TECHNICAL TABLES

1. Bearing materials

1.1 Raceway and rolling element materials

1.1.1 High/mid carbon alloy steel

In general, steel varieties which can be hardened not just on the surface but also deep hardened by the so-called "through hardening method" are used for the raceways and rolling elements of bearings. Foremost among these is high carbon chromium bearing steel, which is widely used.

1.1.2 Mid-carbon chromium steel

Mid-carbon chromium steel incorporating silicone and manganese, which gives it hardening properties comparable to high carbon chromium steel.

1.2 Cage materials

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

1.2.1 Pressed cages

For small and medium sized bearings, pressed cages of cold or hot rolled steel with a low carbon content of approx. 0.1% are used. However, depending on the application, austenitic stainless steel is also used.

1.2.2 Plastic cages

Injection molded plastic cages are now widely used: most are made from fiber glass reinforced heat resistant polyamide resin. Plastic cages are light weight, corrosion resistant and have excellent dampening and sliding properties. Heat resistant polyamide resins now enable the production of cages that perform well in applications ranging between -40℃ - 120℃. However, they are not recommended for use at temperatures exceeding 120°C.

2. Bearing tolerances

2.1 Standard of tolerances

Ball bearing "tolerances" or dimensional accuracy and running accuracy, are regulated by ISO and JIS standards (rolling bearing tolerances). For dimensional accuracy, these standards prescribe the tolerances necessary when installing bearings on shafts or in housings. Running accuracy is defined as the allowable limits for bearing runout during operation.

Table 2.1 Bearings types and applicable tolerance

Bearing type		Applicable standard	Applicable tolerance class				Applicable table
Needle roller bearing		JIS B 1514 ISO 492 (NIKO standard)	class change	class 6	class 5	class 4	Table 3.2
Complex bearing	Radial bearing		class 0	class 6	class 5	—	Table 3.2
	Thrust bearing		NIKO class 0	NIKO class 6	NIKO class 5	NIKO class 4	Table 3.3
Needle roller bearing with double-direction thrustf roller bearing	Radial bearing		—	—	class 5	class 4	Table 3.2
	Thrust bearing		—	—	NIKO class 5	NIKO class 4	Table 3.3
Thrust roller bearings			NIKO class 0	NIKO class 6	NIKO class 5	NIKO class 4	Table 3.3
Roller follower/cam follower			class 0	—	—	—	Table 3.2

Note: JIS B 1514 and ISO 492 have the same specification level.

Table 2.2 Tolerance for radial bearings
Table 2.2.1 Inner rings

Nominal bore diameter *d* mm		Single plane mean bore diameter deviation △*dmp*								Single radial plane bore diameter variation *Vdp*				Mean single plane bore diameter variation *Vdmp*				Inner ring radial runout *Kia*			
		class 0		class 6		class 5		class 4 ❶		class 0	class 6	class 5	class 4	class 0	class 6	class 5	class 4	class 0	class 6	class 5	class 4
over	incl.	high	low	high	low	high	low	high	low	max.				max.				max.			
2.5 ❷	10	0	-8	0	-7	0	-5	0	-4	10	9	5	4	6	5	3	2.0	10	6	4	2.5
10	18	0	-8	0	-7	0	-5	0	-4	10	9	5	4	6	5	3	2.0	10	7	4	2.5
18	30	0	-10	0	-8	0	-6	0	-5	13	10	6	5	8	6	3	2.5	13	8	4	3.0
30	50	0	-12	0	-10	0	-8	0	-6	15	13	8	6	9	8	4	3.0	15	10	5	4.0
50	80	0	-15	0	-12	0	-9	0	-7	19	15	9	7	11	9	5	3.5	20	10	5	4.0
80	120	0	-20	0	-15	0	-10	0	-8	25	19	10	8	15	11	5	4.0	25	13	6	5.0
120	150	0	-25	0	-18	0	-13	0	-10	31	23	13	10	19	14	7	5.0	30	18	8	6.0
150	180	0	-25	0	-18	0	-13	0	-10	31	23	13	10	19	14	7	5.0	30	18	8	6.0
180	250	0	-30	0	-22	0	-15	0	-12	38	28	15	12	23	17	8	6.0	40	20	10	8.0
250	315	0	-35	0	-25	0	-18	—	—	44	31	18	—	26	19	9	—	50	25	13	—
315	400	0	-40	0	-30	0	-23	—	—	50	38	23	—	30	23	12	—	60	30	15	—
400	500	0	-45	0	-35	—	—	—	—	56	44	—	—	34	26	—	—	65	35	—	—

Note: ❶ The dimensional difference △*ds* of the bore diameter to be applied for class 4 is the same as the tolerance of dimensional difference △*dmp* of the average bore diameter
❷ Nominal bore diameter of bearings of 2.5 mm is included in this dimensional division.

(Unit: μm)

Nominal bore diameter *d* mm		Face runout with bore *Sd*			Inner ring axial runout (with side) *Sia* ❸			Inner ring width deviation △*Bs*				Inner ring width variation *VBs*			
		class 0	class 6	class 5	class 0	class 6	class 5	class 0,6		class 5,4		class 0	class 6	class 5	class 4
over	incl.	max.			max.			high	low	high	low	max.			
2.5 ❷	10	7	3	1.5	7	3	1.5	0	-120	0	-40	15	15	5	2.5
10	18	7	3	1.5	7	3	1.5	0	-120	0	-80	20	20	5	2.5
18	30	8	4	1.5	8	4	2.5	0	-120	0	-120	20	20	5	2.5
30	50	8	4	1.5	8	4	2.5	0	-120	0	-120	20	20	5	3.0
50	80	8	5	1.5	8	5	2.5	0	-150	0	-150	25	25	6	4.0
80	120	9	5	2.5	9	5	2.5	0	-200	0	-200	25	25	7	4.0
120	150	10	6	2.5	10	7	2.5	0	-250	0	-250	30	30	8	5.0
150	180	10	6	4.0	10	7	5.0	0	-250	0	-250	30	30	8	5.0
180	250	11	7	5.0	13	8	5.0	0	-300	0	-300	30	30	10	6.0
250	315	13	—	—	15	—	—	0	-350	0	-350	35	35	13	—
315	400	15	—	—	20	—	—	0	-400	0	-400	40	40	15	—
400	500	—		—	—	—	—	0	-450	—	—	50	45	—	—

Note: ❸ To be applied for deep groove ball bearing.

Note: △dmp: deviation of the mean bore diameter from the nominal (△dmp = dmp − d).
Vdp: bore diameter variation; difference between the largest and smallest single bore diameters in one plane.
Vdmp: mean bore diameter variation; difference between the largest and smallest mean bore diameters of one ring or washer.
Kia: radial runout of assembled bearing inner ring and assembled bearing outer ring, respectively.
Sd: side face runout with reference to bore (of inner ring).
Sia: side face runout of assembled bearing inner ring and assembled bearing outer ring, respectively.
△Bs: deviation of single inner ring width or single outer ring width from the nominal (△Bs = Bs − B etc.)
VBs: ring width variation; difference between the largest and smallest single widths of inner ring and of outer ring, respectively.

Table 2.2.2 Outer rings

Nominal outside diameter D mm		Single plane mean outside diameter deviation $\triangle D_{mp}$								Single radial plane outside diameter variation V_{Dp}				Mean single plane outside diameter variation V_{Dmp}				Outer ring radial runout K_{ea}			
		class 0		class 6		class 5		class 4 ❹		class 0	class 6	class 5	class 4	class 0	class 6	class 5	class 4	class 0	class 6	class 5	class 4
over	incl.	high	low	high	low	high	low	high	low	max.				max.				max.			
6 ❺	18	0	-8	0	-7	0	-5	0	-4	10	9	5	4	6	5	3	2.0	15	8	5	3
18	30	0	-9	0	-8	0	-6	0	-5	12	10	6	5	7	6	3	2.5	15	9	6	4
30	50	0	-11	0	-9	0	-7	0	-6	14	11	7	6	8	7	4	3.0	20	10	7	5
50	80	0	-13	0	-11	0	-9	0	-7	16	14	9	7	10	8	5	3.5	25	13	8	5
80	120	0	-15	0	-13	0	-10	0	-8	19	16	10	8	11	10	5	4.0	35	18	10	6
120	150	0	-18	0	-15	0	-11	0	-9	23	19	11	9	14	11	6	5.0	40	20	11	7
150	180	0	-25	0	-18	0	-13	0	-10	31	23	13	10	19	14	7	5.0	45	23	13	8
180	250	0	-30	0	-20	0	-15	0	-11	38	25	15	11	23	15	8	6.0	50	25	15	10
250	315	0	-35	0	-25	0	-18	0	-13	44	31	18	13	26	19	9	7.0	60	30	18	11
315	400	0	-40	0	-28	0	-20	0	-15	50	35	20	15	30	21	10	8.0	70	35	20	13
400	500	0	-45	0	-33	0	-23	—	—	56	41	23	—	34	25	12	—	80	40	23	—
500	630	0	-50	0	-38	0	-28	—	—	63	48	28	—	38	29	14	—	100	50	25	—

Note: ❹ The dimensional difference △Ds of the outer diameter to be applied for class 4 is the same as the tolerance of dimensional difference △Dmp of the average outer diameter.
❺ Nominal outer diameter of bearings of 6 mm is included in this dimensional division.

(Unit: μm)

Nominal outside diameter D mm		Outside surface inclination S_d		Outside ring axial runout S_{ia} ❻		Outer ring width deviation $\triangle C_s$	Outer ring width variation V_{Cs}		
		class 5	class 4	class 5	class 4		class 0,6	class 5	class 4
over	incl.	max.		max.		all type	max.		
6 ❺	18	8	4	8	5	Identical to △B_s of inner ring of same bearing	Identical to △Bs and Vbs of inner ring of same bearing	5	2.5
18	30	8	4	8	5			5	2.5
30	50	8	4	8	5			5	2.5
50	80	8	4	10	5			6	3.0
80	120	9	5	11	6			8	4.0
120	150	10	5	13	7			8	5.0
150	180	10	5	14	8			8	5.0
180	250	11	7	15	10			10	7.0
250	315	13	8	18	10			11	7.0
315	400	13	10	20	13			13	8.0
400	500	15	—	23	—			15	—
500	630	18	—	25	—			18	—

Note: ❻ To be applied for deep groove ball bearings.

Note: △Dmp: deviation of the mean outside diameter from the nominal (△Dmp=Dmp − D).
VDp: outside diameter variation; difference between the largest and smallest single outside diameters in one plane.
Vdmp: mean bore diameter variation; difference between the largest and smallest mean bore diameters of one ring or washer.
Kea: radial runout of assembled bearing inner ring and assembled bearing outer ring, respectively.
Sd: side face runout with reference to bore (of inner ring).
Sia: side face runout of assembled bearing inner ring and assembled bearing outer ring, respectively.
△Cs: deviation of single inner ring width or single outer ring width from the nominal (△Bs = Bs − B etc.)
VCs: ring width variation; difference between the largest and smallest single widths of inner ring and of outer ring, respectively.

Table 2.3 Tolerance of thrust roller bearings

Table 2.3.1 Inner rings **(Unit: μm)**

Nominal outer diameter d mm		Single plane mean bore diameter deviation $\triangle d_{mp}$				Single radial plane bore diameter variation V_{dp}		Thrust bearing shaft washer raceway (or center washer raceway) thickness variation S_i			
		class 0,6,5		class 4		class 0,6,5	class 4	class 0	class 6	class 5	class 4
over	incl.	high	low	high	low	max.		max.			
—	18	0	-8	0	-7	6	5	10	5	3	2
18	30	0	-10	0	-8	8	6	10	5	3	2
30	50	0	-12	0	-10	9	8	10	6	3	2
50	80	0	-15	0	-12	11	9	10	7	4	3
80	120	0	-20	0	-15	15	11	15	8	4	3
120	180	0	-25	0	-18	19	14	15	9	5	4
180	250	0	-30	0	-22	23	17	20	10	5	4
250	315	0	-35	0	-25	26	19	25	13	7	5
315	400	0	-40	0	-30	30	23	30	15	7	5
400	500	0	-45	0	-35	34	26	30	18	9	6
500	630	0	-50	0	-40	38	30	35	21	11	7

Table 2.3.2 Outer rings **(Unit: μm)**

Nominal outside diameter D mm		Single plane mean outside diameter deviation $\triangle D_{mp}$				Single radial plane outside diameter variation V_{Dp}		Thrust bearing housing washer raceway thickness variation S_e			
		class 0,6,5		class 4		class 0,6,5	class 4	class 0	class 6	class 5	class 4
over	incl.	high	low	high	low	max.		max.			
10	18	0	-11	0	-7	8	5	According to the tolerance of S_i against "d" or "d_2" of the same bearings			
18	30	0	-13	0	-8	10	6				
30	50	0	-16	0	-9	12	7				
50	80	0	-19	0	-11	14	8				
80	120	0	-22	0	-13	17	10				
120	180	0	-25	0	-15	19	11				
180	250	0	-30	0	-20	23	15				
250	315	0	-35	0	-25	26	19				
315	400	0	-40	0	-28	30	21				
400	500	0	-45	0	-33	34	25				
500	630	0	-50	0	-38	38	29				
630	800	0	-75	0	-45	55	34				

Note: $\triangle d_{mp}$: deviation of the mean bore diameter from the nominal ($\triangle d_{mp} = d_{mp} - d$).
V_{dp}: bore diameter variation; difference between the largest and smallest single bore diameters in one plane.
S_i: thickness variation, measured from middle of raceway to back (seating) face of shaft washer and of housing washer, respectively (axial runout).
$\triangle D_{mp}$: deviation of the mean outside diameter from the nominal ($\triangle D_{mp} = D_{mp} - D$).
V_{Dp}: outside diameter variation; difference between the largest and smallest single outside diameters in one plane.
S_e: thickness variation, measured from middle of raceway to back (seating) face of shaft washer and of housing washer, respectively (axial runout).

3. Bearing fits

3.1 Interference

For rolling bearings, inner and outer rings are fixed on the shaft or in the housing so that relative movement does not occur between fitted surfaces during operation or under load. This relative movement (referred to as "creep") between the fitted surfaces of the bearing and the shaft or housing can occur in a radial direction, an axial direction, or in the direction of rotation. To help prevent this creeping movement, bearing rings and the shaft or housing are installed with one of three interference fits, a "tight fit" (also called shrink fit), "transition fit," or "loose fit" (also called clearance fit), and the degree of interference between their fitted surfaces varies.

The most effective way to fix the fitted surfaces between a bearing's raceway and shaft or housing is to apply a "tight fit." The advantage of this tight fit for thin walled bearings is that it provides uniform load support over the entire ring circumference without any loss of load carrying capacity. However, with a tight fit, ease of installation and disassembly is lost; and when using a non-separable bearing as the floating-side bearing, axial displacement is not possible. For this reason, a tight fit cannot be recommended in all cases.

3.2 The necessity of a proper fit

In some cases, improper fit may lead to damage and shorten bearing life, therefore it is necessary to make a careful analysis in selecting a proper fit. Some of the negative conditions caused by improper fit are listed below.

- Raceway cracking, early peeling and displacement of raceway
- Raceway cracking, early peeling and displacement of raceway
- Raceway and shaft or housing abrasion caused by creeping and fretting corrosion Seizing caused by loss of internal clearances Increased noise and lowered rotational accuracy due to raceway groove deformation

3.3 Fit selection

Selection of a proper fit is dependent upon thorough analysis of bearing operating conditions, including consideration of:

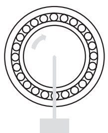
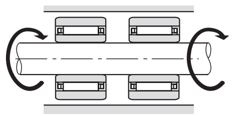
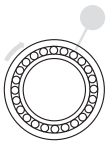
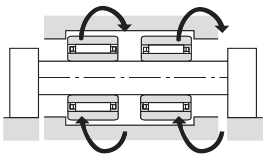
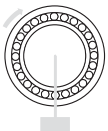
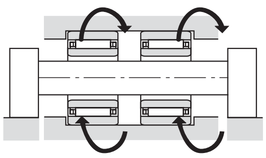
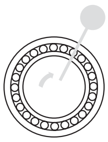
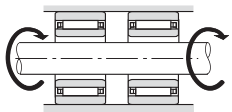
- Shaft and housing material, wall thickness, finished surface accuracy, etc.
- Machinery operating conditions (nature and magnitude of load, rotational speed, temperature, etc.)

3.3.1 "Tight fit," "transition fit," or "loose fit"

For raceways under rotating loads, a tight fit is necessary. (Refer to Table 3.1)
"Raceways under rotating loads" refers to raceways receiving loads rotating relative to their radial direction. For raceways under static loads, on the other hand, a loose fit is sufficient.
(Example) Rotating inner ring load - the direction of the radial load on the inner ring is rotating relatively

For non-separable bearings, such as deep groove ball bearings, it is generally recommended that either the inner ring or outer ring be given a loose fit.

Table 3.1 Radial load and bearing

Illustration	Bearing rotation	Ring load	Fit
Static load 	 Inner ring: Rotating Outer ring: Stationary	Rotating inner ring load	Inner ring: Tight fit
Imbalanced load 	 Inner ring: Stationary Outer ring: Rotating	Static outer ring load	Outer ring: Loose fit
Static load 	 Inner ring: Stationary Outer ring: Rotating	Static inner ring load	Inner ring: Loose fit
Imbalanced load 	 Inner ring: Rotating Outer ring: Stationary	Rotating outer ring load	Outer ring: Tight fit

3.3.2 Recommended Fits

The system of limits and fits define the tolerances of the outside diameter of the shaft or the bore diameter of a housing (the shaft or housing to which a metric bearing is installed). Bearing fit is governed by the selection of tolerances for the shaft outside diameter and housing bore diameter. Fig . 3.1 summarizes the interrelations between shaft outside diameter and bearing bore diameter, and between housing bore diameter and shaft outside diameter.Table 3.2 provides the recommended fits for common radial needle roller bearings (machined ring needle roller bearings with inner ring), relative to dimensions and loading conditions. Table 3.3 is a table of the numerical value of fits.

3.3.3 Interference minimum and maximum values

The following points should be considered when it is necessary to calculate the interference for an application:

- In calculating the minimum required amount of interference keep in mind that:
  1) interference is reduced by radial loads
  2) interference is reduced by differences between bearing temperature and ambient temperature
- 3) interference is reduced by variation of fitted surfaces

Maximum interference should be no more than 1:1000 of the shaft diameter or outer diameter. Required interference calculations are shown below.

3.3.3.1 Fitted surface variation and required interference

Interference between fitted surfaces is reduced by roughness and other slight variations of these surfaces which are flattened in the fitting process. The degree of reduced interference depends upon the finish treatment of these surfaces, but in general it is necessary to assume the following interference reductions.

For ground shafts: 1.0 ~ 2.5 μm
For lathed shafts : 5.0 ~ 7.0 μm

3.3.3.2 Maximum interference

When bearing rings are installed with an interference fit, tension or compression stress may occur along their raceways. If interference is too great, this may cause damage to the rings and reduce bearing life . For these reasons , maximum interference should not exceed the previously mentioned ratio of 1:1,000 of the shaft or outside diameter.

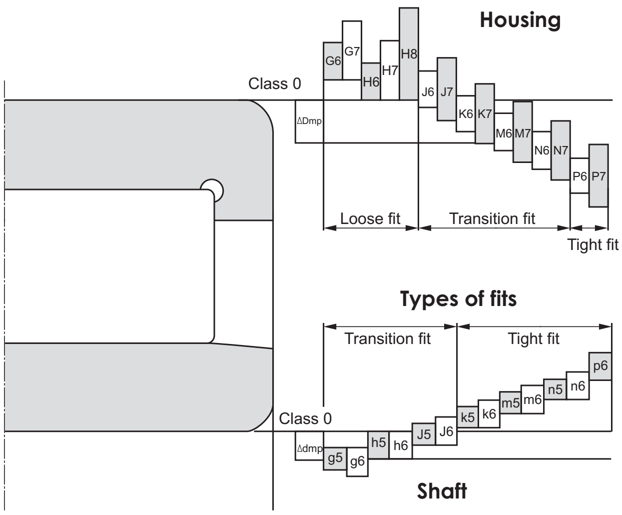


Fig. 4.1

Table 3.2 General standards for needle roller bearing fits

Table 3.2.1 Shaft fits

Nature of load	Fit	Load condition, magnitude	Shaft diameter mm over	Shaft diameter mm incl	Tolerance class	Remarks
Indeterminate direction load Rotating inner ring load	Tight fit/ Transition fit	Light load ❶	~	50	js6	When greater accuracy is required m5 may be substituted for m6.
			50 ~	100	k6	
			100 ~	200	m6	
		Normal load ❶	~	50	k5	
			50 ~	100	m5	
			100 ~	150	m6	
			150 ~	200	n6	
			200 ~		p6	
		Heavy load ❶ or shock load	~	150	n6	When greater accuracy is required m5 may be substituted for m6.
			150 ~		p6	
					r6	
Static inner ring load	Transition fit	Inner ring axial displacement possible	All shaft diameters		g6	When greater accuracy is required use g5. For large bearings, f6 may be used.
		Inner ring axial displacement unnecessary			h6	When greater accuracy is required use h5.
Centric axial load only	Transition fit	All loads	All shaft diameters		js6	General; depending on the fit, shaft and inner rings are not fixed.

❶ Standards for light loads, normal loads, and heavy loads

Light loads : equivalent radial load ⩽ 0.06 *Cr*

Normal loads: 0.06 *Cr* < equivalent radial load ⩽ 0.12 *Cr*

Heavy loads : 0.12 *Cr* <equivalent radial load

Note: All values and fits listed in the above tables are for solid steel shafts.

Table 3.2 .2 Housing fits (Housing of the drawn cup needle roller bearing.)

Nature of load	Housing	Fit	Load condition, magnitude	Tolerance class	Outer ring axial displacement ❷	Remarks
Rotating outer ring load or static outer ring load	Solid housing or split housing	Loose fit	All loads	J7	Displacement possible	G7 also acceptable for large type bearings as well as outer rings and housings with large temperature differences
			Light ❶ to normal load	H7	Displacement possible	—
	Solid housing	Transition or loose fit	High rotation accuracy required with light to normal loads	K6	Displacement not possible(in principle)	Applies primarily to roller bearings
Direction indeterminate load		Tight to transition fit	Light to normal load	J7	Displacement possible	When greater accuracy is required substitute j6 for J7 and K6 for K7.
			Normal to heavy load	K7	Displacement not possible(in principle)	
			Heavy shock load	M7	Displacement not possible	—
Inner ring static load or outer ring rotating load		Tight fit	Light or variable load	M7	Displacement not possible	—
			Normal to heavy load	N7	Displacement not possible	—
			Heavy load (thin wall housing)or heavy shock load	P7	Displacement not possible	—
Centered axial load only - Loose fit		Loose fit	—	Select a tolerance class that will provide clearance between outer ring and housing.		—

❶ Standards for light loads, normal loads, and heavy loads
Light loads: equivalent radial load $\leqslant 0.06\,Cr$
Normal loads: $0.06\,Cr <$ equivalent radial load $\leqslant 0.12\,Cr$
Heavy loads: $0.12\,Cr <$equivalent radial load
❷ Indicates whether or not outer ring axial displacement is possible with non-separable type bearings.

Note 1 : All values and fits listed in the above tables are for cast iron or steel housings.
2 : In cases where only a centered axial load acts on the bearing, select a tolerance class that will provide clearance in the axial direction for the outer ring.

Table 3.3 Numeric value table of fitting for radial bearing of class 0

Table 3.3.1 Fitting against shaft

Nominal bore diameter of bearing *d* mm over	incl.	Single plane mean bore diameter deviation Δdmp high	low	g5 bearing shaft	g6 bearing shaft	h5 bearing shaft	h6 bearing shaft	j5 bearing shaft	js5 bearing shaft	j6 bearing shaft
3	6	0	-8	4T ~ 9L	4T ~ 12L	8T ~ 5L	8T ~ 8L	11T ~ 2L	10.5T ~ 2.5L	14T ~ 2L
6	10	0	-8	3T ~ 11L	3T ~ 14L	8T ~ 6L	8T ~ 9L	12T ~ 2L	11T ~ 3L	15T ~ 2L
10	18	0	-8	2T ~ 14L	2T ~ 17L	8T ~ 8L	8T ~ 11L	13T ~ 3L	12T ~ 4L	16T ~ 3L
18	30	0	-10	3T ~ 16L	3T ~ 20L	10T ~ 9L	10T ~ 13L	15T ~ 4L	14.5T ~ 4.5L	19T ~ 4L
30	50	0	-12	3T ~ 20L	3T ~ 25L	12T ~ 11L	12T ~ 16L	1ST ~ 5L	17.5T ~ 5.5L	23T ~ 5L
50	80	0	-15	5T ~ 23L	5T ~ 29L	15T ~ 13L	15T ~ 19L	21T ~ 7L	21.5T ~ 6.5L	27T ~ 7L
80	120	0	-20	8T ~ 27L	8T ~ 34L	20T ~ 15L	20T ~ 22L	26T ~ 9L	27.5T ~ 7.5L	33T ~ 9L
120	140									
140	160	0	-25	11T ~ 32L	11T ~ 39L	25T ~ 18L	25T ~ 25L	32T ~ 11L	34T ~ 9L	39T ~ 11L
160	180									
180	200									
200	225	0	-30	15T ~ 35L	15T ~ 44L	30T ~ 20L	30T ~ 29L	37T ~ 13L	40T ~ 10L	46T ~ 13L
225	250									
250	280	0	-35	18T ~ 40L	18T ~ 49L	35T ~ 23L	35T ~ 32L	42T ~ 16L	46.5T ~ 11.5L	51T ~ 16L
280	315									
315	355	0	-40	22T ~ 43L	22T ~ 54L	40T ~ 25L	40T ~ 36L	47T ~ 18L	52.5T ~ 12.5L	58T ~ 18L
355	400									
400	450	0	-45	25T ~ 47L	25T ~ 60L	45T ~ 27L	45T ~ 40L	52T ~ 20L	58.5T ~ 13.5L	65T ~ 20L
450	500									

Table 3.3.2 FiTtting against housing

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

Note: T = tight, L = loose

(Unit: μm)

js6 bearing shaft	k5 bearing shaft	k6 bearing shaft	m5 bearing shaft	m6 bearing shaft	n6 bearing shaft	p6 bearing shaft	r6 bearing shaft	Nominal bore diameter of bearing d mm over	 incl
12T ~ 4L	14T ~ 1T	17T ~ 1T	17T ~ 4T	20T ~ 4T	24T ~ 8T	28T ~ 12T	—	3	6
12.5T ~ 4.5L	15T ~ 1T	18T ~ 1T	20T ~ 6T	23T ~ 6T	27T ~ 10T	32T ~ 15T	—	6	10
13.5T ~ 5.5L	17T ~ 1T	20T ~ 1T	23T ~ 7T	26T ~ 7T	31T ~ 12T	37T ~ 18T	—	10	18
16.5T ~ 6.5L	21T ~ 2T	25T ~ 2T	27T ~ 8T	31T ~ 8T	38T ~ 15T	45T ~ 22T	—	18	30
20T ~ 8L	25T ~ 2T	30T ~ 2T	32T ~ 9T	37T ~ 9T	45T ~ 17T	54T ~ 26T	—	30	50
24.5T ~ 9.5L	30T ~ 2T	36T ~ 2T	39T ~ 11T	45T ~ 11T	54T ~ 20T	66T ~ 32T	—	50	80
31T ~11L	38T ~ 3T	45T ~ 2T	48T ~ 13T	55T ~ 13T	65T ~ 23T	79T ~ 37T	—	80	120
37.5T ~ 12.5L	46T ~ 3T	53T ~ 3T	58T ~ 15T	65T ~ 15T	77T ~ 27T	93T ~ 43T	113T ~ 63T	120	140
							115T ~ 65T	140	160
							118T ~ 68T	160	180
44.5T ~ 14.5L	54T ~ 4T	63T ~ 4T	67T ~ 17T	76T ~ 17T	90T ~ 31T	109T ~ 50T	136T ~ 77T	180	200
							139T ~ 80T	200	225
							143T ~ 84T	225	250
51T ~ 16L	62T ~ 4T	71T ~ 4T	78T ~ 20T	87T ~ 20T	101T ~ 34T	123T ~ 56T	161T ~ 94T	250	280
							165T ~ 98T	280	315
58T ~ 18L	69T ~ 4T	80T ~ 4T	86T ~ 21T	97T ~ 21T	113T ~ 37T	138T ~ 62T	184T ~ 108T	315	355
							190T ~ 114T	355	400
65T ~ 20L	77T ~ 5T	90T ~ 4T	95T ~ 23T	108T ~ 23T	125T ~ 40T	153T ~ 68T	211T ~ 126T	400	450
							217T ~ 132T	450	500

(Unit: μm)

K7 housing bearing	M7 housing bearing	N7 housing bearing	P7 housing bearing	Nominal outside diameter of bearing d mm over	 incl
10T ~ 13L	15T ~ 8L	19T ~ 4L	24T ~ 1L	6	10
12T ~ 14L	18T ~ 8L	23T ~ 3L	29T ~ 3L	10	18
15T ~ 15L	21T ~ 9L	28T ~ 2L	35T ~ 5L	18	30
18T ~ 18L	25T ~ 11L	33T ~ 3L	42T ~ 6L	30	50
21T ~ 22L	30T ~ 13L	39T ~ 4L	52T ~ 8L	50	80
25T ~ 25L	35T ~ 15L	45T ~ 5L	59T ~ 9L	80	120
28T ~ 30L	40T ~ 18L	52T ~ 6L	68T ~ 10L	120	150
28T ~ 37L	40T ~ 25L	52T ~ 13L	68T ~ 3L	150	180
33T ~ 43L	46T ~ 30L	60T ~ 16L	79T ~ 3L	180	250
36T ~ 51L	52T ~ 35L	66T ~ 21L	88T ~ 1L	250	315
40T ~ 57L	57T ~ 40L	73T ~ 24L	98T ~ 1L	315	400
45T ~ 63L	63T ~ 45L	80T ~ 28L	108T ~ 0	400	500

4. Bearing internal clearance

Table 4.1 Radial internal clearance of needle roller bearings **(Unit: μm)**

Nominal bore diameter d (mm)		C2		CN		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
-	10	0	30	10	40	25	55	35	65
10	18	0	30	10	40	25	55	35	65
18	24	0	30	10	40	25	55	35	65
24	30	0	30	10	45	30	65	40	70
30	40	0	35	15	50	35	70	45	80
40	50	5	40	20	55	40	75	55	90
50	65	5	45	20	65	45	90	65	105
65	80	5	55	25	75	55	105	75	125
80	100	10	60	30	80	65	115	90	140
100	120	10	65	35	90	80	135	105	160
120	140	10	75	40	105	90	155	115	180
140	160	15	80	50	115	100	165	130	195
160	180	20	85	60	125	110	175	150	215
180	200	25	95	65	135	125	195	165	235
200	225	30	105	75	150	140	215	180	255
225	250	40	115	90	165	155	230	205	280
250	280	45	125	100	180	175	255	230	310
280	315	50	135	110	195	195	280	255	340
315	355	55	145	125	215	215	305	280	370
355	400	65	160	140	235	245	340	320	415
400	450	70	190	155	275	270	390	355	465

5. Lubrication

5.1 Lubrication of rolling bearings

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

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

In order to achieve the above effects, the most effective lubrication method for the operating conditions must be selected. Also, a good quality, reliable lubricant must be selected. In addition, an effectively d&sign&d seating system that prevents the intrusion of damaging elements (Dust, water, etc.) into the bearing interior, removes dust and other impurities from the lubricant, and prevents the lubricant from leaking to the outside, is also a requirement. Almost all rolling bearings use either grease or oil lubrication methods, but in some special applications, a solid lubricant such as molybdenum disulfide or graphite may be used.

5.2 Lubrication methods and characteristics

The lubrication methods come in two general methods: grease or oil, each with their own characteristics. These characteristics are shown in table 5.1

Table 5.1 Comparison of grease lubrication and oil lubrication characteristics

Method / Concern	Grease lubrication	Oil lubrication
Handling	Very good	Fair
Reliability	Good	Very good
Cooling effect	Poor	Good (circulation necessary)
Seal structure	Good	Fair
Power loss	Good	Good
Environment contamination	Good	Fair
High speed rotation	Poor	Good

5.2 Grease lubrication

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

5.2.1 Types and characteristics of grease

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

Standard greases and their characteristics are listed in Table 5.2 as performance characteristics of even the same type of grease will vary widely from brand to brand, it is best to check the manufacturers' data when selecting a grease.

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

However, if different greases must be mixed, at least greases with the same base oil and thickening agent should be selected. But even when greases of the samel base oil and thickening agent are mixed, the quality of U grease may still change due to the difference in additives. For this reason, changes in consistency and other qualities should be checked before being applied.

Table 5.2 Grease varieties and characteristics

Grease name	Lithium grease			Sodium grease (Fiber grease)	Calcium compound base grease
Thickener	Li soap			Na soap	Ca+Na soap Ca+Li soap
Base oil	Mineral oil	Diester oil	Silicone oil	Mineral oil	Mineral oil
Dropping point	170 ~ 190	170 ~ 190	200 ~ 250	150 ~ 180	150 ~ 180
Operating temperature range °C	-30 ~ +130	-50 ~ +130	-50 ~ +160	-20 ~ +130	-20 ~ +120
Mechanical stability	Excellent	Good	Good	Excellent ~ Good	Excellent ~ Good
Pressure resistance	Good	Good	poor	Good	Excellent ~ Good
Water resistance	Good	Good	Good	Good ~ poor	Good ~ poor
Applications	Widest range of applications. Grease used in all types of rolling bearings.	Excellent low temperature and wear characteristics. Suitable for small sized and miniature bearings.	Suitable for high and low temperatures. Unsuitable for heavy load applications due to low oil film strength.	Some emulsification when water is introduced. Excellent characteristics at relatively high temperatures.	Excellent pressure resistance and mechanical stability. Suitable for bearings receiving shock loads.

Grease name	Aluminum grease	Non-soap base grease Thickener	
Thickener	Al soap	Bentone, silica gel, urea, carbon black, fluorine compounds, etc.	
Base oil	Mineral oil	Mineral oil	Synthetic oil
Dropping point	70 ~ 90	250 or above	250 or above
Operating temperature range °C	-10 ~ +80	-10 ~ +130	-50 ~ +200
Mechanical stability	Good ~ poor	Good	Good
Pressure resistance	Good	Good	Good
Water resistance	Good	Good	Good
Applications	Excellent viscosity characteristics. Suitable for bearings subject to vibrations.	Can be used in a wide range of low to high temperatures. Shows excellent heat resistance, cold resistance, chemical resistance, and other characteristics when matched with a suitable base oil and thickener. Grease used in all types of rolling bearings.	

Note: The figures given for operating temperature range are standard characteristic values, and are not guaranteed.

5.2.2 Amount of grease

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

As a general rule, housings and bearings should be only filled from 50% to 80% of their capacities.

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

Moreover, the standard bearing space can be found by below formula (5.1)

$V = K \cdot W$ .... Formula (5.1)

where,
V : Quantity of bearing space open type (approx.) cm^3
K : Bearing space factor (Table 5.3)
W : Mass of bearing kg (See bearing tables)

Table 5.3 Bearings space ratio K

Bearing type	Retainer type	K
Needle roller bearings	Pressed or Machined retainer	35

6. Load rating and life

6.1 Bearing life

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

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

6.2 Basic rated life and basic dynamic load rating

A group of seemingly identical bearings when subjected to identical load and operating conditions will exhibit a wide diversity in their durability. This " life " disparity can be accounted for by the difference in the fatigue of the bearing material itself. This disparity is considered statistically when calculating bearing life, and the basic rated life is defined as follows.

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

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

The relationship between the basic rated life, the basic dynamic load rating and the bearing load is given in formula (6.1).

$L_{10} = \left(\frac{C}{P}\right)^{P}$ .... Formula (6.1)

where,

P = 10/3 ············ For needle roller bearings

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

C : Basic dynamic rating load, N

(C_r: radial bearings, C_a: thrust bearings)

P : Equivalent dynamic load, N

(P_r : radial bearings, P_a: thrust bearings)

The basic rating life can also be expressed in terms of hours of operation (revolution), and is calculated as shown in formula (6.2).

$L_{10h} = 500\ f_h^{\,p}$ .... Formula (6.2)

$f_h = f_n \dfrac{C}{P}$ .... Formula (6.3)

$f_n = \left(\dfrac{33.3}{n}\right)^{1/p}$ .... Formula (6.4)

where,
L_{10} : Basic rating life, h
f_h : Life factor
f_n : Speed factor
n : Rotational speed, r/min

Formula (6.2) can also be expressed as shown in formula (6.5).

$L_{10h} = \dfrac{10^6}{60n}\left(\dfrac{C}{P}\right)^{P}$..................... Formula (6.5)

Roller bearings

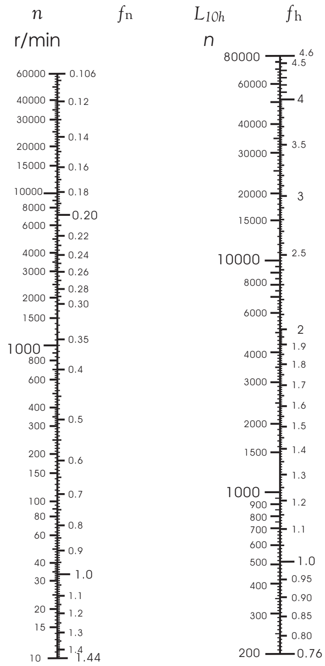


Fig. 6.1 Bearing life rating scale

The relationship between Rotational speed n and speed factor fn as well as the relation between the basic rating life L_{10h} and the life factor fn is shown in Fig. 6.1. When several bearings are incorporated in machines or equipment as complete units, all the bearings in the unit are considered as a whole whencomputing bearing life (see formula 6.6). The total bearing life of the unit is a life rating based on the viable lifetime of the unit before even one of the bearings fails due to rolling contact fatigue.

$L = \dfrac{1}{\left(\dfrac{1}{L_1^{\,e}} + \dfrac{1}{L_2^{\,e}} + \dots \dfrac{1}{L_n^{\,e}}\right)^{1/e}}$ .... Formula (6.6)

where,
e = 9/8.......................For roller bearings
L = Total basic rating life or entire unit, h
$L_1, L_2 \dots L_n$: Basic rating life or individual bearings, 1, 2,...n, h

6.3 Machine applications and requisite life
When selecting a bearing, it is essential that the requisite life of the bearing be established in relation to the operating conditions. The requisite life of the bearing Is usually determined by the type of machine in which the bearing will be used, and duration of service and operational reliability requirements. A general guide to these requisite life criteria is shown in Table 6.1. When determining bearing size, the fatigue life of the bearing is an important factor; however, besides bearing life, the strength and rigidity of the shaft and housing must also be taken into consideration.

6.4 Adjusted life rating factor

The basic bearing life rating (90% reliability factor) can be calculated through the formula mentioned earlier in Section 6.2. However, in some applications a bearing life factor of over 90% reliability may be required. To meet these requirements, bearing life can be lengthened by the use of specially improved bearing materials or special construction techniques. Moreover, according to elastohydrodynamic lubrication theory, it is clear that the bearing operating conditions (lubrication, temperature, speed, etc.) all exert an effect on bearing life.

$L_{na} = a_1 \cdot a_2 \cdot a_3\ (C/P)^P$.......... Formula (6.7)

where,

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

a_1 : Reliability adjustment factor

a_2 : Material adjustment factor

a_3 : Operating condition adjustment factor

6.4.1 Life adjustment factor for reliability a_1

The values for the reliability adjustment factor as (for a reliability factor higher than 90%) can be found in Table 6.1.

Table 6.1 Reliability adjustment factor values a_1

Reliability %	L_n	Reliability factor a_1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

6.4.2 Life adjustment factor for material a_2

The life of a bearing is affected by the material type and quality as well as the manufacturing process. In this regard, the life is adjusted by the use of an a_2 factor.

The basic dynamic load ratings listed in the catalog are based on NIKO's standard material and process, therefore, the adjustment factor $a_2 = 1$. When special materials or processes are used the adjustment factor can be larger than 1.

NIKO bearings can generally be used up to 120°C . If bearings are operated at a higher temperature, the bearing must be specially heat treated (stabilized) so that inadmissible dimensional change does not occur due to changes in the micro-structure. This special heat treatment might cause the reduction of bearing life because of a hardness change.

6.4.3 Life adjustment factor a_3 for operating conditions

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

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

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

As the operating temperature of the bearing increases, the hardness of the bearing material decreases. Thus, the bearing life correspondingly decreases. The operating temperature adjustment values are shown in Fig. 6.2.

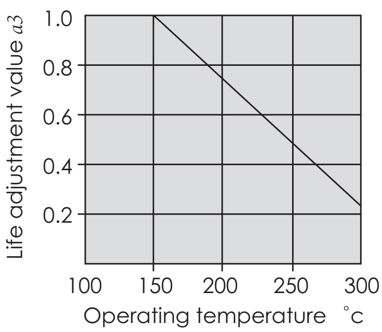


Fig. 6.2 Life adjustment value for operating temperature

6.5 Life of bearing with oscillating motion

The life of a radial bearing with oscillating motion can be calculated according to formula (6.8).

$L_{osc} = \Omega \, L_{Rot}$ .... Formula (6.8)

where,

L_{osc} : life for oscillating bearing

L_{Rot} : rating life at assumed number of rotations same as oscillation cycles

Ω : oscillation factor (Fig.6.3 indicates the relationship between half oscillation angle β and Ω).

Fig. 6.3 is valid only when the amplitude exceeds a certain degree (critical angle 2βc). The critical angle is determined by the internal design of the bearing, in particular by the number of rolling elements in one row. Critical angle values are given in Table 6.3. When the magnitude of the oscillation is less than the critical angle, the life may be shorter than that calculated to be the value in Fig.6.3 It is safer to calculate life with the factor Ω corresponding to the critical angle. For the critical angle of an individual bearing, please consult NIKO Engineering. Where the amplitude of the oscillation 2β is small, it is difficult for a complete lubricant film to form on the contact surfaces of the rings and rolling elements, and fretting corrosion may occur. Therefore it is necessary to exercise extreme care in the selection of bearing type, lubrication and lubricant.

Table 6.3 Critical angle

Number of rolling elements	Half critical angle βc
10	10°
25	4°
40	2.6°

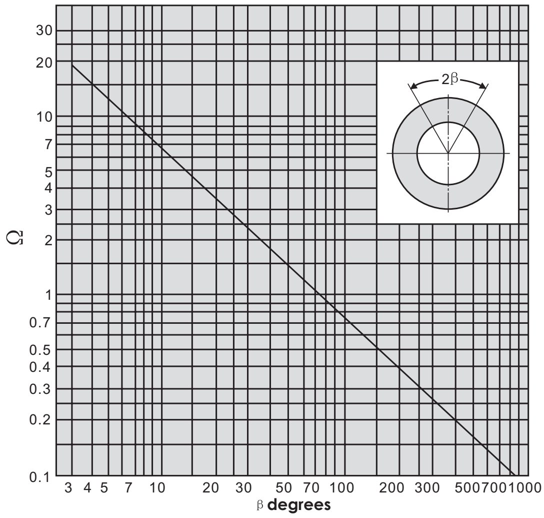


Fig. 6.3 Relationship between half angle β and factor Ω

6.6 Life of bearing with linear motion

With a linear motion bearing such as a linear ball bearing or linear flat roller bearing, the relation among the axial travel distance, bearing load, and load rating is expressed by formulas (6.9).

When the rolling elements are rollers:

$$L = 100 \times \left(\frac{C_r}{P_r}\right)^{\frac{10}{3}} \quad \text{.....................(6.9)}$$

where,

L : Load rating km

Cr : Basic dynamic load rating [kgf]

Pr : Bearing load [kgf]

Fig. 6.4 summarizes the relation between Cr/Pr and L.

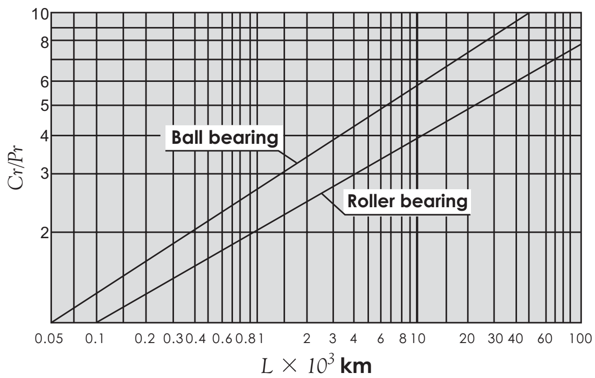


Fig. 6.4 Life of bearing with axial motion

If the cycle and travel distance within a particular travel motion remain constant, the rating life of the bearing can be determined by formulas (6.10) .

$$L_h = \frac{50X10^3}{10 \cdot S}\left(\frac{C_r}{P_r}\right)^{\frac{10}{3}} \text{................ Formula (6.10)}$$

Where,

L_h : Travel life, h

S : Travel distance per minute, m/min.

$S = 2 \cdot L \cdot N$

L : Stroke length, m

n : Stroke cycle, N{kgf}

6.7 Basic static load rating

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

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

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

For roller bearings 4,000 Mpa

6.8 Allowable static equivalent load

Generally the static equivalent load which can be permitted is limited by the basic static rated load as stated in Section 6.7. However, depending on requirements regarding friction and smooth operation, these limits may be greater or lesser than the basic static rated load.

In the following formula (6.11) and Table 6.4 the safety factor S_o can be determined considering the maximum static equivalent load.

$S_o = C_o / P_o$ ...... Formula (6.11)

where,

S_o : Safety factor

C_o : Basic static rated load, N

(radial bearings: C_{or}, thrust bearings: C_{oa})

$P_{o\,max}$: Maximum static equivalent load, N

(radial: $P_{or\,max}$, thrust: $C_{oa\,max}$)

Table 6.4 Minimum safety factor values S_0

Operating conditions	Roller bearings
High rotational accuracy demand	3.0
Normal rotating accuracy demand (Universal application)	1.5
Slight rotational accuracy deterioration permitted (Low speed, heavy loading, etc.)	1.0

Note 1 : For drawn-cup needle roller bearings, min. S_0 value=3.
2 : When vibration and/or shock loads are present, a load factor based on the shock load needs to be included in the P_0 max value.

7. Bearing handling

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

7.1 Bearing storage

Most rolling bearings are coated with a rust preventative before being packed and shipped, and they should be stored at room temperature with a relative humidity of less than 60%.

7.2 Installation

When bearings are being installed on shafts or in housings, the bearing rings should never be struck directly with a hammer or a drift, as shown in Fig. 8.1, because damage to the bearing may result. Any force applied to the bearing should always be evenly distributed over the entire bearing ring face.

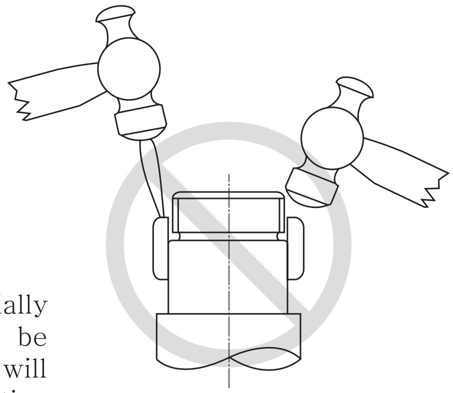


Fig. 7.1

7.2.1 Installation preparations

Bearings should be fitted in a clean, dry work area. Especially for small and miniature bearings, a "clean room" should be provided as any contamination particles in the bearing will greatly affect bearing efficiency. Before installation, all fitting tools, shafts, housings, and related parts should be cleaned and any burrs or cutting chips removed if necessary. Shaft and housing fitting surfaces should also be checked for roughness, dimensional and design accuracy, and to ensure that they are within allowable tolerance limits.

Bearings should not be unwrapped until just prior to installation. Normally, bearings to be usedwith grease lubricant can be installed as is, without removing the rust preventative. However, for bearings which will use oil lubricant, or in cases where mixing the grease and rust preventative would result in loss of lubrication efficiency, the rust preventative should be removed by washing with benzene or petroleum solvent and dried before installation. Bearings should also be washed and dried before installation if the package has been damaged or there are other chances that the bearings have been contaminated. Double shielded bearings and sealed bearings, one way clutches should never be washed.

7.2.2 Installing cylindrical bore bearings

Bearings with relatively small interference fits can be press fit at room temperature by using a sleeve against the inner ring face as shown in Fig. 7.2. Usually, bearings are installed by striking the sleeve with a hammer; however, when installing a large number of bearings, a mechanical or hydraulic press should be used.

When installing non-separable bearings on a shaft and in a housing simultaneously, a pad which distributesth fitting pressure evenly over the inner and outer rings is used as shown in Fig. 7.3. When fitting bearings which have a large inner ring interference fit, or when fitting bearings on shafts that have a large diameter, a considerable amount of force is required to install the bearing at room temperature. Installation can be facilitated by heating and expanding the inner ring beforehand. The required relative temperature difference between the inner ring and the fitting surface depends on the amount of interference and the shaft fitting surface diameter. Fig. 7.4 shows the relation between the bearing inner bore diameter temperature differential and the amount of thermal expansion. In any event, bearings should never be heated above 120°C.

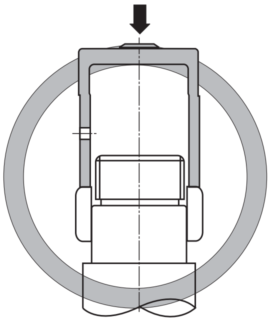


Fig. 7.2 Fitting sleeve pressure against inner ring

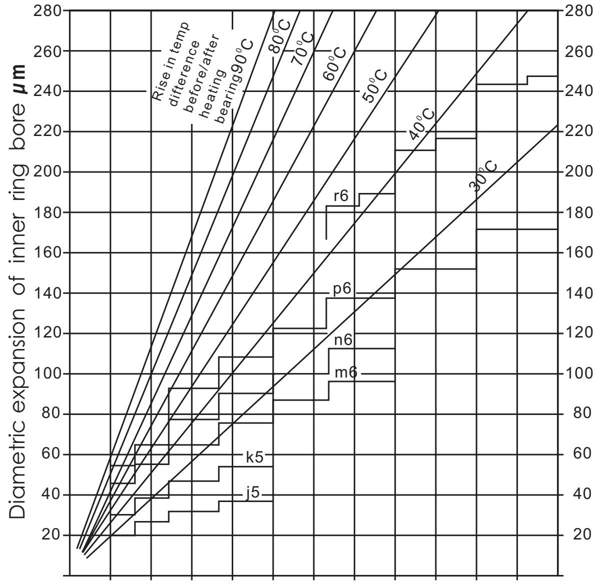


Fig. 7.3 Temperature differential required for shrinkage fit of inner ring

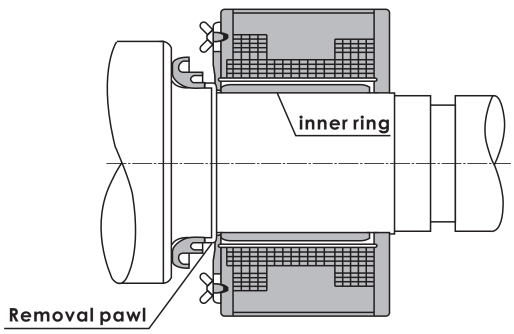


Fig. 7.4 Removal of inner ring using an induction heater

The most commonly used method of heating bearings is to immerse them in hot oil. However, this method should not be used for prelubricated shielded and sealed bearings. To avoid overheating parts of the bearings they should never be brought into direct contact with the heat source, but instead should be suspended inside the heating tank orplacedonawire grid. If bearings are dry-heated with a heating cabinet or hot plate, they can be mounted without drying. An induction heater can be used to quickly heat bearings in a dry state (always demagnetize). When heated bearings are installed on shafts, the inner rings must be held against the shaft abutment until the bearing has been cooled in order to prevent gaps from occurring between the ring and the abutment face.

7.2.3 Installation of outer ring

Even for tight interference fits, the outer rings of small type bearings can be installed by driving them into housings at room temperature. For large type bearings, the housing can be heated before installing the bearing, or the bearing's outer ring can be cooled with dry ice, etc. Before installing. If dry ice or other cooling agent is used, atmospheric moisture will condense on bearing surfaces, and therefore appropriate rust preventative measures are necessary.

7.3 Post installation running test

To insure that the bearing has been properly installed, a running test is performed after installation is completed. The shaft or housing is first rotated by hand and if no problems are observed a low speed, no load power test is performed. If no abnormalities are observed, the load and speed are gradually increased to operating conditions. During the test if any unusual noise, vibration, or temperature rise is observed the test should be stopped and the equipment examined. If necessary, the bearing should be disassembled for inspection. To check bearing running noise, the sound can be amplified and the type of noise ascertained with a listening instrument placed against the housing. A clear, smooth and continuous running sound is normal. A high, metallic or irregular sound indicates some error in function. Vibration can be accurately checked with a vibration measuring instrument, and the amplitude and frequency characteristics measured against a fixed standard. Usually the bearing temperature can be estimated from the housing surface temperature, However, if the bearing outer ring is accessible through oil inlets, etc., the temperature can be more accurately measured. Under normal conditions, bearing temperature rises with rotation time and then reaches a stable operating temperature after a certain period of time. If the temperature does not level off and continues to rise, or if there is a sudden temperature rise, or if the temperature is unusually high, the bearing should be inspected.

7.4 Bearing disassembly

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

7.4.1 Disassembly of bearings with cylindrical bores

For small type bearings, the pullers shown in Fig. 7.5 or the press method shown in Fig. 7.6 can be used for disassembly. When used properly, these methods can improve disassembly efficiency and prevent damage to bearings. To facilitate disassembly procedures, attention should be given to planning the designs of shafts and housings, such as providing extraction grooves on the shaft and housing for puller claws as shown Figs. 7.7 and 7.8. Threaded bolt holes should also be provided in housings to facilitate the pressing out of outer rings as shown in Fig. 7.9.

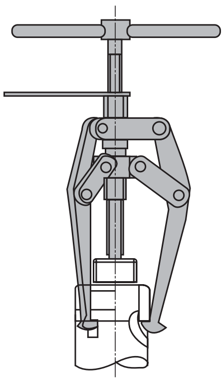


Fig. 7.5 Puller disassembly

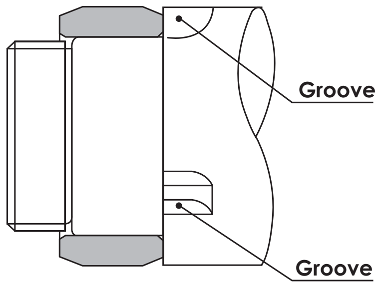


Fig. 7.7 Extracting grooves

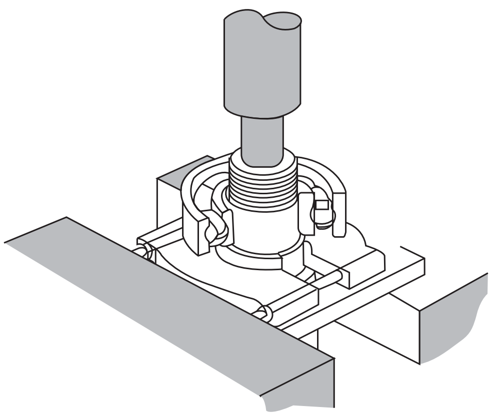


Fig. 7.6 press disassembly

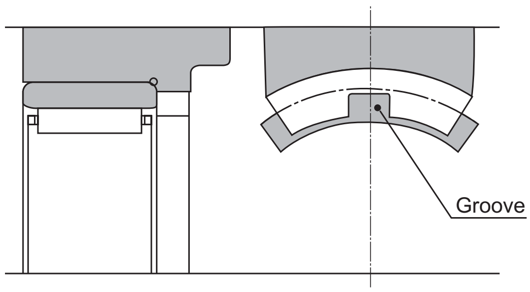


Fig. 7.8 Extraction groove for outer ring disassembly

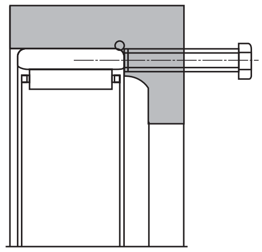


Fig. 7.9 Outer ring disassembly bolt

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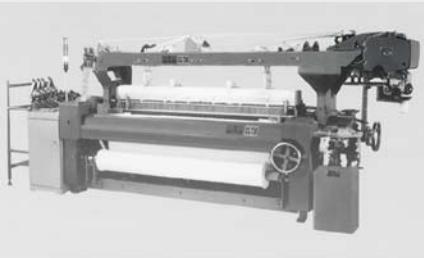

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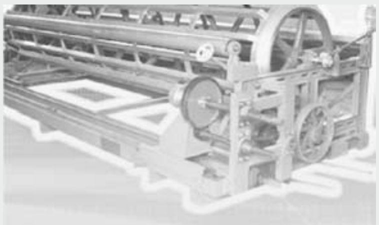

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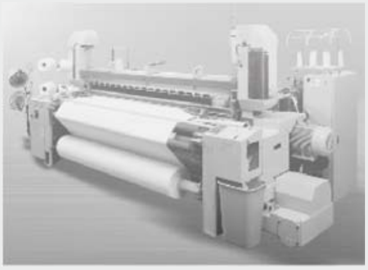

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