calculated using the equations below. In cases where the shaft's own weight is insufficient, preload must be added with a spring or similar.

For minimum axial load in the case of horizontal shafts, please contact NSK.

For thrust ball bearings: $F_{a\,min} = Mn^2$

where $F_{a\,min}$: Minimum axial load (kN)
M : Minimum load factor (listed in bearing tables)
n : Maximum speed during operation (min^{-1})

For thrust cylindrical roller bearings: $F_{a\,min} = 0.0005C_{0a} + Mn^2$

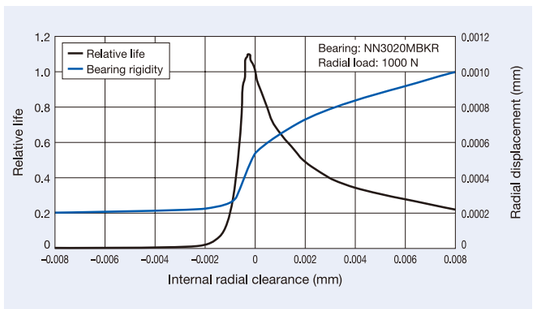
where $F_{a\,min}$: Minimum axial load (kN)
M : Minimum load factor (listed in bearing tables)
C_{0a} : Basic static axial load rating (kN)
n : Maximum speed during operation (min^{-1})

Internal Clearance in Cylindrical Roller Bearings

In order for machine tool spindles to have high running accuracy and rigidity, bearings are used with minimum internal clearance or preload after mounting. When using cylindrical roller bearings, bearings with tapered bores are usually used to allow easier adjustment of internal clearance. In general, cylindrical roller bearings for the front end (fixed end) of the spindle are adjusted upon mounting to have a preload during operation. Bearings for the rear end (free end) are adjusted upon mounting to produce a slight clearance during operation. The amount of internal radial clearance after mounting is determined based on several factors such as speed, load, lubricating method, bearing size, required rigidity, service life, etc.

Fig. 4.9 shows the relationship between internal radial clearance and rolling fatigue life, as well as the relationship between internal radial clearance and radial elastic displacement for a cylindrical roller bearing (NN3020, bore diameter: 100 mm, outside diameter: 150 mm, width: 37 mm).

Fig. 4.9 Influence of Internal Radial Clearance on Rolling Fatigue Life and Rigidity in a Double-Row Cylindrical Roller Bearing



5. Limiting Speeds

Limiting Speeds

Limiting speeds listed in the bearing tables are reference values that assume:

- Use of a single-bearing with appropriate preload applied by spring
- Spindle unbalance is controlled
- Good heat dissipation
- Good lubrication
- Inner ring rotation

These values are not guaranteed for continuous operation over a prolonged period. Especially in the case of grease lubrication, it is recommended to choose suitable speeds taking grease life into consideration.

Limiting speeds with grease assume a suitable amount of high-quality grease is applied appropriately (see p. 256 for details) with sufficient run-in (p. 268).

Limiting speeds for oil lubrication are based on oil-air (or oil-mist) lubrication. It is possible to achieve higher speeds when oil serves to efficiently remove heat, as with jet lubrication.

In order to achieve high-speed operation, it is necessary to control bearing load (including internal load), heat generation and temperature increase. Note that the following factors also exert an influence on limiting speeds:

(1) Preload Method

Under position preload, internal preload increases when speed is increased. As a general rule, limiting speed with position preload will be lower than with constant-pressure preload.

(2) Bearing Positioning and Arrangement

The number of rows in a bearing arrangement affects load distribution. The limiting speed will be determined by the bearing operating under the most stress within an arrangement. In addition, even under the same preload, heat dissipation will be affected by the presence or absence of spacers and by spacer length.

(3) Bearing Load

In general, it is difficult to support high speeds and heavy loads at once. Permissible loads are also affected by speed.

(4) Drive Method

Spindle drive systems can be classified broadly into the following three types, each with their own negative effects on limiting speed:

(a) Coupled drives

If the coupling is significantly misaligned, the shaft will vibrate, resulting in lower limiting speeds.

(b) Built-in motor drives

With higher heat generation within the spindle, bearings near the motor can be easily damaged by heat. When jacket cooling is used, the temperature difference between the inner and outer ring will likely increase, resulting in higher internal load under position preload and lower limiting speeds (see Fig. 5.2).

(c) Gear/belt drives

The driving force itself must also be accounted for, with particular care necessary for high-torue and high-speed spindles.

(5) Mounting Conditions

In position preload, the shaft fit will affect preload. Spacer length and tightening force also affect bearing clearance, thereby changing preload.

Housing fit conditions will greatly influence preload, particularly during operation. Too little clearance between the bearing and housing will lead to an interference fit due to differences in thermal expansion; this increases internal preload (Fig. 5.3).

If cylindrical roller bearings are mounted with excessive negative clearance, the limiting speed will decline due to rising heat generation (page 46).

Table 5.1 Speed Factors for Position Preload

	Arrangement	Preload After Mounting			
		EL	L	M	H
DB		0.85	0.80	0.65	0.55
DBB		0.80	0.75	0.60	0.45
DBD		0.75	0.70	0.55	0.40
DBBD		0.75	0.70	0.55	0.40

Speed Factors for Position Preload

The limiting speed of an arrangement of angular contact ball bearings is calculated by multiplying the limiting speed of a single bearing in the combination by the appropriate adjustment factor listed in Table 5.1. The preload classification in the table refers to the preload after the bearings have been attached to the spindle (i.e. preload after mounting).

Usually, preload after mounting is affected by shaft fit and spacer deformation and may exceed that resulting from initial axial clearance. For example, an EL preloaded bearing mounted with high interference may have a preload after mounting that is equivalent to M preload. It will be necessary to adjust axial clearance using spacers according to the conditions in which the bearings are used.

In order to maintain stable operation over time, it is advisable to reduce the values given in Table 5.1 further by a safety factor of 0.75.

Calculation example:

Target value for 70BNR10HTDB: EL preload after mounting, grease lubrication: (limiting speed) 20000 × (EL) 0.85 × (safety factor) 0.75 = 12 750 min^{-1}

Factors Affecting Preload

Fig. 5.1 Preload Setting Considerations

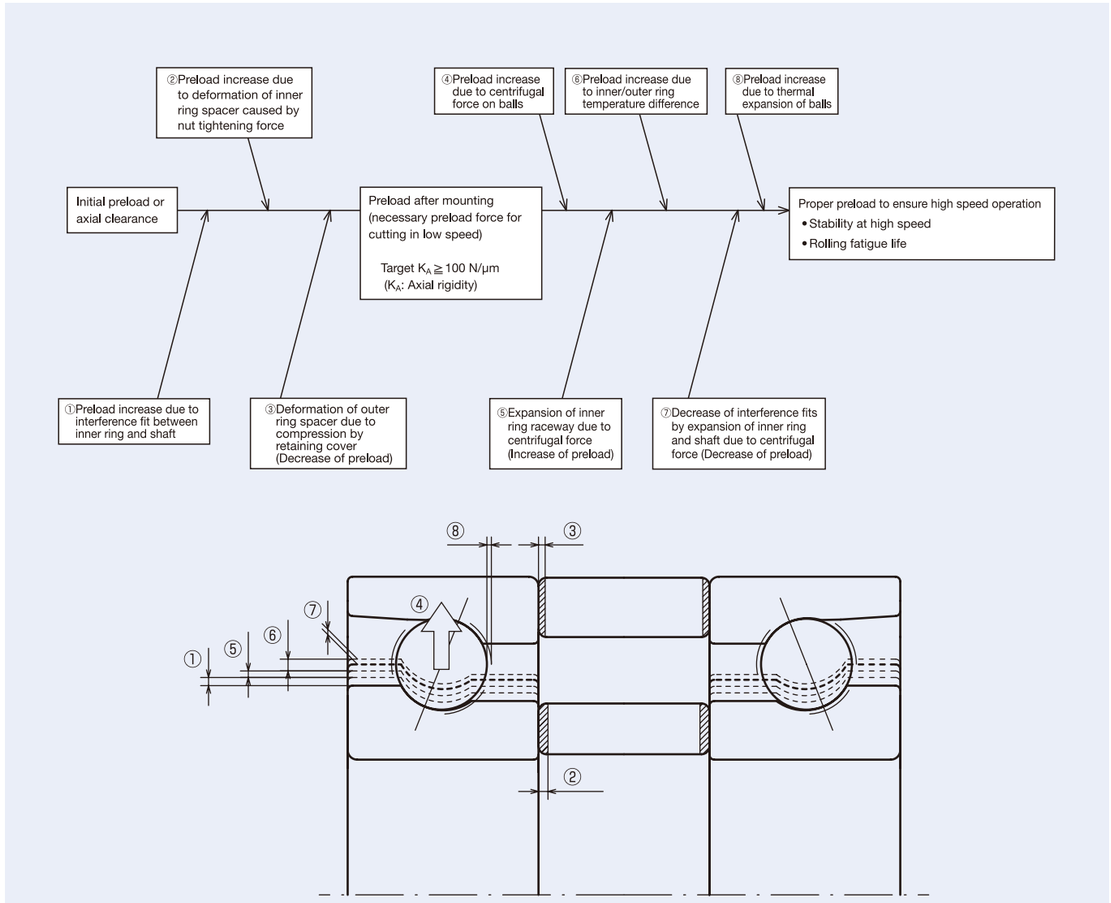


Fig. 5.2 Influence of Jacket Cooling on Limiting Speed

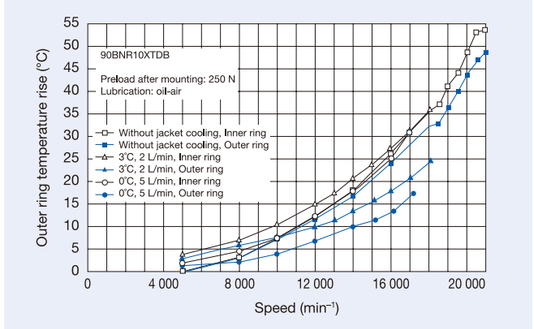
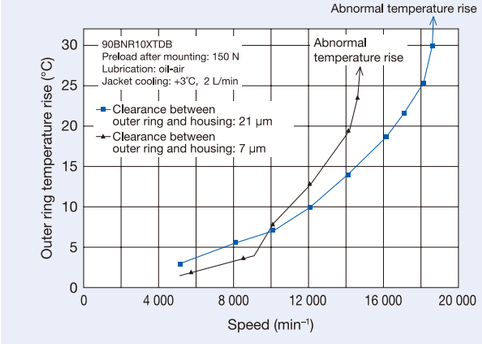


Fig. 5.3 Influence of Clearance Between the Bearing & Housing on Limiting Speed



6. Lubrication

Purposes of Lubrication

The main purpose of lubrication is to reduce friction and wear inside the bearing and thereby prevent seizure. The effects of lubrication may be briefly explained as follows:

(1) Reduction of Friction and Wear

Direct metallic contact between the bearing rings, rolling elements, and cage, which are the basic parts of a bearing, is prevented by an oil film which reduces the friction and wear in contact areas.

(2) Extension of Fatigue Life

Appropriate lubrication of the rolling contact surfaces extends the rolling fatigue life of bearings. This requires suitable viscosity of the lubricant. If the viscosity of the oil is so low that oil-film thickness is insufficient, rolling fatigue life will be shortened.

(3) Dissipation of Heat

Circulating lubrication may be employed to remove frictional heat or heat transferred to the bearing from outside. This prevents the bearing from overheating and the lubricant itself from deteriorating.

(4) Others

Adequate lubrication also helps to prevent foreign material from entering the bearings and guards against corrosion and rusting.

Lubricating Methods

For machine tool spindles in which high accuracy is important, it is necessary to prevent excessive temperature rise of the spindle to reduce thermal deformation.

Bearing heat generation is divided into a load term determined by the bearing type and load, and a speed term determined by the lubricating method and speed.

While the speed term has a greater influence, the load term cannot be ignored when a lubrication method with a small speed term is used. Therefore, it is important to select bearings with low heat generation (load term) and the appropriate type of lubrication (speed term).

Regarding heat generation, both the lubricating method and the quantity of lubricant have important effects. Lubrication using a small amount of grease is common since this method is economical, maintenance-free, and there is little heat generation. Oil-air lubrication minimizes the quantity of oil used and maintains a constant low temperature at high speeds. Refer to Table 6.1 for a comparison of lubricating methods.

Table 6.1 Comparison of Lubricating Methods

Lubricating Method	Advantages	Disadvantages
Grease Lubrication	☐ Cost is low. ☐ Limitation of temperature rise possible. ☐ Maintenance free.	☐ If packed grease deteriorates, seizure may occur. ☐ May allow entry of dust or cutting fluid.
Oil-Mist Lubrication	☐ Since new oil is always fed, no risk of oil deterioration. ☐ Dust and cutting fluid cannot easily enter.	☐ Pollution of environment. ☐ Oil supply quantity varies depending on the oil viscosity and temperature, so control of a small flow rate is difficult. ☐ It is difficult to confirm that oil is actually fed.
Oil-Air Lubrication	☐ Since oil quantity control is possible, the optimum quantity of oil is fed and heat generation is low. ☐ Besides little heat generation, the air provides a cooling effect that keeps temperature rise low. ☐ Since new oil is always fed, no risk of oil deterioration. ☐ Dust/cutting fluid cannot easily enter. ☐ Environmental pollution is slight.	☐ Cost is rather high. ☐ Confirmation of whether oil is actually fed to bearing is difficult.
Jet Lubrication	☐ Since the oil flow rate is high, dust and cutting fluid cannot enter and seizure hardly ever occurs. ☐ Because of cooling by oil, the bearing temperature can be controlled to some degree.	☐ Frictional loss is high. ☐ Since oil leaks, it is difficult to use for vertical spindles. ☐ Cost is high.

The relation between oil quantity, heat generation (frictional loss), and temperature rise is as shown in Fig. 6.1. In Zone A, oil is employed in the minimal quantity necessary. This minimizes agitation resistance and the amount of associated heat generation. However, caution is required since oil quantities lower than Zone A pose the risk of insufficient lubrication causing heat increase and bearing failure.

In Zone B, the oil quantity is higher which means more heat is generated due to oil agitation. However, once a certain quantity is exceeded, temperature decreases as the oil transports some of the heat away from the bearing.

In this case, the amount of oil needed depends on the oil drain of the machine, allowable temperature, heat dissipation, and the heat generation characteristics of the oil due to agitation resistance. The adequate oil amount is thus often determined empirically.

Dynamic friction torque of bearing (heat generation)

$M = Ml + Mv$

- Load term (Determined by bearing type and load)

  $Ml = f_1 F d_m$

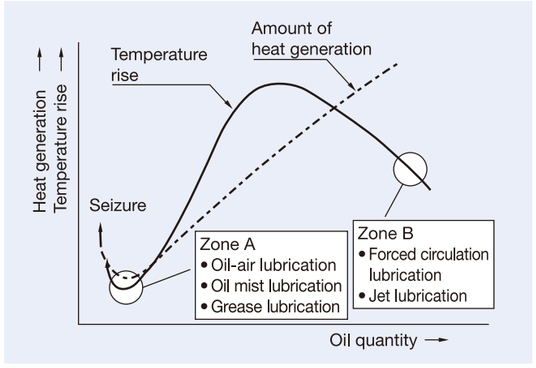
  where f_1 : Coefficient determined by bearing type and load
  F : Load
  d_m : Pitch circle diameter of rolling element

- Speed term (Determined by oil viscosity, oil amount, and speed)

  $Mv = f_0 (\nu_0 n)^{2/3} dm^3$

  where f_0 : Coefficient determined by bearing and lubricating method
  ν_0 : Kinematic viscosity of oil
  n : Speed

Fig. 6.1 Oil Quantity and Temperature Rise



Grease Lubrication

(1) Recommended Greases

For grease lubrication of bearings in high-speed machine tool spindles that require low temperature rise and long life, a grease with Grade 2 or 3 consistency is recommended.

Grease life depends to a large degree upon operating temperature. Therefore, it is necessary to keep the temperature of the bearing (including atmospheric temperature) cool in order to extend grease life.

(2) Grease Quantities for High-Speed Spindle Bearings

To operate bearings at high speed with grease lubrication, the recommended grease fill is 10% to 30% of the internal space. If too much grease is packed, abnormal heat generation occurs, especially during run-in; this may cause the grease to deteriorate. To avoid such a risk, it is necessary to run in spindles for a sufficient time. Based on accumulated experience, NSK determines packing quantities that allow easy run-in and sufficient lubrication. These recommended grease quantities are listed on page 257.

(3) Grease Quantities for Ball Screw Support Bearings

As bearings for ball screw support are usually operated at slow speed, under high load, and in intermittent operation, we recommend a grease quantity of 30% to 55% of the internal space for increased reliability. Recommended grease quantities for each bearing are listed in the bearing tables for ball screw support bearings.

6. Lubrication

Oil Lubrication

When lubricating bearings with oil, highly purified mineral and synthetic oils that perform well under high loads and offer good anti-oxidative and anti-corrosive properties are used.

The most important factor to consider when choosing lubricating oil is the viscosity it will exhibit at operating temperature. If viscosity is too low, the oil film will be insufficient, which may cause abnormal wear and seizure. If viscosity is too high, the agitation resistance of the oil will generate more heat and lead to frictional loss. Oil film formation is also influenced by operating speed and load conditions.

Generally, the higher the operating speed, the lower the viscosity of the oil used. Conversely, the higher the bearing load, or the larger the size of the bearing, the higher the viscosity of the oil used. Table 6.2 shows guideline values for oil viscosity around bearings at temperatures under normal operating conditions.

To aid your selection of lubricating oil, Fig. 6.2 shows the relationship between oil temperature and viscosity.

(1) Oil-Mist and Oil-Air Lubrication (Minimal Oil Quantity Lubrication)

Oil-mist lubrication sprays oil by turning it into a mist using compressed air.

Oil-air lubrication meanwhile feeds oil continuously by injecting oil into a compressed air stream with a mixing valve that intermittently discharges the minimum quantity of oil using a constant-quantity piston. Fig. 6.3 shows the recommended oil quantity for these minimal oil quantity lubrication methods for one bearing. In the case of oil mist lubrication, it is necessary to adjust the oil quantity to accommodate branches in distribution tubing and leakage from the gaps around the spacers. Additionally, note that high-speed operations require oil supply adjustments as high speeds typically increase the quantity of oil needed for proper lubrication.

For the position of the spray nozzle, please refer to pages 240-241.

Example oil-air lubrication system:

Air supply
- Clean and dry compressed air
- Dew point: 3°C or lower
- Air pressure: 0.2 to 0.5 MPa (0.4 to 0.45 recommended)

Lubricating oil
- Clean high-speed spindle oil or turbine oil (With oil line filter, if needed)
- Viscosity: 22 to 68 cSt

Distributing tube for oil-air
- Length of distribution tubing: 1.5 to 5 m (3.5 to 5 m recommended)
- Inner diameter of distribution tubing: 2 to 2.5 mm (transparent urethane tube with 4 mm outside diameter, etc.)

Please refer to the instruction manual of your lubrication device manufacturer for details.

Table 6.2 Required Viscosity of Lubricating Oil by Bearing Type

Bearing Type	Kinematic Viscosity During Operation
Ball bearings, cylindrical roller bearings	13 cSt or higher
Tapered roller bearings	20 cSt or higher

Fig. 6.2 Relationship of Viscosity and Temperature for Lubricating Oil

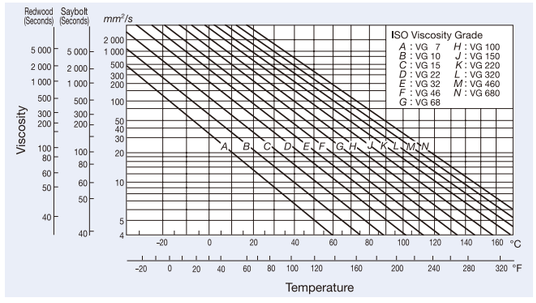


Fig. 6.3 Recommended Oil Quantity by Bearing Bore Size (Minimal Oil Quantity Lubrication)

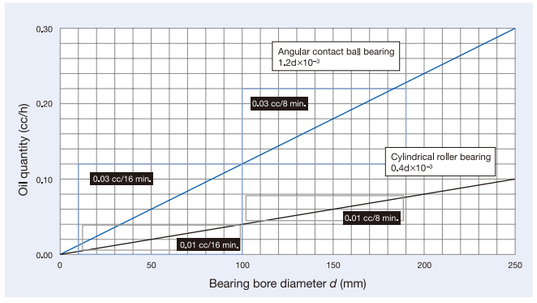
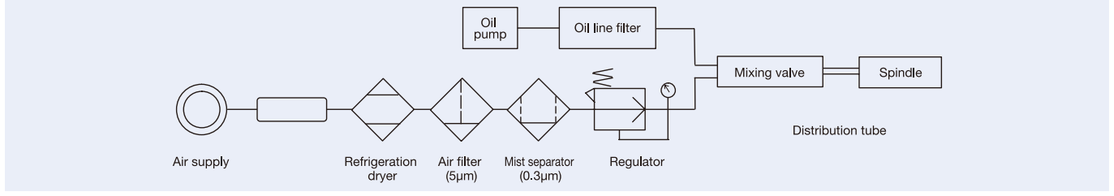


Fig. 6.4 Oil-Air Lubrication System (Example)

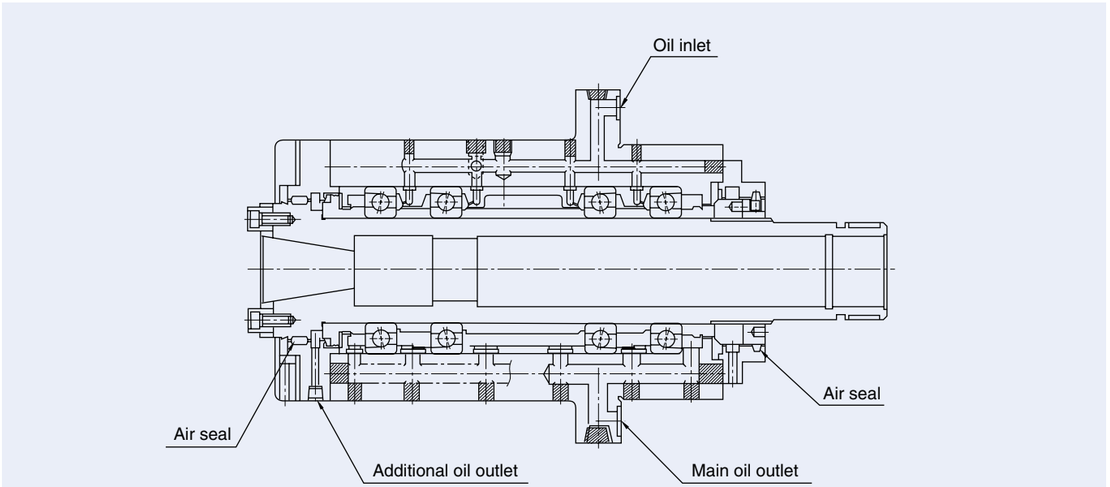


(2) Jet Lubrication

Jet lubrication is mainly used for high-speed bearings with a $d_m n$ value of more than 1.0×10^6. Jets of lubricating oil pass through the bearings via one or several nozzles at a constant pressure. In high-speed applications, the air surrounding the bearing rotates together with the bearing and forms a wall of air. The speed of the jet from each nozzle must exceed the circumferential speed of the inner ring outside surface by at least 20%. To cool the bearings and shaft uniformly, it is preferable to increase the number of nozzles. Enlarging the oil discharge outlet or employing forced discharge should also be considered to improve heat removal.

Though it increases equipment size, this method is used for machine tool spindle bearings in some applications as a means of achieving stable operation at ultra-high speeds (see Fig. 6.5). For the position of the spray nozzle, please refer to pages 240-241.

Fig. 6.5 Jet Lubrication Spindle (Example)



Oil Quantity in Forced-Circulation Lubrication

The equation below can be used to estimate the amount of oil needed for a bearing with forced-circulation lubrication.

$$Q = \frac{0.19 \times 10^{-5}}{T_2 - T_1} d\mu n F \text{ (N)} \quad \cdots\cdots\cdots\cdots\cdots\cdots (1)$$

where Q : Oil supply rate (liters/min)
T_1 : Oil temperature at oil inlet (°C)
T_2 : Oil temperature at oil outlet (°C)
d : Bearing bore (mm)
μ : Dynamic friction coefficient (approximate value determined by bearing type, see Table 6.3)
n : Bearing speed (min^{-1})
F : Bearing load (N)

Table 6.3 Dynamic Friction Coefficient (Approximate Values)

Bearing Type	Coefficient μ
Angular contact ball bearings	0.0015
Thrust ball bearings	0.0011
Cylindrical roller bearings	0.0010
Tapered roller bearings	0.0022
Thrust cylindrical roller bearings	0.0040

The value calculated from equation (1) is a guideline only and may be modified after considering such factors as the restrictions on oil inlet and oil outlet bore sizes.

Note that the oil drain pipe and oil outlet bore must be designed large enough to prevent stagnation of the circulating oil in the housing.

The oil amount obtained from equation (1) is too large for large bearings (bore sizes of more than 200 mm) if they are exposed to heavy loads. In these cases, a value about 2/3 to 1/2 the calculated value should be taken as a guideline and the final value determined after confirming on actual equipment.

7. Bearing Tolerances

The tolerances for boundary dimensions and running accuracy of NSK radial bearings are specified by ISO 492/199/582/1132-1 and JIS B 1514. In addition, NSK manufactures angular contact ball bearings with precision classes ABEC 5, 7, and 9 as specified by American Bearing Manufacturers Association (ABMA) Standard 20.

Rough definitions of items listed for running accuracy and their measuring methods are described in Table 7.1 and Fig 7.1. Further details are available in ISO 5593, JIS B 0104, and JIS B 1515.

Table 7.1

Running Accuracy		Inner Ring	Outer Ring	Dial Gauge
Radial runout of inner ring of assembled bearing	K_{ia}	Rotating	Stationary	A
Radial runout of outer ring of assembled bearing	K_{ea}	Stationary	Rotating	A
Axial runout of inner ring of assembled bearing	S_{ia}	Rotating	Stationary	B_1
Axial runout of outer ring of assembled bearing	S_{ea}	Stationary	Rotating	B_2
Perpendicularity of inner ring face with respect to the bore	S_d	Rotating	Stationary	C
Perpendicularity of outer ring outside surface with respect to the face	S_D	—	Rotating	D
Variation in thickness between shaft washer (housing washer) raceway and back face	S_i, S_e	Only the shaft washer or the housing washer is to be rotated		E

Tolerances for Radial Bearings (Excluding Tapered Roller Bearings)

Inner Ring

Table 7.2 Inner Ring (Class 5) Unit: μm

Nominal Bore Diameter d (mm)		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp} (2)		Variation of Bore Diameter in a Single Plane V_{dsp} (2)		Variation of Mean Bore Diameter V_{dmp} (2)	Radial Runout of Inner Ring of Assembled Bearing K_{ia}	Perpendicularity of Inner Ring Face With Respect to the Bore S_d	Axial Runout of Inner Ring of Assembled Bearing S_{ia} (4)	Deviation of Single Inner Ring Width Δ_{Bs}			Inner Ring Width Variation V_{Bs}
				Diameter Series 9	Diameter Series 0, 2, 3					Single Bearing Matched Bearing	Single Bearing	Matched Bearings (1)	
Over	Incl.	High	Low	Max.		Max.	Max.	Max.	Max.	High	Low		Max.
2.5	**10**	0	− 5	5	4	3	4	7	7	0	− 40	− 250	5
10	**18**	0	− 5	5	4	3	4	7	7	0	− 80	− 250	5
18	**30**	0	− 6	6	5	3	4	8	8	0	− 120	− 250	5
30	**50**	0	− 8	8	6	4	5	8	8	0	− 120	− 250	5
50	**80**	0	− 9	9	7	5	5	8	8	0	− 150	− 250	6
80	**120**	0	− 10	10	8	5	6	9	9	0	− 200	− 380	7
120	**180**	0	− 13	13	10	7	8	10	10	0	− 250	− 380	8
180	**250**	0	− 15	15	12	8	10	11	13	0	− 300	− 500	10
250	**315**	0	− 18	18	14	9	13	13	15	0	− 350	− 500	13
315	**400**	0	− 23	23	18	12	15	15	20	0	− 400	− 630	15

Table 7.3 Inner Ring (Class 4) Unit: μm

Nominal Bore Diameter d (mm)		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp} (2)		Deviation of a Single Bore Dia. Δ_{ds} (2)		Variation of Bore Diameter in a Single Plane V_{dsp} (2)		Variation of Mean Bore Diameter V_{dmp} (2)	Radial Runout of Inner Ring of Assembled Bearing K_{ia}	Perpendicularity of Inner Ring Face With Respect to the Bore S_d	Axial Runout of Inner Ring of Assembled Bearing S_{ia} (4)	Deviation of Single Inner Ring Width Δ_{Bs}			Inner Ring Width Variation V_{Bs}
						Diameter Series 9	Diameter Series 0, 2, 3					Single Bearing Matched Bearing	Single Bearing	Matched Bearings (1)	
Over	Incl.	High	Low	High	Low	Max.		Max.	Max.	Max.	Max.	High	Low		Max.
2.5	**10**	0	− 4	0	− 4	4	3	2	2.5	3	3	0	− 40	− 250	2.5
10	**18**	0	− 4	0	− 4	4	3	2	2.5	3	3	0	− 80	− 250	2.5
18	**30**	0	− 5	0	− 5	5	4	2.5	3	4	4	0	− 120	− 250	2.5
30	**50**	0	− 6	0	− 6	6	5	3	4	4	4	0	− 120	− 250	3
50	**80**	0	− 7	0	− 7	7	5	3.5	4	5	5	0	− 150	− 250	4
80	**120**	0	− 8	0	− 8	8	6	4	5	5	5	0	− 200	− 380	4
120	**180**	0	− 10	0	− 10	10	8	5	6	6	7	0	− 250	− 380	5
180	**250**	0	− 12	0	− 12	12	9	6	8	7	8	0	− 300	− 500	6

Notes for Tables 7.2-7.5:

(1) Applicable to individual rings manufactured for use in bearing arrangements, including SU (universal matching) bearings.
(2) Applicable to bearings with cylindrical bores.
(3) Class 3 is an NSK standard where bearing bore diameter and outer ring outside diameter conform to Class 4 tolerances and others to Class 2.
(4) Applicable to ball bearings.

Remarks

1. The bore diameter "no-go" tolerance limit (high) specified in these tables for cylindrical bore bearings does not necessarily apply within a distance from the ring face to 1/2 times the chamfer dimension r (max.).
2. ANSI/ABMA Std 20: ABEC 5, ABEC 7, and ABEC 9 are equivalent to ISO (JIS) Classes 5, 4, and 2 respectively. ABMA standards apply to angular contact ball bearings.

Fig. 7.1 Measuring Methods for Running Accuracy

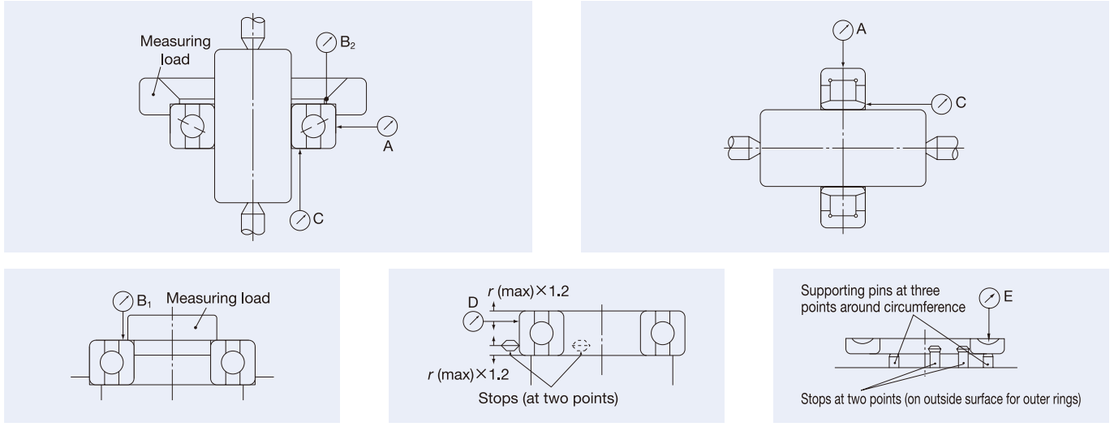


Table 7.4 Inner Ring (Class 3) (3) Unit: μm

Nominal Bore Diameter d (mm)		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp} (2)		Deviation of a Single Bore Dia. Δ_{ds} (2)		Variation of Bore Diameter in a Single Plane V_{dsp} (2)	Variation of Mean Bore Diameter V_{dmp} (2)	Radial Runout of Inner Ring of Assembled Bearing K_{ia}	Perpendicularity of Inner Ring Face With Respect to the Bore S_d	Axial Runout of Inner Ring of Assembled Bearing S_{ia} (4)	Deviation of Single Inner Ring Width Δ_{Bs}			Inner Ring Width Variation V_{Bs}
											Single Bearing Matched Bearing	Single Bearing	Matched (1) Bearing	
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.	Max.	Max.	High	Low		Max.
2.5	**10**	0	− 4	0	− 4	2.5	1.5	1.5	1.5	1.5	0	− 40	−250	1.5
10	**18**	0	− 4	0	− 4	2.5	1.5	1.5	1.5	1.5	0	− 80	−250	1.5
18	**30**	0	− 5	0	− 5	2.5	1.5	2.5	1.5	2.5	0	− 120	−250	1.5
30	**50**	0	− 6	0	− 6	2.5	1.5	2.5	1.5	2.5	0	− 120	−250	1.5
50	**80**	0	− 7	0	− 7	4	2	2.5	1.5	2.5	0	− 150	−250	1.5
80	**120**	0	− 8	0	− 8	5	2.5	2.5	2.5	2.5	0	− 200	−380	2.5
120	**150**	0	− 10	0	− 10	7	3.5	2.5	2.5	2.5	0	− 250	−380	2.5
150	**180**	0	− 10	0	− 10	7	3.5	5	4	5	0	− 250	−380	4
180	**250**	0	− 12	0	− 12	8	4	5	5	5	0	− 300	−500	5

Table 7.5 Inner Ring (Class 2) Unit: μm

Nominal Bore Diameter d (mm)		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp} (2)		Deviation of a Single Bore Dia. Δ_{ds} (2)		Variation of Bore Diameter in a Single Plane V_{dsp} (2)	Variation of Mean Bore Diameter V_{dmp} (2)	Radial Runout of Inner Ring of Assembled Bearing K_{ia}	Perpendicularity of Inner Ring Face With Respect to the Bore S_d	Axial Runout of Inner Ring of Assembled Bearing S_{ia} (4)	Deviation of Single Inner Ring Width Δ_{Bs}			Inner Ring Width Variation V_{Bs}
											Single Bearing Matched Bearing	Single Bearing	Matched (1) Bearing	
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.	Max.	Max.	High	Low		Max.
2.5	**10**	0	− 2.5	0	− 2.5	2.5	1.5	1.5	1.5	1.5	0	− 40	− 250	1.5
10	**18**	0	− 2.5	0	− 2.5	2.5	1.5	1.5	1.5	1.5	0	− 80	− 250	1.5
18	**30**	0	− 2.5	0	− 2.5	2.5	1.5	2.5	1.5	2.5	0	− 120	− 250	1.5
30	**50**	0	− 2.5	0	− 2.5	2.5	1.5	2.5	1.5	2.5	0	− 120	− 250	1.5
50	**80**	0	− 4	0	− 4	4	2	2.5	1.5	2.5	0	− 150	− 250	1.5
80	**120**	0	− 5	0	− 5	5	2.5	2.5	2.5	2.5	0	− 200	− 380	2.5
120	**150**	0	− 7	0	− 7	7	3.5	2.5	2.5	2.5	0	− 250	− 380	2.5
150	**180**	0	− 7	0	− 7	7	3.5	5	4	5	0	− 250	− 380	4
180	**250**	0	− 8	0	− 8	8	4	5	5	5	0	− 300	− 500	5

Inner ring (Class 4Y)

Table 7.6 Tolerances for Bore Diameter of Inner Ring Unit: μm

Bore Diameter (mm)		Class 4		Class 4Y (Median Specification)	
Over	Incl.	High	Low	High	Low
30	**50**	0	− 6	− 1	− 3
50	**80**	0	− 7	− 2	− 5
80	**120**	0	− 8	− 3	− 6
120	**150**	0	− 10	− 3	− 7

* Class 4Y is an NSK proprietary standard in which the tolerances for the bearing bore diameter and outside diameter are in a special class (controlled to a median specification), while other tolerances are per Class 4. Since this minimizes variation in bearing bore diameter and outer ring outside diameter, P4Y is the most suitable tolerance class for universal matching bearings.

Tolerances for Radial Bearings (Excluding Tapered Roller Bearings)

Outer Ring

Table 7.7 Outer Ring (Class 5)

Unit: μm

Nominal Outside Diameter D (mm)		Deviation of Mean Outside Diameter in a Single Plane Δ_{Dmp}		Variation of Outside Diameter in a Single Plane V_{Dsp} Diameter Series		Variation of Mean Outside Diameter V_{Dmp}	Radial Runout of Outer Ring of Assembled Bearing K_{ea}	Perpendicularity of Outer Ring Outside Surface With Respect to the Bore S_D	Axial Runout of Outer Ring of Assembled Bearing S_{ea} (1)	Deviation of Single Outer Ring Width Δ_{Cs}	Outer Ring Width Variation V_{Cs}
Over	Incl.	High	Low	9	0, 2	Max.	Max.	Max.	Max.		Max.
6	**18**	0	− 5	5	4	3	5	8	8	Equal to the value of inner ring (Δ_{Bs}) of the same designation.	5
18	**30**	0	− 6	6	5	3	6	8	8		5
30	**50**	0	− 7	7	5	4	7	8	8		5
50	**80**	0	− 9	9	7	5	8	8	10		6
80	**120**	0	− 10	10	8	5	10	9	11		8
120	**150**	0	− 11	11	8	6	11	10	13		8
150	**180**	0	− 13	13	10	7	13	10	14		8
180	**250**	0	− 15	15	11	8	15	11	15		10
250	**315**	0	− 18	18	14	9	18	13	18		11
315	**400**	0	− 20	20	15	10	20	13	20		13
400	**500**	0	− 23	23	17	12	23	15	23		15
500	**630**	0	− 28	28	21	14	25	18	25		18
630	**800**	0	− 35	35	26	18	30	20	30		20

Table 7.8 Outer Ring (Class 4)

Unit: μm

Nominal Outside Diameter D (mm)		Deviation of Mean Outside Diameter in a Single Plane Δ_{Dmp}		Deviation of Single Outside Dia. Δ_{Ds}		Variation of Outside Diameter in a Single Plane V_{Dsp} Diameter Series		Variation of Mean Outside Diameter V_{Dmp}	Radial Runout of Outer Ring of Assembled Bearing K_{ea}	Perpendicularity of Outer Ring Outside Surface With Respect to the Bore S_D	Axial Runout of Outer Ring of Assembled Bearing S_{ea} (1)	Deviation of Single Outer Ring Width Δ_{Cs}	Outer Ring Width Variation V_{Cs}
Over	Incl.	High	Low	High	Low	9	0, 2	Max.	Max.	Max.	Max.		Max.
6	**18**	0	− 4	0	− 4	4	3	2	3	4	5	Equal to the value of inner ring (Δ_{Bs}) of the same designation.	2.5
18	**30**	0	− 5	0	− 5	5	4	2.5	4	4	5		2.5
30	**50**	0	− 6	0	− 6	6	5	3	5	4	5		2.5
50	**80**	0	− 7	0	− 7	7	5	3.5	5	4	5		3
80	**120**	0	− 8	0	− 8	8	6	4	6	5	6		4
120	**150**	0	− 9	0	− 9	9	7	5	7	5	7		5
150	**180**	0	− 10	0	− 10	10	8	5	8	5	8		5
180	**250**	0	− 11	0	− 11	11	8	6	10	7	10		7
250	**315**	0	− 13	0	− 13	13	10	7	11	8	10		7
315	**400**	0	− 15	0	− 15	15	11	8	13	10	13		8

Notes for Tables 7.7-7.10:

(1) Applicable to ball bearings.

(2) Class 3 is an NSK standard where bearing bore diameter and outer ring outside diameter conform to Class 4 tolerances and others to Class 2.

Remarks

1. The outside diameter "no-go" tolerance limit (high) specified for cylindrical bore bearings in these tables does not necessarily apply within a distance from the ring face to 1/2 times the chamfer dimension r (max.).
2. ANSI/ABMA Std 20: ABEC 5, ABEC 7, and ABEC 9 are equivalent to ISO (JIS) Classes 5, 4, and 2 respectively. ABMA standards apply to angular contact ball bearings.

Table 7.9 Outer Ring (Class 3) (2)

Unit: μm

Nominal Outside Diameter D (mm)		Deviation of Mean Outside Diameter in a Single Plane Δ_{Dmp}		Deviation of Single Outside Dia. Δ_{Ds}		Variation of Outside Diameter in a Single Plane V_{Dsp}	Variation of Mean Outside Diameter V_{Dmp}	Radial Runout of Outer Ring of Assembled Bearing K_{ea}	Perpendicularity of Outer Ring Outside Surface With Respect to the Bore S_D	Axial Runout of Outer Ring of Assembled Bearing S_{ea} (1)	Deviation of Single Outer Ring Width Δ_{Cs}	Outer Ring Width Variation V_{Cs}
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.	Max.	Max.		Max.
6	**18**	0	− 4	0	− 4	2.5	1.5	1.5	1.5	1.5	Equal to the value of inner ring (Δ_{Bs}) of the same designation.	1.5
18	**30**	0	− 5	0	− 5	4	2	2.5	1.5	2.5		1.5
30	**50**	0	− 6	0	− 6	4	2	2.5	1.5	2.5		1.5
50	**80**	0	− 7	0	− 7	4	2	4	1.5	4		1.5
80	**120**	0	− 8	0	− 8	5	2.5	5	2.5	5		2.5
120	**150**	0	− 9	0	− 9	5	2.5	5	2.5	5		2.5
150	**180**	0	− 10	0	− 10	7	3.5	5	2.5	5		2.5
180	**250**	0	− 11	0	− 11	8	4	7	4	7		4
250	**315**	0	− 13	0	− 13	8	4	7	5	7		5
315	**400**	0	− 15	0	− 15	10	5	8	7	8		7

Table 7.10 Outer Ring (Class 2)

Unit: μm

Nominal Outside Diameter D (mm)		Deviation of Mean Outside Diameter in a Single Plane Δ_{Dmp}		Deviation of Single Outside Dia. Δ_{Ds}		Variation of Outside Diameter in a Single Plane V_{Dsp}	Variation of Mean Outside Diameter V_{Dmp}	Radial Runout of Outer Ring of Assembled Bearing K_{ea}	Perpendicularity of Outer Ring Outside Surface With Respect to the Bore S_D	Axial Runout of Outer Ring of Assembled Bearing S_{ea} (1)	Deviation of Single Outer Ring Width Δ_{Cs}	Outer Ring Width Variation V_{Cs}
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.	Max.	Max.		Max.
6	**18**	0	− 2.5	0	− 2.5	2.5	1.5	1.5	1.5	1.5	Equal to the value of inner ring (Δ_{Bs}) of the same designation.	1.5
18	**30**	0	− 4	0	− 4	4	2	2.5	1.5	2.5		1.5
30	**50**	0	− 4	0	− 4	4	2	2.5	1.5	2.5		1.5
50	**80**	0	− 4	0	− 4	4	2	4	1.5	4		1.5
80	**120**	0	− 5	0	− 5	5	2.5	5	2.5	5		2.5
120	**150**	0	− 5	0	− 5	5	2.5	5	2.5	5		2.5
150	**180**	0	− 7	0	− 7	7	3.5	5	2.5	5		2.5
180	**250**	0	− 8	0	− 8	8	4	7	4	7		4
250	**315**	0	− 8	0	− 8	8	4	7	5	7		5
315	**400**	0	− 10	0	− 10	10	5	8	7	8		7

Outer Ring (Class 4Y)

Table 7.11 Tolerances for Outside Diameter of Outer Ring

Unit: μm

Outside Diameter (mm)		Class 4		Class 4Y (Median Specification)	
Over	Incl.	High	Low	High	Low
30	**50**	0	− 6	–2	–6
50	**80**	0	− 7	–2	–6
80	**120**	0	− 8	–2	–6
120	**150**	0	− 9	–3	–7
150	**180**	0	–10	–3	–7
180	**200**	0	–11	–4	–9
200	**Under 215**	0	–11	–2	–9

*Class 4Y is an NSK proprietary standard in which the tolerances for the bearing bore diameter and outside diameter are in a special class (controlled to a median specification), while other tolerances are per Class 4. Since this minimizes variation in the bearing bore diameter and outer ring outside diameter, P4Y is the most suitable tolerance class for universal matching bearings.

Outer / Inner Ring (Class 3W)

Table 7.12 Tolerances for Outside Diameter of Inner / Outer Rings

Unit: μm

Outside Diameter (mm)		Class 3 Matched Bearings		Class 3W (Median Specification) Matched Bearings	
Over	Incl.	High	Low	High	Low
50	**80**	0	–250	0	–100
80	**120**	0	–380	0	–100
120	**140**	0	–380	0	–100
145	**150**	0	–380	0	–100
165	**170**	0	–380	0	–100

*Class 3W is an NSK proprietary standard in which the tolerances for the inner and outer ring width are in a special class, while other tolerances are per Class 3. The width tolerances are the same for the inner and outer rings. This standard can apply to universal matching bearings (SU, DU, DUD, and QU). Minimizing differences in width improves assembly and reduces the need for adjustments.

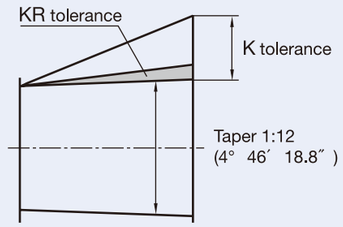
7. Bearing Tolerances

Tolerances for Tapered Bores of Cylindrical Roller Bearings

Tolerances for Tapered Bores

The bore accuracy of tapered bore cylindrical roller bearings is specified by ISO. However, in this standard, the tolerances are rather wide. For super precision cylindrical roller bearings, NSK has established its own narrower tolerances (see Fig. 7.2).

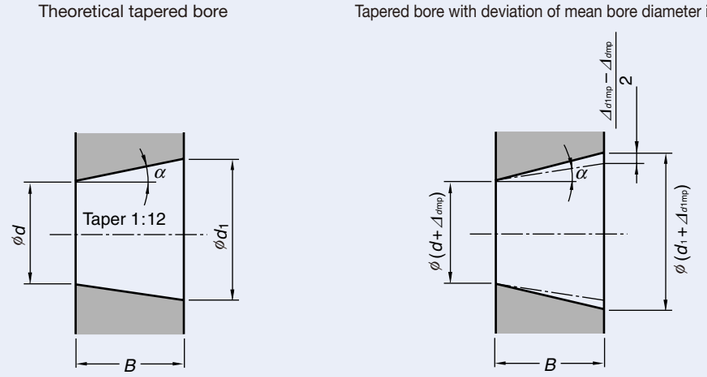
Fig. 7.2 Taper Angle Tolerances



KR : A very narrow range positioned towards the lower limit of ISO tolerance for easier adjustment of taper angle during mounting. KR tolerance is NSK's standard tolerance for bore diameters up to 400 mm.

K : Positioned midrange of ISO tolerance, this accuracy class offers bore dimensional accuracy identical to the ISO specification. K tolerance is NSK's standard tolerance for bore diameters over 400 mm.

Fig. 7.3 Tapered Bore Tolerances



d : Nominal bore diameter

d_1 : Theoretical diameter of larger end of tapered bore
$d_1 = d + \frac{1}{12}B$

Δ_{dmp} : Deviation of mean bore diameter in a single plane in theoretical diameter of smaller end of tapered bore

Δ_{d1mp} : Deviation of mean bore diameter in a single plane in theoretical diameter of larger end of tapered bore

V_{dsp} : Bore diameter variation in a single radial plane

B : Nominal inner ring width

α : Half of taper angle of tapered bore
$\alpha = 2°23'9.4''$
$= 2.38594°$
$= 0.041643\text{rad}$

Table 7.12 KR Tapered Bores

Unit: µm

d (1) (mm)		Δ_{dmp}		(Reference) (2) $\Delta_{d1mp} - \Delta_{dmp}$		V_{dsp} (2)
Over	Incl.	High	Low	High	Low	Max.
18	**30**	+ 13	0	+ 3	0	4
30	**50**	+ 16	0	+ 3	0	5
50	**80**	+ 19	0	+ 4	0	6
80	**120**	+ 22	0	+ 5	0	7
120	**180**	+ 25	0	+ 7	0	9
180	**250**	+ 29	0	+ 9	0	12
250	**315**	+ 32	0	+ 11	0	14
315	**400**	+ 36	0	+ 12	0	16

(1) Please contact NSK for bearings with bores over 400 mm.
(2) Applicable to all radial planes of tapered bores.
(3) Taper angle tolerance is $4°46'18.8''^{+25''}_{0}$

Table 7.13 K Tapered Bores

Unit: µm

d (mm)		Δ_{dmp}		$\Delta_{d1mp} - \Delta_{dmp}$		V_{dsp} (1)
Over	Incl.	High	Low	High	Low	Max.
18	**30**	+ 33	0	+ 21	0	13
30	**50**	+ 39	0	+ 25	0	16
50	**80**	+ 46	0	+ 30	0	19
80	**120**	+ 54	0	+ 35	0	22
120	**180**	+ 63	0	+ 40	0	40
180	**250**	+ 72	0	+ 46	0	46
250	**315**	+ 81	0	+ 52	0	52
315	**400**	+ 89	0	+ 57	0	57
400	**500**	+ 97	0	+ 63	0	63
500	**630**	+ 110	0	+ 70	0	70
630	**800**	+ 125	0	+ 80	0	–

(1) Applicable to all radial planes of tapered bores.

7. Bearing Tolerances

Tolerances for Angular Contact Thrust Ball Bearings

Tolerances for angular contact thrust ball bearings (Class 4A (1)) of BAR, BTR, NSKTAC2xF types)

Table 7.14 Inner ring

Unit: μm

Nominal Bore Diameter d (mm)		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp}		Deviation of a Single Bore Dia. Δ_{ds}		Variation of Bore Diameter in a Single Plane V_{dsp} Diameter Series		Variation of Mean Bore Diameter V_{dmp}	Radial Runout of Inner Ring of Assembled Bearing K_{ia}	Perpendicularity of Inner Ring Face With Respect to the Bore S_d	Inner Ring Face Runout with Raceway S_{ia}	Inner Ring Width Variation V_{Bs}	Deviation of Single Inner Ring Width Δ_{Bs}		
						9	0						Single Bearing Matched Bearing	Single Bearing	Matched Bearing (2)
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.	Max.	Max.	Max.	Max.	High	Low	
–	**50**	0	− 6	0	− 6	6	5	3	4	4	4	3	0	− 120	− 120
50	**80**	0	− 7	0	− 7	7	5	3.5	4	5	5	4	0	− 150	− 150
80	**120**	0	− 8	0	− 8	8	6	4	5	5	5	4	0	− 200	− 200
120	**180**	0	− 10	0	− 10	10	8	5	6	6	7	5	0	− 250	− 250
180	**250**	0	− 12	0	− 12	12	9	6	8	7	8	6	0	− 300	− 300
250	**315**	0	− 14	0	− 14	–	–	–	10	8	10	7	0	− 350	− 500
315	**380**	0	− 18	0	− 18	–	–	–	12	10	14	9	0	− 400	− 630

Table 7.15 Outer ring

Unit: μm

Nominal Outside Diameter D (mm)		Deviation of Mean Outside Diameter in a Single Plane Δ_{Dmp}		Deviation of Single Outside Dia. Δ_{Ds}		Variation of Outside Diameter in a Single Plane V_{dsp} Diameter Series		Variation of Mean Outside Diameter V_{Dmp}	Radial Runout of Outer Ring of Assembled Bearing K_{ea}	Perpendicularity of Outer Ring Outside Surface With Respect to the Bore S_D	Axial Runout of Outer Ring of Assembled Bearing S_{ea}	Outer Ring Width Variation V_{Cs}	Deviation of Single Outer Ring Width Δ_{Cs} Matched Bearing	
						9	0						High	Low
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.	Max.	Max.	Max.	Max.		
–	**80**	− 30	− 37	− 30	− 37	7	5	3.5	5	4	5	3		
80	**120**	− 40	− 48	− 40	− 48	8	6	4	6	5	6	4		
120	**150**	− 50	− 59	− 50	− 59	9	7	5	7	5	7	5		
150	**180**	− 50	− 60	− 50	− 60	10	8	5	8	5	8	5	Equal to the value of inner ring (Δ_{Bs}) of the same designation.	
180	**250**	− 50	− 61	− 50	− 61	11	8	6	10	7	10	7		
250	**315**	− 60	− 73	− 60	− 73	13	10	7	11	8	10	7		
315	**400**	− 60	− 80	− 60	− 80	15	11	8	13	10	13	8		
400	**500**	− 70	− 91	− 70	− 91	–	–	–	15	11	15	10		
500	**560**	− 70	− 94	− 70	− 94	–	–	–	18	13	18	11		

Tolerances for double-direction angular contact thrust ball bearings (Class 7 (3)) of NSKTAC2xD type)

Table 7.16 Tolerances of inner ring, outer ring, and bearing height

Unit: μm

Nominal Bore Diameter d (mm)		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp}		Deviation of the Actual Bearing Width Δ_{Ts}		Radial Runout of Inner (Outer) Ring of Assembled Bearing K_{ia} (K_{ea})	Perpendicularity of Inner Ring Face With Respect to the Bore S_d	Axial Runout of Inner (Outer) Ring of Assembled Bearing S_{ia} (S_{ea})
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.
–	**30**	0	− 5	0	− 300	5	4	3
30	**50**	0	− 5	0	− 400	5	4	3
50	**80**	0	− 8	0	− 500	6	5	5
80	**120**	0	− 8	0	− 600	6	5	5
120	**180**	0	− 10	0	− 700	8	8	5
180	**250**	0	− 13	0	− 800	8	8	6
250	**315**	0	− 15	0	− 900	10	10	6
315	**400**	0	− 18	0	−1 200	10	12	7

Table 7.17 Tolerance of outer ring

Unit: μm

Nominal Outside Diameter D (mm)		Deviation of Single Outside Dia. Δ_{Ds}	
Over	Incl.	High	Low
30	**50**	− 25	− 41
50	**80**	− 30	− 49
80	**120**	− 36	− 58
120	**180**	− 43	− 68
180	**250**	− 50	− 79
250	**315**	− 56	− 88
315	**400**	− 62	− 98
400	**500**	− 68	− 108
500	**630**	− 76	− 120

Notes for Tables 7.14-7.20:

(1) Class 4 uses NSK specification P4A. It is equivalent to ISO Class 4 for radial bearings, excluding the outer ring outside diameter tolerance.

(2) Applicable to individual rings manufactured for use in bearing arrangements.

(3) Class 7 uses NSK specification PN7.

(4) Class 5D uses NSK specification PN5D.

(5) Class 2B uses NSK specification P2B. Rotational accuracy is equivalent to ISO Class 2 for radial bearings.

Tolerances for Angular Contact Thrust Ball Bearings for Ball Screw Support

Tolerances for high-rigidity angular contact thrust ball bearings (Class 7C of NSKTACxxC type)

Table 7.18 NSKTAC C Series

Unit: μm

Nominal Bore or Outside Diameter (mm)		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp}		Deviation of Single Bore Diameter Δ_{ds}		Deviation of Mean Outside Diameter in a Single Plane Δ_{Dmp}		Deviation of Single Outside Diameter Δ_{Ds}		Deviation of Single Inner Ring Width Δ_{Bs}			Axial Runout of Inner (Outer) Ring of Assembled Bearing S_{ia} (S_{ea})
										Single Bearing Matched Bearing	Single Bearing	Matched Bearing (2)	
Over	Incl.	High	Low	High	Low	High	Low	High	Low	High	Low		Max.
10	18	0	− 4	0	− 4	–	–	–	–	0	− 120	− 250	2.5
18	30	0	− 5	0	− 5	–	–	–	–	0	− 120	− 250	2.5
30	50	0	− 6	0	− 6	0	− 6	0	− 6	0	− 120	− 250	2.5
50	80	0	− 7	0	− 7	0	− 7	0	− 7	0	− 150	− 250	2.5
80	120	0	− 8	0	− 8	0	− 8	0	− 8	0	− 200	− 380	2.5

Tolerances for angular contact thrust ball bearings for high-load drive applications (Class 5D (4)) of NSKTAC0xD and NSKTACxx-3 types)

Table 7.19 NSKTAC 03 Series

Unit: μm

Nominal Bore or Outside Diameter (mm)		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp}		Deviation of Mean Outside Diameter in a Single Plane Δ_{Dmp}		Deviation of Single Inner Ring Width Δ_{Bs}		Axial Runout of Inner (Outer) Ring of Assembled Bearing S_{ia} (S_{ea})
Over	Incl.	High	Low	High	Low	High	Low	Max.
10	18	0	− 5	–	–	0	− 80	5
18	30	0	− 6	–	–	0	− 120	5
30	50	0	− 8	0	− 7	0	− 120	5
50	80	0	− 9	0	− 9	0	− 150	8
80	120	0	− 10	0	− 10	0	− 200	8
120	150	0	− 13	0	− 11	0	− 250	10
150	180	0	− 13	0	− 13	0	− 250	10
180	250	–	–	0	− 15	–	–	10
250	315	–	–	0	− 18	–	–	11
315	400	–	–	0	− 20	–	–	13

Tolerances for BSBD Series double-row bearings (Class P2B (5)) of BSF and BSN types)

Table 7.20 BSBD Series double-row bearings (BSF and BSN types)

Unit: μm

Nominal Bore Diameter (mm)		Single Plane Mean Bore Diameter Deviation Δ_{dmp}		Single Plane Mean Outside Diameter Deviation Δ_{Dmp}		Inner Ring Face Runout with Raceway S_{ia}	Radial Runout of Inner Ring K_{ia}	Width Tolerance	
Over	Incl.	High	Low	High	Low	Max.	Max.	High	Low
10	18	0	− 5	0	− 10	1.5	1.5	0	− 250
18	30	0	− 5	0	− 10	2.5	2.5	0	− 250
30	50	0	− 5	0	− 10	2.5	2.5	0	− 250
50	80	0	− 8	0	− 15	2.5	2.5	0	− 250

Tolerances for Metric Series Tapered Roller Bearings

Inner Ring

Table 7.21 Inner ring (Class 5) Unit: μm

Nominal Bore Diameter (mm) d		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp}		Variation of Bore Diameter in a Single Plane V_{dsp}	Variation of Mean Bore Diameter V_{dmp}	Radial Runout of Inner Ring of Assembled Bearing K_{ia}	Perpendicularity of Inner Ring Face With Respect to the Bore S_d
Over	Incl.	High	Low	Max.	Max.	Max.	Max.
10	18	0	− 7	5	5	3.5	7
18	30	0	− 8	6	5	4	8
30	50	0	− 10	8	5	5	8
50	80	0	− 12	9	6	5	8
80	120	0	− 15	11	8	6	9
120	180	0	− 18	14	9	8	10
180	250	0	− 22	17	11	10	11
250	315	0	− 25	—	—	13	13
315	400	0	− 30	—	—	15	15
400	500	0	− 35	—	—	18	19

Table 7.22 Inner ring (Class 4) Unit: μm

Nominal Bore Diameter (mm) d		Deviation of Mean Bore Diameter in a Single Plane Δ_{dmp}		Deviation of Single Bore Diameter Δ_{ds}		Variation of Bore Diameter in a Single Plane V_{dsp}	Variation of Mean Bore Diameter V_{dmp}	Radial Runout of Inner Ring of Assembled Bearing K_{ia}	Perpendicularity of Inner Ring Face With Respect to the Bore S_d	Axial Runout of Inner Ring of Assembled Bearing S_{ia}
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.	Max.	Max.
10	18	0	− 5	0	− 5	4	4	2.5	3	3
18	30	0	− 6	0	− 6	5	4	3	4	4
30	50	0	− 8	0	− 8	6	5	4	4	4
50	80	0	− 9	0	− 9	7	5	4	5	4
80	120	0	− 10	0	− 10	8	5	5	5	5
120	180	0	− 13	0	− 13	10	7	6	6	7
180	250	0	− 15	0	− 15	11	8	8	7	8
250	315	0	− 18	0	− 18	—	—	10	8	10
315	400	0	− 23	0	− 23	—	—	12	10	14
400	500	0	− 27	0	− 27	—	—	14	13	17

Remarks: 1. The outside diameter "no-go side" tolerances (low) specified in this table do not necessarily apply within a distance of 1.2 times the chamfer dimension r (max.) from the ring face.
2. Some of the tolerances refer to NSK's own specification.

Outer Ring

Table 7.23 Outer ring (Class 5) Unit: μm

Nominal Outside Diameter (mm) D		Deviation of Mean Outside Diameter in a Single Plane Δ_{Dmp}		Outside Diameter Variation in a Single Radial Plane V_{Dp}	Mean Outside Diameter Variation V_{Dmp}	Radial Runout of Outer Ring K_{ea}	Variation of Outside Surface Generatrix Inclination with Face S_D
Over	Incl.	High	Low	Max.	Max.	Max.	Max.
18	30	0	− 8	6	5	6	8
30	50	0	− 9	7	5	7	8
50	80	0	− 11	8	6	8	8
80	120	0	− 13	10	7	10	9
120	150	0	− 15	11	8	11	10
150	180	0	− 18	14	9	13	10
180	250	0	− 20	15	10	15	11
250	315	0	− 25	19	13	18	13
315	400	0	− 28	22	14	20	13
400	500	0	− 33	—	—	23	15
500	630	0	− 38	—	—	25	18

Table 7.24 Outer ring (Class 4) Unit: μm

Nominal Outside Diameter (mm) D		Single Plane Mean Outside Diameter Deviation Δ_{Dmp}		Deviation of Single Outside Diameter Δ_{Ds}		Outside Diameter Variation in a Single Radial Plane V_{Dp}	Mean Outside Diameter Variation V_{Dmp}	Radial Runout of Outer Ring K_{ea}	Variation of Outside Surface Generatrix Inclination with Face S_D	Outer Ring Face Runout with Raceway S_{ea}
Over	Incl.	High	Low	High	Low	Max.	Max.	Max.	Max.	Max.
18	30	0	− 6	0	− 6	5	4	4	4	5
30	50	0	− 7	0	− 7	5	5	5	4	5
50	80	0	− 9	0	− 9	7	5	5	4	5
80	120	0	− 10	0	− 10	8	5	6	5	6
120	150	0	− 11	0	− 11	8	6	7	5	7
150	180	0	− 13	0	− 13	10	7	8	5	8
180	250	0	− 15	0	− 15	11	8	10	7	10
250	315	0	− 18	0	− 18	14	9	11	8	10
315	400	0	− 20	0	− 20	15	10	13	10	13
400	500	0	− 23	0	− 23	—	—	15	11	15
500	630	0	− 28	0	− 28	—	—	18	13	18

Remarks: 1. The cylindrical bore diameter "no-go side" tolerance limit (high) specified in this table does not necessarily apply within a distance of 1.2 times the chamfer dimension r (max.) from the ring face.
2. Some of the tolerances refer to NSK's own specification.

Tolerances for Thrust Ball Bearings

Shaft Washer

Table 7.25 Shaft washer (Class 5) Unit: μm

Nominal Bore Diameter (mm) d		Single Plane Mean Bore Diameter Deviation Δ_{dmp}		Bore Diameter Variation in a Single Radial Plane V_{dp}	Shaft Washer (Housing Washer) Raceway to Back Face Thickness Variation S_i (S_e)	(Reference) Deviation of Bearing Height ΔT_s	
Over	Incl.	High	Low	Max.	Max.	High	Low
18	30	0	− 10	8	3	0	− 75
30	50	0	− 12	9	3	0	− 100
50	80	0	− 15	11	4	0	− 125
80	120	0	− 20	15	4	0	− 150
120	180	0	− 25	19	5	0	− 175
180	250	0	− 30	23	5	0	− 200
250	315	0	− 35	26	7	0	− 225
315	400	0	− 40	30	7	0	− 300
400	500	0	− 45	34	9	0	− 350
500	630	0	− 50	38	11	0	− 450
630	800	0	− 75	—	13	0	− 550
800	1000	0	− 100	—	15	0	− 700
1000	1250	0	− 125	—	18	0	− 900

Table 7.26 Shaft washer (Class 4) Unit: μm

Nominal Bore Diameter (mm) d		Single Plane Mean Bore Diameter Deviation Δ_{dmp}		Bore Diameter Variation in a Single Radial Plane V_{dp}	Shaft Washer (Housing Washer)Raceway to Back Face Thickness Variation S_i (S_e)	(Reference) Deviation of Bearing Height ΔT_s	
Over	Incl.	High	Low	Max.	Max.	High	Low
18	30	0	− 8	6	2	0	− 75
30	50	0	− 10	8	2	0	− 100
50	80	0	− 12	9	3	0	− 125
80	120	0	− 15	11	3	0	− 150
120	180	0	− 18	14	4	0	− 175
180	250	0	− 22	17	4	0	− 200
250	315	0	− 25	19	5	0	− 225
315	400	0	− 30	23	5	0	− 300
400	500	0	− 35	26	6	0	− 350
500	630	0	− 40	30	7	0	− 450
630	800	0	− 50	—	8	0	− 550

Housing Washer

Table 7.27 Housing washer (Class 5) Unit: μm

Nominal Outside Diameter (mm) D		Single Plane Mean Outside Diameter Deviation Δ_{Dmp}		Outside Diameter Variation in a Single Radial Plane V_{Dp}
Over	Incl.	High	Low	Max.
30	50	0	− 16	12
50	80	0	− 19	14
80	120	0	− 22	17
120	180	0	− 25	19
180	250	0	− 30	23
250	315	0	− 35	26
315	400	0	− 40	30
400	500	0	− 45	34
500	630	0	− 50	38
630	800	0	− 75	55
800	1000	0	− 100	75
1000	1250	0	− 120	—
1250	1600	0	− 160	—

Table 7.28 Housing washer (Class 4) Unit: μm

Nominal Outside Diameter (mm) D		Single Plane Mean Outside Diameter Deviation Δ_{Dmp}		Outside Diameter Variation in a Single Radial Plane V_{Dp}
Over	Incl.	High	Low	Max.
30	50	0	− 9	7
50	80	0	− 11	8
80	120	0	− 13	10
120	180	0	− 15	11
180	250	0	− 20	15
250	315	0	− 25	19
315	400	0	− 28	21
400	500	0	− 33	25
500	630	0	− 38	29
630	800	0	− 45	34

Fits of Shafts and Housings

In order to take full advantage of the capabilities of super precision bearings in terms of running accuracy, high-speed performance, and low heat generation, a high level of accuracy is required of shafts, housings, and surrounding parts.

When the inner or outer ring is mounted onto a shaft or into a housing with interference, the shape of the shaft or the housing (out-of-roundness) is transferred to the bearing raceway surfaces and will affect running accuracy. When multiple angular contact ball bearings are used in an arrangement, cylindricity affects the distribution of preload between the bearings. Therefore, the parts that mate with bearing surfaces should be as accurate as possible. Inaccurate mating of parts may cause the formation of peaks or ridges along the shaft, which can affect the quality of finished work, especially in precision lathes.

Tables 8.1 through 8.4 list recommended fit values for bearings operated in standard conditions at speeds of less than 800 000 $d_m n$ (Tables 8.1 and 8.2 refer to bearings for machine tool spindles, while Tables 8.3 and 8.4 refer to ball screw support bearings).

For thrust cylindrical roller bearings and thrust ball bearings, we recommend a shaft fit of h6 and a housing fit of G7 as a target. Clearance fits are commonly used for both shaft washers and housing washers. However, particularly in thrust ball bearings where alignment of both washers with the shaft is required, small values are preferable within the range that assembly allows.

Table 8.1 Fits on Shafts

Bearing Type	Shaft Outside Diameter (mm)		Tolerance of Shaft Outside Diameter (mm)		Target Fit (1)(2) (mm)	
	Over	Incl.	Min.	Max.	Min.	Max.
Bearings for machine tool spindles	**10**	**18**	− 0.003	0	0	0.002T
	18	**50**	− 0.004	0	0	0.003T
	50	**80**	− 0.005	0	0	0.003T
	80	**120**	− 0.003	0.003	0	0.004T
	120	**180**	− 0.004	0.004	0	0.006T
	180	**250**	− 0.005	0.005	0	0.008T
	250	**315**	− 0.008	0.008	0	0.010T
	315	**400**	− 0.009	0.009	0	0.013T
	400	**500**	− 0.010	0.010	0	0.015T

(1) Aim for target fit values when matching the bearing with the shaft and housing.
(2) T refers to a tight (or interference) fit, and L to a loose or clearance fit.
(3) Applicable to tapered roller bearings only if the outer ring is inserted into the housing separately.

Table 8.2 Fits on Housings

Bearing Type	Housing Bore Diameter (mm)		Tolerance of Housing bore (mm)		Target Fit (1)(2) (mm)	
	Over	Incl.	Min.	Max.	Min.	Max.
Angular contact ball bearings (Fixed end)	**18**	**50**	− 0.002	0.002	0.002L	0.005L
	50	**80**	− 0.0025	0.0025	0.002L	0.005L
	80	**120**	− 0.003	0.003	0.002L	0.006L
	120	**180**	− 0.004	0.004	0.003L	0.009L
	180	**250**	− 0.005	0.005	0.004L	0.012L
	250	**315**	− 0.006	0.006	0.005L	0.015L
	315	**400**	− 0.007	0.007	0.007L	0.02L
	400	**500**	− 0.008	0.008	0.008L	0.023L
	500	**630**	− 0.008	0.008	0.008L	0.024L
	630	**800**	− 0.009	0.009	0.009L	0.027L
Angular contact thrust ball bearings (if used together with radial bearings)	**50**	**80**	− 0.005	0	0.025L	0.037L
	80	**120**	− 0.007	0	0.033L	0.047L
	120	**150**	− 0.008	0	0.042L	0.059L
	150	**180**	− 0.011	0	0.039L	0.059L
	180	**250**	− 0.012	0	0.038L	0.061L
	250	**315**	− 0.013	0	0.047L	0.073L
	315	**400**	− 0.015	0	0.045L	0.080L
	400	**500**	− 0.016	0	0.054L	0.091L
	500	**630**	− 0.018	0	0.052L	0.094L
Angular contact ball bearings (Free end)	**18**	**50**	0.004	0.007	0.006L	0.009L
	50	**80**	0.005	0.008	0.007L	0.01L
	80	**120**	0.006	0.010	0.008L	0.012L
	120	**180**	0.008	0.014	0.011L	0.017L
	180	**250**	0.010	0.018	0.014L	0.022L
	250	**315**	0.012	0.022	0.017L	0.027L
	315	**400**	0.013	0.026	0.021L	0.034L
	400	**500**	0.015	0.030	0.024L	0.039L
	500	**630**	0.016	0.032	0.026L	0.042L
	630	**800**	0.018	0.036	0.031L	0.049L
Cylindrical roller bearings, Tapered roller bearings (3)	**50**	**80**	− 0.005	0	0.002L	0.002T
	80	**120**	− 0.007	0	0.002L	0.002T
	120	**180**	− 0.008	0	0.003L	0.003T
	180	**250**	− 0.011	0	0.004L	0.004T
	250	**315**	− 0.012	0	0.005L	0.005T
	315	**400**	− 0.013	0	0.007L	0.007T
	400	**500**	− 0.015	0	0.008L	0.008T
	500	**630**	− 0.016	0	0.008L	0.008T
	630	**800**	− 0.018	0	0.009L	0.009T

8. Design of Shafts and Housings

Recommended interference values for standard operating conditions of ball screws are listed in Tables 8.3 and 8.4.

When moment loads are high with single-end support using high load capacity angular contact thrust ball bearings for ball screw support, it is advisable to increase shaft interference as required by choosing k5 or similar.

In precision class applications, ensure the following specifications:

- Inclination: 1/2000 or less (target 1/5000 or less)
- Eccentricity: 0.020 mm or less

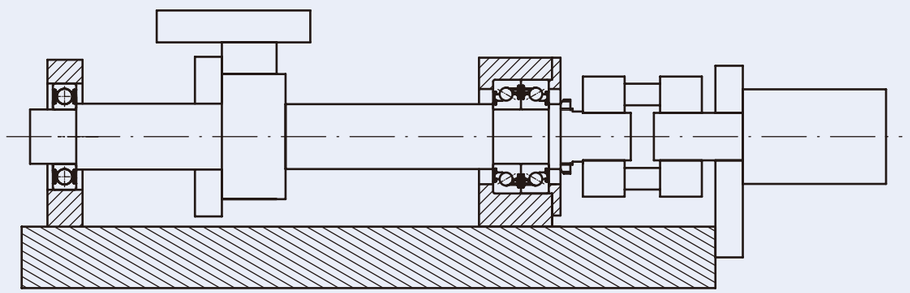
Table 8.3 Fits on Shafts

Bearing Type	Shaft Outside Diameter (mm)		Tolerance of Shaft Outside Diameter (mm)		
	Over	Incl.		Min.	Max.
High-rigidity angular contact thrust ball bearings for ball screw support	**10**	**18**	h5	− 0.008	0
	18	**30**		− 0.009	0
	30	**50**		− 0.011	0
	50	**80**		− 0.013	0
	80	**120**		− 0.015	0
High load capacity angular contact thrust ball bearings for ball screw support	**10**	**18**	js5	− 0.004	0.004
	18	**30**		− 0.0045	0.0045
	30	**50**		− 0.0055	0.0055
	50	**80**		− 0.0065	0.0065
	80	**120**		− 0.0075	0.0075

Table 8.4 Fits on Housings

Bearing Type	Housing Bore Diameter (mm)		Tolerance of Housing Bore Diameter (mm)		
	Over	Incl.		Min.	Max.
Angular contact thrust ball bearings for ball screw support	**30**	**50**	H6	0	0.016
	50	**80**		0	0.019
	80	**120**		0	0.022
	120	**180**		0	0.025
	180	**250**		0	0.029
	250	**315**		0	0.032

Fig. 8.1 Arrangement of Ball Screw Support Bearings (Example)



Recommended accuracy and surface roughness for bearing seats in machine tool spindle applications are shown in the tables below.

When using cylindrical roller bearings with tapered bores, if the taper of the inner ring does not match that of the shaft, misalignment of the inner ring groove will cause irregular movement of the rollers. With double-row cylindrical roller bearings, a difference in residual clearance between the rows will occur. Consequently, load will not be sustained equally, and rigidity will decline.

We recommend matching the tapered parts to be mated with bearings. Contact should cover more than 80% of the total surface area that is dyed blue. Please refer to page 248 for a description and details on matching shaft tapers.

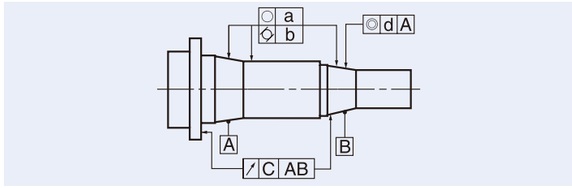


Table 8.5 Tolerances for and Mean Roughness of Shafts

Shaft Diameter (mm)		Tolerance Grades and Mean Roughness (μm)							
		Out-of-Roundness (○)		Cylindricity (⌭)	Runout (↗)		Coaxiality (◎)		Roughness
		a		b	c		d		R_a
		Bearing Accuracy		Bearing Accuracy	Bearing Accuracy		Bearing Accuracy		Bearing Accuracy
Over	Incl.	P5	P4, P3, P2	All accuracy classes	P5	P4, P3, P2	P5	P4, P3, P2	All accuracy classes
—	**10**	0.7	0.5	0.7	2	1.2	4	2.5	0.2
10	**18**	1	0.6	1	2.5	1.5	5	3	0.2
18	**30**	1.2	0.7	1.2	3	2	6	4	0.2
30	**50**	1.2	0.7	1.2	3.5	2	7	4	0.2
50	**80**	1.5	1	1.5	4	2.5	8	5	0.2
80	**120**	2	1.2	2	5	3	10	6	0.4
120	**180**	2.5	1.7	2.5	6	4	12	8	0.4
180	**250**	3.5	2.2	3.5	7	5	14	10	0.4
250	**315**	4	3	4	8	6	16	12	0.4
315	**400**	4.5	3.5	4.5	9	6.5	18	13	0.8
400	**500**	5	4	5	10	7.5	20	15	0.8

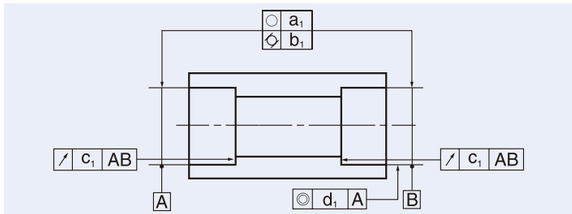
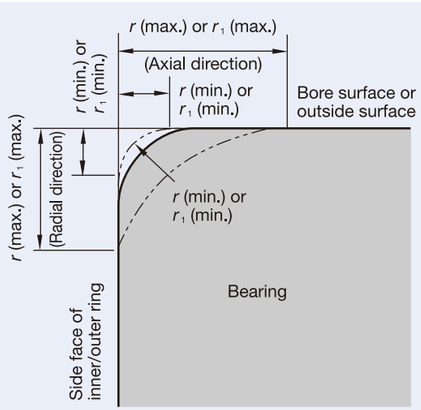


Table 8.6 Tolerances for and Mean Roughness of Housings

Housing Bore Diameter (mm)		Tolerance Grades and Mean Roughness (μm)							
		Out-of-Roundness (○)		Cylindricity (⌭)	Runout (↗)		Coaxiality (◎)		Roughness
		a_1		b_1	c_1		d_1		R_a
		Bearing Accuracy		Bearing Accuracy	Bearing Accuracy		Bearing Accuracy		Bearing Accuracy
Over	Incl.	P5	P4, P3, P2	All accuracy classes	P5	P4, P3, P2	P5	P4, P3, P2	All accuracy classes
10	**18**	1	0.6	1	2.5	1.5	5	3	0.4
18	**30**	1.2	0.7	1.2	3	2	6	4	0.4
30	**50**	1.2	0.7	1.2	3.5	2	7	4	0.4
50	**80**	1.5	1	1.5	4	2.5	8	5	0.4
80	**120**	2	1.2	2	5	3	10	6	0.8
120	**180**	2.5	1.7	2.5	6	4	12	8	0.8
180	**250**	3.5	2.2	3.5	7	5	14	10	0.8
250	**315**	4	3	4	8	6	16	12	1.6
315	**400**	4.5	3.5	4.5	9	6.5	18	13	1.6
400	**500**	5	4	5	10	7.5	20	15	1.6
500	**630**	5.5	4.5	5.5	11	8	22	16	1.6
630	**800**	6.5	5	6.5	12.5	9	25	18	1.6

Chamfer Dimension Limits and Corner Radius of Shaft or Housing

Fig. 8.2 Chamfer Dimensions



r : Chamfer dimension of inner/outer ring
r_1: Chamfer dimension of inner/outer ring (Front side)

Remarks: The precise shape of chamfer surfaces has not been specified but its profile in the axial plane must not intersect an arc of radius *r* (min.) or r_1 (min.) touching the side face of an inner ring and the bore surface, or the side face of an outer ring and the outside surface.

Table 8.7 Chamfer Dimension Limits for Radial Bearings (Excluding Tapered Roller Bearings)

Unit: mm

Smallest permissible chamfer dimension	Nominal bore diameter *d*		Largest permissible chamfer dimension		For reference
			r (max.) or r_1 (max.)		Corner radius of shaft or housing r_a
r (min.) or r_1 (min.)	Over	Incl.	Radial direction	Axial direction (1)	Max.
0.15	—	—	0.3	0.6	0.15
0.3	—	40	0.6	1	0.3
0.3	40	—	0.8	1	0.3
0.6	—	40	1	2	0.6
0.6	40	—	1.3	2	0.6
1	—	50	1.5	3	1
1	50	—	1.9	3	1
1.1	—	120	2	3.5	1
1.1	120	—	2.5	4	1
1.5	—	120	2.3	4	1.5
1.5	120	—	3	5	1.5
2	—	80	3	4.5	2
2	80	220	3.5	5	2
2	220	—	3.8	6	2
2.1	—	280	4	6.5	2
2.1	280	—	4.5	7	2
2.5	—	100	3.8	6	2
2.5	100	280	4.5	6	2
2.5	280	—	5	7	2
3	—	280	5	8	2.5
3	280	—	5.5	8	2.5
4	—	—	6.5	9	3
5	—	—	8	10	4
6	—	—	10	13	5

Table 8.8 Chamfer Dimension Limits for Tapered Roller Bearings

Unit: mm

Smallest permissible chamfer dimension	Nominal bore diameter or nominal outside diameter *d* or *D*		Largest permissible chamfer dimension *r* (max.)		For reference
					Corner radius of shaft or housing r_a
r (min.)	Over	Incl.	Radial direction	Axial direction (1)	Max.
0.6	—	40	1.1	1.7	0.6
0.6	40	—	1.3	2	0.6
1	—	50	1.6	2.5	1
1	50	—	1.9	3	1
1.5	—	120	2.3	3	1.5
1.5	120	250	2.8	3.5	1.5
1.5	250	—	3.5	4	1.5
2	—	120	2.8	4	2
2	120	250	3.5	4.5	2
2	250	—	4	5	2
2.5	—	120	3.5	5	2
2.5	120	250	4	5.5	2
2.5	250	—	4.5	6	2
3	—	120	4	5.5	2.5
3	120	250	4.5	6.5	2.5
3	250	400	5	7	2.5
3	400	—	5.5	7.5	2.5
4	—	120	5	7	3
4	120	250	5.5	7.5	3
4	250	400	6	8	3
4	400	—	6.5	8.5	3
5	—	180	6.5	8	4
5	180	—	7.5	9	4
6	—	180	7.5	10	5
6	180	—	9	11	5

(1) For inner ring chamfer, locate *d* in "Over"/"Incl." columns; for outer ring chamfer, locate *D* in "Over"/"Incl." columns.

Table 8.9 Chamfer Dimension Limits for Thrust Bearings

Unit: mm

Smallest permissible chamfer dimension	Largest permissible chamfer dimension *r* (max.)	For reference
		Corner radius of shaft or housing r_a
r (min.) or r_1 (min.)	Radial and axial direction	Max.
0.6	1.5	0.6
1	2.2	1
1.1	2.7	1
1.5	3.5	1.5
2	4	2
2.1	4.5	2
3	5.5	2.5
4	6.5	3
5	8	4
6	10	5
7.5	12.5	6
9.5	15	8
12	18	10
15	21	12
19	25	15

Spacer Dimensions

The dimensions of standard spacers for open angular contact ball bearings (19, 29, 10, 20 and 02 Series) are listed below.

Notes

- Spacer material: Bearing steel or carbon steel
- Reccomended spacer end face parallelism:
  Spacers with bores up to 300 mm: 0.003 mm or less
  Spacers with bores over 300 mm: 0.004 mm or less

19 or 29 Series

Standard Spacers for Dimension Series 19 or 29 (79, BNR19, BER19, BNR29, BER29, BSR19)

Unit: mm

Bore number	Nominal bearing bore diameter *d*	Bearing outside diameter *D*	Outer ring spacer		Inner ring spacer		Spacer chamfer
			Outside diameter (1)	Bore diameter	Outside diameter	Bore diameter (2)	
00	10	22	21.5	17.5	14.5	10.5	0.2
01	12	24	23.5	19.5	16.5	12.5	0.2
02	15	28	27.5	23.5	19.5	15.5	0.2
03	17	30	29.5	25.5	21.5	17.5	0.2
04	20	37	36.5	31.5	26	20.5	0.2
05	25	42	41.5	36	31	25.5	0.2
06	30	47	46.5	41	36	30.5	0.2
07	35	55	54.5	48	42	35.5	0.3
08	40	62	61.5	54.5	47.5	40.5	0.3
09	45	68	67.5	60	53	45.5	0.3
10	50	72	71.5	66	56	50.5	0.3
11	55	80	79.5	72	64	55.5	0.5
12	60	85	84.5	77	68	60.5	0.5
13	65	90	89.5	82	73	65.5	0.5
14	70	100	99.5	91.5	79	70.5	0.5
15	75	105	104.5	96.5	84	75.5	0.5
16	80	110	109.5	101.5	89.5	80.5	0.5
17	85	120	119.5	110	95	85.5	0.5
18	90	125	124.2	116	100	90.5	0.5
19	95	130	129.2	120	106	95.5	0.5
20	100	140	139.2	129	112	100.5	0.5
21	105	145	144.2	133	117	105.5	0.5
22	110	150	149.2	138	122	110.5	0.5
24	120	165	164.2	152	133	120.5	0.5
26	130	180	179.2	166	144	130.8	0.8
28	140	190	189.2	176	154	140.8	0.8
30	150	210	209.2	193	167	150.8	1.0
32	160	220	219.2	203	175	160.8	1.0
34	170	230	229.2	214	188	170.8	1.0
36	180	250	249.2	231	200	180.8	1.0
38	190	260	259.2	242	206	190.8	1.0
40	200	280	279.2	255	225	200.8	1.0
44	220	300	299.2	275	245	220.8	1.2
48	240	320	319.2	297	263	240.8	1.2
52	260	360	359.2	322	290	260.8	1.2
56	280	380	379.2	348	312	280.8	1.2
60	300	420	419.2	386	335	300.8	1.2
64	320	440	439.2	400	360	320.8	1.2
68	340	460	459.2	425	375	340.8	1.2
72	360	480	479.2	441	399	360.8	1.2
76	380	520	519.2	475	425	380.8	1.5
80	400	540	539.2	494	446	400.8	1.5

(1) When using oil mist lubrication or jet lubrication, we recommend that the outside diameter of the outer ring spacer is the same as the bearing outside diameter, with a tolerance of g5 or better.

(2) For high-speed operations exceeding 700 000 $d_m n$, we recommend that the bore diameter of the inner ring spacer is the same as the bearing bore diameter, with a tolerance of F6 or better.

9. Spacers

Notes

- Spacer material: Bearing steel or carbon steel
- Reccomended spacer end face parallelism:
  Spacers with bores up to 300 mm: 0.003 mm or less
  Spacers with bores over 300 mm: 0.004 mm or less

10 or 20 Series

Standard Spacers for Dimension Series 10 or 20 (70, BNR10, BER10, BNR20, BER20, BSR10, BNRD10, BERD10)

Unit: mm

Bore number	Nominal bearing bore diameter *d*	Bearing outside diameter *D*	Outer ring spacer		Inner ring spacer		Spacer chamfer
			Outside diameter (1)	Bore diameter	Outside diameter	Bore diameter (2)	
00	10	26	25.5	21.5	14.5	10.5	0.2
01	12	28	27.5	23.5	17	12.5	0.2
02	15	32	31.5	27	20	15.5	0.2
03	17	35	34.5	29.5	23	17.5	0.2
04	20	42	41.5	35	27	20.5	0.3
05	25	47	46.5	40.5	32	25.5	0.3
06	30	55	54.5	47.5	38	30.5	0.5
07	35	62	61.5	54	43	35.5	0.5
08	40	68	67.5	60	48	40.5	0.5
09	45	75	74.5	66	55	45.5	0.5
10	50	80	79.5	71	60	50.5	0.5
11	55	90	89.5	81	66	55.5	0.5
12	60	95	94.5	86	69	60.5	0.5
13	65	100	99.5	91	74	65.5	0.5
14	70	110	109.5	98	83	70.5	0.5
15	75	115	114.5	105	85	75.5	0.5
16	80	125	124.2	112	93	80.5	0.5
17	85	130	129.2	117	99	85.5	0.5
18	90	140	139.2	126	104	90.5	0.8
19	95	145	144.2	131	109	95.5	0.8
20	100	150	149.2	136	114	100.5	0.8
21	105	160	159.2	144	121	105.5	1.0
22	110	170	169.2	153	128	110.5	1.0
24	120	180	179.2	166	136	120.5	1.0
26	130	200	199.2	177	150	130.8	1.0
28	140	210	209.2	190	160	140.8	1.0
30	150	225	224.2	203	172	150.8	1.2
32	160	240	239.2	217	183	160.8	1.2
34	170	260	259.2	230.5	199.5	170.8	1.2
36	180	280	279.2	250	210	180.8	1.2
38	190	290	289.2	261	221	190.8	1.2
40	200	310	309.2	278	232	200.8	1.2
44	220	340	339.2	305	255	220.8	1.2
48	240	360	359.2	325	275	240.8	1.2
52	260	400	399.2	345	304	260.8	1.5
56	280	420	419.2	380	320	280.8	1.5
60	300	460	459.2	412	352	300.8	1.5
64	320	480	479.2	440	360	320.8	1.5
68	340	520	519.2	470	390	340.8	2.0
72	360	540	539.2	490	410	360.8	2.0
76	380	560	559.2	502	438	380.8	2.0
80	400	600	599.2	536	464	400.8	2.0

(1) When using oil mist lubrication or jet lubrication, we recommend that the outside diameter of the outer ring spacer is the same as the bearing outside diameter, with a tolerance of g5 or better.
(2) For high-speed operations exceeding 700 000 $d_m n$, we recommend that the bore diameter of the inner ring spacer is the same as the bearing bore diameter, with a tolerance of F6 or better.

02 Series

Standard Spacers for Dimension Series 02 (72, BSR02)

Unit: mm

Bore number	Nominal bearing bore diameter *d*	Bearing outside diameter *D*	Outer ring spacer		Inner ring spacer		Spacer chamfer
			Outside diameter (1)	Bore diameter	Outside diameter	Bore diameter (2)	
00	10	30	29.5	25	17	10.5	0.3
01	12	32	31.5	27	18	12.5	0.3
02	15	35	34.5	29	21	15.5	0.3
03	17	40	39.5	33	24	17.5	0.3
04	20	47	46.5	39	28	20.5	0.5
05	25	52	51.5	44	33	25.5	0.5
06	30	62	61.5	53	40	30.5	0.5
07	35	72	71.5	62	46	35.5	0.5
08	40	80	79.5	68	52	40.5	0.5
09	45	85	84.5	75	56	45.5	0.5
10	50	90	89.5	80	60	50.5	0.5
11	55	100	99.5	90	65	55.5	0.8
12	60	110	109.5	95	75	60.5	0.8
13	65	120	119.5	105	80	65.5	0.8
14	70	125	124.2	110	85	70.5	0.8
15	75	130	129.2	115	90	75.5	0.8
16	80	140	139.2	125	95	80.5	1.0
17	85	150	149.2	135	105	85.5	1.0
18	90	160	159.2	140	110	90.5	1.0
19	95	170	169.2	150	115	95.5	1.0
20	100	180	179.2	160	125	100.5	1.0
21	105	190	189.2	170	132	105.5	1.0
22	110	200	199.2	175	135	110.5	1.0
24	120	215	214.2	190	145	120.5	1.0
26	130	230	229.2	203	157	130.8	1.2
28	140	250	249.2	220	170	140.8	1.2
30	150	270	269.2	233	189	150.8	1.2
32	160	290	289.2	255	195	160.8	1.2
34	170	310	309.2	270	210	170.8	1.5
36	180	320	319.2	277	223	180.8	1.5
38	190	340	339.2	300	235	190.8	1.5
40	200	360	359.2	320	250	200.8	1.5

(1) When using oil mist lubrication or jet lubrication, we recommend that the outside diameter of the outer ring spacer is the same as the bearing outside diameter, with a tolerance of g5 or better.
(2) For high-speed operations exceeding 700 000 $d_m n$, we recommend that the bore diameter of the inner ring spacer is the same as the bearing bore diameter, with a tolerance of F6 or better.

Nozzle Position

The following table lists nozzle positions for oil-air, oil-mist, and oil-jet lubrication systems.

Unit: mm

Bore number	Nominal bearing bore	79 Series		70 Series		72 Series		N10XXMR (Standard Series)		N10XXR (NSKROBUST Series)	
		ϕA (1)	B	ϕA (1)	B	ϕA	B	ϕA	B	ϕA	B
00	10	14.5	0.4	16.1	0.5	18.1	0.5	–	–	–	–
01	12	16.5	0.4	18.3	0.5	19.6	0.5	–	–	–	–
02	15	20.0	0.5	21.3	0.5	22.6	0.7	–	–	–	–
03	17	21.8	0.5	23.5	1.0	25.9	0.7	–	–	–	–
04	20	26.1	0.5	28.2	1.0	30.5	1.0	–	–	–	–
05	25	31.1	0.5	32.9	1.0	35.5	1.0	–	–	–	–
06	30	36.1	0.5	39.5	1.0	41.8	1.0	39.7	1.2	–	–
07	35	42.6	0.5	44.6	1.0	48.6	0.7	45.4	1.5	–	–
08	40	47.9	0.5	50.0	1.0	54.6	0.7	50.6	1.5	–	–
09	45	53.4	0.5	55.6	1.0	59.4	0.7	56.5	2.0	60.0	1.2
10	50	57.9	0.5	60.6	1.0	64.4	1.0	61.5	2.0	64.5	1.3
11	55	64.0	0.5	67.3	1.0	70.8	1.0	69.2	2.5	71.0	1.2
12	60	69.0	0.5	72.5	1.0	77.4	0.7	74.3	2.5	76.5	1.2
13	65	74.0	0.5	77.5	1.0	84.6	0.7	79.2	2.5	81.5	1.2
14	70	80.9	0.7	83.7	1.0	89.4	0.7	86.6	3.0	89.0	1.5
15	75	85.5	0.7	89.4	1.0	94.5	0.7	90.0	2.5	94.5	1.5
16	80	90.5	0.7	96.5	1.0	101.4	0.7	98.5	3.0	101.0	2.0
17	85	98.8	0.7	101.5	1.0	109.8	1.0	103.5	3.0	106.0	2.0
18	90	102.8	0.7	108.6	1.0	116.7	1.0	109.0	3.0	–	–
19	95	107.7	0.7	113.3	1.0	123.6	1.0	115.5	2.5	–	–
20	100	116.0	0.7	118.6	1.0	130.6	1.0	119.0	2.5	–	–
21	105	119.5	0.7	125.1	0.7	137.4	1.0	125.5	3.0	–	–
22	110	124.5	0.7	131.9	0.7	144.4	1.0	134.0	3.0	–	–
24	120	136.3	0.7	142.3	0.7	156.3	1.0	142.0	3.0	–	–
26	130	149.3	0.7	156.2	1.0	168.9	1.0	156.1	4.5	–	–
28	140	158.1	0.7	165.7	2.5	182.6	1.0	168.0	4.5	–	–
30	150	171.8	0.7	178.1	2.5	196.5	1	–	–	–	–
32	160	181.8	0.7	190.4	2.5	210	1	190	5	–	–
34	170	191.8	0.7	203.4	2.5	223	1	203	5	–	–
36	180	205.6	0.7	217.1	2.5	233	1	–	–	–	–
38	190	215.4	0.7	227.1	2.5	248	1	–	–	–	–
40	200	229	0.7	240.9	2.5	262	1	242	6.5	–	–
44	220	249	0.7	264.3	5	290	1.5	–	–	–	–
48	240	271.4	0.7	287	5	320	2	–	–	–	–

(1) ϕA of 79 and 70 Series bearings are in compliance with DIN Standard 628-6.

Unit: mm

Nominal bearing bore	BNR19 BER19		BNR10 BER10 BNRD10 BERD10		BSR19		BSR10		BSR02		BAR10 BTR10	
	ϕA	B	ϕA	B	ϕA	B	ϕA	B	ϕA	B	ϕA	B
6	–	–	–	–	–	–	9.0	0.4	–	–	–	–
7	–	–	–	–	–	–	10.5	0.4	–	–	–	–
8	–	–	–	–	–	–	12.0	0.5	–	–	–	–
10	–	–	–	–	13.5	0.4	14.5	0.5	17.0	1.0	–	–
12	–	–	–	–	15.5	0.4	16.5	0.5	18.0	0.5	–	–
15	–	–	–	–	18.5	0.5	20.0	1.0	21.0	1.0	–	–
17	–	–	–	–	20.5	0.5	22.5	1.5	24.0	0.5	–	–
20	–	–	–	–	25.0	0.8	26.5	0.8	28.3	0.5	–	–
25	31.0	0.5	–	–	30.0	0.8	31.5	0.8	33.2	1.0	–	–
30	35.5	0.5	39.0	1.0	–	–	–	–	–	–	–	–
35	42.0	0.5	44.5	1.2	–	–	–	–	–	–	–	–
40	48.0	0.5	50.0	1.5	–	–	–	–	–	–	–	–
45	53.0	0.5	55.5	1.7	–	–	–	–	–	–	–	–
50	57.5	0.5	60.5	1.7	–	–	–	–	–	–	60.5	0.9
55	63.5	0.5	67.5	1.5	–	–	–	–	–	–	67.5	0.7
60	68.5	0.5	73.0	1.5	–	–	–	–	–	–	73.5	0.7
65	73.5	0.5	77.5	1.5	–	–	–	–	–	–	77.5	0.7
70	80.5	0.7	84.0	1.7	–	–	–	–	–	–	84.0	0.7
75	85.0	0.7	89.0	1.7	–	–	–	–	–	–	89.0	0.7
80	90.5	0.7	96.0	1.7	–	–	–	–	–	–	96.0	0.9
85	98.5	0.7	102.0	1.7	–	–	–	–	–	–	102.0	0.9
90	102.0	0.7	109.0	1.7	–	–	–	–	–	–	108.5	1.2
95	107.0	0.7	112.0	1.7	–	–	–	–	–	–	112.5	1.2
100	113.5	0.7	118.5	2.5	–	–	–	–	–	–	118.5	1.7
105	119.0	0.7	125.0	1.7	–	–	–	–	–	–	126.0	1.4
110	124.0	0.7	132.5	1.7	–	–	–	–	–	–	132.5	1.2
120	136.0	0.7	143.0	1.7	–	–	–	–	–	–	142.5	1.2
130	149.0	0.7	156.5	1.7	–	–	–	–	–	–	155.5	1.7
140	157.5	0.7	166.0	1.7	–	–	–	–	–	–	167	1.7
150	171.5	0.7	178.5	1.7	–	–	–	–	–	–	179.5	1.9
160	181.8	0.7	190	2	–	–	–	–	–	–	190	2.0
170	191.8	0.7	–	–	–	–	–	–	–	–	205	1.7
180	205.6	0.7	–	–	–	–	–	–	–	–	218	2.0
190	215.4	0.7	–	–	–	–	–	–	–	–	228	2.0
200	229	0.7	–	–	–	–	–	–	–	–	242	2.2

Notes

- Positioning the nozzle parallel to the spindle axis is sufficient for normal operating speeds. For bearings usually operated at high speeds, it is advantageous to position the nozzle bore so that the lubricant is directed into the bearing at an angle of about 15° to 20°.
- Appropriate oil drain is necessary to prevent oil from accumulating in the spindle, which leads to increased heat in sliding sections and damages bearings.
- We recommend filtering the oil to 5 micron or less before it enters the lubrication system. Refer to page 220 for an example of an oil-air lubrication system.

Fig. 9.1 Angle of Spray Nozzle

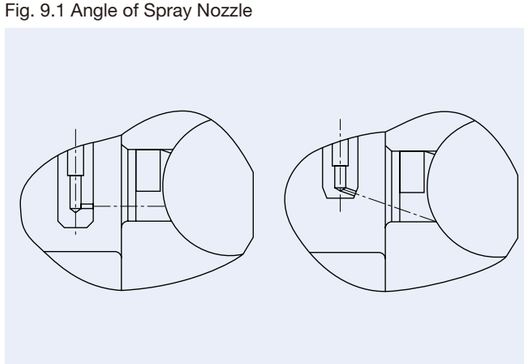
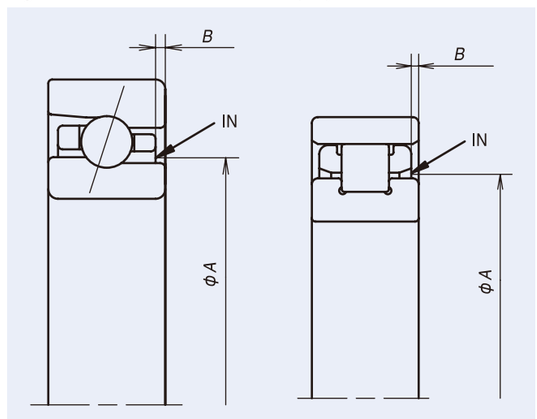


Fig. 9.2 Position and Direction of Spray Nozzle





Part 6

Bearing Handling

1. Mounting ·····244-265
1. Cleaning Bearings and Related Parts
2. Checking Dimensions of Related Parts
3. Mounting Procedures
4. Inspection After Mounting
2. Operational Inspection ·····266-267
3. Run-in Procedures ·····268-269

Mounting

How rolling bearings are mounted strongly affects their accuracy, life, and performance. We recommend that designers and engineers fully analyze the handling procedures for bearings and establish standards with regard to the following procedures:

(1) Cleaning the bearings and related parts
(2) Checking the dimensions of related parts
(3) Mounting procedures
(4) Inspection after mounting

Precautions for Proper Handling of Bearings

Super precision bearings are highly accurate, high-quality machine parts and must be handled accordingly. Even if high-performance, high-accuracy bearings are used, their expected performance cannot be achieved if they are not handled properly. The main precautions to be observed are as follows:

■ Keep bearings and surrounding area clean

Foreign particles, even if invisible to the naked eye, have harmful effects on bearings. Take care to prevent the ingress of dirt and debris into the bearing by maintaining a clean working environment.

■ Handle bearings carefully

Avoid any heavy shocks during handling. Shock loads can scratch or dent a bearing, possibly resulting in failure. Excessive impacts may cause brinelling or cracks. Even very small damage can lead to noise and should be avoided.

■ Use proper tools

Always use the proper equipment when handling bearings and avoid general-purpose tools.

■ Prevent corrosion

Handling bearings with bare hands can corrode the bearing surfaces because of perspiration or various other contaminants on the skin. Wear gloves if possible. Take measures to prevent bearings rusting due to corrosive gasses.

Storage Method

■ Although bearings are coated with an anti-corrosion agent and then wrapped and packed, it is impossible to completely avoid exposure to the air surrounding the bearings. Store the bearings in a dry location and avoid exposure to moisture and humidity.

■ Bearings should be stored in a clean, dry, and well-ventilated location that also provides protection from direct sunlight. Store the bearings in a locker or on shelves that are at least 30 cm from the floor.

■ When bearings are unpacked for acceptance inspection, take measures to prevent rust and contamination. After inspection, follow the guidelines above to ensure proper storage.

Section Outline

(See the relevant sections on subsequent pages for further details.)

1. Cleaning the Bearings and Related Parts

Bearings should not be unpacked until immediately before mounting. Delivered bearings are coated with an anti-corrosion agent to protect against dust and rust during transport. In general, bearings for high-speed operation should be cleaned with clean filtered solvent to remove the anti-corrosion agent. That applies in particular to bearings that need packing with grease; any anti-corrosion agent remaining on the raceway surfaces may lead to grease outflow. After the bearings are cleaned with filtered solvent, they should be protected to prevent corrosion. Refer to pages 246-247 for a detailed description of the cleaning process.

Some products such as bearings prepacked with grease or sealed bearings do not require cleaning. Instead wipe off the anti-corrosion agent from the outside surface with a clean cloth before assembly.

Shafts and housings should be cleaned and the mating surfaces (bearing seats, spacer seats) checked for flaws or burrs.

2. Checking the Dimensions of Related Parts

Inspection of shaft and housing

■ Verify that the shaft and housing dimensions match the fit with the bore and outside diameter of the bearing. See p. 232-234 for recommended fits.

■ Perform measurements and mounting in a temperature-controlled environment. After parts have reached a stable temperature, use a micrometer or cylinder gauge at several points to ensure there are no significant differences in measurements. See p. 235 for recommended tolerances.

■ We recommend that the parallelism of spacers arranged on a spindle be controlled to 0.003 mm or less (0.004 mm or less for bores over 300 mm). Poor parallelism may result in bearing misalignment, causing accuracy issues and noise.

■ When using cylindrical roller bearings with tapered bores, be sure to measure all dimensions before applying lubrication (even when using grease) to properly control radial clearance/preload. Prepare spacers accordingly for the adjustment of radial clearance. See p. 249 for details on measurement and adjustment methods.

3. Mounting Procedures

When using grease lubrication, pack bearings with grease before mounting. With oil-air and oil mist lubrication, mount bearings after cleaning and degreasing.

Procedures for mounting vary according to the fit requirements of the inner and outer rings.

Typically, the inner ring rotates in machine tool bearings, so bearings with cylindrical bores are usually mounted by heating them to expand the inner ring (shrink fit). Bearings with tapered bores can be mounted directly onto a tapered shaft.

Outer rings are usually mounted with some clearance. The housing can be heated to make mounting easier. In cases where outer rings are to be mounted with interference fit, press fitting is feasible and may be combined with the housing heating method.

4. Inspection After Mounting

After mounting the bearings, check shaft runout (see p. 263) and preload (p. 264) Then, correct shaft unbalance and begin run-in procedures.

Shaft unbalance can be static or dynamic (p. 266). To check for static unbalance, position the shaft horizontally on a smooth rail or jig. If the distribution of mass is uneven, the heavier side will face down. Static unbalance can be corrected by adding or removing mass in one plane.

Dynamic unbalance can only be detected using specialized equipment when the shaft is rotating. Corrections are made in two planes, at the front and rear of the shaft.

Run-in is a process that gradually brings a spindle up to stable rotation conditions (p. 268). Run-in is particularly important for grease-lubricated spindles as it expels excess grease while ensuring grease is evenly distributed.

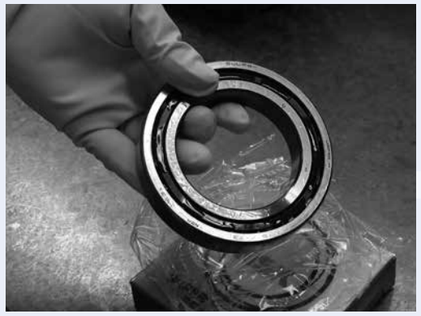
See p. 275 for a list of common operating irregularities, causes, and countermeasures.

1. Mounting

1. Cleaning the Bearings

After preparing the shaft and surrounding components, follow this guide to properly clean and handle bearings before mounting.

1 Take the bearing out of its package.



2 Without rotating it, pour cleaning oil over the bearing to remove the anti-corrosion agent.



3 Rotate the bearing while allowing cleaning oil to flow between the inner and outer ring to remove the anti-corrosion agent inside. Make sure the bearing runs smoothly after cleaning.

4 Degrease by rinsing the bearing with clean organic solvent.



Cleaning Complete

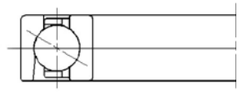
Handle bearings with care

The way you hold angular contact ball bearings is important: When holding the inner ring, make sure the bearing's outer ring back face is facing up. When holding the outer ring, make sure the bearing's outer ring front face is facing up.

When holding the inner ring

Outer ring back face

Top

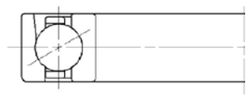


Bottom

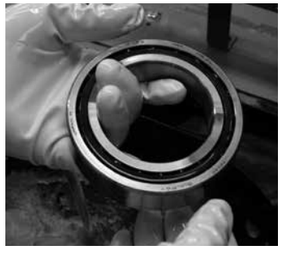
When holding the outer ring

Outer ring front face

Top



Bottom





Cleaning device and cleaning oil

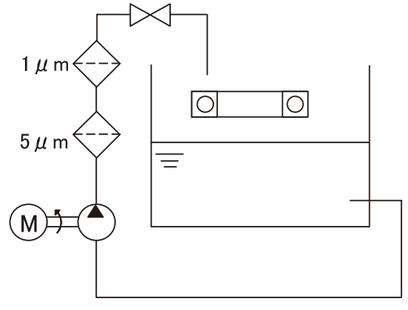
Always use a circulating filtration system to filter cleaning oil.

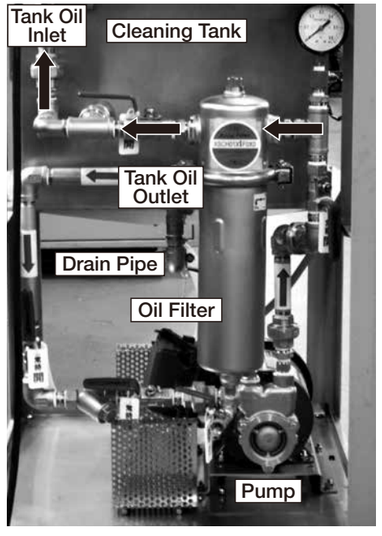
A pump is used to circulate the cleaning oil in the cleaning tank. The oil drain pipe is equipped with an oil filter that must be changed at regular intervals to prevent clogging.

Two-step filtration will efficiently remove large and small particles and extend the service life of the filters.

Replace the cleaning oil at regular intervals.

Circulating filtration system (Example)





Note: Shown here is an example of a filtration system with one oil filter.

Degrease with caution

(1) After using organic solvent for degreasing, do not re-use the waste fluid.

(2) Do not rotate bearings during or after contact with degreasing solvent as this may damage the raceways and balls.

(3) Avoid the use of compressed air as particles suspended in the surrounding air may enter the bearing.

(4) For grease lubrication, pack bearings with grease after degreasing. For oil lubrication, apply clean lubrication oil to the bearing interior prior to mounting. Alternatively, coat the bearing by rinsing with lubricating oil diluted with degreasing solvent (oil plating method).

Precautions for handling kerosene and organic solvents

When working with kerosene and organic solvents, be aware of the hazards of fire and toxicity. Always take appropriate measures in line with applicable industrial safety laws and regulations to prevent accidents. Some simple precautions include grounding cleaning equipment to prevent sparks from static electricity and ensuring adequate ventilation.

Note that this content is meant as an aid for cleaning methods and NSK assumes no responsibility for any associated risks.

2. Checking the Dimensions of Related Parts

2.1 Gauging and adjusting tapered shafts

For precision cylindrical roller bearings to perform as expected, proper mounting is essential.

If bearings are mounted with a mismatch between the taper angle of the bearing bore and the taper angle of the shaft, inner ring expansion will be uneven along the spindle axis, which will impede bearing performance during operation. This detrimental impact increases in double-row cylindrical roller bearings due to their wider inner ring.

Fig 1.1 illustrates how uneven inner ring expansion during fitting caused by a mismatch of taper angle leads to a difference in radial clearance between the two rows. A large imbalance of load capacity ensues that results in lower rigidity; misaligned inner ring grooves result in irregular movement of the rollers, etc.

Even with single-row cylindrical roller bearings, misalignment of inner ring grooves causes unevenness of the contact points (running traces) of the rollers with the inner and outer ring, resulting in irregular roller motion.

Actual examples of such defects are discussed on page 272.

■ Taper angle

Please refer to pages 226-227 for taper angle values of cylindrical roller bearings with tapered bores.

■ Checking the shaft taper angle

We recommend matching the shaft taper with the bore taper of the actual bearing. as follows:

- Clean and degrease the tapered bore of the bearing and the tapered section of the shaft, then coat either tapered surface with blue dye very thinly and evenly.
- Push the bearing inner ring onto the shaft.
- Slightly rotate the inner ring, then withdraw the inner ring carefully from the shaft without rotating it (Fig. 1.2).
- Assess the blue area. The blue dye will have adhered to the shaft where the inner ring and shaft taper make contact.
- The area of the tapered shaft dyed blue should cover at least 80% of the bearing width.

  If the contact area is insufficient, modify the shaft taper accordingly.

  Red lead or other appropriate marking compounds may be used in place of blue dye.

■ Types of taper contact

When checking the taper angle as described, uniform taper contact along the entire bearing bore surface is ideal, but this in rare in practice due to limitations in machining accuracy.

In Fig. 1.3, the shade of blue indicates the intensity of contact between bearing bore and shaft.

Contact at the large end of the shaft taper (a) means that contact is strongest in the area where the inner ring is thinnest. Therefore, row A will expand to a larger degree, and row B to a smaller degree. The result is a large imbalance of radial clearance, as shown in Fig. 1.1.

We recommend contact at the small end of the shaft taper (b) since the thick wall of the inner ring means row B is less likely to expand and the effect of taper mismatch can be absorbed more easily.

Fig. 1.1 Unevenness of Radial Clearance Due to Taper Mismatch

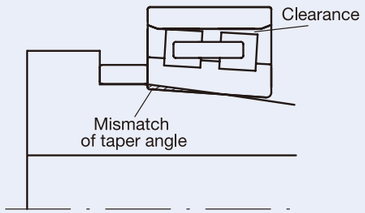


Fig. 1.2 Confirmation of Taper Angle with Dye

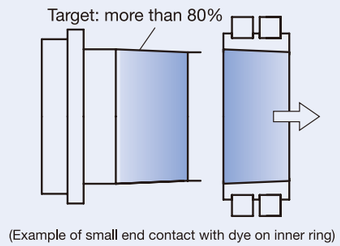
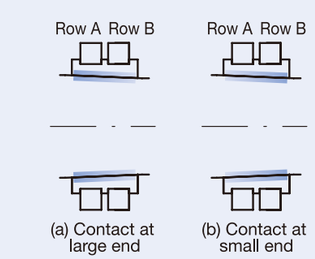


Fig. 1.3 Types of Taper Contact



2.2 Adjusting the radial clearance of cylindrical roller bearings

The internal radial clearance (preload) of cylindrical roller bearings is adjusted by pushing the inner ring up the tapered shaft, thereby expanding the inner ring. This process can be divided in four steps as explained below (Fig. 1.4).

Step 1: Push inner ring onto tapered shaft
Step 2: Measure radial clearance
Step 3: Measure the distance between inner ring and shaft shoulder
Step 4: Adjust spacer width as required to achieve target radial clearance

Step 1: Push inner ring onto tapered shaft

After cleaning and degreasing, slide the inner ring onto the shaft. Mount the inner ring by giving it a light tap using a spacer (Fig. 1.5). Tighten the shaft locknut. For grease-lubricated bearings, do not apply grease yet.

Please refer to pages 246-247 for details on bearing cleaning.

Step 2: Measure radial clearance

Radial clearance is measured either (1) using a special instrument called a GN gauge or (2) without a GN gauge (calculation method). Descriptions of each measuring process start on page 250 and page 252, respectively.

Step 3: Measure the distance between inner ring and shaft shoulder

Using gauge blocks or feeler gauges etc., measure the distance L from the inner ring end face to the shaft shoulder (Fig. 1.6).

Take three or more measurements around the circumference and use the average in order to account for possible shaft misalignment during adjustment and for the accuracies of surrounding parts.

Step 4: Adjust spacer width as required to achieve target radial clearance

Based on the calculation of the width measured in step 3 and the other measurement values, determine the spacer width that will obtain the targeted radial clearance. Finish the spacer to this width.

Fig. 1.4 Adjusting the Radial Clearance of Cylindrical Roller Bearings

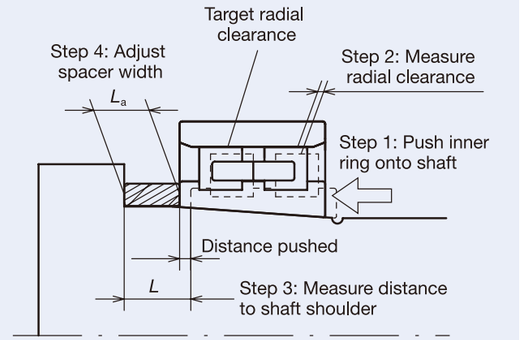


Fig. 1.5 Pushing the Inner Ring in Place

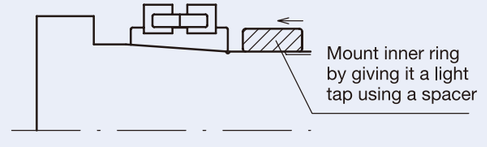
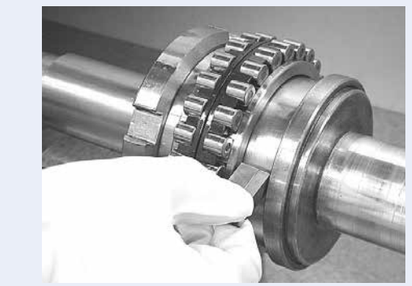
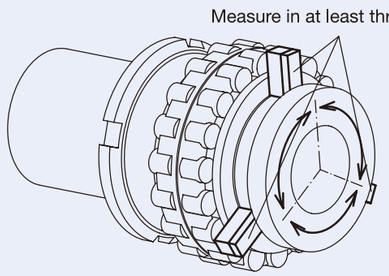


Fig. 1.6 Measuring the Required Spacer Width





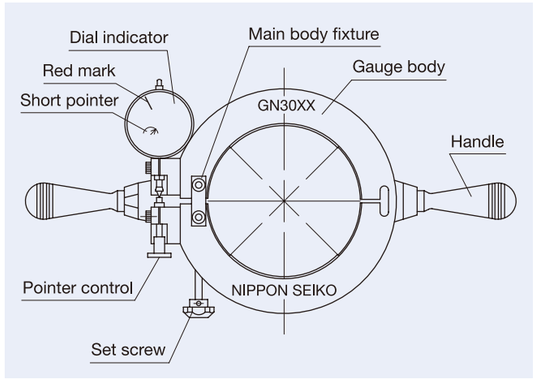
1. Mounting

(1) Measuring Radial Clearance of Cylindrical Roller Bearings (GN Gauge Method)

When mounting a cylindrical roller bearing with a tapered bore onto a machine tool spindle, it is essential to accurately match the shaft taper with that of the bearing and to ensure that the desired radial internal clearance is attained after mounting. A GN gauge is a special instrument that allows precise measurement and control of mounted radial clearance during this process. GN gauges are especially effective when cylindrical roller bearings are used with radial preload.

Fig. 1.7 describes the components of GN gauges.

Fig. 1.7 Components of a GN Gauge



How to use a GN gauge

① Insert outer ring into housing

Recommended values for the fit between the outer ring and housing are listed on page 233.

② Set cylinder gauge to zero

Ensure that the temperatures are the same for the outer ring (inserted into the housing), the inner ring, and the shaft. Then, measure the bore diameter of the outer ring at about four different locations using a cylinder gauge (dial bore gauge) and set the dial of the cylinder gauge to zero using the average value of the measurements (Fig. 1.8).

Fig. 1.8 Setting the Cylinder Gauge to Zero



③ Adjust the bore diameter of the GN gauge

Loosen the bolt of the main body fixture on the GN gauge. Place the cylinder gauge in the bore of the GN gauge and adjust the set screw until the dial on the cylinder gauge reads zero. The idea is to set the bore of the GN gauge to replicate that of the outer ring. Use the GN gauge in an upright position during this process to avoid inaccuracies in the cylindricity of the GN gauge bore due to its own weight (Fig 1.9).

Fig. 1.9 Adjusting the GN Gauge Bore



④ Correct the GN gauge

Next, use the pointer control on the dial indicator to adjust the main pointer to the red mark for gauge correction, as shown in Fig. 1.10. Confirm that the short pointer is close to 2 on the secondary dial.

Gauge correction compensates for the elastic deformation of the rollers caused by the measuring pressure of the GN gauge. The amount of correction is different for each GN gauge. It is determined when the gauge is shipped.

⑤ Set the GN gauge

Expand the GN gauge by adjusting the set screw. Place centrally over the inner ring rollers (Fig 1.11) and release the set screw to allow the GN gauge to spring closed over the rollers (so that contact deformation is applied to the rollers by the elasticity of the GN gauge only).

⑥ Read the dial

Read the main pointer on the dial indicator. The geometrical structure of the GN gauge (Fig 1.12) results in the amount shown on the dial gauge being twice the deviation from zero at measuring point A; in other words, the actual amount of clearance or preload is ½ the reading.

Example 1:
A reading of 4 (clockwise) indicates a radial clearance of +2 µm

Example 2:
A reading of 2 (anti-clockwise) indicates a radial clearance of -1 µm

⑦ Readjust the amount the inner ring is pushed up the shaft

As in ⑤, widen the GN gauge by turning the set screw until the dial shows play. Take the GN gauge off the inner ring. Tighten the shaft locknut. Then measure radial clearance again. Repeat these steps until the GN gauge indicates the target clearance value.

⑧ Measure the required spacer width

Measure the distance between the inner ring end face and the shaft shoulder as described on page 249, step 3.

⑨ Adjust the spacer width

Finish the spacer width to the dimension obtained in ⑧.

Fig. 1.10 Correcting the GN Gauge



Fig. 1.11 Setting the GN Gauge

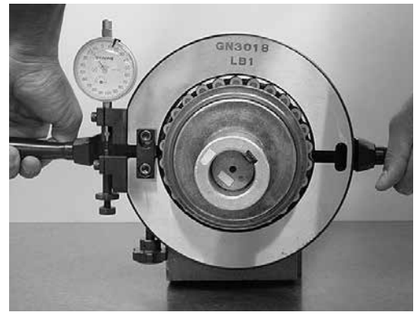
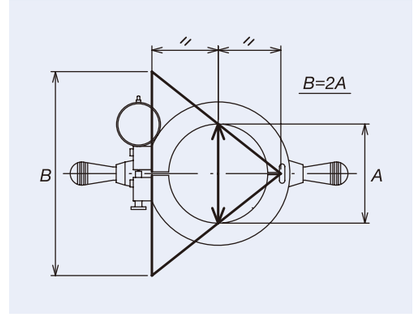


Fig. 1.11 Geometrical Structure of the GN Gauge



(2) Adjusting Radial Clearance of Cylindrical Roller Bearings Without a GN Gauge (Calculation Method)

This section briefly explains how to measure radial clearance without a GN gauge and how to calculate the spacer width L_a required to obtain the target mounted clearance.

When not using a GN gauge, the outer ring needs to be measured before inserting it into the housing. Furthermore, the amount of shrinkage of the outer ring raceway diameter due to fitting must be taken into account when determining the required dimension of the spacer for clearance adjustment.

①Based upon measurement of the housing bore and the dimension of the bearing outer ring, calculate the amount of shrinkage of the outer ring raceway diameter owing to the fit of the outer ring with the housing, Δr_e. If the outer ring is mounted with positive clearance, there is no need for this calculation (in this case, $\Delta r_e = 0$) (Fig. 1.13).

$$\Delta r_e = (D_h\text{-}D) \times H$$

(if this value is positive, a value of $\Delta r_e = 0$ is adopted)

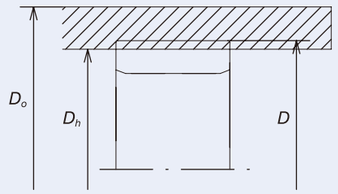
Table 1.1 Housing Wall Thickness and Shrinkage Factor *H*

Housing wall thickness ratio h_O (=D_O/D_h)	Shrinkage factor *H*
110%	0.32
120%	0.47
130%	0.56
140%	0.61
150%	0.65
160%	0.68
170%	0.70
180%	0.71

②Place the outer ring over the rollers. Apply a dial gauge to the outside diameter of the outer ring. Then lightly tighten the shaft locknut (Fig. 1.14 above).

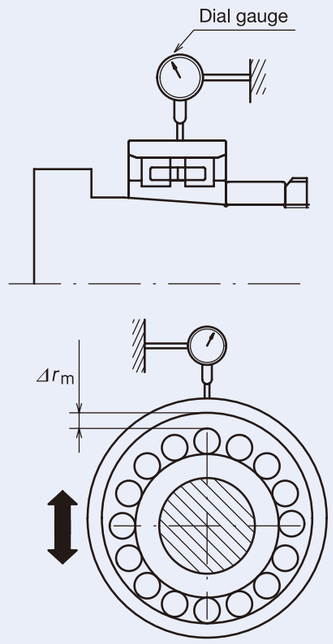
③Moving the outer ring up and down by hand, measure the free radial play Δr_m with the dial gauge.
Body temperature may increase the temperature of the outer ring, resulting in an erroneous measurement. Therefore we recommend you wear gloves and finish the measurement quickly (Fig. 1.14 below).

Fig 1.13 Explanation of Variables



D_o : Outside diameter of housing
D_h : Housing bore
D : Outside diameter of outer ring (Refer to bearing inspection sheet)
H : Outer ring raceway shrinkage factor

Fig 1.14 Measuring the Free Radial Play of the Outer Ring



④Repeat ② and ③ until Δr_m = 5 µm
Note: We adopted 5 µm as a guideline because if the free radial play Δr_m is too large, deformation of the outer ring due to measuring force may result in an incorrect measurement. Smaller values are acceptable if the target radial clearance Δr is negative. However, consider that the radial clearance will be further reduced due to shrinkage of the outer ring by Δr_e as determined in ①, so the shaft locknut should not be tightened excessively (Fig. 1.15).

⑤Measure the distance between the inner ring end face and the shaft shoulder *L* as described on page 249, step 3.

⑥Using the following equations, determine the inner ring expansion Δr_i required to achieve the target radial clearance Δr, as well as the necessary distance ΔL_i that the inner ring needs to be pushed (drive-up distance) to achieve expansion Δr_i (Fig. 1.16).

$$\Delta r_i = \Delta r_m + \Delta r_e - \Delta r$$
$$\Delta L_i = K\,\Delta r_i$$

Table 1.2 Hollow Shaft Ratio k_O and Coefficient *K*

Hollow shaft ratio k_O (=d_O/d)	Coefficient *K*	Hollow shaft ratio k_O (=d_O/d)	Coefficient *K*
0%	14.1	50%	15.2
10%	14.1	60%	16.0
20%	14.2	70%	17.4
30%	14.4	80%	20.2
40%	14.7		

⑦Calculate the required length L_a for the clearance adjustment spacer according to the results obtained in ⑤ and ⑥:

$$L_a = L - \Delta L_i$$

Fig 1.15 Deformation of the Outer Ring Due to Measuring Force

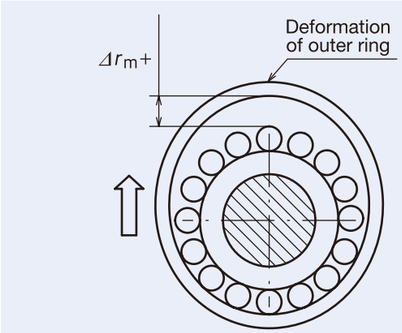
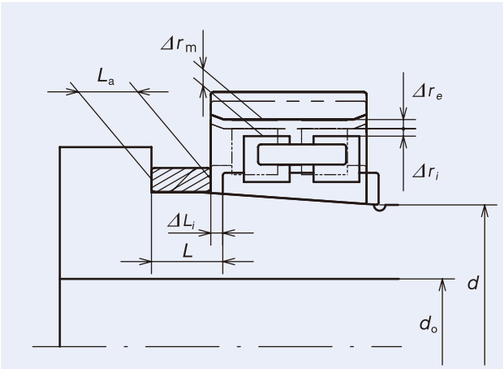


Fig 1.16 Explanation of Variables



Example calculation
To set radial clearance after mounting to Δr = -0.002 mm for NN3020MBKR:

$h_O = D_O/D$ = 225/150 = 150% ⇒ H = 0.65

Shrinkage of outer ring raceway diameter due to fit Δr_e = (149.993-149.998) × 0.65 = -0.003 mm

Inner ring expansion necessary for target clearance Δr_i = 0.005-0.003- (-0.002) = 0.004 mm

$k_O = d_O/d$ = 50/100 = 50% ⇒ K = 15.2

Inner ring drive-up distance ΔL_i =15.2×0.004 = 0.061 mm

Width of clearance adjustment spacer L_a = 20.55-0.061 = 20.489 mm

Values used in calculation example (mm)

Δr	-0.002
Δr_m	0.005
D_o	225
D	149.998
Dh	149.993
d	100
d_o	50
L	20.55

2.2 Securing the Outer Ring

A retaining cover held by bolts is generally used to secure the bearing outer ring axially. If a bolt is tightened excessively or a combination of bolts is tightened unevenly, the bearing outer ring may become deformed. Therefore, the dimensions of the surrounding parts should be checked and the clearance between retaining cover and housing controlled prior to mounting the bearings into the housing.

Measure the bearing width, as well as the width of the housing section where the bearing seats are and the amount the retaining cover projects (stick-out); if necessary, machine the retaining cover to adjust the tightening between the retaining cover and housing (Fig. 1.17). In a DB arrangement, we recommend to adjust the clearance between the retaining cover and housing end face at this point to about 0.01 to 0.05 mm according to the bearing size as listed in Table 1.5 on page 261 before the bolts are completely tightened.

Figures 1.18 through 1.20 show examples of deformation of the outer ring raceway caused by securing the outer ring.

Fig. 1.17 Adjusting the Retaining Cover

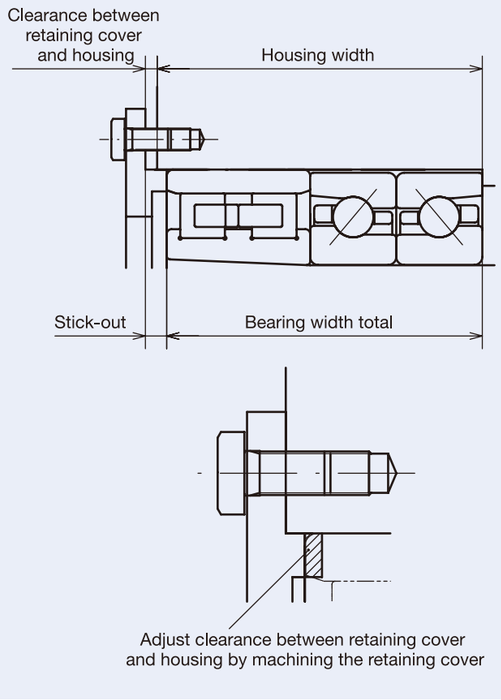


Fig. 1.18 Example Outer Ring Deformation Due to Clearance Between Retaining Cover and Housing

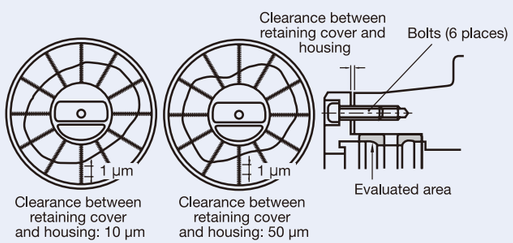


Fig. 1.19 Outer Ring Deformation Caused by Tightening of Retaining Cover

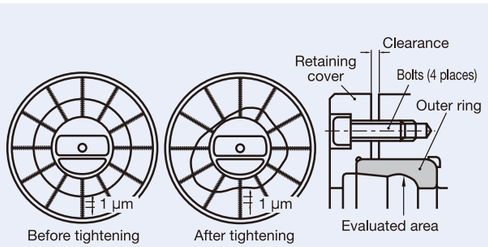
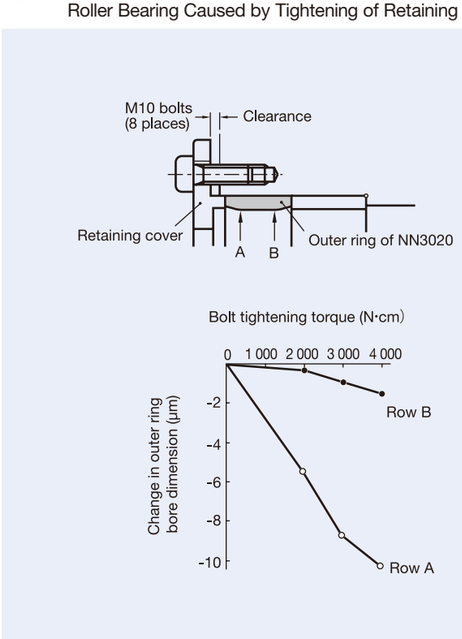


Fig. 1.20 Outer Ring Deformation of a Double-Row Cylindrical Roller Bearing Caused by Tightening of Retaining Cover



Note regarding adjustments with DF arrangements

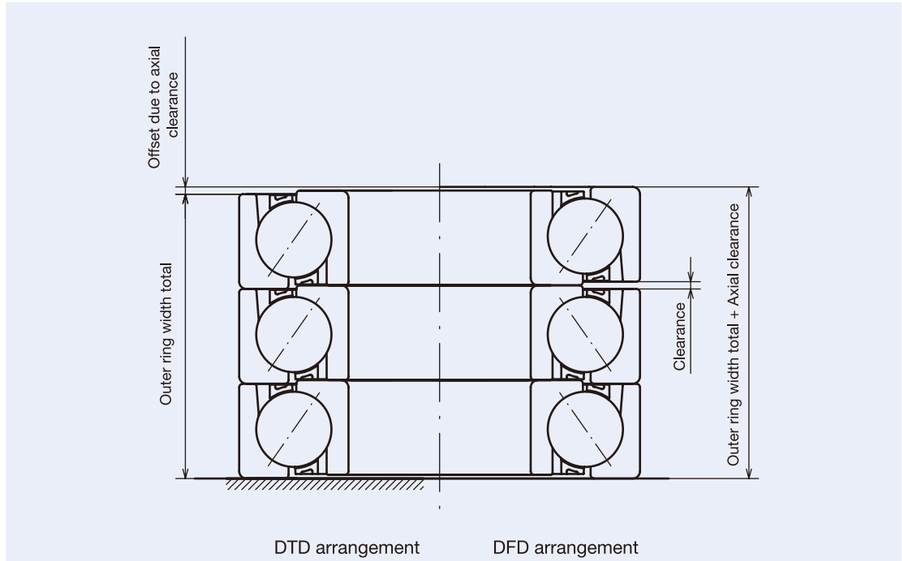
When using bearings in DF arrangements (including DFD, DFF, DFT, etc.), there is axial clearance on the outer rings before the outer rings are secured. Therefore, the outer ring width total cannot simply be measured as such. Some methods of working out the exact outer ring width are:

① Measure the outer ring width of each bearing individually
Note when using universal matching bearings, the outer ring width of each bearing is listed on the bearing inspection sheet, so there is no need for measurement.

② Rearrange bearings to have no axial clearance and measure (e.g.. use DT, DTT, etc.)
With this method, take extra care when mounting not to mistake the proper order or direction of the bearings.

③ Measure outer ring width of the arrangement, then correct for the axial clearance amount
For matched bearings, axial clearance is listed on the bearing inspection sheet. Subtract this value after measuring the outer ring width of the arrangement.

Fig. 1.21 Influence of Arrangement Type on Measurement of Outer Ring Width Total



3. Bearing Mounting Procedures

3.1 Packing grease

After selecting a grease suited to operating conditions, it needs to be properly packed into the bearing. Packing bearings incorrectly or with too much grease may result in a much longer run-in period or cause unusual temperature rise leading to seizure. Follow proper procedures for packing and using appropriate grease amounts as follows:

1. Pre-Inspection

Check to ensure there is no foreign matter in the bearing interior. Bearings for high-speed spindles should be cleaned and degreased before packing with grease. For other applications, it is advisable to remove the anti-corrosion agent adhering to interior surfaces of the bearings to prevent grease outflow.

2. Grease dispensers

Use a grease dispenser, such as a plastic syringe for precision dispensing. If possible, use a dispenser that comes with a gauge for packing accurate amounts of grease. Alternatively, measure the weight of the bearing before and after packing.

3. Amount of grease

Use the following recommended amounts of grease (grease fill) for super precision bearings:

Angular contact ball bearings for high-speed spindles:

Fill 15-30% of the free internal space.

Cylindrical roller bearings for high-speed spindles:

Fill 10-15% of the free internal space.

Increasing the amount of grease extends grease life. However, more time will be required for run-in and the bearing will exhibit higher temperature rise. The advantages and disadvantages need to be considered and the grease amount chosen to suit operating conditions.

For small bearings up to 30 mm bore diameters, setting the grease amount to 15% of internal space results in a very small total amount of grease. In such cases, we suggest 20% for increased reliability.

Recommended grease quantities for each bearing type are listed in Table 1.3.

■ Packing method for ball bearings

(1) Pack grease evenly between the balls. If a ring guided cage is used, apply a light coating of grease on the guided surface of the cage.

(2) Rotate the bearing by hand to spread the grease evenly. In doing so, take care not to expose angular contact ball bearings to reverse thrust loads.

■ Packing method for cylindrical roller bearings

(1) Coat about 80% of the grease amount evenly on the rolling surfaces of the rollers. Avoid putting too much grease on the cage bore. Grease on the cage bore is difficult to disperse during run-in, which can result in a rapid rise in temperature or a long run-in period.

(2) Spread the grease applied to the rolling surfaces thinly to the roller end faces, roller cage contact points, and face edges of each cage pocket.

(3) Using the remaining 20%, apply a thin film of grease to the raceway surface of the mounted outer ring (to the raceway surface of the inner ring for NNU type bearings).

Fig. 1.22 Grease-Packed Angular Contact Ball Bearing



Fig. 1.23 Grease-Packed Cylindrical Roller Bearing



Table 1.3 Recommended Grease Quantities for High-Speed Spindle Bearings

Unit: g/bearing

Bore number	Nominal bearing bore dia. (mm)	Angular contact ball bearings: 15% of free internal space						Cylindrical roller bearings: 10% of free internal space			
		79 Series BNR19 BER19 BSR19	70 Series BNRD10 BERD10	72 Series	BSR10 BNR10 BER10 BAR10 BTR10	TAC29F TAC29D	TAC20F TAC20D	NN49 Series NNU49 series	NN39 Series	NN30 Series	N10 Series
5	5	–	–	0.02	–	–	–	–	–	–	–
6	6	–	0.04	0.07	0.03	–	–	–	–	–	–
7	7	–	0.07	–	0.04	–	–	–	–	–	–
8	8	–	0.11	0.09	0.09	–	–	–	–	–	–
00	10	0.05	0.12	0.14	0.10	–	–	–	–	–	–
01	12	0.06	0.13	0.22	0.11	–	–	–	–	–	–
02	15	0.11	0.17	0.27	0.17	–	–	–	–	–	–
03	17	0.12	0.22	0.38	0.22	–	–	–	–	–	–
04	20	0.21	0.41	0.63	0.41	–	–	–	–	–	–
05	25	0.25	0.48	0.79	0.50	–	–	–	–	0.3	–
06	30	0.29	0.64	1.2	0.53	–	–	–	–	0.6	0.4
07	35	0.44	0.91	1.6	0.72	–	–	–	–	0.7	0.5
08	40	0.70	1.1	2.0	0.85	–	–	–	–	0.9	0.6
09	45	0.77	1.4	2.5	1.1	–	–	–	–	1.2	0.9
10	50	0.84	1.5	2.8	1.1	–	1.6	–	–	1.3	1.0
11	55	1.1	2.2	3.7	1.6	–	2.2	–	–	1.8	1.3
12	60	1.1	2.4	4.5	1.7	–	2.3	–	–	2.0	1.4
13	65	1.2	2.5	5.3	1.8	–	2.5	–	–	2.0	1.5
14	70	2.0	3.4	6.0	2.6	–	3.7	–	–	3.0	2.2
15	75	2.1	3.4	6.5	2.7	–	3.9	–	–	3.3	2.4
16	80	2.3	4.7	8.1	3.6	–	4.9	–	–	4.4	3.3
17	85	3.3	4.9	10	3.8	–	5.1	–	–	4.5	3.4
18	90	3.4	6.2	12	5.1	–	7.1	–	–	6.0	4.2
19	95	3.3	6.4	15	5.3	–	7.4	–	–	6.2	4.3
20	100	4.6	6.7	18	5.7	4.9	7.7	5.0	4.2	6.3	4.5
21	105	4.8	8.4	21	7.1	–	9.3	5.2	–	8.7	5.4
22	110	4.9	11	25	8.5	5.3	11	5.4	4.4	10	6.9
24	120	7.3	11	29	9.1	7.0	12	7.8	6.1	12	7.8
26	130	8.4	17	32	14	8.9	18	11	7.9	17	12
28	140	9.2	19	39	16	–	41	11	8.7	18	13
30	150	13	23	49	20	30	51	18	13	22	15
32	160	16	29	–	25	34	62	18	14	27	20
34	170	16	40	–	30	36	79	19	14	35	25
36	180	24	49	–	42	50	108	26	22	48	33
38	190	26	50	–	47	47	118	28	22	50	35
40	200	35	66	–	57	74	145	41	32	65	43
44	220	39	87	–	66	77	184	45	35	88	59
48	240	41	96	–	71	89	199	48	38	92	63
52	260	72	142	–	103	147	278	82	65	130	94
56	280	75	151	–	–	158	–	89	69	146	100

Remarks:

· When using the greases listed on p. 219, divide the grease fill listed above by 0.93 (the specific gravity) to convert to volume.

· Values for angular contact ball bearings are based on filling 15% of the internal space (X fill). For 20% fill (K fill), multiply values by 1.3 and for 30% (L fill), multiply by 2.

· Refer to p. 146 and p. 148 for recommended quantities for angular contact thrust ball bearings for ball screw support.

· For bore diameters over 280 mm, please contact NSK for details.

3.2 Mounting of Bearings with Cylindrical Bores

(1) Press Fit

Press fits tend to be used for smaller bearings. A mounting tool is placed on the inner ring as shown in Fig. 1.24 and the bearing is slowly pressed on the shaft with a press until the side of the inner ring rests against the shoulder of the shaft. When press mounting the inner ring, the mounting tool must never be placed on the outer ring, as it would cause dents or other damage to the raceway surfaces.

Before mounting, lightly greasing or oiling the mating parts is recommended for smooth insertion. Never use a hammer when mounting super precision bearings.

In the case of separable bearings, such as cylindrical roller bearings and tapered roller bearings, the inner and outer rings can be mounted onto the shaft and into the housing as separate units. When assembling the two units, take extra care to align the inner and outer rings correctly. Careless or forced assembly may cause scratches on the rolling contact surfaces.

(2) Shrink Fit

Press fitting large bearings or press fitting bearings with high interference requires a great deal of force. Therefore, a shrink fit is widely used. The inner rings of the bearings are heated to expand them before mounting. This method prevents excessive force from being imposed on the bearings and enables mounting in a short time.

The temperature required for shrink fitting depends on the bearing bore size and the selected interference. To aid in selecting the right temperature, Fig. 1.25 shows the expansion of the inner ring for various temperature differences and bearing sizes.

The following precautions need to be taken when shrink fitting:

1. Do not heat bearings to more than 120 ˚C.
2. Heat the bearings to a temperature 20 to 30 ˚C higher than the lowest temperature required for mounting without interference since the inner ring will cool slightly during mounting.
3. After mounting, the bearings will shrink in the axial direction as well as the radial direction while cooling. Therefore, after confirming that the bearing has sufficiently cooled, it is advisable to apply the mounting tool again or further tighten the locknut to eliminate any clearance between the bearing and shoulder (Fig 1.26).

Fig 1.24 Press Fitting the Inner Ring

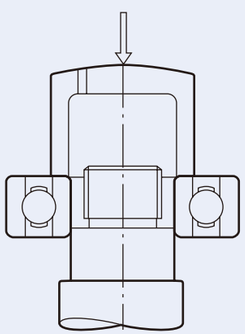
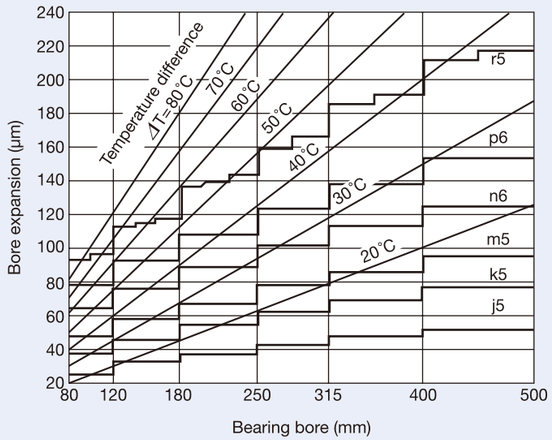


Fig 1.25 Temperature and Thermal Expansion of Inner Ring



Using NSK Bearing Heaters makes the shrink fit process more efficient. For more information, refer to pages 182-183.

Fig 1.26 Additional Pressing

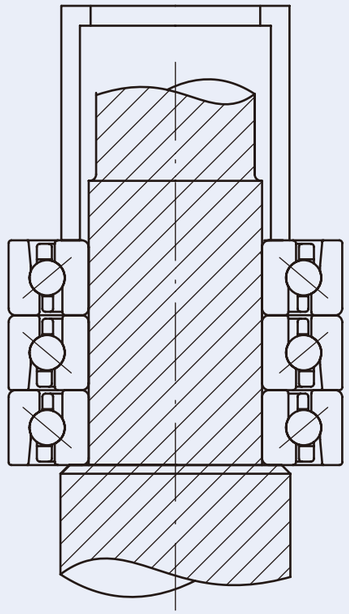
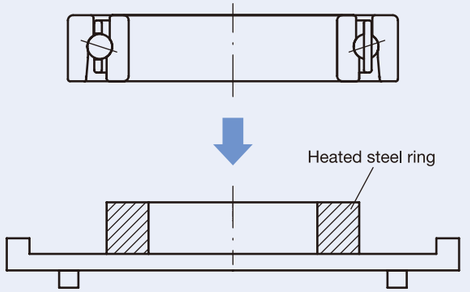


Fig 1.27 Heating the Inner Ring by Hot Plate



3.3 Securing the Bearing

(1) Securing the inner ring

The inner ring is usually secured onto the shaft by tightening the locknut on the threaded shaft, which is why perpendicularity of the threads and the end face of the locknut are very important. Even if accuracy of a single component is good, a gap between the shaft and locknut can result in a slightly tilted locknut, causing shaft misalignment, which in turn results in misalignment of the bearing. Therefore, adjustments are necessary to ensure consistent running accuracy. It is important that the locknut is completely tightened so as to eliminate any risk of it becoming loose.

Tightening torque information for shaft locknuts is shown in Table 1.5.

To reduce the risk of unbalance due to locknut misalignment and clearance between the shaft thread and locknut thread, stepped sleeves are widely used to achieve an interference fit between the shaft and the sleeve bore. However, compared to a locknut, a stepped sleeve is more likely to become loose after continuous operation, so it must be checked periodically.

When a wide spacer is used between matched bearings with position preload, and the tightening torque of the locknut is excessive, the inner ring spacer may become deformed and alter the preload to a level higher than expected. It is necessary to consider this deformation when determining the preload.

Bearings for machine tool spindles

Refer to Table 1.5 for recommended tightening force values.

Please note that when shaft interference increases, insertion pressure (press-fit force) increases. Add an allowance to the insertion pressure value to obtain the tightening force.

Extra caution is required for high-speed operation in particular.

Angular contact thrust ball bearings for ball screw support

Recommended locknut tightening force values for face-to-face arrangements (DF, DFD, DFT, etc.) are listed in Table 1.5.

For back-to-back arrangements (DB, DBD, DBT etc.), we recommend adding preload to the tightening force value from Table 1.5 to obtain the recommended locknut tightening force. However, compare the value obtained to the insertion pressure. If the insertion pressure value is higher, use the insertion pressure value and add an allowance to obtain the locknut tightening force.

(2) Securing the outer ring

Adjust the clearance between the retaining cover and housing as shown in Table 1.5, then tighten the bolts. Take care when tightening the bolts since uneven tightening forces will increase out-of-roundness of the outer ring.

Reference tightening torque values for each bolt size are listed in Table 1.4.

Table 1.4 Bolt Tightening Torque for Securing the Outer Ring

Nominal thread dimension	Tightening torque N·m
M3×0.5	1.7
M4×0.7	3.9
M5×0.8	7.9
M6×1	13.5
M8×1.25	32.8
M10×1.5	65.0
M12×1.75	114
M14×2	180
M16×2	281
M18×2.5	387
M20×2.5	549
M22×2.5	747
M24×3	949

Note: Tightening torque is affected by various conditions such as material and surface conditions. Therefore, the values in this table are reference values only.

Table 1.5 Shaft Locknut Tightening Force and Clearance Between Retaining Cover and Housing

Nominal bearing bore (mm)	Locknut tightening force (N)	Locknut tightening torque, reference (N·m)	Clearance between retaining cover and housing (mm)
6	1 500	2	0.01 – 0.03
8	1 500	2	0.01 – 0.03
10	1 500	3	0.01 – 0.03
12	3 000	7	0.01 – 0.03
15	3 000	8	0.01 – 0.03
17	3 000	9	0.01 – 0.03
20	4 900	17	0.01 – 0.03
25	4 900	21	0.01 – 0.03
30	4 900	25	0.01 – 0.03
35	9 800	57	0.01 – 0.03
40	9 800	64	0.01 – 0.03
45	9 800	72	0.01 – 0.03
50	9 800	80	0.01 – 0.03
55	14 700	132	0.01 – 0.03
60	14 700	142	0.01 – 0.03
65	14 700	153	0.01 – 0.03
70	14 700	166	0.01 – 0.03
75	14 700	176	0.01 – 0.03
80	19 600	251	0.01 – 0.03
85	19 600	267	0.01 – 0.03
90	19 600	281	0.01 – 0.03
95	19 600	296	0.01 – 0.03
100	19 600	311	0.01 – 0.03
105	19 600	327	0.01 – 0.03
110	19 600	343	0.01 – 0.03
120	19 600	371	0.01 – 0.03
130	19 600	403	0.01 – 0.03
140	29 400	649	0.03 – 0.05
150	29 400	695	0.03 – 0.05
160	29 400	745	0.03 – 0.05
170	29 400	796	0.03 – 0.05
180	29 400	841	0.03 – 0.05
190	29 400	886	0.03 – 0.05
200	29 400	932	0.03 – 0.05
220	39 200	–	0.03 – 0.05
240	39 200	–	0.03 – 0.05
260	39 200	–	0.03 – 0.05
280	39 200	–	0.03 – 0.05
300	39 200	–	0.03 – 0.05

Note: For bearing bores larger than 300 mm, please contact NSK.

■ Conversion equation of locknut tightening torque

$$T = 0.5F\{d_p \cdot \tan(p^* + \beta) + d_w \cdot \mu_w\} \times 10^{-3} \quad [\mathrm{N \cdot m}]$$

The locknut tightening torque values in the table have been calculated using a friction coefficient of 0.15.

T : Locknut tightening torque [N·m]
F : Locknut tightening force [N]
d_p : Effective diameter of locknut [mm]
p^* : Friction angle of locknut surface
$p^* = \tan^{-1}\mu_S$
μ_S : Friction coefficient of locknut surface
d_w : Frictional torque equivalent diameter at locknut seating surface [mm]
μ_w : Friction coefficient of locknut seating surface
β : Lead angle of nut
$\beta = \tan^{-1}(\mathrm{pitch}/(3.142\, d_p))$

■ Equation of press-fit force (insertion pressure)

$$K = \mu \cdot p_m \cdot \pi \cdot d \cdot B \ [\mathrm{N}]$$

$$P_m = \frac{E}{2} \frac{\Delta d}{d} \frac{(1-k^2)(1-k_0^2)}{1-k^2 k_0^2}$$

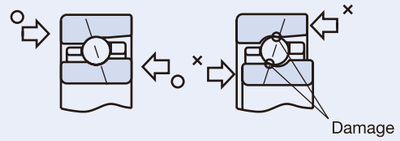
μ : Friction coefficient at fitting surface [=0.12]
p_m : Surface pressure [MPa]
d : Shaft diameter [mm]
B : Bearing width [mm]
Δd : Effective interference [mm]
E : Young's modulus of steel [MPa]
k : Wall thickness ratio of inner ring ($k = d/D_i$)
D_i : Raceway diameter of inner ring [mm]
k_0 : Wall thickness ratio of hollow shaft ($k_0 = d_0/d$)
d_0 : Bore diameter of hollow shaft [mm]

3.4 Precautions for Mounting Angular Contact Ball Bearings

Due to design restrictions, an angular contact ball bearing can sustain loads in only one direction. Therefore, when mounting angular contact ball bearings onto the shaft or into the housing, it is important not to apply any load in the wrong direction.

Pay special attention to the mounting order of matched bearings, as it is different for back-to-back and face-to-face arrangements.

Fig 1.28 Direction of Load for Angular Contact Ball Bearings

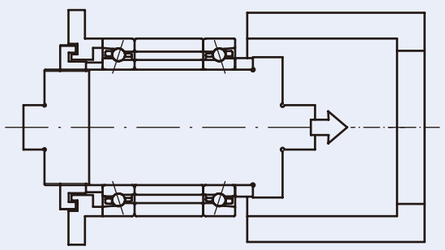


Load applied in direction “x” may cause damage to the balls.
○ : Right ✕ : Wrong

Back-to-back arrangement:

① Press bearings onto shaft.
② Tighten shaft locknut for preload.
③ Insert shaft with bearings into the housing and attach the retaining cover.

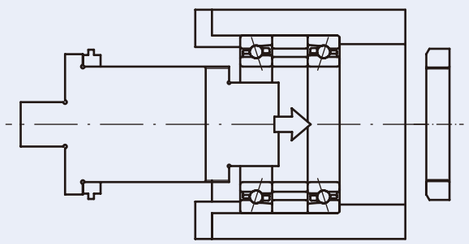
Fig 1.29 Mounting of Back-to-Back Arrangement



Face-to-face arrangement:

① Insert bearings into housing.
② Secure the retaining cover for preload.
③ Insert shaft into the inner rings and tighten the shaft locknut.

Fig 1.30 Mounting of Face-to-Face Arrangement



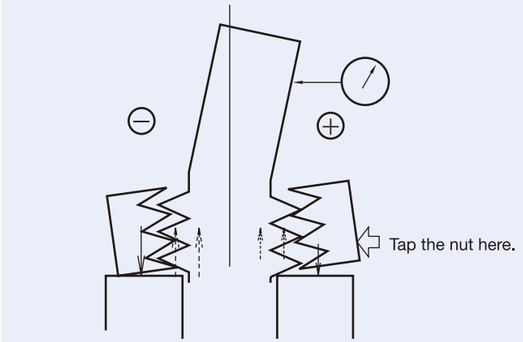
Reverse the order of each step for dismounting.

4. Inspection after Mounting

4.1 Runout Accuracy

Spindle accuracy requires accurate components as well as an accurate assembly process. For example, misalignment of the locknut can cause the shaft and bearing to bend when tightened, as depicted in Fig. 1.31.

Fig 1.31 Adjusting Misaligned Shaft Locknut

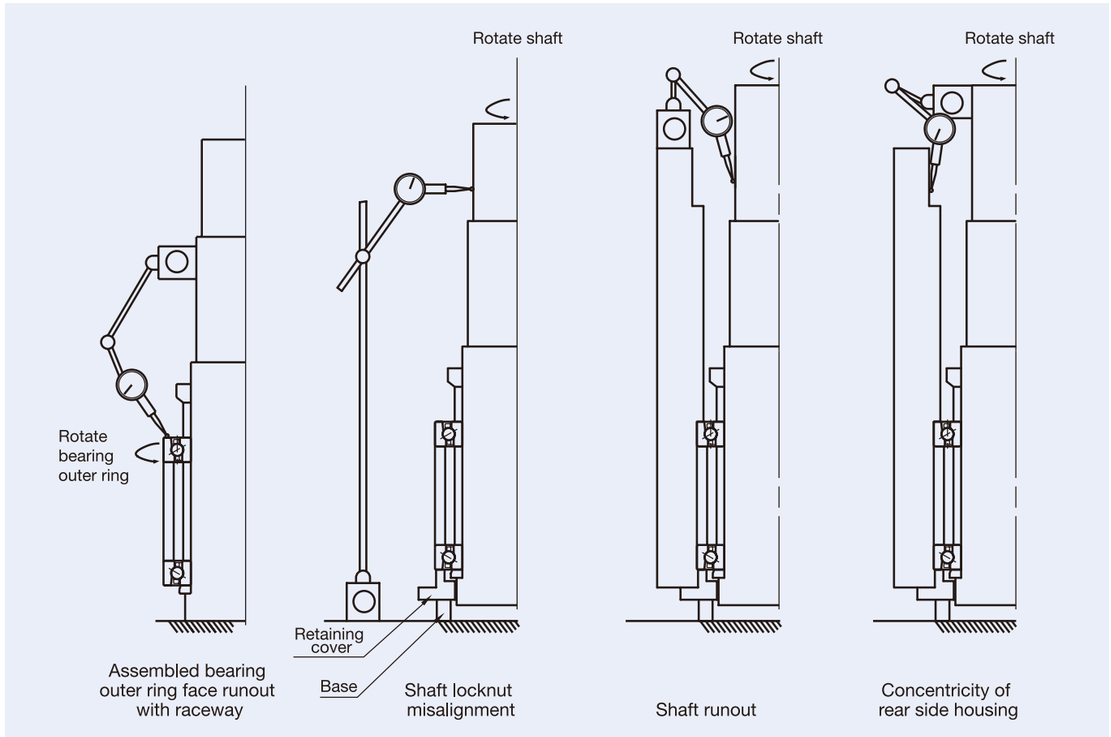


Check the following during inspection (note that these values are provided as an example):

Assembled bearing outer ring face runout with raceway for angular contact ball bearings:
Adjust to 0.002 mm or less by lightly tapping on the outer ring end face.
Shaft locknut misalignment:
Adjust shaft nut misalignment to 0.005 mm or less (Fig. 1.31)
Shaft runout:
0.005 mm or less
Concentricity of rear side housing:
0.015 mm or less

Define target accuracy values based on spindle size and application. If the target accuracy values cannot be met, disassemble and check the accuracy of the components again.

Fig 1.32 Measuring Spindle Runout



4.2 Control of Preload After Mounting of Bearings

Setting a higher preload will increase the rigidity of the bearing, but heat generation will also increase. In extreme cases, it may cause seizure. Therefore, it is necessary to set the optimum preload carefully in accordance with the operating conditions and to control mounted preload. Methods for measuring preload are detailed below.

For cylindrical roller bearings, we recommend a GN gauge to control mounted radial preload (refer to Page 250).

Measuring preload

There are three methods for checking the preload of bearings after they have been mounted onto a spindle: the starting torque method, the force deflection method (thrust static rigidity method), and the natural frequency method. The features of each method are summarized in Table 1.6.

For tapered roller bearings, starting torque method is suitable because the axial rigidity of tapered roller bearings is not sensitive to preload amount.

Table 1.6 Methods of Measuring Mounted Preload

	Starting torque method	Force deflection method	Natural frequency method
Advantages	**Suitable for heavy preload** Measurement error is small if starting torque is high	**Suitable for light preload**	**High measuring accuracy** **Excellent repeatability**
Disadvantages	**Not suitable for light preload** Variation of measurement is large if starting torque is small	**Not suitable for heavy preload** May require large loading device Measurement may be affected by deformation of related parts	Easily influenced by measuring jig structure and spindle orientation

(1) Starting torque method

Measure the initial tangential force required to start the spindle from standstill using a push-pull gauge (Fig. 1.33). The product of the initial tangential force F and the shaft radius at the point of measurement ($d/2$) yields the starting torque T.

$$T = F \times d/2$$

Obtain the level of preload from the relationship between preload and starting torque (Fig. 1.34).

With the starting torque method, measurement is simple. However, measurement accuracy varies depending on the level of starting torque. As the starting torque is low in most high-speed machine tool spindles, measurement error is higher.

If oil film formation in the rolling contact areas is unstable during measurement, stick-slip may occur: this is where rotation initially does not start even though tangential force is applied, then rotation suddenly starts as the tangential force is increased gradually. In this case, the measured force tends to be higher than the actual force. Therefore, take a sufficient number of measurements and exclude any such exaggerated measurements.

Fig. 1.33 Starting Torque Method

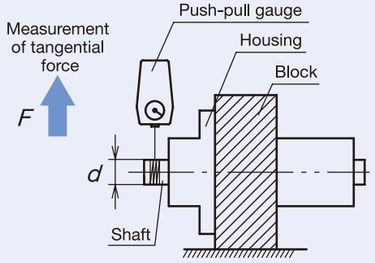
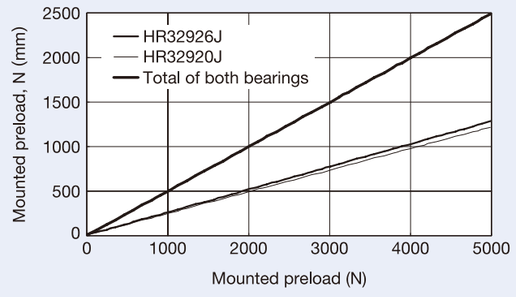


Fig. 1.34 Relationship Between Starting Torque and Preload



(2) Force deflection method

Apply a thrust load to the shaft and measure its axial displacement (Fig. 1.35). Determine preload from the relationship between axial displacement and preload (Fig. 1.36).

This method is suitable for ball bearings since they exhibit a large change in rigidity in response to a change in preload. It is useful for spindles with moderate axial rigidity as force deflection can easily be measured. By contrast, when measuring spindles with high axial rigidity, it may be necessary to use special hydraulic equipment to apply a large enough axial load. For example, if the axial rigidity is 200 N/μm, an axial load of 2000 N will be required to deflect the spindle by 10μm.

If excessive load is applied, elastic deformation can occur in both the bearing internals and associated machine parts; in this case, the measured value tends to be lower than the actual preload.

(3) Natural frequency method

Vibrate the shaft in the axial direction and measure its resonance frequency (Fig. 1.37).

The natural frequency method offers the highest sensitivity and repeatability for angular contact ball bearings. However, results can be affected by the measuring jig structure. In other words, if the mounting conditions and/or spindle orientation change, measured results will also change. Therefore, we recommend setting initial control values by comparing values obtained with those from other measurement methods.

Measurement of resonance frequency (F_Z) of shaft in axial direction

↓

Axial spring constant (K_a) of bearing

↓

Mounted preload

$$K_a = m\left(\frac{\pi \cdot F_z}{500}\right)^2$$

K_a : Axial spring constant of bearing (N/μm)
F_z : Resonance frequency (Hz)
m : Mass of rotating body

Fig. 1.35 Force Deflection Method

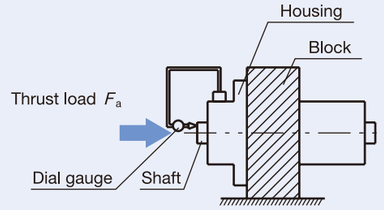


Fig. 1.36 Relationship Between Axial Displacement and Preload

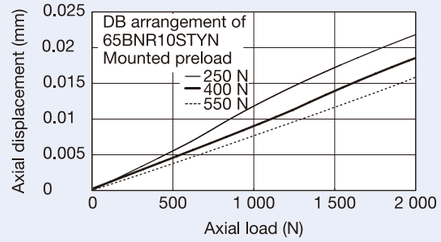
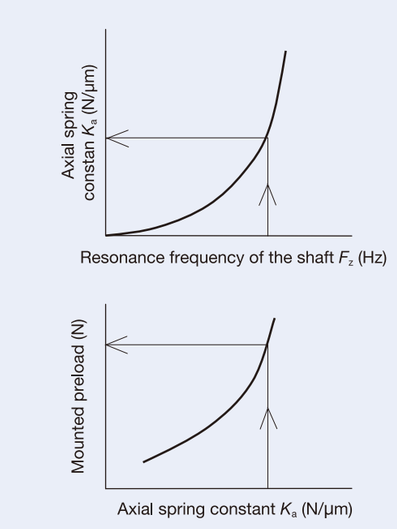


Fig. 1.37 Natural Frequency Method

Fig. 1.38 Relationship Between Resonance Frequency of Shaft and Axial Spring Constant of Bearing



2. Operational Inspection

Preparing for Operation

After spindle assembly has been completed, connect the spindle to the power source.

■ Connection with motor

When using V-belt drives, adjust parallelism and misalignment of the spindle pulley and the motor pulley (Fig. 2.1).

Coupled drives need to be well-balanced and checked for offset and misalignment (Fig. 2.2).

Fig. 2.1 V-Belt Drive

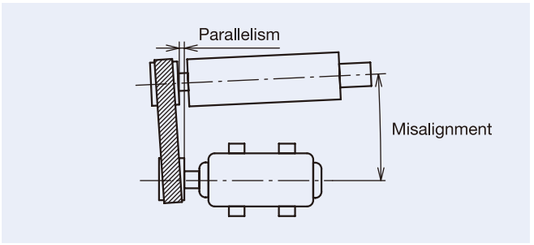
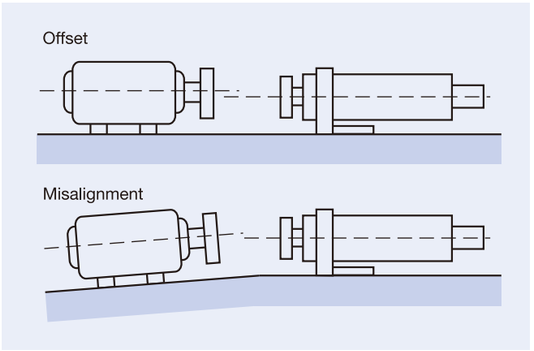


Fig. 2.2 Coupled Drive



■ Balance

Any unbalance in rotating components will cause an eccentric load during rotation due to centrifugal force. This results in vibration, decreased machining accuracy, and decreased service life of the bearings (Fig 2.3).

Recommended balance quality grades for different types of rotors are listed in JIS B 0905. Table 2.1 and Fig 2.4 show an excerpt of this standard. Use specialized equipment when measuring ε (permissible residual specific unbalance) to apply the values in these tables.

Fig. 2.3 Bearing Load Due to Unbalance

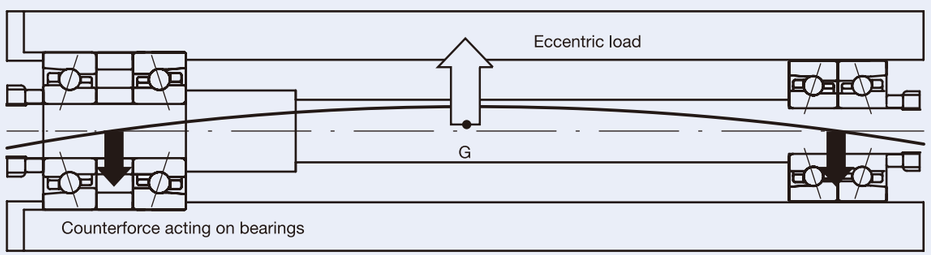
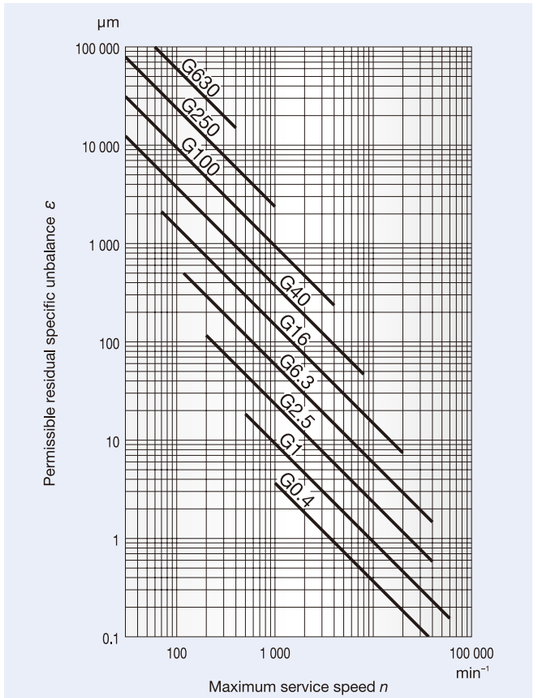


Table 2.1 Balance Quality Grades and Examples of Rotor Types

Balance Quality Grade	Rotor Type Example
G0.4	Precision grinder
G1.0	Grinding machine spindles, Small motors with special requirements
G2.5	Machine tool spindles, Medium and large motors with special requirements
G6.3	Components of machine tools and general machinery

Fig. 2.4 Permissible Residual Specific Unbalance Based on Balance Quality Grade



Operational Inspection

After mounting has been completed, conduct a test run to determine if the bearing has been mounted correctly. Small equipment may be manually operated to assure that it rotates smoothly. As this point, check for sticking due to contamination, surface damage or dents, uneven torque caused by improper mounting or an improperly machined mounting surface, and excessive torque caused by an inadequate clearance, mounting errors, or seal friction. If there are no abnormalities, run-in can be started.

As large equipment is impossible to turn by hand, start up without load and switch power off again immediately to allow the spindle to coast. Check for vibration, noise, and abnormalities in the contact of the rotating components.

Start run-in slowly without load, then gradually increase the speed until the maximum speed is reached (page 268-269). During the test run, check for abnormal noise, excessive rise of bearing temperature, leakage or discoloration of lubricants, etc. If any abnormality is found, stop the test run immediately and inspect the machinery. If necessary, dismount the bearings for examination.

Although the temperature of the outside surface of the housing can generally help determine bearing temperature, it is better to directly measure the temperature of the outer ring using oil holes for access.

Bearing temperature should rise gradually to a steady level within one or two hours after starting operation. If problems arise in a bearing or if an error was made in mounting, bearing temperature may increase rapidly and become abnormally high. Possible causes of such an abnormal temperature include an excessive amount of lubricant, insufficient bearing clearance, incorrect mounting, or excessive friction of the seals. In high-speed operations, an incorrect selection of bearing type or lubricating method may also cause abnormal temperature rise.

Bearing noise can be checked with a sound detector or other instruments. Abnormal conditions are indicated by a loud metallic sound or other irregular noise. Possible causes include incorrect lubrication, insufficient accuracy of the shaft and housing, damaged bearings, or ingress of contamination into the bearing.

Possible causes and countermeasures for irregularities are listed on page 275.

3. Run-in Procedures

Run-in

If operating speed is suddenly increased after the bearings are mounted, bearings may be damaged due to insufficient lubrication or the lubricant may deteriorate.

Proper run-in with gradual increases of operating speed is indispensable, especially for grease-lubricated bearings where grease must be allowed to spread evenly. Run-in should be conducted at room temperature (15°C to 25°C) while monitoring bearing temperature.

Maximum operating temperature of the bearing should be targeted at about 60°C (50 °C at the spindle housing exterior). If temperature exceeds this target by 5°C or more, temporarily stop the run-in process or decelerate to lower the temperature.

Some spindle assemblies incorporate both cylindrical roller bearings and angular contact ball bearings. Since cylindrical roller bearings tend to require more time than ball bearings for temperature to level off, the timing of speed increases must be set to correspond with the cylindrical roller bearings.

Note

Spindle assemblies operating under oil-mist and oil-air lubrication are at risk of a sudden temperature rise at initial operation or after the spindle assembly has not been operated for a long time. Excess oil that has collected in the oil lines of the lubrication system may suddenly flood the bearing interior, causing a temperature spike. Run-in for bearings with these lubricating systems requires much less time than for grease-based systems and is highly recommended.

(1) Continuous run-in method

Continuous run-in works by gradually increasing the operating speed from the low speed zone. Divide the maximum operating speed evenly into a number of stages and increase speed in increments:

1. Begin at a reasonably low operating speed.
2. Monitor temperature rise.
3. When temperature has stabilized, increase speed to the next target speed.
4. Continue repeating steps 2 and 3 until the maximum operating speed is reached.

Maximum operating speed is commonly divided into ten stages. Allow between 30 min. and two hours for the temperature to stabilize before increasing speed to the next stage. Fig. 3.2 shows patterns of temperature development to help determine whether speed may be increased or not. Determine the target speeds that are optimally suited for your application while monitoring the actual temperature on the spindle.

Although somewhat time-consuming, this procedure helps detect potential spindle defects, thus avoiding costly damage to the bearings.

Fig. 3.1 Bearing Temperature Change During Continuous Run-in

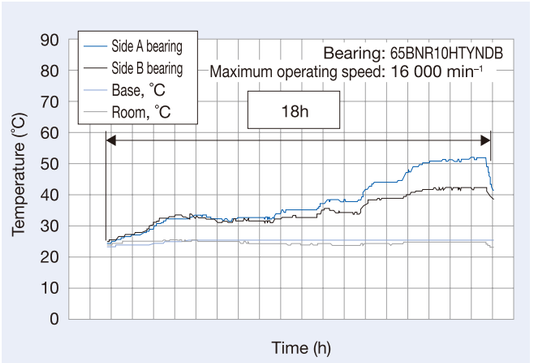
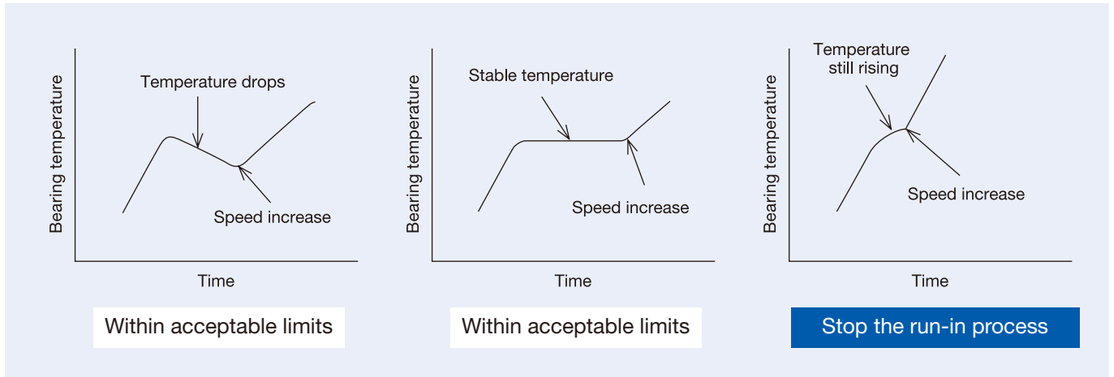


Fig. 3.2 Bearing Temperature Change During Run-in



Increase operating speed when temperature rise characteristics are within limits.

(2) Intermittent run-in method

At first, run the spindle continuously at about 500 min^{-1} (100 min^{-1} for larger machines) for 15 minutes to allow the grease to settle. Take the maximum operating speed and divide it into eight to ten stages.

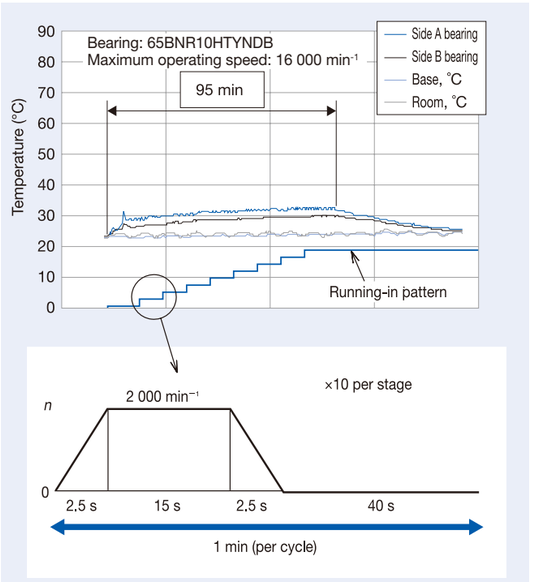
1. Each stage is divided into 10 cycles that are approximately one minute long.
2. During each cycle, rapidly accelerate the spindle to the target speed for the current stage, decelerate back to zero, and then rest for a period of 40 seconds.
3. Repeat this cycle about 10 times.
4. Continue moving up through the stages, following the above procedures, until the maximum operating speed.

The temperature data shown in Fig 3.3 were gained during intermittent run-in with a maximum operating speed of 16 000 min^{-1}, 8 speed stages, and 10 cycles per stage. After the maximum operating speed is reached, we recommend continuously running the spindle at that speed for about 1 hour.

Speed increase causes a sudden supply of grease to the bearing's interior, resulting in a sharp temperature rise. During intermittent run-in, the spindle is stopped to allow the temperature to stabilize. This saves time compared to continuous run-in.

The number of target speed stages and the number of cycles to be performed in each speed stage vary according to spindle design and arrangement. Please determine appropriate values on actual equipment.

Fig. 3.3 Cycle Structure and Bearing Temperature Change During Intermittent Run-in



Part 7

Bearing Failure Diagnosis

1. Bearing Failure and Countermeasures ·····272-277
2. Diagnosis with Sound and Vibration ·····278-279

1. Bearing Failure and Countermeasures

Maintenance, Inspection, and Corrective Measures

Proper maintenance and inspection help to maintain the original performance of a bearing for as long as possible, avoid bearing problems, increase reliability and productivity, and keep operating costs low. We suggest periodic maintenance with specified procedures that include the supervision of operating conditions, replenishment or replacement of lubricants, and the inspection of parts.

Items that should regularly be checked during operation include bearing noise, vibration, temperature, and lubrication. If any irregularity is found during operation, the cause should be determined and the proper corrective action taken after referring to Table 1.2 on page 275. If necessary, bearings should be dismounted and examined in detail.

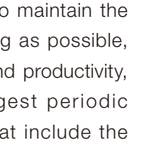
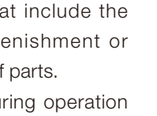
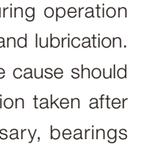
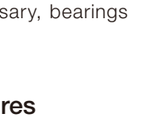
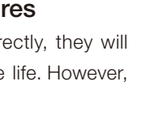
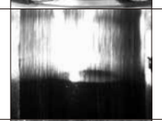
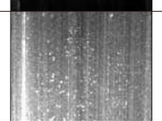
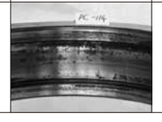
Bearing Failure and Countermeasures

In general, if rolling bearings are used correctly, they will continue to perform to their predicted fatigue life. However, bearings often fail prematurely due to avoidable mistakes. In contrast to fatigue life, this premature failure is caused by improper mounting, handling, or lubrication; ingress of contamination; or abnormal heat generation.

Possible causes of rib scoring in roller bearings, as one example, include the use of improper lubricant, a faulty lubrication system, ingress of contamination, bearing mounting errors, excessive shaft deflection, or any combination of these.

Therefore, it is difficult to determine the real cause of some premature failures by looking only at the bearing itself. If all the conditions at the time of failure and prior to failure are known, including the application, operating conditions, and details regarding the spindle structure around bearings, then by studying the bearing damage and its probable causes, the risk of similar future failures can be reduced. The most frequent types of bearing failure, along with probable causes and countermeasures, are listed in Table 1.1.

Table 1.1 Causes and Measures for Bearing Failures

Type of Failure	Symptoms	Photo	Probable Causes	Countermeasures
Flaking	Flaking on one side of raceway of radial bearing.		·Abnormal axial load (sliding failure of free-side bearing)	·Select a looser fit to allow axial expansion of the shaft.
	Flaking pattern inclined relative to raceway in radial ball bearings. Flaking near edge of raceway and rolling surface in roller bearings.		·Improper mounting, bending of shaft ·Inadequate centering ·Inadequate tolerances for shaft and housing	·Use care in mounting and centering. ·Select a bearing with a larger clearance. ·Correct perpendicularity of shaft and housing shoulder.
	Flaking of raceway with same spacing as rolling elements.		·Large shock load during mounting ·Rust while bearing is out of operation for prolonged period ·Damage to cylindrical roller bearings during mounting	·Use care in mounting. ·Apply a rust preventative when operation is suspended for a long time.
	Premature flaking of raceway and rolling elements.		·Insufficient clearance ·Excessive load ·Improper lubrication, rust, etc.	·Select proper fit, bearing clearance, and lubricant.
	Premature flaking of matched bearings.		·Excessive preload	·Adjust the preload.
Scoring	Scoring or smearing between raceway and rolling surface.		·Inadequate initial lubrication ·Excessively hard grease ·High acceleration when starting operation	·Use a softer grease. ·Avoid rapid acceleration.
Scoring	Scoring or smearing between the end face of the rollers and guide rib.		·Inadequate lubrication ·Incorrect mounting ·Large axial load	·Select proper lubricant. ·Modify the mounting.
Cracks	Crack in outer or inner ring.		·Excessive shock load ·Excessive interference in fitting ·Poor shaft cylindricity ·Large fillet radius ·Development of thermal cracks ·Increased flaking	·Examine load conditions. ·Modify the fit of bearing and sleeve. ·Improve accuracy in machining shaft. ·Correct fillet radius (the fillet radius must be smaller than the bearing chamfer).
	Crack in rolling element or broken rib.		·Progression from flaking ·Shock applied to rib during mounting ·Dropped during handling	·Use care in mounting and handling bearings.
	Fracture of cage.		·Abnormal load on cage due to incorrect mounting ·Improper lubrication	·Correct mounting. ·Examine the lubrication method and lubricant.
Dents	Indentations on raceway with same spacing as rolling elements (Brinelling).		·Shock load during mounting ·Excessive load while stationary	·Use care in handling bearings.
	Indentations on raceway and rolling elements.		·Ingress of contamination such as metallic particles or grit	·Clean the housing. ·Improve the seals and use clean lubricant.
Abnormal Wear	False brinelling (phenomenon similar to brinelling).		·Vibration of a stationary bearing during transport ·Oscillating motion with small amplitude	·Secure the shaft and housing during transport. ·Select a suitable lubricant. ·Reduce vibration by applying preload.
	Fretting, localized wear with reddish-brown wear dust at fitting surface.		·Sliding wear from small clearance at fitting surface	·Increase interference. ·Apply oil.
	Wear on raceway, rolling surfaces, rib, and cage.		·Ingress of contamination ·Incorrect lubrication and rust	·Improve sealing capabilities. ·Clean the housing. ·Use a clean lubricant.
	Creep, scoring wear at fitting surface.		·Insufficient interference ·Insufficiently secured sleeve	·Modify fit. ·Tighten the sleeve properly.
Seizure	Wear of raceway, rolling elements, rib, and cage. Discoloration and melting of rib surface.		·Insufficient clearance ·Incorrect lubrication ·Improper mounting	·Examine fit and internal clearance of bearings. ·Supply an adequate amount of suitable lubricant. ·Examine the mounting method and quality of related parts.
Corrosion and Rust	Corrosion and rust at bearing interior or fitting surface.		·Condensation due to humidity, fretting ·Entry of corrosive substances (especially varnish solvents etc.)	·Store carefully when in a moist or hot climate. ·Take rust prevention measures for periods of non-running. ·Select suitable varnish and grease.

●Please contact NSK when bearings show these kind of symptoms.

Running Traces and Applied Loads

As the bearing rotates, the raceways of the inner ring and outer ring make contact with the rolling elements. This results in a dull trace on both the rolling elements and raceways. Running traces are a source of information about load conditions and should be carefully studied when the bearing is dismounted.

If the running traces are clearly defined, it is possible to detect whether the bearing was subjected to radial loads, axial loads, or moment loads, or whether there was excessive variation in housing rigidity. Running traces may point to major mounting errors, unexpected loads, or other causes of bearing damage.

Fig. 1.1 shows the running traces generated in deep groove ball bearings under various load conditions. Pattern (a) is the most common running trace generated when the inner ring rotates under a radial load only. Patterns (e) through (h) indicate adverse conditions which usually lead to shortened service life.

Similarly, Fig 1.2 shows running traces for cylindrical roller bearings. The outer ring traces of pattern (i) indicate properly applied radial load with inner ring rotation. Running trace (j) shows variations in shade across its width and a diagonal cutoff at the end of the loaded zone. This indicates misalignment of inner and outer rings relative to each other or shaft bending. Patterns (k) and (l) show outer ring running traces of double-row cylindrical roller bearings with inner ring rotation. While pattern (k) occurs with radial loads applied evenly to both rows, pattern (l) indicates a mismatch of bearing taper and shaft taper angle that results in uneven preload between the two rows.

Fig. 1.1 Typical Running Traces of Deep Groove Ball Bearings

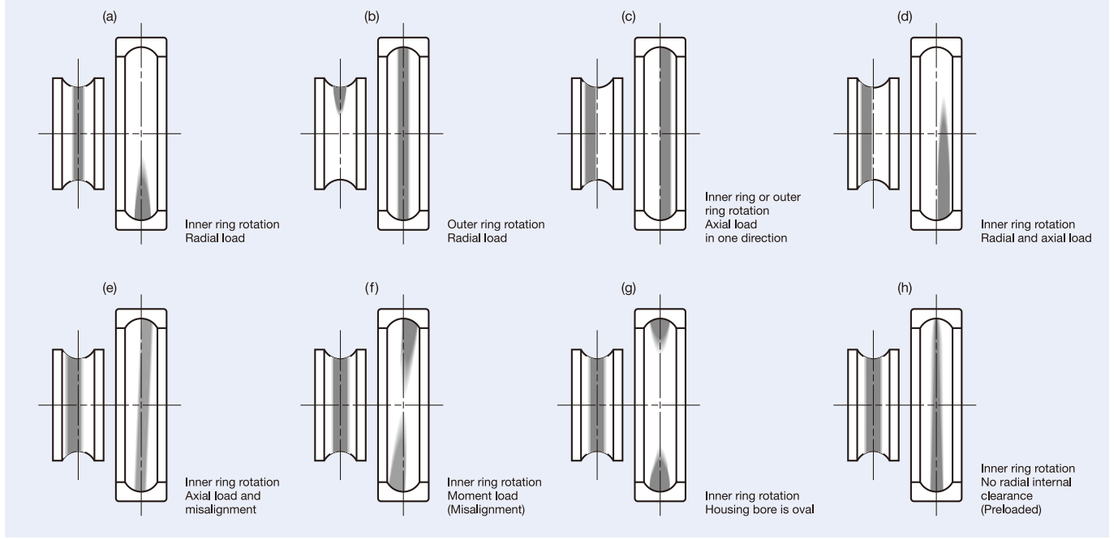


Fig. 1.2 Typical Running Traces of Roller Bearings

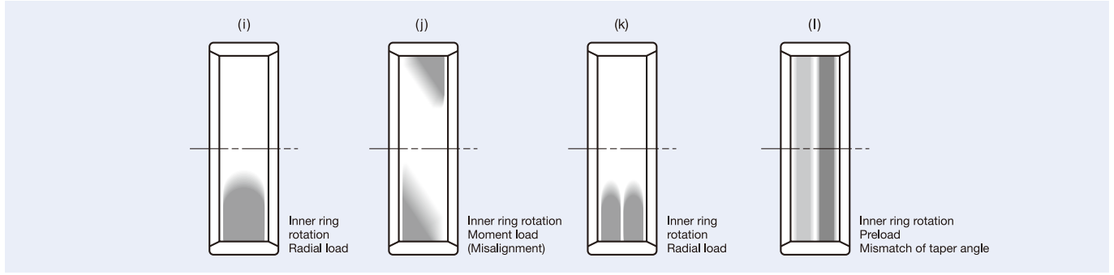


Table 1.2 Causes and Countermeasures for Operating Irregularities

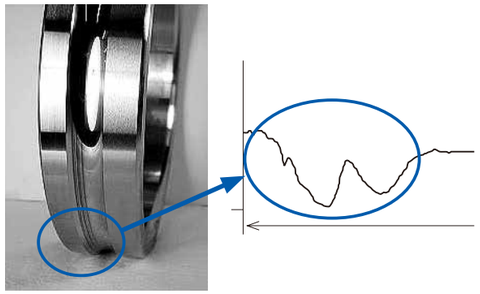
Irregularities		Possible Causes	Countermeasures
Noise	High-pitched metallic sound (1)	Abnormal load	Improve fit, examine internal clearance, adjust preload, correct position of housing shoulder, etc.
		Incorrect mounting	Improve machining accuracy of shaft and housing, improve mounting accuracy, review mounting method
		Insufficient or improper lubricant	Replenish lubricant, use suitable lubricant
		Contact of rotating parts	Modify the contact area of labyrinth seal, etc.
	Regular sound	Dents, corrosion or scratches on raceways due to contamination	Replace bearing, clean related parts, improve seals, use clean lubricant
		Brinelling	Replace bearing, handle bearings with care
		Flaking on raceway	Replace bearing
	Irregular sound	Excessive clearance	Examine fit and clearance, correct preload
		Ingress of contamination	Consider bearing replacement, clean related parts, improve seals, use clean lubricant
		Scratches or flaking on balls	Replace bearing
Abnormal temperature rise		Excessive amount of lubricant	Reduce amount of lubricant
		Insufficient or improper lubricant	Replenish lubricant, use suitable lubricant
		Abnormal load	Improve fit, examine clearance, adjust preload, correct position of housing shoulder, etc.
		Incorrect mounting	Improve machining accuracy of shaft and housing, improve mounting accuracy, review mounting method
		Creep on fitted surface, excessive seal friction	Replace bearing, examine fit, correct shaft and housing, change seal type
Vibration (Whirling of shaft)		Brinelling	Replace bearing, handle bearings with care
		Flaking	Replace bearing
		Incorrect mounting	Correct perpendicularity of shaft shoulder, housing shoulder, and spacer end faces
		Ingress of contamination	Replace bearing, clean related parts, improve seals, etc.
Leakage or discoloration of lubricant		Excessive amount of lubricant, ingress of contamination, abrasion with ingress of abrasion dust	Adjust amount of lubricant, select a different type of lubricant, clean the housing and related parts, replace the bearing

(1) Intermittent squealing or high-pitched noises may be heard in medium- to large-sized cylindrical roller bearings or ball bearings operating under grease lubrication in low-temperature environments. Generally, even with such noises, bearing temperature may not rise and fatigue/grease life are not affected; therefore, it is safe to continue using the bearings.
If quieter operation properties are needed, please contact NSK.

1. Bearing Failure and Countermeasures

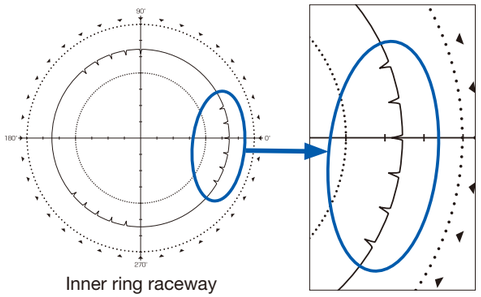
Some examples of bearing damage are shown in detail below.

Example 1



Type : Angular contact ball bearing
Application : Machining center spindle
Lubrication : Grease
Symptom : Unusual noise
Probable cause : Abnormal wear due to excessive preload
Countermeasure : Adjust preload

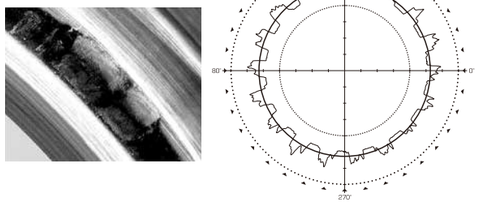
Example 2



Inner ring raceway out-of-roundness

Type : Angular contact ball bearing
Application : Machining center spindle
Lubrication : Grease
Symptom : Unusual noise
Probable cause : Brinell indentations caused by a shock load
Countermeasure : Review operating conditions

Example 3



Outer ring raceway out-of-roundness

Type : Angular contact ball bearing for ball screw support
Application : Machine tool feed shaft
Lubrication : Grease
Symptom : Unusual noise
Probable cause : Fretting caused by small vibration
Countermeasures : Change to ceramic ball bearing
Review stroke length
Add a re-organization stroke for grease and balls (dummy stroke)

Example 4



Outer ring raceway



Inner ring raceway

Type : Double-row cylindrical roller bearing
Application : Lathe spindle
Lubrication : Grease
Symptom : Seizure
Probable cause : Mismatch of shaft taper angle resulting in excessive load on one row and subsequent interruption of oil film
Countermeasure : Adjust shaft taper angle

●Please contact NSK when bearings show these kind of symptoms.

Table 1.3 Bearing Diagnostic Chart

Damage	Location (Phenomenon)	Cause: Handling: Storage, Transport	Handling: Mounting	Bearing Surroundings: Shaft, Housing	Bearing Surroundings: Seal, Water, Contaminants	Bearing Surroundings: Temperature	Lubrication: Lubricant	Lubrication: Lubrication Method	Load: Excessive Load, Shock Load	Load: Moment Load	Load: Ultra-Small Load	Speed: High Speed, High Accelaration	Speed: Shaking, Vibration, Stationary	Bearing Selection	Remarks
01. Flaking	Raceway, Rolling surface		○	○	○		○	○	○	○				○	
02. Peeling	Raceway, Rolling surface				○		○	○			○	○			
	Bearing outside surfaces (Rolling contact)			○*	○		○	○							*Mating part
03. Scoring	Roller end face, Rib surface		○	○	○		○	○	○	○		○			
	Cage guide surface, Pocket surface		○		○		○	○							
04. Smearing	Raceway, Rolling surface				○		○	○			○	○			
05. Chips	Ribs, Rollers	○	○	○					○	○					
06. Cracks	Raceway rings, Rolling elements		○	○		○			○	○					
	Rib surface, Roller end face, Cage guide surface (Thermal crack)			○				○	○	○					
07. Cage damage	(Deformation), (Fracture)		○	○					○	○					
	(Wear)		○		○		○	○	○	○		○			
08. Dents	Raceway, Rolling surface (Indentation at rolling element pitch)				○			○							
	Raceway (Dents on the rolling element pitch)	○	○						○				○		
09. Pitting	Raceway, Rolling surface				○		○	○							
10. Wear	Raceway, Rolling surface, Rib surface, Roller end face		○		○		○	○							
11. Fretting	Raceway, Rolling surface	○	○	○			○	○	○			○	○		
	Bearing outside and bore, side surface (Contact with housing and shaft)		○	○					○						
12. False brinelling	Raceway, Rolling surface	○					○	○					○		
13. Creep	Fitting surface		○	○		○	○*	○*	○			○			*Loose fit
14. Seizure	Rings, Rolling elements, Cage		○	○	○		○	○	○	○		○		○	
15. Electrical erosion	Raceway, Rolling surface		○*	○*											*Electricity passing through the rolling element
16. Rust and corrosion	Rings, Rolling elements, Cage	○	○		○	○	○	○							
17. Mounting flaws	Raceway, Rolling surface		○	○											
18. Discoloration	Raceway ring, Rolling element, Cage					○	○	○							

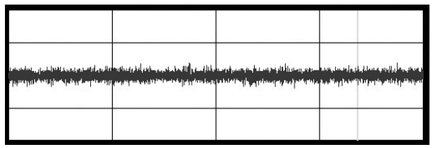
Remark: This chart is not comprehensive; it lists only the more common types of damage, causes, and locations.

2. Diagnosis with Sound and Vibration

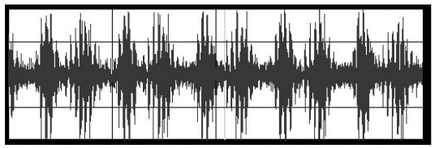
Classification of Sounds and Vibrations

Sound and vibration accompany the rotation of rolling bearings. The tone and amplitude of such sound and vibration vary depending on the type of bearing, mounting conditions, operating conditions, etc. The sound and vibration of a rolling bearing can be classified under the following four chief categories and each category can be further classified into several sub-categories, as described in Table 2.1 below. However, the distinctions between the categories are not absolute. Even if some types of sound or vibration are inherent in the bearings, the volume might be related to the manufacturing process, while some types of sound or vibration, even if they are due to manufacturing, cannot be eliminated even under normal conditions.

By recording the sounds and vibrations of a rotating machine and analyzing them, it is possible to determine their likely cause. As shown in the figures on the next page, a mechanically normal bearing shows a stable waveform. However, a bearing with a scratch, for example, shows a waveform with wide swings indicating large-amplitude sounds at regular intervals.



Sound waveform of a normal bearing



Sound waveform of a scratched bearing

Frequency analysis of bearing with damage on inner ring raceway

Bore diameter: 100 mm, Method of recording and analysis: Envelope analysis of the sounds of a test machine recorded by microphone, Speed: 50 min^{-1}

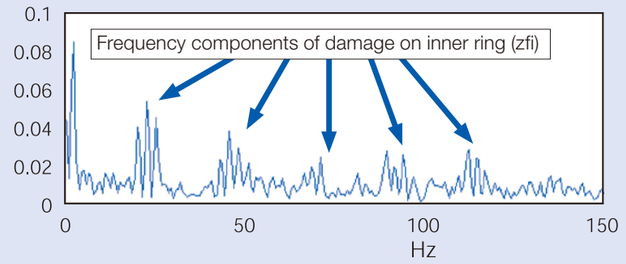


Table 2.1 Classification of Sounds and Vibrations in Rolling Bearings

	Type of Sound	Vibration		Features	Generated Frequency (Frequency Analysis) FFT of original wave: Radial (angular) direction	FFT of original wave: Axial direction	FFT after envelope (basic No.)	Source	Countermeasures
Structural	Race noise	Free vibration of raceway ring		Continuous noise: basic unavoidable noise which all bearings generate	f_{RiN}, f_{MI}	f_{AiN}, f_{AM}	—	Selective resonance from waviness (rolling friction)	Improve rigidity around bearings, provide appropriate radial clearance, use high-viscosity lubricant and high-quality bearings.
	Click noise	Free vibration of raceway ring, free vibration of cage		Regular noise at a certain interval: found in large bearings with horizontal shafts or radial loads at low rpm	f_{RiN}, f_{MI} Natural frequency of cage	f_{AiN}, f_{AM}	Zf_c	Collision of rolling elements with inner ring or cage	Reduce radial clearance, apply preload, use high-viscosity luxuriant.
	Squeal noise	Free vibration of raceway ring		Intermittent or continuous: generally found in large cylindrical roller bearings or with radial loads under grease lubrication at particular speeds	($\approx f_{R2N}$, f_{R3N})	—	?	Self-induced vibration caused by sliding friction at rolling surface	Reduce radial clearance, apply preload, change grease, replace with bearings with countermeasures.
	Cage noise "CK" noise (Kacha-kacha)	Free vibration of cage		Regular noise at a set interval: generated by all bearing types	Natural frequency of cage		f_c	Collision of cage with rolling elements or rings	Apply preload, use high-viscosity lubricant, reduce mounting error.
	Cage noise "CG" noise (Gaga-gaga)	Vibration of cage		Intermittent or continuous: found with particular types of grease	Natural frequency of cage		?	Self-induced vibration caused by friction at cage guide surface	Change grease brand, replace with cage with countermeasures.
	Cage noise Tapping noise	Free vibration of cage		Set interval: slightly irregular under radial load and during initial stage	Natural frequency of cage		Zf_c	Collision of cage and rolling element caused by grease resistance	Reduce radial clearance, apply preload, use low-viscosity lubricant.
	Rumbling	Rolling element passage vibration		Continuous noise: found in all bearings under radial load	Zf_c	—	—	Displacement of inner ring due to rolling element passage	Reduce radial clearance, apply preload.
Manufacturing	Chatter noise	Vibration due to waviness	Inner ring	Continuous noise	$nZf_i \pm f_r$ ($nZ \pm 1$ peaks)	nZf_i (nZ peaks)	—	Inner ring raceway waviness, irregularity of shaft exterior	Use high-quality bearings, improve shaft accuracy.
			Outer ring		nZf_c ($nZ \pm 1$ peaks)	nZf_c (nZ peaks)	—	Outer ring raceway waviness, irregular housing bore	Use high-quality bearings, improve housing bore accuracy.
			Rolling element	Continuous noise, may pause occasionally (ball bearings)	$2nf_b \pm f_c$ ($2n$ peaks)	$2nf_b$ ($2n$ peaks)	—	Rolling element waviness	Use high-quality bearings.
Handling	Flaw noise	Vibration due to flaw	Inner ring	Regular noise at a certain interval	f_{RiN}, f_{MI}	f_{AiN}, f_{AM}	Zf_i	Nicks, dents, rust, flaking on inner ring raceway	Replace bearing and take care when handling.
			Outer ring				Zf_c	Nicks, dents, rust, flaking on inner ring raceway	Replace bearing and take care when handling.
			Rolling element	Regular noise, may pause occasionally (ball bearings)			$2f_b$	Nicks, dents, rust, flaking on rolling elements	Replace bearing and take care when handling.
	Contamination noise	Vibration due to contamination		Irregular	f_{RiN}, f_{MI}	f_{AiN}, f_{AM}	Irregular	Entry of dirt and debris	Clean the bearing, improve sealing quality.
Others	Seal noise	Free vibration of seal		Occurs in contact seals	Natural frequency of seal		(f_r)	Self-induced vibration due to friction at seal contact area	Change the seal, change the grease.
	Lubricant noise	—		Irregular	?	?	Irregular	Lubricant or lubricant bubbles crushed between rolling elements and raceways	Change the grease.
	Rumbling	Shaft whirl	f_r	Continuous	f_r	—	—	Irregular inner ring cross-section	Use high-quality bearings.
			f_c	Continuous	f_c	—	—	Displacement of inner ring (shaft) from ball variation or unequal rolling element spacing	Use high-quality bearings.
			f_r-$2f_c$	Continuous	f_r-$2f_c$	—	—	Nonlinear vibration from anisotropic rigidity due to ball variation	Use high-quality bearings.

n: Positive integer (1, 2, 3...)
Z: Number of rolling elements
f_{RIN}: Ring natural frequency in radial bending mode (Hz)
f_{MI}: Natural frequency in the mode of angular vibration in inertia of outer ring-spring system (Hz)
f_r: Rotation frequency of inner ring (Hz)
f_c: Orbital revolution frequency of rolling elements (Hz)
f_{AiN}: Ring natural frequency in axial bending mode (Hz)
f_{AM}: Natural frequency in the mode of axial vibration in mass of outer ring-spring system (Hz)
f_i: $f_i = f_r - f_c$ (Hz)
f_b: Rotation frequency of rolling element around its center (Hz)

Part 8

Appendices

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Shaft Tolerance Classes

Unit: μm

Diameter Classification (mm)		d6	e6	f6	g5	g6	h5	h6	h7	h8	h9	h10	js5	js6
Over	Incl.													
3	6	− 30 − 38	− 20 − 28	− 10 − 18	− 4 − 9	− 4 − 12	0 − 5	0 − 8	0 − 12	0 − 18	0 − 30	0 − 48	± 2.5	± 4
6	10	− 40 − 49	− 25 − 34	− 13 − 22	− 5 − 11	− 5 − 14	0 − 6	0 − 9	0 − 15	0 − 22	0 − 36	0 − 58	± 3	± 4.5
10	18	− 50 − 61	− 32 − 43	− 16 − 27	− 6 − 14	− 6 − 17	0 − 8	0 − 11	0 − 18	0 − 27	0 − 43	0 − 70	± 4	± 5.5
18	30	− 65 − 78	− 40 − 53	− 20 − 33	− 7 − 16	− 7 − 20	0 − 9	0 − 13	0 − 21	0 − 33	0 − 52	0 − 84	± 4.5	± 6.5
30	50	− 80 − 96	− 50 − 66	− 25 − 41	− 9 − 20	− 9 − 25	0 − 11	0 − 16	0 − 25	0 − 39	0 − 62	0 −100	± 5.5	± 8
50	80	−100 −119	− 60 − 79	− 30 − 49	− 10 − 23	− 10 − 29	0 − 13	0 − 19	0 − 30	0 − 46	0 − 74	0 −120	± 6.5	± 9.5
80	120	−120 −142	− 72 − 94	− 36 − 58	− 12 − 27	− 12 − 34	0 − 15	0 − 22	0 − 35	0 − 54	0 − 87	0 −140	± 7.5	± 11
120	180	−145 −170	− 85 −110	− 43 − 68	− 14 − 32	− 14 − 39	0 − 18	0 − 25	0 − 40	0 − 63	0 −100	0 −160	± 9	± 12.5
180	250	−170 −199	−100 −129	− 50 − 79	− 15 − 35	− 15 − 44	0 − 20	0 − 29	0 − 46	0 − 72	0 −115	0 −185	± 10	± 14.5
250	315	−190 −222	−110 −142	− 56 − 88	− 17 − 40	− 17 − 49	0 − 23	0 − 32	0 − 52	0 − 81	0 −130	0 −210	± 11.5	± 16
315	400	−210 −246	−125 −161	− 62 − 98	− 18 − 43	− 18 − 54	0 − 25	0 − 36	0 − 57	0 − 89	0 −140	0 −230	± 12.5	± 18
400	500	−230 −270	−135 −175	− 68 −108	− 20 − 47	− 20 − 60	0 − 27	0 − 40	0 − 63	0 − 97	0 −155	0 −250	± 13.5	± 20
500	630	−260 −304	−145 −189	− 76 −120	—	− 22 − 66	—	0 − 44	0 − 70	0 −110	0 −175	0 −280	—	± 22
630	800	−290 −340	−160 −210	− 80 −130	—	− 24 − 74	—	0 − 50	0 − 80	0 −125	0 −200	0 −320	—	± 25
800	1 000	−320 −376	−170 −226	− 86 −142	—	− 26 − 82	—	0 − 56	0 − 90	0 −140	0 −230	0 −360	—	± 28
1 000	1 250	− 350 − 416	−195 −261	− 98 −164	—	− 28 − 94	—	0 − 66	0 −105	0 −165	0 −260	0 −420	—	± 33
1 250	1 600	−390 −468	−220 −298	−110 −188	—	− 30 −108	—	0 − 78	0 −125	0 −195	0 −310	0 −500	—	± 39
1 600	2 000	−430 −522	−240 −332	−120 −212	—	− 32 −124	—	0 − 92	0 −150	0 −230	0 −370	0 −600	—	± 46

j5	j6	j7	k5	k6	k7	m5	m6	n6	p6	r6	r7	Diameter Classification (mm)	
												Over	Incl.
− 3 − 2	− 6 − 2	− 8 − 4	+ 6 + 1	+ 9 + 1	+ 13 + 1	+ 9 + 4	+ 12 + 4	+ 16 + 8	+ 20 + 12	+ 23 + 15	+ 27 + 15	3	6
+ 4 − 2	+ 7 − 2	+ 10 − 5	+ 7 + 1	+ 10 + 1	+ 16 + 1	+ 12 + 6	+ 15 + 6	+ 19 + 10	+ 24 + 15	+ 28 + 19	+ 34 + 19	6	10
+ 5 − 3	+ 8 − 3	+ 12 − 6	+ 9 + 1	+ 12 + 1	+ 19 + 1	+ 15 + 7	+ 18 + 7	+ 23 + 12	+ 29 + 18	+ 34 + 23	+ 41 + 23	10	18
+ 5 − 4	+ 9 − 4	+ 13 − 8	+ 11 + 2	+ 15 + 2	+ 23 + 2	+ 17 + 8	+ 21 + 8	+ 28 + 15	+ 35 + 22	+ 41 + 28	+ 49 + 28	18	30
+ 6 − 5	+ 11 − 5	+ 15 − 10	+ 13 + 2	+ 18 + 2	+ 27 + 2	+ 20 + 9	+ 25 + 9	+ 33 + 17	+ 42 + 26	+ 50 + 34	+ 59 + 34	30	50
+ 6 − 7	+ 12 − 7	+ 18 − 12	+ 15 + 2	+ 21 + 2	+ 32 + 2	+ 24 + 11	+ 30 + 11	+ 39 + 20	+ 51 + 32	+ 60 + 41	+ 71 + 41	50	65
										+ 62 + 43	+ 73 + 43	65	80
+ 6 − 9	+ 13 − 9	+ 20 − 15	+ 18 + 3	+ 25 + 3	+ 38 + 3	+ 28 + 13	+ 35 + 13	+ 45 + 23	+ 59 + 37	+ 73 + 51	+ 86 + 51	80	100
										+ 76 + 54	+ 89 + 54	100	120
+ 7 − 11	+ 14 − 11	+ 22 − 18	+ 21 + 3	+ 28 + 3	+ 43 + 3	+ 33 + 15	+ 40 + 15	+ 52 + 27	+ 68 + 43	+ 88 + 63	+103 + 63	120	140
										+ 90 + 65	+105 + 65	140	160
										+ 93 + 68	+108 + 68	160	180
+ 7 − 13	+ 16 − 13	+ 25 − 21	+ 24 + 4	+ 33 + 4	+ 50 + 4	+ 37 + 17	+ 46 + 17	+ 60 + 31	+ 79 + 50	+106 + 77	+123 + 77	180	200
										+109 + 80	+126 + 80	200	225
										+113 + 84	+130 + 84	225	250
+ 7 − 16	± 16	± 26	+ 27 + 4	+ 36 + 4	+ 56 + 4	+ 43 + 20	+ 52 + 20	+ 66 + 34	+ 88 + 56	+126 + 94	+146 + 94	250	280
										+130 + 98	+150 + 98	280	315
+ 7 − 18	± 18	+ 29 − 28	+ 29 + 4	+ 40 + 4	+ 61 + 4	+ 46 + 21	+ 57 + 21	+ 73 + 37	+ 98 + 62	+144 +108	+165 +108	315	355
										+150 +114	+171 +114	355	400
+ 7 − 20	± 20	+ 31 − 32	+ 32 + 5	+ 45 + 5	+ 68 + 5	+ 50 + 23	+ 63 + 23	+ 80 + 40	+108 + 68	+166 +126	+189 +126	400	450
										+172 +132	+195 +132	450	500
—	—	—	—	+ 44 0	+ 70 0	—	+ 70 + 26	+ 88 + 44	+122 + 78	+194 +150	+220 +150	500	560
										+199 +155	+225 +155	560	630
—	—	—	—	+ 50 0	+ 80 0	—	+ 80 + 30	+100 + 50	+138 + 88	+225 +175	+255 +175	630	710
										+235 +185	+265 +185	710	800
—	—	—	—	+ 56 0	+ 90 0	—	+ 90 + 34	+112 + 56	+156 +100	+266 +210	+300 +210	800	900
										+276 +220	+310 +220	900	1 000
—	—	—	—	+ 66 0	+105 0	—	+106 + 40	+132 + 66	+186 +120	+316 +250	+355 +250	1 000	1 120
										+326 +260	+365 +260	1 120	1 250
—	—	—	—	+ 78 0	+125 0	—	+126 + 48	+156 + 78	+218 +140	+378 +300	+425 +300	1 250	1 400
										+408 +330	+455 +330	1 400	1 600
—	—	—	—	+ 92 0	+150 0	—	+150 + 58	+184 + 92	+262 +170	+462 +370	+520 +370	1 600	1 800
										+492 +400	+550 +400	1 800	2 000

Housing Bore Tolerance Classes

Unit: μm

Diameter Classification (mm)		E6	F6	F7	G6	G7	H6	H7	H8	J6	J7	JS6	JS7
Over	Incl.												
10	**18**	+ 43 + 32	+ 27 + 16	+ 34 + 16	+ 17 + 6	+ 24 + 6	+ 11 0	+ 18 0	+ 27 0	+ 6 − 5	+ 10 − 8	± 5.5	± 9
18	**30**	+ 53 + 40	+ 33 + 20	+ 41 + 20	+ 20 + 7	+ 28 + 7	+ 13 0	+ 21 0	+ 33 0	+ 8 − 5	+ 12 − 9	± 6.5	± 10.5
30	**50**	+ 66 + 50	+ 41 + 25	+ 50 + 25	+ 25 + 9	+ 34 + 9	+ 16 0	+ 25 0	+ 39 0	+ 10 − 6	+ 14 − 11	± 8	± 12.5
50	**80**	+ 79 + 60	+ 49 + 30	+ 60 + 30	+ 29 + 10	+ 40 + 10	+ 19 0	+ 30 0	+ 46 0	+ 13 − 6	+ 18 − 12	± 9.5	± 15
80	**120**	+ 94 + 72	+ 58 + 36	+ 71 + 36	+ 34 + 12	+ 47 + 12	+ 22 0	+ 35 0	+ 54 0	+ 16 − 6	+ 22 − 13	± 11	± 17.5
120 **150**	**150** **180**	+ 110 + 85	+ 68 + 43	+ 83 + 43	+ 39 + 14	+ 54 + 14	+ 25 0	+ 40 0	+ 63 0	+ 18 − 7	+ 26 − 14	± 12.5	± 20
180	**250**	+ 129 + 100	+ 79 + 50	+ 96 + 50	+ 44 + 15	+ 61 + 15	+ 29 0	+ 46 0	+ 72 0	+ 22 − 7	+ 30 − 16	± 14.5	± 23
250	**315**	+ 142 + 110	+ 88 + 56	+ 108 + 56	+ 49 + 17	+ 69 + 17	+ 32 0	+ 52 0	+ 81 0	+ 25 − 7	+ 36 − 16	± 16	± 26
315	**400**	+ 161 + 125	+ 98 + 62	+ 119 + 62	+ 54 + 18	+ 75 + 18	+ 36 0	+ 57 0	+ 89 0	+ 29 − 7	+ 39 − 18	± 18	± 28.5
400	**500**	+ 175 + 135	+ 108 + 68	+ 131 + 68	+ 60 + 20	+ 83 + 20	+ 40 0	+ 63 0	+ 97 0	+ 33 − 7	+ 43 − 20	± 20	± 31.5
500	**630**	+ 189 + 145	+ 120 + 76	+ 146 + 76	+ 66 + 22	+ 92 + 22	+ 44 0	+ 70 0	+ 110 0	—	—	± 22	± 35
630	**800**	+ 210 + 160	+ 130 + 80	+ 160 + 80	+ 74 + 24	+ 104 + 24	+ 50 0	+ 80 0	+ 125 0	—	—	± 25	± 40
800	**1 000**	+ 226 + 170	+ 142 + 86	+ 176 + 86	+ 82 + 26	+ 116 + 26	+ 56 0	+ 90 0	+ 140 0	—	—	± 28	± 45
1 000	**1 250**	+ 261 + 195	+ 164 + 98	+ 203 + 98	+ 94 + 28	+ 133 + 28	+ 66 0	+ 105 0	+ 165 0	—	—	± 33	± 52.5
1 250	**1 600**	+ 298 + 220	+ 188 + 110	+ 235 + 110	+ 108 + 30	+ 155 + 30	+ 78 0	+ 125 0	+ 195 0	—	—	± 39	± 62.5
1 600	**2 000**	+ 332 + 240	+ 212 + 120	+ 270 + 120	+ 124 + 32	+ 182 + 32	+ 92 0	+ 150 0	+ 230 0	—	—	± 46	± 75
2 000	**2 500**	+ 370 + 260	+ 240 + 130	+ 305 + 130	+ 144 + 34	+ 209 + 34	+ 110 0	+ 175 0	+ 280 0	—	—	± 55	± 87.5

K5	K6	K7	M5	M6	M7	N5	N6	N7	P6	P7	Diameter Classification (mm)	
											Over	Incl.
+ 2 − 6	+ 2 − 9	+ 6 − 12	− 4 − 12	− 4 − 15	0 − 18	− 9 − 17	− 9 − 20	− 5 − 23	− 15 − 26	− 11 − 29	**10**	**18**
+ 1 − 8	+ 2 − 11	+ 6 − 15	− 5 − 14	− 4 − 17	0 − 21	− 12 − 21	− 11 − 24	− 7 − 28	− 18 − 31	− 14 − 35	**18**	**30**
+ 2 − 9	+ 3 − 13	+ 7 − 18	− 5 − 16	− 4 − 20	0 − 25	− 13 − 24	− 12 − 28	− 8 − 33	− 21 − 37	− 17 − 42	**30**	**50**
+ 3 −10	+ 4 − 15	+ 9 − 21	− 6 − 19	− 5 − 24	0 − 30	− 15 − 28	− 14 − 33	− 9 − 39	− 26 − 45	− 21 − 51	**50**	**80**
+ 2 −13	+ 4 − 18	+ 10 − 25	− 8 − 23	− 6 − 28	0 − 35	− 18 − 33	− 16 − 38	− 10 − 45	− 30 − 52	− 24 − 59	**80**	**120**
+ 3 −15	+ 4 − 21	+ 12 − 28	− 9 − 27	− 8 − 33	0 − 40	− 21 − 39	− 20 − 45	− 12 − 52	− 36 − 61	− 28 − 68	**120**	**180**
+ 2 −18	+ 5 − 24	+ 13 − 33	− 11 − 31	− 8 − 37	0 − 46	− 25 − 45	− 22 − 51	− 14 − 60	− 41 − 70	− 33 − 79	**180**	**250**
+ 3 −20	+ 5 − 27	+ 16 − 36	− 13 − 36	− 9 − 41	0 − 52	− 27 − 50	− 25 − 57	− 14 − 66	− 47 − 79	− 36 − 88	**250**	**315**
+ 3 −22	+ 7 − 29	+ 17 − 40	− 14 − 39	− 10 − 46	0 − 57	− 30 − 55	− 26 − 62	− 16 − 73	− 51 − 87	− 41 − 98	**315**	**400**
+ 2 −25	+ 8 − 32	+ 18 − 45	− 16 − 43	− 10 − 50	0 − 63	− 33 − 60	− 27 − 67	− 17 − 80	− 55 − 95	− 45 −108	**400**	**500**
—	0 − 44	0 − 70	—	− 26 − 70	− 26 − 96	—	− 44 − 88	− 44 −114	− 78 −122	− 78 −148	**500**	**630**
—	0 − 50	0 − 80	—	− 30 − 80	− 30 −110	—	− 50 −100	− 50 −130	− 88 −138	− 88 −168	**630**	**800**
—	0 − 56	0 − 90	—	− 34 − 90	− 34 −124	—	− 56 −112	− 56 −146	−100 −156	−100 −190	**800**	**1 000**
—	0 − 66	0 −105	—	− 40 −106	− 40 −145	—	− 66 −132	− 66 −171	−120 −186	−120 −225	**1 000**	**1 250**
—	0 − 78	0 −125	—	− 48 −126	− 48 −173	—	− 78 −156	− 78 −203	−140 −218	−140 −265	**1 250**	**1 600**
—	0 − 92	0 −150	—	− 58 −150	− 58 −208	—	− 92 −184	− 92 −242	−170 −262	−170 −320	**1 600**	**2 000**
—	0 −110	0 −175	—	− 68 −178	− 68 −243	—	−110 −220	−110 −285	−195 −305	−195 −370	**2 000**	**2 500**

Appendices

Values of IT Standard Tolerance Grades

Basic Size (mm)		Standard Grades								
		IT1	IT2	IT3	IT4	IT5	IT6	IT7	IT8	IT9
Over	Incl.	Tolerances (μm)								
—	3	0.8	1.2	2	3	4	6	10	14	25
3	6	1	1.5	2.5	4	5	8	12	18	30
6	10	1	1.5	2.5	4	6	9	15	22	36
10	18	1.2	2	3	5	8	11	18	27	43
18	30	1.5	2.5	4	6	9	13	21	33	52
30	50	1.5	2.5	4	7	11	16	25	39	62
50	80	2	3	5	8	13	19	30	46	74
80	120	2.5	4	6	10	15	22	35	54	87
120	180	3.5	5	8	12	18	25	40	63	100
180	250	4.5	7	10	14	20	29	46	72	115
250	315	6	8	12	16	23	32	52	81	130
315	400	7	9	13	18	25	36	57	89	140
400	500	8	10	15	20	27	40	63	97	155
500	630	9	11	16	22	32	44	70	110	175
630	800	10	13	18	25	36	50	80	125	200
800	1 000	11	15	21	28	40	56	90	140	230
1 000	1 250	13	18	24	33	47	66	105	165	260
1 250	1 600	15	21	29	39	55	78	125	195	310
1 600	2 000	18	25	35	46	65	92	150	230	370
2 000	2 500	22	30	41	55	78	110	175	280	440
2 500	3 150	26	36	50	68	96	135	210	330	540

Standard Grades									Basic Size (mm)	
IT10	IT11	IT12	IT13	IT14	IT15	IT16	IT17	IT18		
		Tolerances (mm)							Over	Incl.
40	60	0.10	0.14	0.25	0.40	0.60	1.00	1.40	—	3
48	75	0.12	0.18	0.30	0.48	0.75	1.20	1.80	3	6
58	90	0.15	0.22	0.36	0.58	0.90	1.50	2.20	6	10
70	110	0.18	0.27	0.43	0.70	1.10	1.80	2.70	10	18
84	130	0.21	0.33	0.52	0.84	1.30	2.10	3.30	18	30
100	160	0.25	0.39	0.62	1.00	1.60	2.50	3.90	30	50
120	190	0.30	0.46	0.74	1.20	1.90	3.00	4.60	50	80
140	220	0.35	0.54	0.87	1.40	2.20	3.50	5.40	80	120
160	250	0.40	0.63	1.00	1.60	2.50	4.00	6.30	120	180
185	290	0.46	0.72	1.15	1.85	2.90	4.60	7.20	180	250
210	320	0.52	0.81	1.30	2.10	3.20	5.20	8.10	250	315
230	360	0.57	0.89	1.40	2.30	3.60	5.70	8.90	315	400
250	400	0.63	0.97	1.55	2.50	4.00	6.30	9.70	400	500
280	440	0.70	1.10	1.75	2.80	4.40	7.00	11.00	500	630
320	500	0.80	1.25	2.00	3.20	5.00	8.00	12.50	630	800
360	560	0.90	1.40	2.30	3.60	5.60	9.00	14.00	800	1 000
420	660	1.05	1.65	2.60	4.20	6.60	10.50	16.50	1 000	1 250
500	780	1.25	1.95	3.10	5.00	7.80	12.50	19.50	1 250	1 600
600	920	1.50	2.30	3.70	6.00	9.20	15.00	23.00	1 600	2 000
700	1 100	1.75	2.80	4.40	7.00	11.00	17.50	28.00	2 000	2 500
860	1 350	2.10	3.30	5.40	8.60	13.50	21.00	33.00	2 500	3 150

Remarks 1. Standard tolerance grades IT14 to IT18 must not be used for basic sizes less than or equal to 1 mm.
2. Values for standard tolerance grades IT1 to IT5 for basic sizes over 500 mm are included for experimental use.

Appendices

Physical and Mechanical Properties of Materials

Application	Material	Heat treatment	Density g/cm³	Specific heat KJ/(kg K)	Thermal conductivity W/(m K)	Electric resistance μΩ·cm	Coefficient of linear expansion (0°–100°C) ×10⁻⁶/°C	Young's modulus MPa	Yield point MPa	Tensile strength MPa	Elongation %	Hardness HB
Shaft	Carbon steel for machine structure S45C	Quenching, 650 °C tempering	7.83	0.48	47	18	12.8	207 000	440	735	25	217
	Chrome steel SCr430	Quenching, 520 to 620 °C fast cooling				22	12.5		Min. *637	Min. *784	Min. *18	*229 –293
	Chrome steel SCr440	Quenching, 520 to 620 °C fast cooling			45	23		208 000	Min. *784	Min. *930	Min. *13	*269 –331
	Chrome Molybdenum steel SCM420	Quenching, 150 to 200 °C air cooling		0.47	48	21	12.8		—	Min. *930	Min. *14	*262 –352
	Nickel-chromium-molybdenum steel SNCM439	Quenching, 650 °C tempering			38	30	11.3	207 000	920	1 030	18	320
	Low carbon cast steel SC46	Normalizing	—	—	—	—	—	206 000	294	520	27	143
	Martensitic stainless steel SUS420J2	1038 °C oil cooling, 400 °C air cooling.	7.75	0.46	22	55	10.4	200 000	1 440	1 650	10	400
Housing	Gray cast iron FC200	Casting	7.3	0.50	43	—		98 000	—	Min. *200	—	Max. *217
	Spheroidal graphite cast iron FCD400	Casting	7.0	0.48	20	—	11.7		Min. *250	Min. *400	Min. *12	Max. *201
	Pure aluminum A1100	Annealing	2.69	0.90	222	3.0	23.7	70 600	34	78	35	—
	Aluminum alloy for sand casting AC4C	Casting	2.68	0.88	151	4.2	21.5	72 000	88	167	7	—
	Aluminum alloy for die casting ADC10	Casting	2.74	0.96	96	7.5	22.0	71 000	167	323	4	—
	Austenitic stainless steel SUS304	Annealing	8.03	0.50	15	72	15.7–16.8	193 000	245	588	60	150

Notes *Standard JIS value or reference value.

Physical and Mechanical Properties of Materials

Application	Material	Heat treatment	Density g/cm³	Specific heat KJ/(kg K)	Thermal conductivity W/(m K)	Electric resistance μΩ·cm	Coefficient of linear expansion (0°–100°C) ×10⁻⁶/°C	Young's modulus MPa	Yield point MPa	Tensile strength MPa	Elongation %	Hardness HB
Bearing	High-carbon chrome bearing steel No.2 SUJ2	Quenching, tempering	7.83	0.47	46	22	12.5	208 000	1 370	1 570 –1 960	Max. 0.5	650 –740
		Spheroidizing annealing	7.86				11.9		420	647	27	180
	Chrome steel SCr420	Quenching, Low temp. tempering	7.83		48	21	12.8		882	1 225	15	370
	Nickel-chromium-molybdenum steel SAE4320 (SNCM420)	Quenching, Low temp. tempering			44	20	11.7		902	1 009	16	**293 –375
	Nickel-chromium-molybdenum steel SNCM815	Quenching, Low temp. tempering	7.89		40	35	—		—	Min. *1 080	Min. *12	*311 –375
	Martensitic stainless steel SUS440C	Quenching, Low temp. tempering	7.68	0.46	24	60	10.1	200 000	1 860	1 960	—	**580
	Cold rolled steel plate SPCC	Annealing	7.86	0.47	59	15	11.6	206 000	—	Min. *275	Min. *32	—
	Carbon steel for machine structure S25C	Annealing		0.48	50	17	11.8		323	431	33	120
	High-tension brass CAC301 (HB_SC1)		8.5	0.38	123	6.2	19.1	103 000	—	Min. *431	Min. *20	—

Notes *Standard JIS value or reference value.

**The hardness of such materials is usually expressed using the Rockwell C scale, but it is listed here as Brinell hardness for ease of comparison.

Reference The proportional limits of SUJ2 and SCr420 are 833 MPa (85kgf/mm²) and 440 MPa (45kgf/mm²) respectively.

Appendices

Conversion Tables from SI (International Unit) System

The values accompanied by (*) in the marked boxes denote SI units.

Comparison of SI, CGS, and Engineering Units (Gravitational Metric System)

Units / Unit System	Length	Mass	Time	Temperature	Acceleration	Force	Stress	Pressure	Energy	Power
SI	m	kg	s	K, °C	m/s^2	N	Pa	Pa	J	W
CGS System	cm	g	s	°C	Gal	dyn	dyn/cm^2	dyn/cm^2	erg	erg/s
Engineering Units	m	$kgf \cdot s^2/m$	s	°C	m/s^2	kgf	kgf/m^2	kgf/m^2	kgf·m	kgf·m/s

Prefixes Used in SI

Multiples	Prefix	Code	Multiples	Prefix	Code
10^{18}	Exa	E	10^{-1}	Deci	d
10^{15}	Peta	P	10^{-2}	Centi	c
10^{12}	Tera	T	10^{-3}	Milli	m
10^{9}	Giga	G	10^{-6}	Micro	μ
10^{6}	Mega	M	10^{-9}	Nano	n
10^{3}	Kilo	k	10^{-12}	Pico	p
10^{2}	Hecto	h	10^{-15}	Femto	f
10	Deca	da	10^{-18}	Ato	a

Force

*N	dyn	kgf
1	1×10^{5}	1.01972×10^{-1}
1×10^{-5}	1	1.01972×10^{-6}
9.80665	9.80665×10^{5}	1

Stress

*Pa	*MPa or N/mm^2	kgf/mm^2	kgf/cm^2
1	1×10^{-6}	1.01972×10^{-7}	1.01972×10^{-5}
1×10^{6}	1	1.01972×10^{-1}	1.01972×10
9.80665×10^{6}	9.80665	1	1×10^{2}
9.80665×10^{4}	9.80665×10^{-2}	1×10^{-2}	1

Pressure

*Pa	bar	kgf/cm^2	atm	mmH_2O	mmHg or Torr
1	1×10^{-5}	1.01972×10^{-5}	9.86923×10^{-6}	1.01972×10^{-1}	7.50062×10^{-3}
1×10^{5}	1	1.01972	9.86923×10^{-1}	1.01972×10^{4}	7.50062×10^{2}
9.80665×10^{4}	9.80665×10^{-1}	1	9.67841×10^{-1}	1×10^{4}	7.35559×10^{2}
1.01325×10^{5}	1.01325	1.03323	1	1.03323×10^{4}	7.60000×10^{2}
9.80665	9.80665×10^{-5}	1×10^{-4}	9.67841×10^{-5}	1	7.35559×10^{-2}
1.33322×10^{2}	1.33322×10^{-3}	1.35951×10^{-3}	1.31579×10^{-3}	1.35951×10	1

Note: $1Pa=1N/m^2$

Viscosity

*Pa・s	cP	P
1	1×10^{3}	1×10
1×10^{-3}	1	1×10^{-2}
1×10^{-1}	1×10^{2}	1

Thermal Conductivity

*W/(m·K)	$kcal_{IT}/(h\cdot m\cdot°C)$
1	8.59845×10^{-1}
1.16300	1

Note: $1Pa=1dyn\cdot s/cm^2=1g/cm\cdot s$, $1Pa\cdot s=1N\cdot s/m^2$, 1cP=1mPa·s

Kinematic Viscosity

$*m^2/s$	cSt	St
1	1×10^{6}	1×10^{4}
1×10^{-6}	1	1×10^{-2}
1×10^{-4}	1×10^{2}	1

Heat Transfer Coefficient

$*W/(m^2\cdot K)$	$kcal_{IT}/(h\cdot m^2\cdot °C)$
1	8.59845×10^{-1}
1.16300	1

Note: $1St=1cm^2/s$, $1cSt=1mm^2/s$,

Energy, Work, Amount of Heat

*J	kW·h	kgf·m	$kcal_{IT}$
1	2.77778×10^{-7}	1.01972×10^{-1}	2.38846×10^{-4}
3.600×10^{6}	1	3.67098×10^{5}	8.5985×10^{2}
9.80665	2.272407×10^{-6}	1	2.34228×10^{-3}
4.18680×10^{3}	1.16300×10^{-3}	4.26935×10^{2}	1

Note: 1J=1W·s, 1W·h=3600W·s $1cal_{IT}=4.186\ 8J$

Power, Thermal Flow

*W	kgf·m/s	PS	$kcal_{IT}/h$
1	1.01972×10^{-1}	1.35962×10^{-3}	8.59845×10^{-1}
9.80665	1	1.33333×10^{-2}	8.43220
7.355×10^{2}	7.5×10	1	6.32415×10^{2}
1.16300	1.18593×10^{-1}	1.58124×10^{-3}	1

Note: 1W=1J/s, PS: Metric horsepower

Bearing Specifications Request Form

Feel free to copy and use this form to request bearing specifications from your nearest NSK sales office.

Specification Request for Super Precision Bearings

Company name: _____

Background/reason for request

(Example: To increase spindle rigidity, to avoid previous bearing damage, etc.)

Attachments: ☐ Yes ☐ No

●Specifications:

Machine type: ☐ Machining center ☐ Lathe ☐ Grinder ☐ Other ( )

Model No.: _____

Orientation: ☐ Vertical ☐ Horizontal ☐ Swiveling

Drive type: ☐ Built-in motor spindle ☐ Belt drive ☐ Direct drive ☐ Gear drive

Maximum speed: _____ min^{-1} Unclamping force: _____ N

Lubrication: ☐ Grease ☐ Oil-air ☐ Oil mist ☐ Circulating lubrication

☐ Other ( )

Bearing arrangement: (Example: Angular contact ball bearing ∅ Cylindrical roller bearing □□ , □ Spacers ∅=∅ Spring WW∅)

(Include the designation of any bearings in use)

Front side bearings | Rear side bearings

Shaft material: _____ Housing material: _____

Front-side (fixed side) bearing	Rear-side (free side) bearing
Designation:	Designation:
Bore diameter: ㎜ Outside diameter: ㎜	Bore diameter: ㎜ Outside diameter: ㎜
Combined width: ㎜ Accuracy class:	Combined width: ㎜ Accuracy class:
Clearance/preload: (e.g. EL, 800N, CC9)	Clearance/preload: (e.g. EL, 800N, CC9)
Jacket cooling: ☐ Yes ☐ No Sleeve: ☐ Yes ☐ No	Jacket cooling: ☐ Yes ☐ No Sleeve: ☐ Yes ☐ No
Spacer width: 1st Spacer ㎜, 2nd Spacer ㎜, 3rd Spacer ㎜	Spacer width: 1st Spacer ㎜, 2nd Spacer ㎜, 3rd Spacer ㎜
Shaft bore: ㎜ Housing outside diameter: ㎜	Shaft bore: ㎜ Housing outside diameter: ㎜
Inner ring temp.: °C Outer ring temp.: °C	Inner ring temp.: °C Outer ring temp.: °C
Shaft fit:	Shaft fit:
Housing fit:	Housing fit:

● Priorities, Additional Comments, etc.

[Example (in order of importance):
1. Rolling fatigue life: Over 20 000 h
2. Mounted axial rigidity: 100 N/μm, etc.]

(Note that it may not be possible to fulfill all requested criteria.)

*To request life calculations under external load, include details on load conditions (acting load position, amount/direction of load, rotational speed, and duty cycle rate) and spindle structure (dimensions and distance between bearings).