Koyo®

Ball & Roller Bearings



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BALL & ROLLER BEARINGS

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Publication of Rolling Bearing Catalog

Today's technology-based society, in order to utilize the earth's limited resources effectively and protect the environment, must strive to develop new technologies and alternate energy sources, and in that connection it continues to pursue new targets in various fields. To achieve such targets, technically advanced and highly functional rolling bearings with significantly greater compactness, lighter weight, longer life and lower friction as well as higher reliability during use in special environments are sought.

This new-edition catalog is based on the results of wide-ranging technical studies and extensive R&D efforts and will enable the reader to select the optimal bearing for each application.

JTEKT is confident that you will find this new catalog useful in the selection and use of rolling bearings. JTEKT is grateful for your patronage and look forward to continuing to serve you in the future.

★The contents of this catalog are subject to change without prior notice. Every possible effort has been made to ensure that the data herein is correct; however, JTEKT cannot assume responsibility for any errors or omissions.

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1. Rolling bearing structures and types

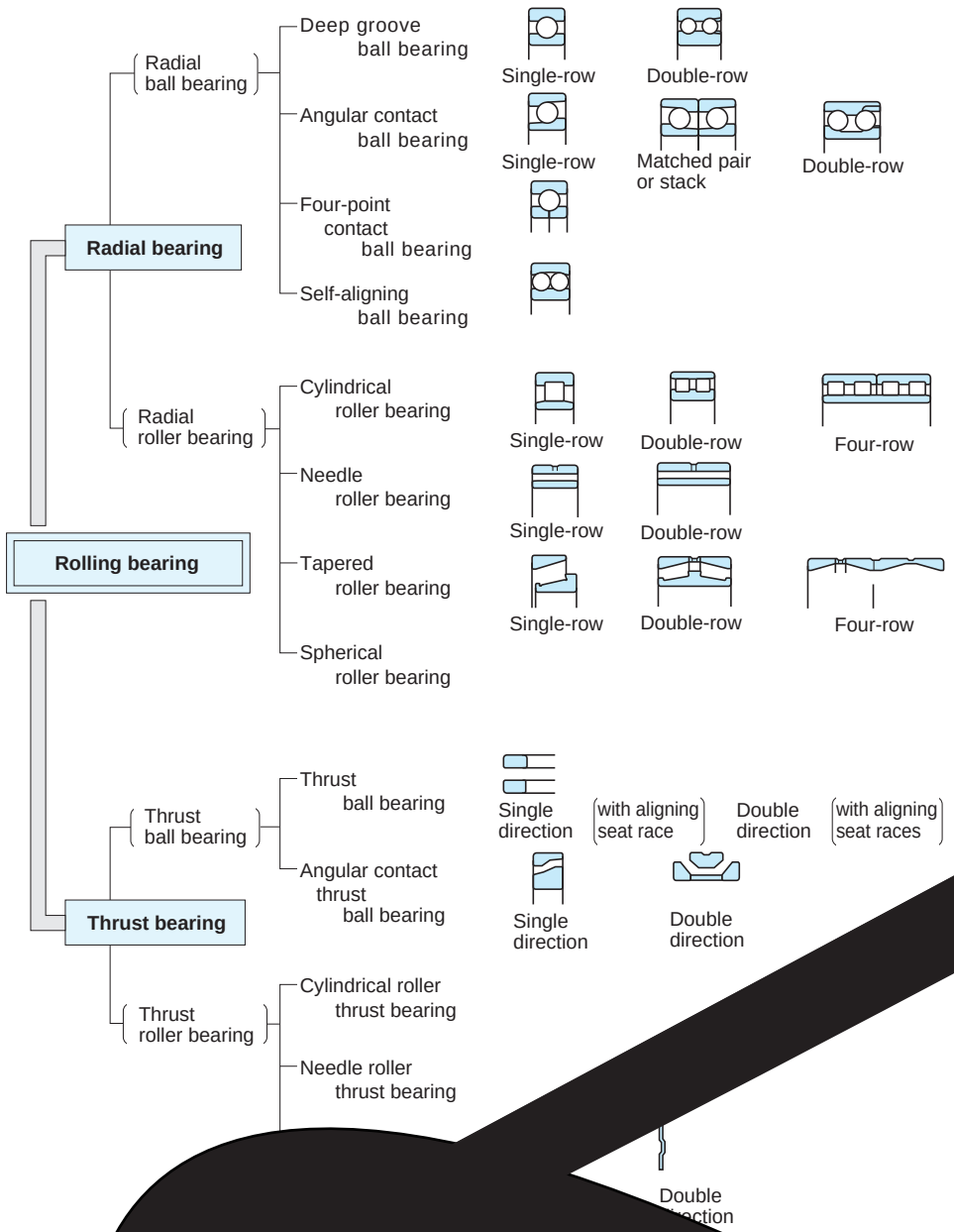
1-1 Structure

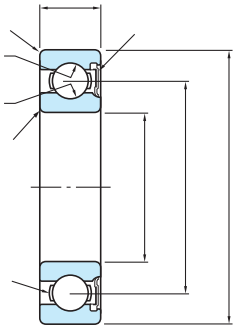
Rolling bearings (bearings hereinafter) normally comprise bearing rings, rolling elements and a cage. (see Fig. 1-1)

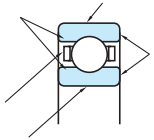
Rolling elements are arranged between inner and outer rings with a cage, which retains the rolling elements in correct relative position, so they do not touch one another. With this structure, a smooth rolling motion is maintained during operation.

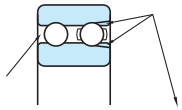
Bearings are classified as follows the number of rows of rolling elements : single-row, double-row, or multi-row (three-row, four-row) bearings.

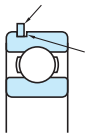



Rolling bearing
Radial bearing
Radial ball bearing
Deep groove ball bearing
Single-row
Double-row
Angular contact ball bearing
Single-row
Matched pair or stack
Double-row
Four-point contact ball bearing
Self-aligning ball bearing
Radial roller bearing
Cylindrical roller bearing
Single-row
Double-row
Four-row
Needle roller bearing
Single-row
Double-row
Tapered roller bearing
Single-row
Double-row
Four-row
Spherical roller bearing
Thrust bearing
Thrust ball bearing
Thrust ball bearing
Single direction
with aligning seat race
Double direction
with aligning seat races
Angular contact thrust ball bearing
Single direction
Double direction
Thrust roller bearing
Cylindrical roller thrust bearing
Needle roller thrust bearing
Double

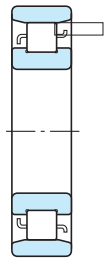


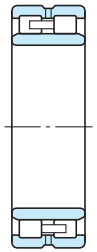




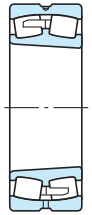












2. Outline of bearing selection



(● **Operating conditions to be considered**)

- Installation space
- Load magnitude, types and direction of application
- Rotational speed
- Running accuracy
- Rigidity
- Misalignment
- Mounting ease

— A 16

- Bearing arrangement — A 20
- Noise characteristics, friction torque
- Marketability, cost performance

(● **Other data**)

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

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

- Bearing boundary dimensions — A 52
- Basic dynamic load rating — A 24
- Basic static load rating — A 42
- Allowable axial load — A 44

(for cylindrical roller bearing with rib)

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

- Bearing tolerances — A 58

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

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

- Rotational speed
- Noise characteristics

- Conditions of application site

- Lubrication

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

(Reference) & ceramic bearing series — C 57

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

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

4. Selection of bearing arrangement

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

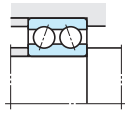
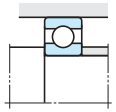
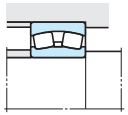
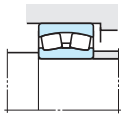
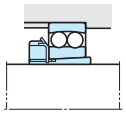
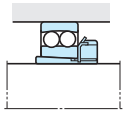
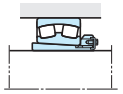
In many cases, in order to locate shaft positions in the axial direction, one bearing is mounted on the fixed side first, then the other bearing is mounted on the free side.

Table 4-1 Bearings on fixed and free sides

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

Table 4-2 (1) Example bearing arrangements

Table 4-2 (2) Example bearing arrangements

Example	Bearing arrangement		Recommended application	Application example
	Fixed side	Free side		
Ex. 8			) This is recommended for operations with relatively heavy axial load in both directions.) Some applications use matched pair angular contact ball bearings on fixed side instead of double-row angular contact ball bearings.	Worm gear speed reducers
Ex. 9			) This is the optimum arrangement for applications with possible mounting errors or shaft deflection.) Bearings in this arrangement can accommodate partial axial load, as well as heavy radial load.	Steel manufacturing table roller speed reducers, overhead crane wheels
Ex. 10			) This is optimum arrangement for applications with possible mounting errors or shaft deflection.) Ease of mounting and dismounting, ensured by use of adaptor, makes this arrangement suitable for long shafts which are neither stepped nor threaded.) This arrangement is not recommended for applications requiring axial load capability.	General industrial equipment counter shafts
Ex. 11			) This is the optimum arrangement for applications with possible mounting errors or shaft deflection.) This is recommended for operations under impact load or radial load heavier than that in Ex. 10.) This arrangement can accommodate partial axial load as well as radial load.	Steel manufacturing table rollers

Arrangement in which fixed and free sides are not distinguished	Recommended application	Application example
Ex. 12		

5. Selection of bearing dimensions

f_n

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f_h

L

5-2-3 Correction of basic dynamic load

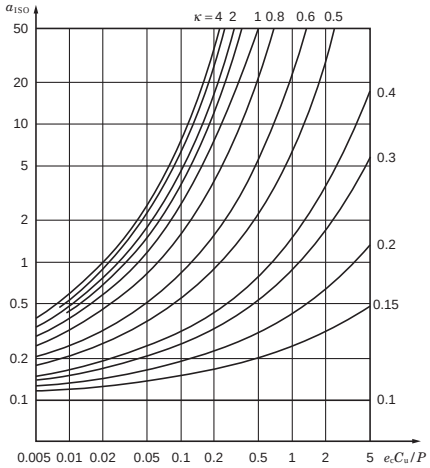


Fig. 5-2 Life modification factor a_{ISO} (Radial ball bearings)

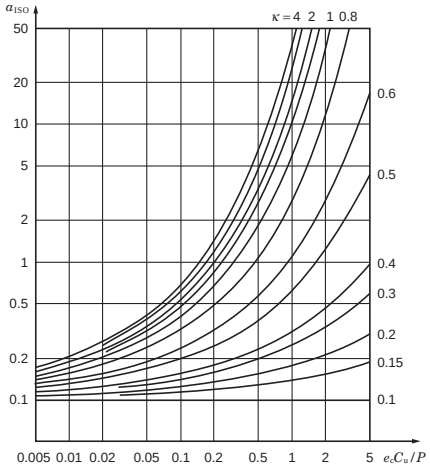


Fig. 5-3 Life modification factor a_{ISO} (Radial roller bearings)

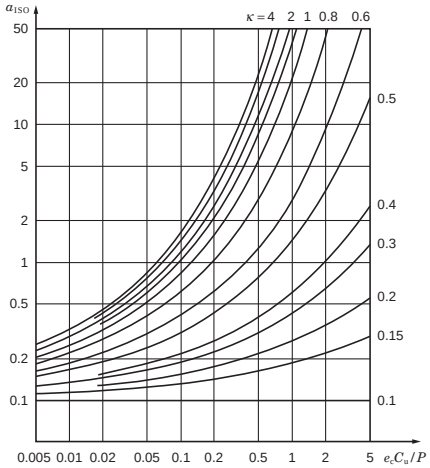


Fig. 5-4 Life modification factor a_{ISO} (Thrust ball bearings)

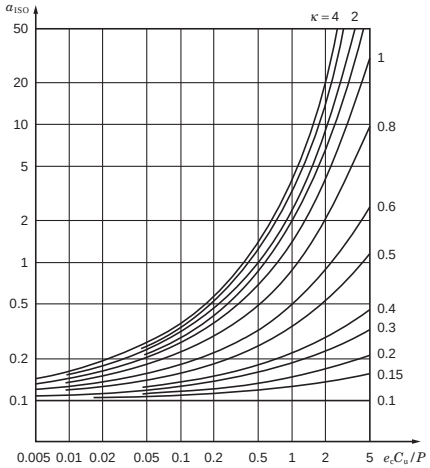


Fig. 5-5 Life modification factor a_{ISO} (Thrust roller bearings)

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

b) Fatigue load limit C_u

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

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

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

c) Contamination factor e_c

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

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

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

Table 5-4 Values of contamination factor e_c

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

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

d) Viscosity ratio κ

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

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

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

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

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

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



5-3 Calculation of loads

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

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

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

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

5-3-1 Load coefficient

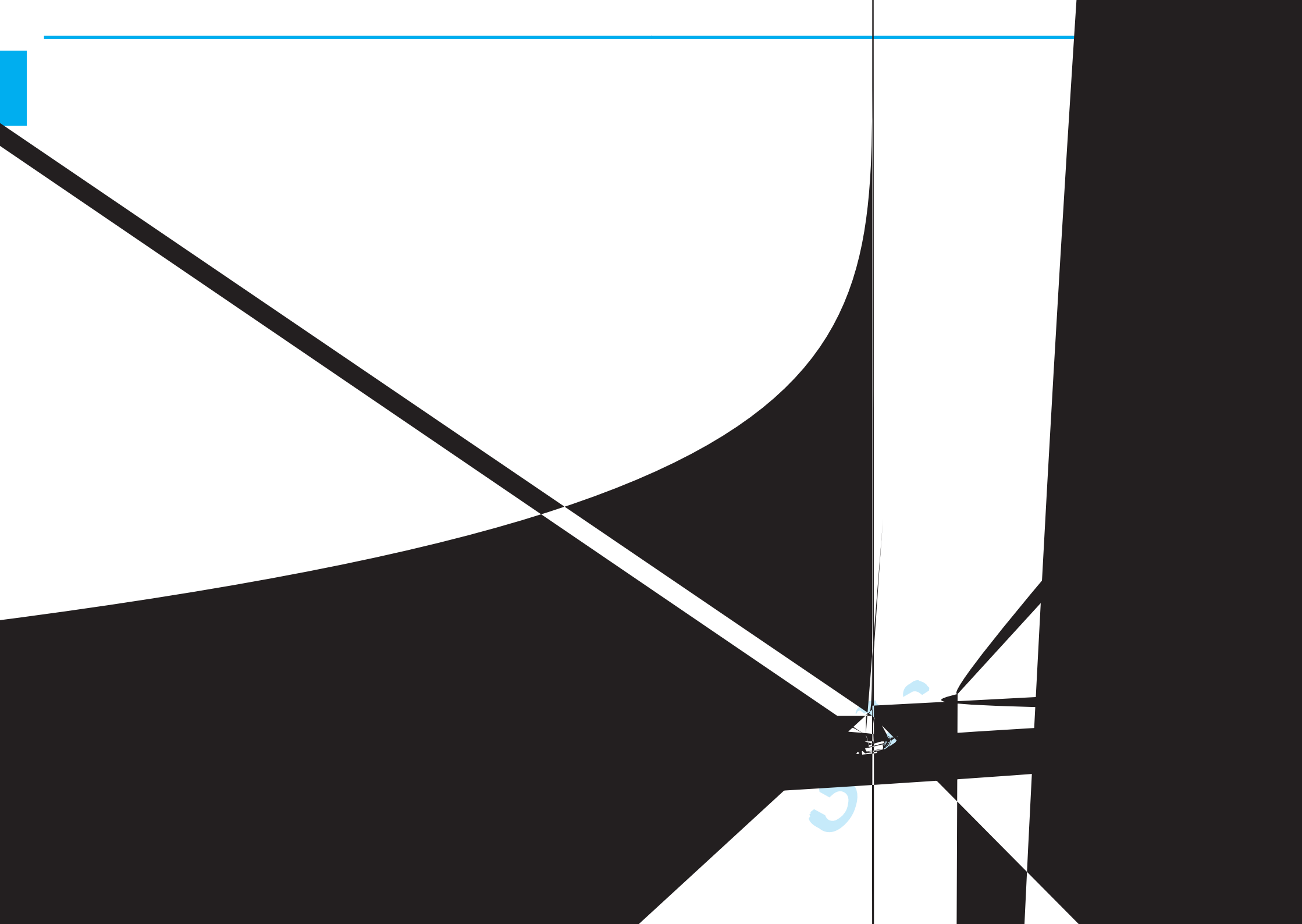
Even if radial and axial loads are obtained through general dynamic calculation, the actual load becomes greater than the calculated value due to vibration and impact during operation.

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

Table 5-6 Values of load coefficient f_w

Operating condition1	S58 cm s /T133 i5 1 Tw 3cc2on4/tpl tho det3 w /GS5 gs.136 exay sint	

w



Example 5 Simultaneous application of gear load and other load

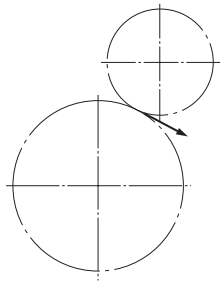
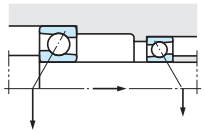
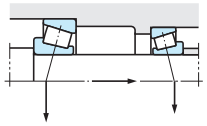


Table 5-9 Dynamic equivalent load calculation : when a pair of single-row angular contact ball bearings or tapered roller bearings is arranged face-to-face or back-to-back.

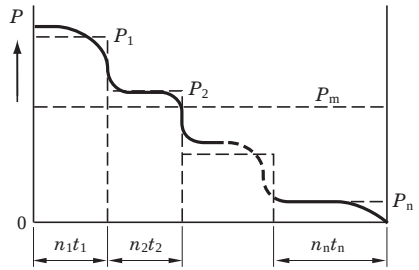
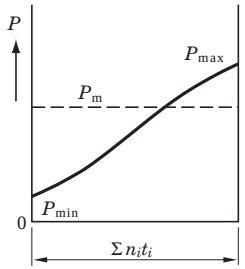
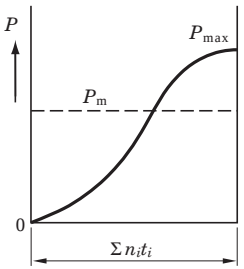
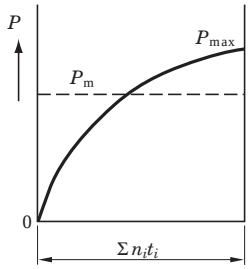
Paired mounting			Loading condition	Bearing	Axial load	Dynamic equivalent load
Back-to-back arrangement	Face-to-face arrangement					
						
						

5-4-2 Mean dynamic equivalent load

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

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

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

(1) Staged fluctuation	(2) Stageless fluctuation	(3) Fluctuation forming sine curve	(4) Fluctuation forming sine curve (upper half of sine curve)
			
$P_m = \sqrt[p]{\dfrac{P_1{}^p n_1t_1 + P_2{}^p n_2t_2 + \cdots\cdots + P_n{}^p n_nt_n}{n_1t_1 + n_2t_2 + \cdots\cdots + n_nt_n}}$ ····· (5-35)	$P_m = \dfrac{P_{min} + 2P_{max}}{3}$ ····· (5-36)	$P_m = 0.68\,P_{max}$ ····· (5-37)	$P_m = 0.75\,P_{max}$ ····· (5-38)

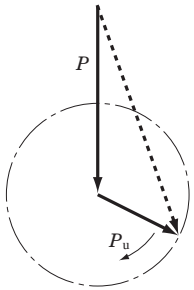
Symbols for Graphs (1) to (4)

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

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

$$n_m = \frac{n_1t_1 + n_2t_2 + \cdots\cdots + n_nt_n}{t_1 + t_2 + \cdots\cdots + t_n}$$

(5) Stationary load and rotating load acting simultaneously



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

where :

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

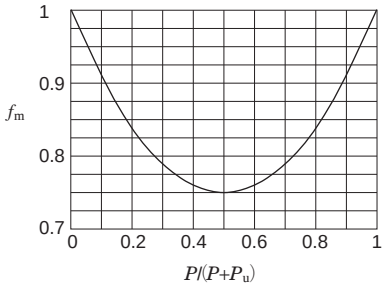
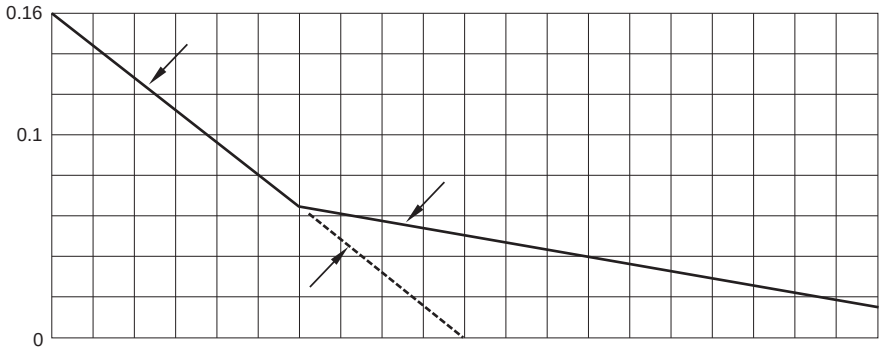
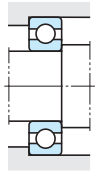


Fig. 5-12 Coefficient f_m


0.16
0.1
0

[Example 6] Bearing size selection

(Conditions)
Deep groove ball bearing :
63 series
Required service life :
more than 15 000 h
Radial load $F_r = 4\,000$ N
Axial load $F_a = 2\,400$ N
Rotational speed $n = 1\,000$ min^{-1}



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

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

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

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

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

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

$$= 6\,080 \times \left(15\,000 \times \frac{60 \times 1\,000}{10^6} \right)^{1/3}$$

$$= 58\,700 \text{ N}$$

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

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

Values obtained using the proportional interpolation are :

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

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

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

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

$$P_r = XF_r + YF_a$$

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

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

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

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

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

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

(Conditions)
Single-row cylindrical roller bearing : NUP 310
Rotational speed $n = 1\,500$ min^{-1}
Oil lubrication
Axial load is intermittently applied.

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

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

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

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

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

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

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

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

$$F_{ap} = 9.8 f_a \cdot f_b \cdot f_p \cdot d_m^2$$

$$= 9.8 \times 2 \times 1.0 \times 0.062 \times 80^2$$

$$\fallingdotseq \underline{7\,780 \text{ N}}$$

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

(Conditions)

Tapered roller bearing

Bearing A : 32309 JR

Bearing B : 32310 JR

Gear type : spur gear (normally machined)

Gear pressure angle $\alpha_1 = \alpha_2 = 20°$

Gear pitch circle diameter $D_{p1} = 360$ mm

$D_{p2} = 180$ mm

Transmission power $W = 150$ kW

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

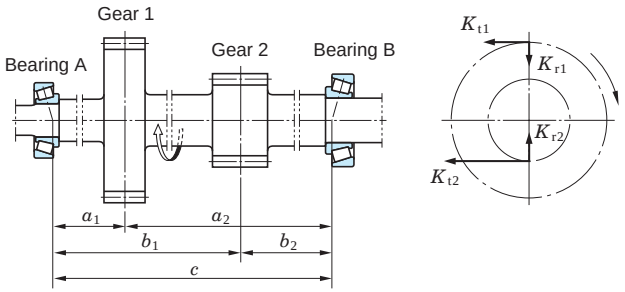
Operating condition: accompanied by impact

Installation locations

$a_1 = 95$ mm, $a_2 = 265$ mm,

$b_1 = 245$ mm, $b_2 = 115$ mm,

$c = 360$ mm



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

[Gear 1]

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

$$K_{r1} = K_{t1} \tan \alpha_1 = 2\,896 \text{ N}$$

[Gear 2]

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

$$K_{r2} = K_{t2} \tan \alpha_2 = 5\,793 \text{ N}$$

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

[Bearing A]

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

$$K_{tA} = f_w f_g \left(\frac{a_2}{c} K_{t1} + \frac{b_2}{c} K_{t2} \right)$$

$$= 1.5 \times 1.2 \times \left(\frac{265}{360} \times 7\,958 + \frac{115}{360} \times 15\,917 \right) = 19\,697 \text{ N}$$

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

$$K_{rA} = f_w f_g \left(\frac{a_2}{c} K_{r1} - \frac{b_2}{c} K_{r2} \right)$$

$$= 1.5 \times 1.2 \times \left(\frac{265}{360} \times 2\,896 - \frac{115}{360} \times 5\,793 \right) = 506 \text{ N}$$

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

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

[Bearing B]

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

$$K_{tB} = f_w f_g \left(\frac{a_1}{c} K_{t1} + \frac{b_1}{c} K_{t2} \right)$$

$$= 1.5 \times 1.2 \times \left(\frac{95}{360} \times 7\,958 + \frac{245}{360} \times 15\,917 \right) = 23\,278 \text{ N}$$

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

$$K_{rB} = f_w f_g \left(\frac{a_1}{c} K_{r1} - \frac{b_1}{c} K_{r2} \right)$$

$$= 1.5 \times 1.2 \times \left(\frac{95}{360} \times 2\,896 - \frac{245}{360} \times 5\,793 \right) = -5\,721 \text{ N}$$

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

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

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

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

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

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

Considering this fact, the axial load applied from the shaft and peripheral parts is to be calculated :

(Equation 5-33, Table 5-9)

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

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

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

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

$$P_{rB} = F_{rB} = 23\,971 \text{ N}$$

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

[Bearing A]

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

[Bearing B]

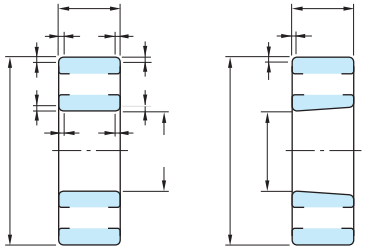
$$L_{10hB} = \frac{10^6}{60n} \left(\frac{C_{rB}}{P_B} \right)^p = \frac{10^6}{60 \times 1\,000} \times \left(\frac{221 \times 10^3}{23\,971} \right)^{10/3} \fallingdotseq \underline{27\,400 \text{ h}}$$

Reference

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

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

6. Boundary dimensions and bearing numbers



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

(2) Tapered roller bearing



(3) Thrust bearing
(single/double direction)

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

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

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

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

Table 6-2 Bearing number configuration

	Basic number			Supplementary code									
Order of arrengement	Bearing series code	Bore diameter No.	Contact angle code	Internal design code, cage guide code	Shield/seal code	Ring shape code, lubrication hole/groove code	Material code, special treatment code	Matched pair or stack code	Internal clearance code, preload code	Spacer code	Cage material/ shape code	Tolerance code	Grease code

68
69
60

7. Bearing tolerances

7-1 Tolerances and tolerance classes for bearings

Bearing tolerances and permissible values for the boundary dimensions and running accuracy of bearings are specified.

These tolerances are prescribed in JIS B 1514-1, JIS B 1514-2, and JIS B 1514-3 (roller bearings - bearing tolerances part 1: radial bearings, part 2: thrust bearings, and part 3: permissible values for chamfer dimensions). (These JIS standards are based on ISO standards.)

Bearing tolerances are standardized by classifying bearings into the following six classes (accuracy in tolerances becomes higher in the order described): 0, 6X, 6, 5, 4 and 2.

Class 0 bearings offer adequate performance for general applications; and, bearings of class 5 or higher are required for demanding applications and operating conditions including those described in Table 7-1.

These tolerances follow ISO standards, but some countries use different names for them. Tolerances for each bearing class, and organizations concerning bearings are listed in Table 7-2.

■ Boundary dimension accuracy
(items on shaft and housing mounting dimensions)
* Tolerances for bore diameter, outside diameter, ring width, assembled bearing width
* Tolerances for set bore diameter and set outside diameter of rollers
* Tolerance limits for chamfer dimensions
* Permissible values for width variation
* Tolerance and permissible values for tapered bore

■ Running accuracy
(items on runout of rotating elements)
* Permissible values for radial and axial runout of inner and outer rings
* Permissible values for perpendicularity of inner ring face
* Permissible values for perpendicularity of outer ring outside surface
* Permissible values for thrust bearing raceway thickness

Accuracies for dimensions and running of each bearing type are listed in Tables 7-3 through 7-10; and, tolerances for tapered bore and limit values for chamfer dimensions of radial bearings are in Tables 7-11 and 7-12.

Table 7-1 High precision bearing applications

Required performance	Applications	Tolerance class
High accuracy in runout is required for rolling elements.	Acoustic / visual equipment spindles (VTR, tape recorders)	P 5, P 4
	Radar / parabola antenna slewing shafts	P 4
	Machine tool spindles	P 5, P 4, P 2, ABEC 9
	Computers, magnetic disc spindles	P 5, P 4, P 2, ABEC 9
	Aluminum foil roll necks	P 5
	Multi-stage mill backing bearings	P 4
High speed rotation	Dental spindles	P 2, ABMA 5P, ABMA 7P
	Superchargers	P 5, P 4
	Jet engine spindles and accessories	P 5, P 4
	Centrifugal separators	P 5, P 4
	LNG pumps	P 5
	Turbo molecular pump spindles and touch-down	P 5, P 4
	Machine tool spindles	P 5, P 4, P 2, ABEC 9
	Tension reels	P 5, P 4
Low friction or low friction variation is required.	Control equipment (synchronous motors, servomotors, gyro gimbals)	P 4, ABMA 7P
	Measuring instruments	P 5
	Machine tool spindles	P 5, P 4, P 2, ABEC 9

Table 7-2 Bearing type and tolerance class

(Reference) Standards and organizations concerned with bearings

JIS : Japanese Industrial Standard
BAS : The Japan Bearing Industrial Association Standard
ISO : International Organization for Standardization
ANSI : American National Standards Institute, Inc.
ABMA : American Bearing Manufactures Association
DIN : Deutsches Institut für Normung
BS : British Standards Institution
NF : Association Francaise de Normalisation

Table 7-3 (1) Radial bearing tolerances (tapered roller bearings excluded)

= JIS B 1514-1 =

(1) Inner ring (bore diameter)

Unit : μm

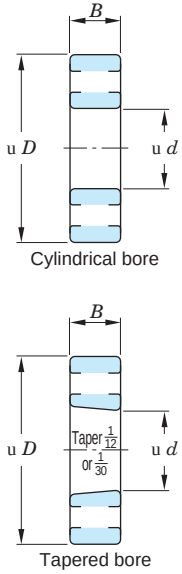


Table 7-3 (2) Radial bearing tolerances (tapered roller bearings excluded)

(3) Outer ring (outside diameter)

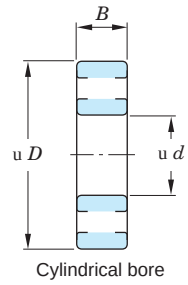
Unit : μm

[Notes]

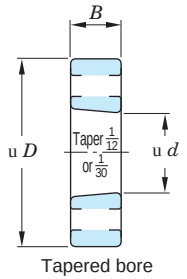
1) These shall be applied to bearings of diameter series 0, 1, 2, 3 and 4.
2) Shall be applied when locating snap ring is not fitted.
3) These shall be applied to deep groove ball bearings and angular contact ball bearings.
4) These shall not be applied to flanged bearings.
5) These shall not be applied to shielded bearings and sealed bearings.

[Remark]

Values in Italics are prescribed in JTEKT standards.



Cylindrical bore



Tapered bore

d : nominal bore diameter
D : nominal outside diameter
B : nominal assembled bearing width

(Refer.) Table 7-4 Tolerances for measuring instrument ball bearings (inch series)

= ANSI/ABMA standards = (reference)

(1) Inner ring and outer ring width

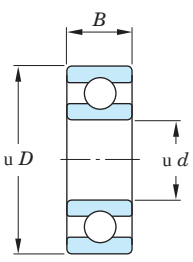
Unit : μm

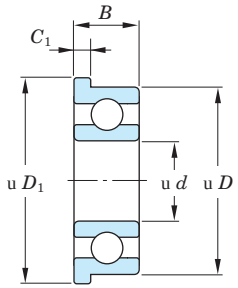
Nominal bore dia. d mm		Single plane mean bore diameter deviation 3_{dmp}				Single bore diameter deviation 3_{ds}				Single plane bore diameter variation V_{dsp}		Mean bore diameter variation V_{dmp}		Radial runout of assembled bearing inner ring K_{ia}			Axial runout of assembled bearing inner ring S_{ia}			Perpendicularity of inner ring face with respect to the bore S_d			Single inner or outer ring width deviation 3_{Bs} , 3_{Cs}		Inner or outer ring width variation V_{Bs} , V_{Cs}		
		classes 5P, 7P		class 9P		classes 5P, 7P		class 9P		classes 5P, 7P	class 9P	classes 5P, 7P	class 9P	class 5P	class 7P	class 9P	class 5P	class 7P	class 9P	class 5P	class 7P	class 9P	classes 5P, 7P, 9P		class 5P	class 7P	class 9P
over	up to	upper	lower	upper	lower	upper	lower	upper	lower	max.		max.		max.			max.			max.			upper	lower	max.		
–	10	0	– 5.1	0	– 2.5	0	– 5.1	0	– 2.5	2.5	1.3	2.5	1.3	3.8	2.5	1.3	7.6	2.5	1.3	7.6	2.5	1.3	0	– 25.4	5.1	2.5	1.3
10	18	0	– 5.1	0	– 2.5	0	– 5.1	0	– 2.5	2.5	1.3	2.5	1.3	3.8	2.5	1.3	7.6	2.5	1.3	7.6	2.5	1.3	0	– 25.4	5.1	2.5	1.3
18	30	0	– 5.1	0	– 2.5	0	– 5.1	0	– 2.5	2.5	1.3	2.5	1.3	3.8	3.8	2.5	7.6	3.8	1.3	7.6	3.8	1.3	0	– 25.4	5.1	2.5	1.3

(2) Outer ring

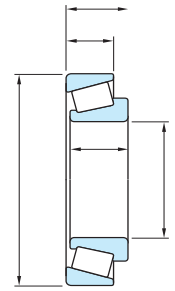
Unit : μm

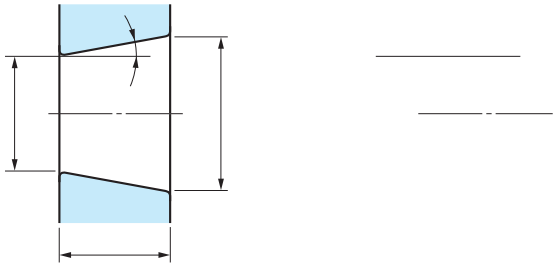
Nominal outside dia. D mm		Single plane mean outside diameter deviation 3_{Dmp}				Single outside diameter deviation 3_{Ds}						Single plane outside diameter variation V_{Dsp}			Mean outside diameter variation V_{Dmp}			Radial runout of assembled bearing outer ring K_{ea}			Axial runout of assembled bearing outer ring S_{ea}			Perpendicularity of outer ring outside surface with respect to the face S_D			Single outer ring flange outside diameter deviation 3_{D1s}		Single outer ring flange width deviation 3_{C1s}	
		classes 5P, 7P		class 9P		classes 5P, 7P				class 9P		classes 5P, 7P		class 9P	classes 5P, 7P		class 9P	class 5P	class 7P	class 9P	class 5P	class 7P	class 9P	class 5P	class 7P	class 9P	classes 5P, 7P		classes 5P, 7P	
						Open type		Shielded/ sealed type		Open type		Open type	Shielded/ sealed type	Open type	Open type	Shielded/ sealed type	Open type													
over	up to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	max.			max.			max.			max.			max.			upper	lower	upper	lower
–	18	0	– 5.1	0	– 2.5	0	– 5.1	+ 1	– 6.1	0	– 2.5	2.5	5.1	1.3	2.5	5.1	1.3	5.1	3.8	1.3	7.6	5.1	1.3	7.6	3.8	1.3	0	– 25.4	0	– 50.8
18	30	0	– 5.1	0	– 3.8	0	– 5.1	+ 1	– 6.1	0	– 3.8	2.5	5.1	2	2.5	5.1	2	5.1	3.8	2.5	7.6	5.1	2.5	7.6	3.8	1.3	0	– 25.4	0	– 50.8
30	50	0	– 5.1	0	– 3.8	0	– 5.1	+ 1	– 6.1	0	– 3.8	2.5	5.1	2	2.5	5.1	2	5.1	5.1	2.5	7.6	5.1	2.5	7.6	3.8	1.3	0	– 25.4	0	– 50.8

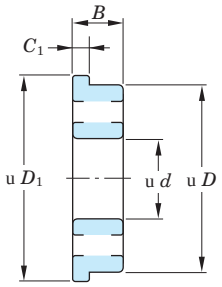




d : nominal bore diameter
D : nominal outside diameter
B : nominal assembled bearing width
D_1 : nominal outer ring flange outside diameter
C_1 : nominal outer ring flange width







d : nominal bore diameter
D : nominal outside diameter
B : nominal assembled bearing width
D_1 : nominal outer ring flange outside diameter
C_1 : nominal outer ring flange width

Table 7-13 Permissible values for chamfer dimensions = JIS B 1514-3 =

(1) Radial bearing (tapered roller bearings excluded)

Unit : mm

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

[Remarks]

1. Value of $r_{\max}$ or $r_{1\max}$ in the axial direction of bearings with nominal width lower than 2 mm shall be the same as the value in radial direction.
2. There shall be no specification for the accuracy of the shape of the chamfer surface, but its outline in the axial plane shall not be situated outside of the imaginary circle arc with a radius of $r_{\min}$ or $r_{1\min}$ which contacts the inner ring side face and bore, or the outer ring side face and outside surface.

(2) Radial bearings with locating snap ring (snap ring groove side) and cylindrical roller bearings (separete thrust collar and loose rib side)

Unit : mm

$r_{1\min}$	Nominal bore dia. or nominal outside dia. d or D		$r_{1\max}$	
	over	up to	Radial direction	Axial direction
0.2	–	–	0.5	0.5
0.3	–	40	0.6	0.8
	40	–	0.8	0.8
0.5	–	40	1	1.5
	40	–	1.3	1.5
0.6	–	40	1	1.5
	40	–	1.3	1.5
1	–	50	1.5	2.2
	50	–	1.9	2.2
1.1	–	120	2	2.7
	120	–	2.5	2.7
1.5	–	120	2.3	3.5
	120	–	3	3.5
2	–	80	3	4
	80	220	3.5	4
	220	–	3.8	4
2.1	–	280	4	4.5
	280	–	4.5	4.5
2.5	–	100	3.8	5
	100	280	4.5	5
	280	–	5	5
3	–	280	5	5.5
	280	–	5.5	5.5
4	–	–	6.5	6.5
5	–	–	8	8
6	–	–	10	10

[Remark] There shall be no specification for the accuracy of the shape of the chamfer surface, but its outline in the axial plane shall not be situated outside of the imaginary circle arc with a radius of $r_{1\min}$ which contacts the inner ring side face and bore, or the outer ring side face and outside surface.

(3) Cylindrical roller bearings (non-rib side) and angular contact ball bearings (front face side)

Unit : mm

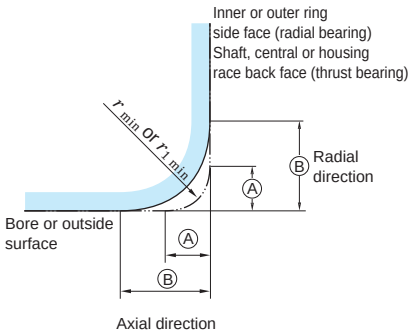
$r_{1\min}$	Nominal bore dia. or nominal outside dia. d or D		$r_{1\max}$	
	over	up to	Radial direction	Axial direction
0.1	–	–	0.2	0.4
0.15	–	–	0.3	0.6
0.2	–	–	0.5	0.8
0.3	–	40	0.6	1
	40	–	0.8	1
0.6	–	40	1	2
	40	–	1.3	2
1	–	50	1.5	3
	50	–	1.9	3
1.1	–	120	2	3.5
	120	–	2.5	4
1.5	–	120	2.3	4
	120	–	3	5
2	–	80	3	4.5
	80	220	3.5	5
	220	–	3.8	6

[Remark] There shall be no specification for the accuracy of the shape of the chamfer surface, but its outline in the axial plane shall not be situated outside of the imaginary circle arc with a radius of $r_{1\min}$ which contacts the inner ring side face and bore, or the outer ring side face and outside surface.

(4) Metric series tapered roller bearing

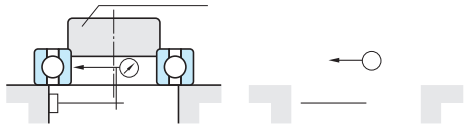
Unit : mm

r				



(Ⓐ : $r_{\min}$ or $r_{1\min}$
Ⓑ : $r_{\max}$ or $r_{1\max}$)

Bore diameter (d) **Cylindrical bore bearings**	Obtain the maximum value ($d_{sp\ max}$) and the minimum value ($d_{sp\ min}$) of the bore diameter (d_s) acquired in a single radial plane.d
Bore diameter (d) **Tapered bore bearings**	
Outside diameter (D)	

Radial runout of assembled bearing inner ring (K_{ia})		The radial runout of the inner ring (K_{ia}) shall be obtained as the difference between the maximum value and the minimum value of the readings of the measuring instrument, when the inner ring has been rotated through one rotation. [Note] The measurement of the radial runout of the inner ring of cylindrical roller bearings, machined ring needle roller bearings, self-aligning ball bearings and spherical roller bearings shall be carried out by fixing the outer ring with ring supports.
Radial runout of assembled bearing outer ring (K_{ea})		The measurement of outer ring runout (K_{ea}) shall be obtained as the difference between the maximum value and the minimum value of the readings of the measuring instrument, when the outer ring has been rotated through one rotation. [Note] The measurement of the radial runout of the outer ring of cylindrical roller bearings, machined ring needle roller bearings, self-aligning ball bearings and spherical roller bearings shall be carried out by fixing the inner ring with ring supports.
Axial runout of assembled bearing inner ring (S_{ia})		The axial runout of the inner ring (S_{ia}) shall be obtained as the difference between the maximum value and the minimum value of the readings of the measuring instrument, when the inner ring has been rotated through one rotation.
Axial runout of assembled bearing outer ring (S_{ea})		The axial runout of the outer ring (S_{ea}) shall be obtained as the difference between the maximum value and the minimum value of the readings of the measuring instrument, when the outer ring has been rotated through one rotation.

9. Bearing fits

9-1 Purpose of fit

The purpose of fit is to securely fix the inner or outer ring to the shaft or housing, to preclude detrimental circumferential sliding on the fitting surface.

Such detrimental sliding (referred to as "creep") will cause abnormal heat generation, wear of the fitting surface, infiltration of abrasion metal particles into the bearing, vibration, and many other harmful effects, which cause a

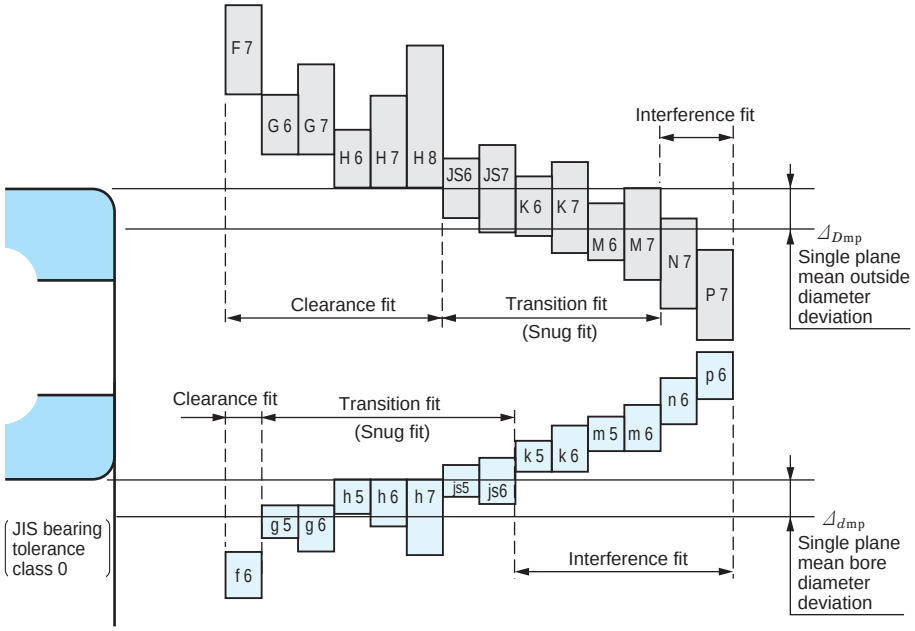


Fig. 9-1 Relationship between tolerances for shaft/housing bore diameters and fits (bearings of class 0 tolerance)

2) Effect of load magnitude

When a radial load is applied, the inner ring will expand slightly. Since this expansion enlarges the circumference of the bore minutely, the initial interference is reduced.

The reduction can be calculated by the following equations :

[In the case of $F_r \leqq 0.25\ C_0$]

$$\Delta_{dF} = 0.08 \sqrt{\frac{d}{B} \cdot F_r} \times 10^{-3} \quad \cdots\cdots (9\text{-}1)$$

[In the case of $F_r > 0.25\ C_0$]

$$\Delta_{dF} = 0.02 \frac{F_r}{B} \times 10^{-3} \quad \cdots\cdots (9\text{-}2)$$

where:

Δ_{dF}	: reduction of inner ring interference	mm
d	: nominal bore diameter of bearing	mm
B	: nominal inner ring width	mm
F_r	: radial load	N
C_0	: basic static load rating	N

Consequently, when the radial load, exceeds the C_0 value by more than 25 %, greater interference is needed.

Much greater interference is needed, when impact loads are expected.

3) Effect of fitting surface roughness

The effective interference obtained after fitting differs from calculated interference due to plastic deformation of the ring fitting surface. When the inner ring is fitted, the effective interference, subject to the effect of the fitting surface finish, can be approximated by the following equations :

[In the case of a ground shaft]

$$\Delta_{deff} \fallingdotseq \frac{d}{d+2} \Delta_d \quad \cdots\cdots (9\text{-}3)$$

[In the case of a turned shaft]

$$\Delta_{deff} \fallingdotseq \frac{d}{d+3} \Delta_d \quad \cdots\cdots (9\text{-}4)$$

where:

Δ_{deff}	: effective interference	mm
Δ_d	: calculated interference	mm

5 0 0 17.0866 58.4646 66. 7.7953 36.8504 93Q cS 1 SC 0.3 /GS5 0 0 m 164.409 0 1 223.937 346.761 cm /CS47 0 0 m 1m3644 Td d T 1.033 3 1.033 a c 9 112.53

d

Table 9-2 Maximum fitting-generated stress in bearings

Shaft & inner ring	Housing bore & outer ring
(In the case of hollow shaft) $\sigma = \frac{E}{2} \cdot \frac{\Delta_{deff}}{d} \cdot \frac{\left(1 - \frac{d_0^2}{d^2}\right)\left(1 + \frac{d^2}{D_i^2}\right)}{\left(1 - \frac{d_0^2}{D_i^2}\right)}$ (In the case of solid shaft) $\sigma = \frac{E}{2} \cdot \frac{\Delta_{deff}}{d} \cdot \left(1 + \frac{d^2}{D_i^2}\right)$	(In the case of $D_h \neq \infty$) $\sigma = E \cdot \frac{\Delta_{Deff}}{D} \cdot \frac{\left(1 - \frac{D^2}{D_h^2}\right)}{\left(1 - \frac{D_e^2}{D_h^2}\right)}$ (In the case of $D_h = \infty$) $\sigma = E \cdot \frac{\Delta_{Deff}}{D}$

where :

σ	: maximum stress	MPa
d	: nominal bore diameter (shaft diameter)	mm
D_i	: raceway contact diameter of inner ring { ball bearing ····· $D_i \fallingdotseq 0.2\ (D + 4d)$ { roller bearing ··· $D_i \fallingdotseq 0.25\ (D + 3d)$	mm
Δ_{deff}	: effective interference of inner ring	mm
d_0	: bore diameter of hollow shaft	mm
D_e	: raceway contact diameter of outer ring { ball bearing ····· $D_e \fallingdotseq 0.2\ (4D + d)$ { roller bearing ··· $D_e \fallingdotseq 0.25\ (3D + d)$	mm
D	: nominal outside diameter (bore diameter of housing)	mm
Δ_{Deff}	: effective interference of outer ring	mm
D_h	: outside diameter of housing	mm
E	: young's modulus	2.08×10^5 MPa

[Remark] The above equations are applicable when the shaft and housing are steel. When other materials are used, JTEKT should be consulted.

9-4 Recommended fits

As described in Section 9-3, the characteristics / magnitude of the bearing load, temperature, mounting / dismounting methods and other conditions must be considered to choose proper fits.

Past experience is also valuable.

Table 9-3 shows standard fits for the metric series bearings; Tables 9-4 to 9-8 tabulate the most typical and recommended fits for different bearings types.

Table 9-3 Standard fits for metric series bearings [1)]

(1) Fits for bore diameter [2)] **of radial bearings**

Class of bearing	Rotating inner ring load or indeterminate direction load							Stationary inner ring load		
	Class of shaft tolerance range									
Classes 0, 6X, 6	r 6	p 6	n 6	m 6 m 5	k 6 k 5	js 6 js 5	h 5	h 6 h 5	g 6 g 5	f 6
Class 5	–	–	–	m 5	k 4	js 4	h 4	h 5	–	–
Fit	Interference fit					Transition fit				Clearance fit

(2) Fits for outside diameter [2)] **of radial bearings**

Class of bearing	Stationary outer ring load				Indeterminate direction load or rotating outer ring load				
	Class of housing bore tolerance range								

Table 9-6 (2) Recommended housing fits for metric J series tapered roller bearings

■ Bearing tolerance : class PK, class PN

Load type		Nominal outside diameter D mm		Class of housing bore diameter tolerance range	Remarks
		over	up to		
Rotating inner ring load	Used for free or fixed side	18 315	315 400	G 7 F 6	Outer ring is easily displaceable in axial direction.
	Position of outer ring is adjustable (in axial direction)	18	400	J 7	Outer ring is displaceable in axial direction.
	Position of outer ring is not adjustable (in axial direction)	18	400	P 7	Outer ring is fixed in axial direction.
Rotating outer ring load	Position of outer ring is not adjustable (in axial direction)	18 120 180	120 180 400	R 7	Outer ring is fixed in axial direction.

■ Bearing tolerance : class PC, class PB

Table 9-7 (2) Recommended housing fits for inch series tapered roller bearings

■ Bearing tolerance : class 4, class 2

Load type		Nominal outside diameter D mm (1/25.4) over	up to	Deviation of a single outside diameter Δ_{Ds}, μm upper	lower	Dimensional tolerance of housing bore diameter μm upper	lower	Remarks
Rotating inner ring load	Used for free or fixed side.	–	76.2 (3.0)	+ 25	0	+ 76	+ 51	
		76.2 (3.0)	127.0 (5.0)	+ 25	0	+ 76	+ 51	
		127.0 (5.0)	304.8 (12.0)	+ 25	0	+ 76	+ 51	
		304.8 (12.0)	609.6 (24.0)	+ 51	0	+152	+102	
		609.6 (24.0)	914.4 (36.0)	+ 76	0	+229	+152	

10. Bearing internal clearance

Bearing internal clearance is defined as the total distance either inner or outer ring can be moved when the other ring is fixed.

If movement is in the radial direction, it is called radial internal clearance; if in the axial direction, axial internal clearance.
(Fig. 10-1)

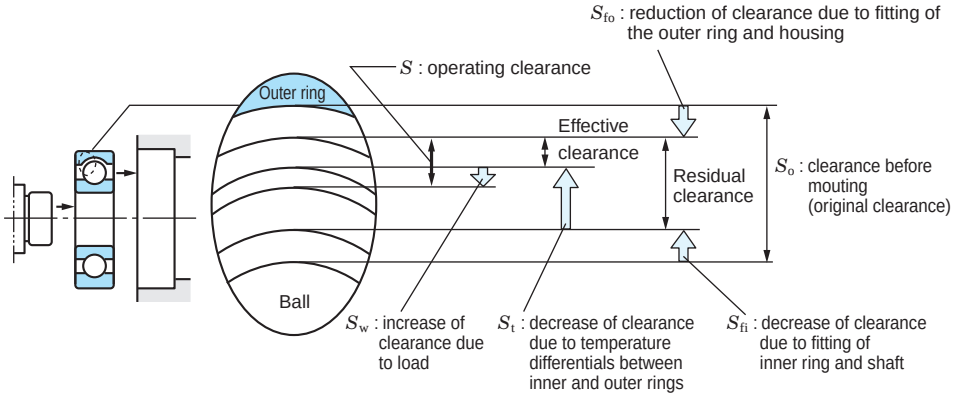
Bearing performance depends greatly upon internal clearance during operation (also referred to as operating clearance); inappropri-
ate-1.182 TD (ate-3);pon)Tj12.674 l -1 0 -1.187ating clearance); inappropri-
ate-1.182 TD (ate-3);pon)Tj12.674 l -1 0 -1.187ating clearance); inappropri-
r10/529900 0 2 l 01 0 0 0 c77_0.13(Lo4 0 232e441.0 0 360527181)3e2g (r10/529900 0 2 l 01 0 0 0 c77_0.13(Lo4 0

10-2 Operating clearance

Table 10-1 shows how to determine the operating clearance when the shaft and housing are made of steel.

Tables 10-2 to 10-10 show standard values for bearing internal clearance before mounting.
Table 10-11 shows examples of clearance selection excluding CN clearance.

Table 10-1 How to determine operating clearance



Operating clearance (S)	$S = S_o - (S_f + S_{t1} + S_{t2}) + S_w$ *	* S_w (increase of clearance due to load) is generally small, and thus may be ignored, although there is an equation for determining the value.
Decrease of clearance due to fitting (S_f)	(In the case of hollow shaft) $S_{fi} = \Delta_{deff} \frac{d}{D_i} \cdot \frac{\left(1 - \frac{d_0^2}{d^2}\right)}{\left(1 - \frac{d_0^2}{D_i^2}\right)}$ (In the case of solid shaft) $S_{fi} = \Delta_{deff} \frac{d}{D_i}$	(In the case of $D_h \neq \infty$) $S_{fo} = \Delta_{Deff} \frac{D_e}{D} \cdot \frac{\left(1 - \frac{D^2}{D_h^2}\right)}{\left(1 - \frac{D_e^2}{D_h^2}\right)}$ (In the case of $D_h = \infty$) $S_{fo} = \Delta_{Deff} \frac{D_e}{D}$
Decrease of clearance due to temperature differentials between inner and outer rings (S_{t1})	The amount of decrease varies depending on the state of housing; however, generally the amount can be approximated by the following equation on the assumption that the outer ring will not expand : $S_{t1} = \alpha (D_i \cdot t_i - D_e \cdot t_e)$	where : $D_e = D_i + 2D_w$ Consequently, $S_{t1} + S_{t2}$ will be determined by the following equation : $S_{t1} + S_{t2} = \alpha \cdot D_i \cdot t_1 + 2\,\alpha \cdot D_w \cdot t_2$ Temperature differential between the inner and outer rings, t_1, can be expressed as follows : $t_1 = t_i - t_e$ Temperature differential between the rolling element and outer ring, t_2, can be expressed as follows : $t_2 = t_w - t_e$
Decrease of clearance due to temperature rise of rolling element (S_{t2})	$S_{t2} = 2\,\alpha \cdot D_w \cdot t_w$	

In Table 10-1,

S	: operating clearance	mm	Δ_{Deff}	: effective interference of outer ring	mm
S_o	: clearance before mounting	mm	D_h	: outside diameter of housing	mm
S_f	: decrease of clearance due to fitting	mm	D_e	: outer ring raceway contact diameter	mm
S_{fi}	: expansion of inner ring raceway contact diameter	mm		(ball bearing ····· $D_e \fallingdotseq 0.2(4D + d)$ roller bearing ··· $D_e \fallingdotseq 0.25(3D + d)$)	
S_{fo}	: contraction of outer ring raceway contact diameter	mm	D	: nominal outside diameter	mm
S_{t1}	: decrease of clearance due to temperature differentials between inner and outer rings	mm	α	: linear expansion coefficient of bearing steel (12.5×10^{-6})	1/°C
S_{t2}	: decrease of clearance due to temperature rise of the rolling elements	mm	D_w	: average diameter of rolling elements	mm
S_w	: increase of clearance due to load	mm		(ball bearing ····· $D_w \fallingdotseq 0.3(D - d)$ roller bearing ··· $D_w \fallingdotseq 0.25(D - d)$)	
Δ_{deff}	: effective interference of inner ring	mm	t_i	: temperature rise of the inner ring	°C
d	: nominal bore diameter (shaft diameter)	mm	t_e	: temperature rise of the outer ring	°C
d_0	: bore diameter of hollow shaft	mm	t_w	: temperature rise of rolling elements	°C
D_i	: inner ring raceway contact diameter	mm			
	(ball bearing ····· $D_i \fallingdotseq 0.2(D + 4d)$ roller bearing ··· $D_i \fallingdotseq 0.25(D + 3d)$)				

■ Bearings are sometimes used with a non-steel shaft or housing.
In the automotive industry, a statistical method is often incorporated for selection of clearance.
In these cases, or when other special operating conditions are involved, JTEKT should be consulted.

Table 10-4 Axial internal clearance of matched pair angular contact ball bearings (measurement clearance) [1)]

Unit : μm

Nominal bore diameter d, mm		Contact angle : 15°				Contact angle : 30°							
		C 2		C N		C 2		C N		C 3		C 4	
over	up to	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
–	**10**	13	33	33	53	3	14	10	30	30	50	50	70
10	**18**	15	35	35	55	3	16	10	30	30	50	50	70
18	**24**	20	40	45	65	3	20	20	40	40	60	60	80
24	**30**	20	40	45	65	3	20	20	40	40	60	60	80
30	**40**	20	40	45	65	3	20	25	45	45	65	70	90
40	**50**	20	40	50	70	3	20	30	50	50	70	75	95
50	**65**	30	55	65	90	9	27	35	60	60	85	90	115
65	**80**	30	55	70	95	10	28	40	65	70	95	110	135
80	**100**	35	60	85	110	10	30	50	75	80	105	130	155
100	**120**	40	65	100	125	12	37	65	90	100	125	150	175
120	**140**	45	75	110	140	15	40	75	105	120	150	180	210
140	**160**	45	75	1_5 1 Tf -7.66		1.4(160)]TJ /T1_T1_68 0 Td [(45)-c788 -1.48 Td [(140)-2121.4(160)]TJ /T1							

(2) Tapered bore bearing

Unit : μm

Nominal bore diameter d, mm		Non-interchangeable clearance													
		C 9 NA[1)]		C 1 NA		C 2 NA		C N NA		C 3 NA		C 4 NA		C 5 NA	
over	up to	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
12	14	5	10	–	–	–	–	–	–	–	–	–	–	–	–
14	24	5	10	10	20	20	30	35	45	45	55	55	65	75	85
24	30	5	10	10	25	25	35	40	50	50	60	60	70	80	95
30	40	5	12	12	25	25	40	45	55	55	70	70	80	95	110
40	50	5	15	15	30	30	45	50	65	65	80	80	95	110	125
50	65	5	15	15	35	35	50	55	75	75	90	90	110	130	150
65	80	10	20	20	40	40	60	70	90	90	110	110	130	150	170
80	100	10	25	25	45	45	70	80	105	105	125	125	150	180	205
100	120	10	25	25	50	50	80	95	120	120	145	145	170	205	230
120	140	15	30	30	60	60	90	105	135	135	160	160	190	230	260
140	160	15	35	35	65	65	100	115	150	150	180	180	215	260	295
160	180	15	35	35	75	75	110	125	165	165	200	200	240	285	320
180	200	20	40	40	80	80	120	140	180	180	220	220	260	315	355
200	225	20	45	45	90	90	135	155	200	200	240	240	285	350	395
225	250	25	50	50	100	100	150	170	215	215	265	265	315	380	430
250	280	25	55	55	110	110	165	185	240	240	295	295	350	420	475
280	315	30	60	60	120	120	180	205	265	265	325	325	385	470	530
315	355	30	65	65	135	135	200	225	295	295	360	360	430	520	585
355	400	35	75	75	150	150	225	255	330	330	405	405	480	585	660
400	450	45	85	85	170	170	255	285	370	370	455	455	540	650	735
450	500														

Table 10-9 Radial internal clearance of spherical roller bearings

(1) Cylindrical bore bearing

Unit : μm

Nominal bore diameter d, mm		Clearance									
		C 2		C N		C 3		C 4		C 5	
over	up to	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
14	18	10	20	20	35	35	45	45	60	60	75
18	24	10	20	20	35	35	45	45	60	60	75
24	30	15	25	25	40	40	55	55	75	75	95
30	40	15	30	30	45	45	60	60	80	80	100
40	50	20	35	35	55	55	75	75	100	100	125
50	65	20	40	40	65	65	90	90	120	120	150
65	80	30	50	50	80	80	110	110	145	145	180
80	100	35	60	60	100	100	135	135	180	180	225
100	120	40	75	75	120	120	160	160	210	210	260
120	140	50	95	95	145	145	190	190	240	240	300
140	160	60	110	110	170	170	220	220	280	280	350
160	180	65	120	120	180	180	240	240	310	310	390
180	200	70	130	130	200	200	260	260	340	340	430
200	225	80	140	140	220	220	290	290	380	380	470
225	250	90	150	150	240	240	320	320	420	420	520
250	280	100	170	170	260	260	350	350	460	460	570
280	315	110	190	190	280	280	370	370	500	500	630
315	355	120	200	200	310	310	410	410	550	550	690
355	400	130	220	220	340	340	450	450	600	600	750
400	450	140	240	240	370	370	500	500	660	660	820
450	500	140	260	260	410	410	550	550	720	720	900
500	560	150	280	280	440	440	600	600	780	780	1 000
560	630	170	310	310	480	480	650	650	850	850	1 100
630	710	190	350	350	530	530	700	700	920	920	1 190
710	800	210	390	390	580	580	770	770	1 010	1 010	1 300
800	900	230	430	430	650	650	860	860	1 120	1 120	1 440
900	1 000	260	480	480	710	710	930	930	1 220	1 220	1 570

(2) Tapered bore bearing

Unit : μm

Nominal bore diameter d, mm		Clearance									
		C 2		C N		C 3		C 4		C 5	
over	up to	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
18	24	15	25	25	35	35	45	45	60	60	75
24	30	20	30	30	40	40	55	55	75	75	95
30	40	25	35	35	50	50	65	65	85	85	105
40	50	30	45	45	60	60	80	80	100	100	130
50	65	40	55	55	75	75	95	95	120	120	160
65	80	50	70	70	95	95	120	120	150	150	200
80	100	55	80	80	110	110	140	140	180	180	230
100	120	65	100	100	135	135	170	170	220	220	280
120	140	80	120	120	160	160	200	200	260	260	330
140	160	90	130	130	180	180	230	230	300	300	380
160	180	100	140	140	200	200	260	260	340	340	430
180	200	110	160	160	220	220	290	290	370	370	470
200	225	120	180	180	250	250	320	320	410	410	520
225	250	140	200	200	270	270	350	350	450	450	570
250	280	150	220	220	300	300	390	390	490	490	620
280	315	170	240	240	330	330	430	430	540	540	680
315	355	190	270	270	360	360	470	470	590	590	740
355	400	210	300	300	400	400	520	520	650	650	820
400	450	230	330	330	440	440	570	570	720	720	910
450	500	260	370	370	490	490	630	630	790	790	1 000
500	560	290	410	410	540	540	680	680	870	870	1 100
560	630	320	460	460	600	600	760	760	980	980	1 230
630	710	350	510	510	670	670	850	850	1 090	1 090	1 360
710	800	390	570	570	750	750	960	960	1 220	1 220	1 500
800	900	440	640	640	840	840	1 070	1 070	1 370	1 370	1 690
900	1 000	490	710	710	930	930	1 190	1 190	1 520	1 520	1 860

Table 10-11 Examples of non-standard clearance selection

Service conditions	Applications	Examples of clearance selection
In the case of heavy/impact load, large interference	Railway rolling stock axle journals	

12. Bearing lubrication

* Thrust ball bearing (contact angle : 90°)

$$F_{a\,min} = 5.1 \left(\frac{n}{1\,000}\right)^2 \cdot \left(\frac{C_{0a}}{1\,000}\right)^2 \times 10^{-3} \quad \text{(11-1)}$$

* Spherical thrust roller bearing (the higher value determined by the two equations should be taken.)

$$F_{a\,min} = \frac{C_{0a}}{2\,000} \quad \text{(11-2)}$$

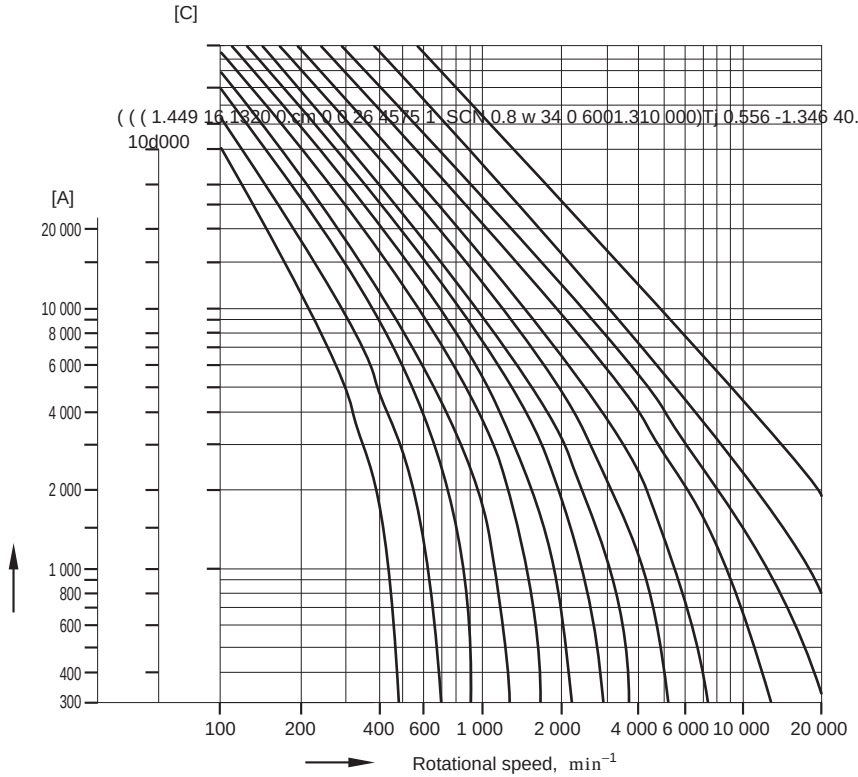
$$F_{a\,min} = 1.8F_r + 1.33 \left(\frac{n}{1\,000}\right)^2 \cdot \left(\frac{C_{0a}}{1\,000}\right)^2 \times 10^{-4} \quad \text{(11-3)}$$

where :

$F_{a\,min}$: minimum necessary axial load N

n : rotational speed min^{-1}

C



(((1.449 16.1320 0 cm 0 0 26 4575 1 SCN 0.8 w 34 0 6001.310 000)Tj 0.556 -1.346 40.368 26.081 l -2.228826 000)Tj 0 -2.51 TD3 1 000)Tj 0 -4.15 TD29(2 000)Tj 0 -4.254 201(10 000 4 0 -4.254 20091 000)Tj 0.834 TD7571 Td 90.368 10d000

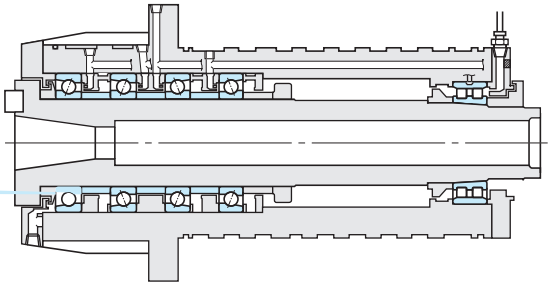
⑦ **Oil/air lubrication**	* A proportioning pump sends forth a small quantity of oil, which is mixed with compressed air by a mixing valve. The admixture is supplied continuously and stably to the bearing. * This method enables quantitative control of oil in extremely small amounts, always supplying new lubricating oil. It is thus suitable for machine tools and other applications requiring high speed. * Compressed air and lubricating oil are supplied to the spindle, increasing the internal pressure and helping prevent dirt, cutting-liquid, etc. from entering. As well, this method allows the lubricating oil to flow through a feeding pipe, minimizing atmospheric pollution. ■ JTEKT produces an oil/air lubricator and, air cleaner, as well as a spindle unit incorporating the oil/air lubrication system. Please refer to brochure "oil/air lubricator & air clean unit". 

Table 12-3 Characteristics of respective greases

	Lithium grease			Calcium grease (cup grease)	Sodium grease (fiber grease)		Complex base grease		Non-soap base grease			
Thickener	Lithium soap			Calcium soap	Sodium soap		Lithium complex soap	Calcium complex soap	Bentone	Urea compounds	Fluorine compounds	Thickener
Base oil	Mineral oil	Synthetic oil (diester oil)	Synthetic oil (silicon oil)	Mineral oil	Mineral oil		Mineral oil	Mineral oil	Mineral oil	Mineral/ synthetic oil	Synthetic oil	Base oil
Dropping point (°C)	170 to 190	170 to 230	220 to 260	80 to 100	160 to 180		250 or higher	200 to 280	–	240 or higher	250 or higher	Dropping point (°C)
Operating temperature range (°C)	– 30 to + 120	– 50 to + 130	– 50 to + 180	– 10 to + 70	0 to + 110		– 30 to + 150	– 10 to + 130	– 10 to + 150	– 30 to + 150	– 40 to + 250	Operating temperature range (°C)
Rotation speed range												

Table 12-5 Typical examples of standard grease for JTEKT bearings

Table 12-8 Proper kinematic viscosities by bearing operating conditions

[Remarks] 1. $d_m n = \frac{D+d}{2} \times n \cdots \{D$: nominal outside diameter (mm), d : nominal bore diameter (mm), n : rotational speed (min^{-1})$\}$

2. Refer to refrigerating machine oil (JIS K 2211), turbine oil (JIS K 2213), gear oil (JIS K 2219), machine oil (JIS K 2238) and bearing oil (JIS K 2239).

3. Please contact with JTEKT if the bearing operating temperature is under −30°C or over 150°C .

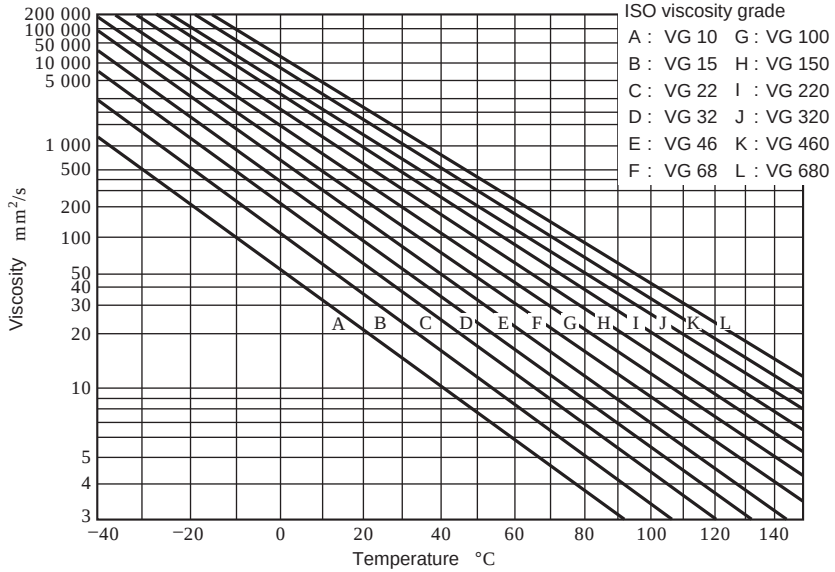


Fig. 12-3 Relationship between lubricating oil viscosity and temperature (viscosity index :100)

13. Bearing materials

Bearing materials include steel for bearing rings and rolling elements, as well as steel sheet, steel, copper alloy and synthetic resins for cages.

These bearing materials should possess the following characteristics :

1) High elasticity, durable under high partial contact stress. — Bearing rings
2) High strength against rolling contact fatigue due to large repetitive contact load. — Rolling elements
3) Strong hardness
4) High abrasion resistance
5) High toughness against impact load — Bearing rings, Rolling elements
6) Excellent dimensional stability — Cages

13-1 Bearing rings and rolling elements materials

1) High carbon chromium bearing steel

High carbon chromium bearing steel specified in JIS is used as a general material in bearing rings (inner rings, outer rings) and rolling elements (balls, rollers).

Their chemical composition classified by steel type is given in Table 13-1.

Among these steel types, SUJ 2 is generally

2) Case carburizing bearing steel (case hardened steel)

When a bearing receives heavy impact loads, the surface of the bearing should be hard and the inside soft.

Such materials should possess a proper amount of carbon, dense structure, and carburizing case depth on their surface, while having proper hardness and fine structure internally.

For this purpose, chromium steel and nickel-chromium-molybdenum steel are used as materials.

Typical steel materials are shown in Table 13-2.

3) Steel for Standard JTEKT Specification Bearings

In general terms, it is known that the non-metallic inclusions contained in materials are harmful to the rolling contact fatigue life.

At JTEKT, to reduce the amount of non-metallic inclusions, which are harmful to the fatigue life, we set the chemical compounds of the bearing steel in a proprietary manner. As a result, JTEKT standard bearings have a life that is approximately twice as long as the general bearings that are targeted by JIS B 1518 (and ISO 281).

Table 13-2 Chemical composition of case carburizing bearing steel

13-2 Materials used for cages

Since the characteristics of materials used for cages greatly influence the performance and reliability of rolling bearings, the choice of materials is of great importance.

It is necessary to select cage materials in accordance with required shape, ease of lubrication, strength, and abrasion resistance.

Typical materials used for metallic cages are shown in Tables 13-3 and 13-4.

In addition, phenolic resin machined cages and other synthetic resin molded cages are often used.

Materials typically used for molded cages are polyacetal, polyamide (Nylon 6.6, Nylon 4.6), and polymer containing fluorine, which are strengthened with glass and carbon fibers.

Table 13-3 Chemical compositions of pressed cage steel sheet (A) and machined cage carbon steel (B)

14. Shaft and housing design

Table 14-1 Recommended accuracy and roughness of shafts and housings

Item	Bearing class	Shaft	Housing bore
Roundness tolerance	classes 0, 6	IT 3 – IT 4	IT 4 – IT 5
	classes 5, 4	IT 2 – IT 3	IT 2 – IT 3
Cylindrical form tolerance	classes 0, 6	IT 3 – IT 4	IT 4 – IT 5
	classes 5, 4	IT 2 – IT 3	IT 2 – IT 3
Shoulder runout tolerance	classes 0, 6	IT 3	IT 3 – IT 4
	classes 5, 4	IT 3	IT 3
Roughness of fitting surfaces Ra	Small size bearings Large size bearings	0.8 a 1.6 a	1.6 a 3.2 a

[Remark] Refer to the figures listed in the attached table when the basic tolerance IT is required.

14-2 Mounting dimensions

Mounting dimensions mean the necessary dimensions to mount bearings on shafts or housings, which include the fillet radius or shoulder diameters.

Standard values are shown in Table 14-2. (The mounting related dimensions of each bearing are given in the bearing specification table.)

The grinding undercut dimensions for ground shafts are given in Table 14-3.

shafts are gi9ts are gi9ts 7N-6f each bear-
1vd

14-3 Shaft design

When bearings are mounted on shafts, locating method should be carefully determined. Shaft design examples for cylindrical bore bearings are given in Table 14-4, and those for bearings with a tapered bore in Table 14-5.

Table 14-4 Mounting designs for cylindrical bore bearings

14-4-2 Contact type sealing devices

This type provides a sealing effect by means of the contact of its end with the shaft and are manufactured from synthetic rubber, synthetic resin, or felt.

The synthetic rubber oil seal is most popular.

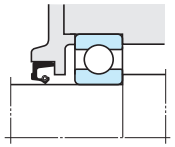
1) Oil seals

Many types and sizes of oil seals, as a finished part, have been standardized.

JTEKT produces various oil seals.

The names and functions of each oil seal part are shown in Fig. 14-8 and Table 14-7. Table 14-8 provides a representative example.

Oil seals without minor lips are mounted in different directions according to their operating conditions (shown in Fig. 14-9).



15. Handling of bearings

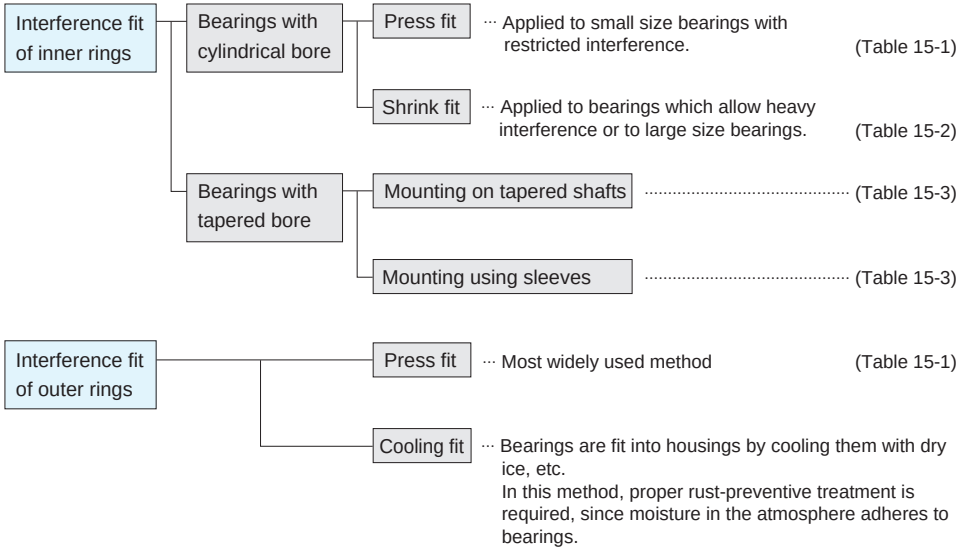

Interference fit of inner rings
Bearings with cylindrical bore
Press fit
··· Applied to small size bearings with restricted interference.
(Table 15-1)
Shrink fit
··· Applied to bearings which allow heavy interference or to large size bearings.
(Table 15-2)
Bearings with tapered bore
Mounting on tapered shafts
(Table 15-3)
Mounting using sleeves
(Table 15-3)
Interference fit of outer rings
Press fit
··· Most widely used method
(Table 15-1)
Cooling fit
··· Bearings are fit into housings by cooling them with dry ice, etc.
In this method, proper rust-preventive treatment is required, since moisture in the atmosphere adheres to bearings.

Table 15-2 Shrink fit of cylindrical bore bearings

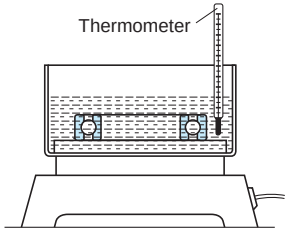
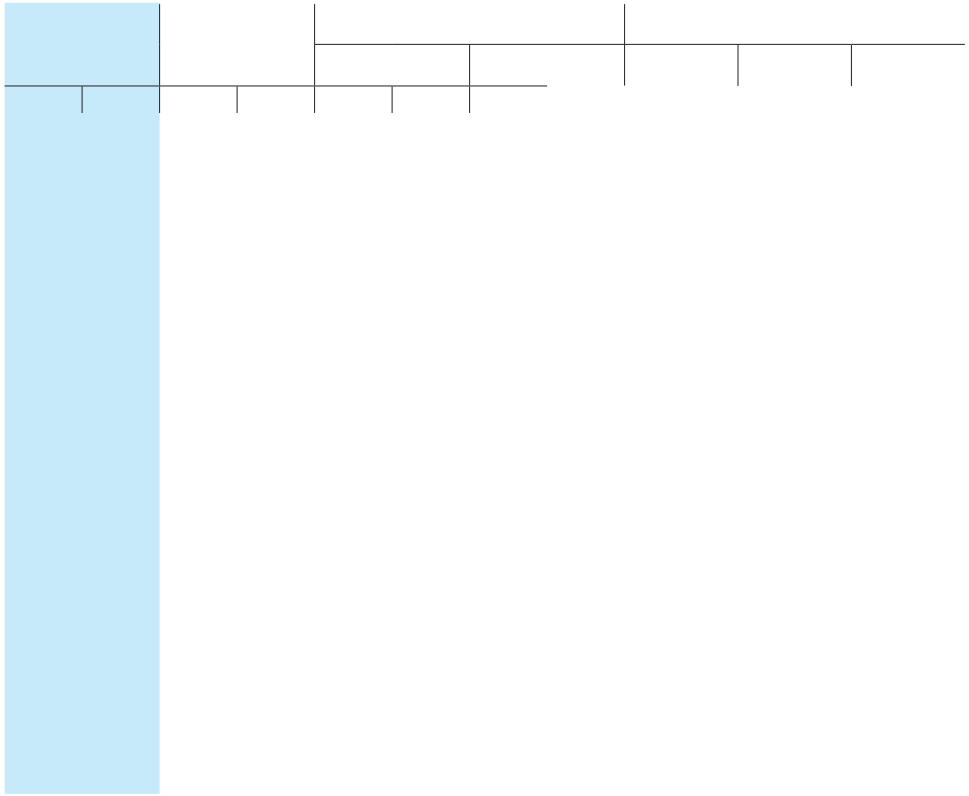
Shrink fit	Descriptions
 (a) Heating in an oil bath  (b) Induction heater	■ This method, which expands bearings by heating them in oil, has the advantage of not applying too much force to bearings and taking only a short time.

Table 15-4 Mounting tapered bore spherical roller bearings



15-6 Maintenance and inspection of bearings

Periodic and thorough maintenance and inspection are indispensable to drawing full performance from bearings and lengthening their useful life.

Besides, prevention of accidents and down time by early detection of failures through maintenance and inspection greatly contributes to the enhancement of productivity and profitability.

15-6-1 Cleaning

Before dismounting a bearing for inspection, record the physical condition of the bearing, including taking photographs.

Cleaning should be done after checking the amount of remaining lubricant and collecting lubricant as a sample for examination.

- A dirty bearing should be cleaned using two cleaning processes, such as rough cleaning and finish cleaning.
  It is recommended that a net be set on the bottom of cleaning containers.
- In rough cleaning, use brushes to remove grease and dirt. Bearings should be handled carefully. Note that raceway surfaces may be damaged by foreign matter, if bearings are rotated in cleaning oil.
- During finish cleaning, clean bearings carefully by rotating them slowly in cleaning oil.

In general, neutral water-free light oil or kerosene is used to clean bearings, a warm alkali solution can also be used if necessary. In any case, it is essential to keep oil clean by filtering it prior to cleaning.

Apply anti-corrosion oil or rust preventive grease on bearings immediately after cleaning.

15-6-2 Inspection and analysis

Before determining that dismounted bearings will be reused, the accuracy of their dimensions and running, internal clearance, fitting surfaces, raceways, rolling contact surfaces, cages and seals must be carefully examined, so as to confirm that no abnormality is present.

It is desirable for skilled persons who have sufficient knowledge of bearings to make decisions on the reuse of bearings.

Criteria for reuse differs according to the performance and importance of machines and inspection frequency.

If the following defects are found, replace the bearing with a new one.

- Cracks and chips in bearing components
- Flaking on the raceway surfaces and the rolling contact surfaces
- Other failures of a serious degree described in the following section "16. Examples of bearing failures."

15-7 Methods of analyzing bearing failures

It is important for enhancing productivity and profitability, as well as for accident prevention that abnormalities in bearings are detected during operation.

Representative detection methods are described in the following section.

1) Noise checking

Since the detection of abnormalities in bearings from noises requires ample experience, sufficient training must be given to inspectors. Given this, it is recommended that specific persons be assigned to this work in order to gain this experience.

Attaching hearing aids or listening rods on housings is effective for detecting bearing noise.

2) Checking of operating temperature

Since this method utilizes change in operating temperature, its application is limited to relatively stable operations.

For detection, operating temperatures must be continuously recorded.

If abnormalities occur in bearings, operating temperature not only increase but also change irregularly.

It is recommended that this method be employed together with noise checking.

3) Lubricant checking

This method detects abnormalities from the foreign matter, including dirt and metallic powder, in lubricants collected as samples.

This method is recommended for inspection of bearings which cannot be checked by close visual inspection, and large size bearings.

16. Examples of bearing failures

Table 16-1 (1) Bearing failures, causes and countermeasures

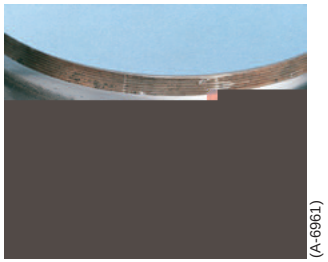
Failures	Characteristics	Damages	Causes	Countermeasures
1 Flaking	 (A-6961)  (A-6476) Flaking is a phenomenon when material is removed in flakes from a surface layer of the bearing raceways or rolling elements due to rolling fatigue. This phenomenon is generally attributed to the approaching end of bearing service life. However, if flaking occurs at early stages of bearing service life, it is necessary to determine causes and adopt countermeasures. [Reference] Pitting Pitting is another type of failure caused by rolling fatigue, in which minute holes of approx. 0.1 mm in depth are generated on the raceway surface.	Flaking occurring at an incipient stage	· Too small internal clearance · Improper or insufficient lubricant · Too much load · Rust	· Provide proper internal clearance. · Select proper lubricating method or lubricant.
		Flaking on one side of radial bearing raceway	· Extraordinarily large axial load	· Fitting between outer ring on the free side and housing should be changed to clearance fit.
		Symmetrical flaking along circumference of raceway	· Inaccurate housing roundness	· Correct processing accuracy of housing bore. (Especially for split housings, care should be taken to ensure processing accuracy.)
		Slanted flaking on the radial ball bearing raceway	· Improper mounting · Shaft deflection · Inaccuracy of the shaft and housing	· Correct centering. · Widen bearing internal clearance. · Correct squareness of shaft or housing shoulder.
		Flaking occurring near the edge of the raceway or rolling contact surface of roller bearings		
		Flaking on the raceway surface at the same interval as rolling element spacing	· Heavy impact load during mounting · A flaw of cylindrical roller bearings or tapered roller bearings caused when they are mounted. · Rust gathered while out of operation	· Improve mounting procedure. · Provide rust prevention treatment before long cessation of operation.
2 Cracking, chipping	 (A-6395)	Cracking in outer ring or inner ring	· Excessive interference · Excessive fillet on shaft or housing · Heavy impact load · Advanced flaking or seizure	· Select proper fit. · Adjust fillet on the shaft or in the housing to smaller than that of the bearing chamfer dimension. · Re-examine load conditions.
		Cracking on rolling elements	· Heavy impact load · Advanced flaking	· Improve mounting and handling procedure. · Re-examine load conditions.
		Cracking on the rib	· Impact on rib during mounting · Excessive axial impact load	· Improve mounting procedure. · Re-examine load conditions.
3 Brinelling, nicks	 (A-6617) (Brinelling) · Brinelling is a small surface indentation generated either on the raceway through plastic deformation at the contact point between the raceway and rolling elements, or on the rolling surfaces from insertion of foreign matter, when heavy load is applied while the bearing is stationary or rotating at a low rotation speed. · Nicks are those indentations produced directly by rough handling such as hammering.	Brinelling on the raceway or rolling contact surface	· Entry of foreign matter	· Clean bearing and its peripheral parts. · Improve sealing devices.
		Brinelling on the raceway surface at the same interval as the rolling element spacing	· Impact load during mounting · Excessive load applied while bearing is stationary	· Improve mounting procedure. · Improve machine handling.
		Nicks on the raceway or rolling contact surface	· Careless handling	· Improve mounting and handling procedure.

Table 16-1 (2) Bearing failures, causes and countermeasures

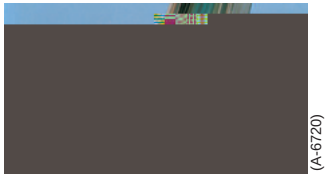
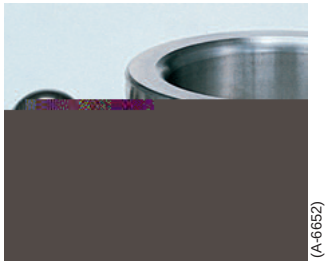
Failures	Characteristics	Damages	Causes	Countermeasures
4 Pear skin, discoloration	 (A-6720) (Discoloration) · Pear skin is a phenomenon in which minute brinell marks cover the entire rolling surface, caused by the insertion of foreign matter. This is characterized by loss of luster and a rolling surface that is rough in appearance. In extreme cases, this is accompanied by discoloration due to heat generation. · Discoloration is a phenomenon in which the surface color changes because of staining or heat generation during rotation. Color change caused by rust and corrosion is generally separate from this phenomenon.	Indentation similar to pear skin on the raceway and rolling contact surface.	· Entry of minute foreign matter	· Clean the bearing and its peripheral parts. · Improve sealing device.
		Discoloration of the raceway, surface rolling contact surface, rib face, and cage riding land.	· Too small bearing internal clearance · Improper or insufficient lubricant · Quality deterioration of lubricant due to aging, etc.	· Provide proper internal clearance. · Select proper lubricating method or lubricant.
5 Scratches, scuffing	 (A-6459) (Scuffing) · Scratches are relatively shallow marks generated by sliding contact, in the same direction as the sliding. This is not accompanied by apparent melting of material. · Scuffing refers to marks, the surface of which are partially melted due to higher contact pressure and therefore a greater heat effect. · Generally, scuffing may be regarded as a serious case of scratches.	Scratches on raceway or rolling contact surface	· Insufficient lubricant at initial operation · Careless handling	· Apply lubricant to the raceway and rolling contact surface when mounting. · Improve mounting procedure.
		Scuffing on rib face and roller end face	· Improper or insufficient lubricant · Improper mounting · Excessive axial load	· Select proper lubricating method or lubricant. · Correct centering of axial direction.
6 Smearing	 (A-6640) Smearing is a phenomenon in which a cluster of minute seizures cover the rolling contact surface. Since smearing is caused by high temperature due to friction, the surface of the material usually melts partially ; and, the smeared surfaces appear very rough in many cases.	Smearing on raceway or rolling contact surface	· Improper or insufficient lubricant · Slipping of the rolling elements (This occurs due to the break down of lubricant film when an abnormal self rotation causes slip of the rolling elements on the raceway.)	· Select proper lubricating method or lubricant. · Provide proper preload.
7 Rust, corrosion	 (A-7130) · Rust is a film of oxides, or hydroxides, or carbonates formed on a metal surface due to chemical reaction. · Corrosion is a phenomenon in which a metal surface is eroded by acid or alkali solutions through chemical reaction (electrochemical reaction such as chemical combination and battery formation) ; resulting in oxidation or dissolution. (It often occurs when sulfur or chloride contained in the lubricant additives is dissolved at high temperature.)	Rust partially or completely covering the bearing surface.	· Improper storage condition · Dew formation in atmosphere	· Improve bearing storage conditions. · Improve sealing devices. · Provide rust preventive treatment before long cessation of operation.
		Rust and corrosion at the same interval as rolling element spacing	· Contamination by water or corrosive matter	· Improve sealing devices.
8 Electric pitting	 (A-6652) When an electric current passes through a bearing while in operation, it can generate sparks between the raceway and rolling elements through a very thin oil film, resulting in melting of the surface metal in this area. This phenomenon appears to be pitting at first sight. (The resultant flaw is referred to as a pit.) When the pit is magnified, it appears as a hole like a crater, indicating that the material melted when it was sparking. In some cases, the rolling surface becomes corrugated by pitting.	Pitting or a corrugated surface failure on raceway and rolling contact surface (The bearings must be replaced, if the corrugated texture is found by scratching the surface with a fingernail or if pitting can be observed by visual inspection.)	· Sparks generated when electric current passes through bearings	· Providing a bypass which prevents current from passing through bearings. · Insulation of bearings.

Table 16-1 (3) Bearing failures, causes and countermeasures

Failures	Characteristics		Damages	Causes	Countermeasures
9 Wear	 (A-4719)				
10					