8. Application of Bearings

8. Application of Rolling Contact Bearings

8.1 Fits and Clearance

8.1.1 Importance of Fit

To get the best performance from a rolling contact bearing, the fit between the inner ring and shaft, and outer ring and housing must be correct. If the mating surfaces lack interference, the bearing ring may move circumferentially on the shaft or in the housing. This phenomenon is called creep. Once mating surfaces start to creep, the bearing ring will begin to wear excessively and the shaft and/or housing may be damaged. Abrasive debris may enter the bearing to cause abnormal heating or vibration.
Creep is often impossible to prevent by mere fastening of the bearing in an axial direction. To prevent creep, the bearing rings that support the rotating load must be provided with necessary interference. The bearing rings that support stationary load normally do not require interference unless contact corrosion from vibration is a concern.

8.1.2 Selection of Fit

To select the most appropriate fit, the following items must be considered:

- direction of load
- characteristics of load
- magnitude of load
- temperature conditions
- mounting, and dismounting conditions

For general recommendations see Table 8.1.
For mounting bearings in a thin-walled housing or on a hollow shaft, large interference than normal must be provided.
Split-housing applications requiring high precision or tight housing bore fits are not recommended.(A split housing may cause the outer ring to deform).
For application of bearings subjected to vibration, an interference fit should be applied to both inner and outer rings.
Tables 8.2 through 8.14 describe general fit recommendations. For fits not covered by these tables, please contact NACHI.

Table 8.1 Fits vs. Load Characteristics

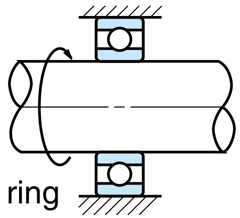
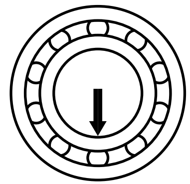
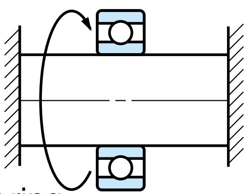
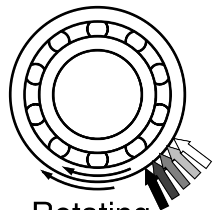
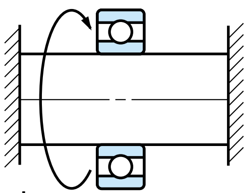
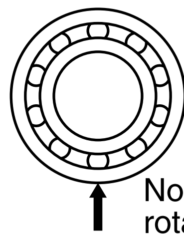
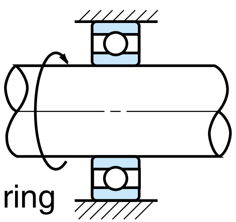
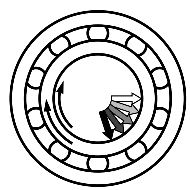
Rotating condition	Type of load	Load conditions	Fit	
			Inner ring	Outer ring
 inner ring	 Non-rotating	Rotating inner ring load	Interference fit	Loose fit
 outer ring	 Rotating	Stationary outer ring load		
 outer ring	 Non-rotating	Rotating outer ring load	Loose fit	Interference fit
 inner ring	 Rotating	Stationary inner ring load		
Load direction not constant because of fluctuation unbalanced load	Rotating or Non-rotating	Indeterminate direction load	Interference fit	Interference fit

Table 8.2.1 Bearing Bore [1] Fits for Radial Bearings

Bearing tolerance class	Fit class vs. load type								
	For rotating inner ring load and indeterminate direction load							For rotating outer ring load	
Class 0, class 6	r 6	p 6	n 6	m 5 m 6	k 5 k 6	j 5 j 6 js 6	h 5	h 5 h 6	g 5 g 6
Class 5, class 4	–	–	–	m 5	k 4	js 4	h 4	h 5	–

Table 8.2.2 Bearing Outside Diameter [1] Fits for Radial Bearings

Bearing tolerance class	Fit class vs. load type									
	For rotating inner ring load				For indeterminate direction load			For rotating outer ring load		
Class 0, class 6	–	J 6 J 7	H 6 H 7	G 7	M 7	K 6 K 7	J 6 J 7	P 7	N 7	M 7
Class 5, class 4	K 5	Js 5	H 5	–	–	–	–	–	–	M 5

Table 8.3.1 Bearing Bore or Center Washer Bore (1) Fits for Thrust Bearings

Bearing tolerance class	Fit class vs. load type				
	For centric axial load	For composite load (spherical roller thrust bearing)			
Class 0	j 6 js 6	n 6	m 6	k 6	j 6 js 6

Table 8.3.2 Bearing Outside Diameter (1) Fits for Thrust Bearings

Bearing tolerance class	Fit class vs. load type		
	For centric axial load	For composite load (spherical roller thrust bearing)	
Class 0	–	M 7	H 7

Note: (1) These dimensional fits are based on JIS B 1514.

Table 8.4 Shaft Tolerances (1) for Radial Bearings

Operating conditions		Shaft diameter (mm)			Tolerance symbols	Remarks	Examples of application (Reference)
		Bell bearings	Cylindrical roller bearings Tapered roller bearings	Spherical roller bearings			
Bearings with cylindrical bore							
Rotating outer ring load	When the inner ring is required to move on the shaft easily	For all shaft diameters			g6	When high precision is required, adopt g5 and h5 respectively. For large bearings, f6 is adopted because of easy bearing movement in axial direction.	Driven wheel
	When the inner ring is required to move on the shaft easily	For all shaft diameters			h6		Tension pully, rope sheave

Note: (1) Shaft tolerances in this table are applied to solid steel shaft for bearings with tolerance class 0 and 6.

Remarks: Heavy load $P > 0.12Cr$, Normal Load $0.06Cr < P \leq 0.12Cr$, Light Load $P \leq 0.06Cr$ Cr: Basic Dynamic Load Rating

Table 8.4 Shaft Tolerances (1) for Radial Bearings

Operating conditions		Shaft diameter (mm) Bell bearings	Shaft diameter (mm) Cylindrical roller bearings Tapered roller bearings	Shaft diameter (mm) Spherical roller bearings	Tolerance symbols	Remarks	Examples of application (Reference)
Rotating inner ring load or indeterminate direction load	Light load or fluctuating load	18 under and incl.	–	–	h5	When high precision is required, adopt j5, k5 and m5 instead of j6, k5 and m6 respectively	Electrical appliance, machining tool, pump, blower, haulage car
		18 Over 100 Incl.	40 under and incl.	–	j6		
		100 Over 200 Incl.	40 Over 140 Incl.	–	k6		
		–	140 Over 200 Incl.	–	m6		
	Normal load or heavy load	18 under and incl.	–	–	j5	The tolerances of k6 and m6 instead of k5 and m5 can be used for single row tapered roller bearings and single row angular contact ball bearings.	Electric motor, turbine, pump, internal combustion engine, wood working machine, bearing application in general.
		18 Over 100 Incl.	40 under and incl.	40 under and incl.	k5		
		100 Over 200 Incl.	40 Over 100 Incl.	40 Over 65 Incl.	m5		
		–	100 Over 140 Incl.	65 Over 100 Incl.	m6		
		–	140 Over 200 Incl.	100 Over 140 Incl.	n6		
		–	200 Over 400 Incl.	140 Over 200 Incl.	p6		
		–	–	280 Over	r6		
	Composite load	–	50 Over 140 Incl.	50 Over 100 Incl.	n6	A bearing with an internal clearance larger than the normal clearance is required	Axles of locomotive and passenger train, traction motor
		–	140 Over 200 Incl.	100 Over 140 Incl.	p6		
		–	200 Over	140 Over	r6		

Note: (1) Shaft tolerances in this table are applied to solid steel shaft for bearings with tolerance class 0 and 6.

Remarks: Heavy load $P > 0.12Cr$, Normal Load $0.06Cr < P \leq 0.12Cr$, Light Load $P \leq 0.06Cr$ Cr: Basic Dynamic Load Rating

Table 8.4 Shaft Tolerances (1) for Radial Bearings

Operating conditions	Shaft diameter (mm)			Tolerance symbols	Remarks	Examples of application (Reference)
	Bell bearings	Cylindrical roller bearings Tapered roller bearings	Spherical roller bearings			
Centric axial load	250 under and incl.			j6	–	–
	250 Over			js6, j6		
Bearing with tapered bore (with sleeve)						
For all load condition	For all shaft condition			h9/IT5	h10/IT7 instead of h9/IT5 can be used for power transmission shaft. IT5 and IT7 mean the form error (out of roundness, taper) should be limited within the tolerance ranges of IT5 and IT7	Railroad car axle, bearing application in general

Note: (1) Shaft tolerances in this table are applied to solid steel shaft for bearings with tolerance class 0 and 6.

Remarks: Heavy load $P > 0.12Cr$, Normal Load $0.06Cr < P \leq 0.12Cr$, Light Load $P \leq 0.06Cr$ Cr: Basic Dynamic Load Rating

Table 8.5 Shaft Tolerances for Thrust Bearings

Operating conditions		Shaft diameter (mm)	Tolerance symbols
Centric axial load (Thrust ball bearings and spherical roller thrust bearings)		250 under and incl.	j6
		250 Over	js6, j6
Composite load (Spherical roller thrust bearings)	Rotating outer ring load	250 under and incl.	j6
		250 Over	js6, j6
	Rotation inner ring load or indeterminate direction load	200 under and incl.	k6
		200 Over 400 Incl.	m6
		400 Over	n6

Table 8.7 Housing Bore Tolerances for Thrust Bearings

Operating conditions		Tolerance symbols	Remarks
Centric axial load (All thrust bearings)	Thrust ball bearing	H8	When high accuracy is not required, radial clearance will be provided between outer ring (housing washer)/aligning housing washer and housing
	Spherical roller thrust bearing; When housing is located in radial direction by another bearing.	–	0.001D is recommended as a radial clearance between outer ring and housing. D: outside diameter of housing washer
Composite load (Spherical roller thrust bearings)	Stationary outer ring load or indeterminate direction load	H7 J7	–
	Rotating outer ring load	K7 M7	In case when the radial load is comparatively large, bearing application in general

Table 8.6 Housing Bore Tolerances (1) for Radial Bearings (Except Inch-series Tapered Roller Bearings) (1/2)

Operating conditions			Tolerance symbols	Outer ring movement (2)	Examples of application (Reference)
Monoblock housing	Rotating outer ring load	When a heavy load is applied to a thin-walled housing or impact load	P7	Outer ring can not be moved in axial direction.	Automotive wheel (roller bearing)
Monoblock housing	Rotating outer ring load	Normal load or heavy load	N7	Outer ring can not be moved in axial direction.	Automotive wheel (ball bearing)
Monoblock housing	Rotating outer ring load	Light load or fluctuating load	M7	Outer ring can not be moved in axial direction.	Conveyer roller, pulley, tension pulley
Monoblock housing	Indeterminate direction load	Heavy impact load	M7	Outer ring can not be moved in axial direction.	Traction motor
Monoblock housing	Indeterminate direction load	Heavy load or normal load; When the outer ring is not required to move in axial direction	K7	Outer ring can not be moved in axial direction as a rule.	Electric motor, pump, crank shaft
Monoblock or split housing	Indeterminate direction load	Normal load or light load; When it is desirable that the outer ring can be moved in axial direction	J7	Outer ring can be moved in axial direction.	Electric motor, pump, crank shaft
Monoblock or split housing	Rotating inner ring load	Impact load; When no-load condition occurs instantaneously	J7	Outer ring can be moved in axial direction.	Railroad car axle
Monoblock or split housing	Rotating inner ring load	All kinds of load	H7	Outer ring can be moved easily in axial direction.	Railroad car axle, bearing application in general
Monoblock or split housing	Rotating inner ring load	Normal load or light load	H8	Outer ring can be moved easily in axial direction.	Gear transmission
Monoblock or split housing	Rotating inner ring load	When thermal conduction through the shaft is caused	G7	Outer ring can be moved easily in axial direction.	Paper mill (Drying cylinder)

Note: (1) The tolerances in this table are applied to cast iron or steel housing for bearings with tolerance class 0 and 6. Tighter fit is adopted for light alloy housing.
(2) Outer ring of non-separable bearing

Table 8.6 Housing Bore Tolerances (1) for Radial Bearings (Except Inch-series Tapered Roller Bearings) (2/2)

Operating conditions			Tolerance symbols	Outer ring movement (2)	Examples of application (Reference)
Monoblock housing	When extremely high accuracy is required	Fluctuating load; When extremely accurate rotation and high rigidity are required	N6	Outer ring can not be moved in axial direction.	Main shaft of machine tool (roller bearing, outside diameter is over 125 mm)
			M6		Main shaft of machine tool (roller bearing outside diameter is under and including 125 mm)
		Indeterminate direction light load; When extremely accurate rotation is required.	K6	Outer ring can not be moved in axial direction as a rule.	Main shaft of grinding machine, ball bearing on grinding wheel side High speed centrifugal compressor, clamping side bearing
		When extremely accurate rotation is required and it is desirable that the outer ring can be moved in axial direction.	J6	Outer ring can be moved in axial direction.	Main shaft of grinding machine, ball bearing on driving side High speed centrifugal compressor, floating side bearing

Note: (1) The tolerances in this table are applied to cast iron or steel housing for bearings with tolerance class 0 and 6. Tighter fit is adopted for light alloy housing.
(2) Outer ring of non-separable bearing

Table 8.8 Fits of Inch Series Tapered Roller Bearings with Shafts

Table 8.8.1 For Bearings with ABMA Classes 4 and 2

Unit: μm

Operating conditions			Bearing bore diameter Nominal d (mm)		Bearing bore deviation		Shaft diameter deviation		Amounts (1)	
			Over	Incl.	High	Low	High	Low	Max	Min
Rotating inner ring load	Normal load No impact		–	76.2	+13	0	+ 38	+ 26	38T	12T
			76.2	304.8	+25	0	+ 64	+ 38	64T	13T
			304.8	609.6	+51	0	+127	+ 76	127T	25T
			609.6	914.4	+76	0	+191	+114	191T	38T
	Heavy load High speed rotation Impact load		–	76.2	+13	0	+ 64	+ 38	64T	25T
			76.2	304.8	+25	0			(2)	
			304.8	609.6	+51	0				
			609.6	914.4	+76	0	+381	+305	381T	229T
Rotating outer ring load	Normal load No impact	Non-ground shaft (When the inner ring is not required to move in axial direction.)	–	76.2	+13	0	+ 13	0	13T	13L
			76.2	304.8	+25	0	+ 25	0	25T	25L
			304.8	609.6	+51	0	+ 51	0	51T	51L
			609.6	914.4	+76	0	+ 76	0	76T	76L
	Normal load No impact	Ground shaft (When the inner ring is required to move in axial direction)	–	76.2	+13	0	0	– 13	0	26L
			76.2	304.8	+25	0	0	– 25	0	51L
			304.8	609.6	+51	0	0	– 51	0	102L
			609.6	914.4	+76	0	0	– 76	0	152L

Note:

(1) T: Tight fit L: Loose fit

(2) Mean amounts of tight fits are d/2000 mm

Table 8.8.2 For Bearings with ABMA Classes 3 and 0

Unit: μm

Operating conditions		Bearing bore diameter Nominal d (mm)		Bearing bore deviation		Shaft diameter deviation		Amounts (1)	
		Over	Incl.	High	Low	High	Low	Max	Min
Rotating inner ring load	Main shaft of precision machine tool	–	304.8	+13	0	+ 38	+18	31T	5T
		304.8	609.6	+25	0	+ 64	+38	64T	13T
		609.6	914.4	+38	0	+102	+63	102T	25T
	Heavy load High speed rotation Impact load	–	76.2	+13	0	(2)			
		76.2	304.8	+13	0				
		304.8	609.6	+25	0				
		609.6	914.4	+38	0				
Rotating outer ring load	Main shaft of precision machine tool	–	304.8	+13	0	+ 13	+18	31T	5T
		304.8	609.6	+25	0	+ 64	+38	64T	13T
		609.6	914.4	+38	0	+102	+63	102T	25T

Note:

(1) T: Tight fit L: Loose fit

(2) Mean amounts of tight fits are d/4000mm

(3) This table is not applied to the bearing with tolerance class 0 whose bore diameter is over 241.3 mm

Table 8.9 Fits of Inch Series Tapered Roller Bearings with Housings

Table 8.9.1 For Bearings with ABMA Classes 4 and 2

Unit: μm

Operating conditions		Bearing outside diameter Nominal D (mm)		Bearing outside diameter deviation		Housing bore diameter deviation		Amounts (1)	
		Over	Incl.	High	Low	High	Low	Max	Min
Rotating inner ring load	Floating side or Clamping side	–	76.2	+25	0	+ 76	+ 50	25L	76L
		76.2	127.0	+25	0	+ 76	+ 50	25L	76L
		127.0	304.8	+25	0	+ 76	+ 50	25L	76L
		304.8	609.6	+51	0	+152	+102	51L	152L
		609.6	914.4	+76	0	+229	+152	76L	229L
	Outer ring location in axial direction can be adjusted	–	76.2	+25	0	+ 25	0	25T	25L
		76.2	127.0	+25	0	+ 25	0	25T	25L
		127.0	304.8	+25	0	+ 51	0	25T	51L
		304.8	609.6	+51	0	+ 76	+ 26	25T	76L
		609.6	914.4	+76	0	+127	+ 51	25T	127L
	Outer ring location in axial direction can not be adjusted	–	76.2	+25	0	− 13	− 39	64T	13T
		76.2	127.0	+25	0	− 25	− 51	76T	25T
		127.0	304.8	+25	0	− 25	− 51	76T	25T
		304.8	609.6	+51	0	− 25	− 76	127T	25T
		609.6	914.4	+76	0	− 25	−102	178T	25T
Rotating outer ring load	Outer ring location in axial direction can not be adjusted	–	76.2	+25	0	− 13	− 39	64T	13T
		76.2	127.0	+25	0	− 25	− 51	76T	25T
		127.0	304.8	+25	0	− 25	− 51	76T	25T
		304.8	609.6	+51	0	− 25	− 76	127T	25T
		609.6	914.4	+76	0	− 25	−102	178T	25T

Note: (1) T: Tight fit L: Loose fit

Table 8.9.2 For Bearings with ABMA Classes 3 and 0

Unit: μm

Operating conditions		Bearing outside diameter Nominal D (mm)		Bearing outside diameter deviation		Housing bore diameter deviation		Amounts (1)	
		Over	Incl.	High	Low	High	Low	Max	Min
Rotating inner ring load	Floating side	–	152.4	+13	0	+38	+26	13L	38L
		152.4	304.8	+13	0	+38	+26	13L	38L
		304.8	609.6	+25	0	+64	+38	13L	64L
		609.6	914.4	+38	0	+89	+51	13L	89L
	Clamping side	–	152.4	+13	0	+25	+13	0	25L
		152.4	304.8	+13	0	+25	+13	0	25L
		304.8	609.6	+25	0	+51	+25	0	51L
		609.6	914.4	+38	0	+76	+38	0	76L
	Outer ring location in axial direction can be adjusted	–	152.4	+13	0	+13	0	13T	13L
		152.4	304.8	+13	0	+25	0	13T	25L
		304.8	609.6	+25	0	+25	0	25T	25L
		609.6	914.4	+38	0	+38	0	38T	38L
	Outer ring location in axial direction can not be adjusted	–	152.4	+13	0	0	–12	25T	0
		152.4	304.8	+13	0	0	–25	38T	0
		304.8	609.6	+25	0	0	–26	51T	0
		609.6	914.4	+38	0	0	–38	76T	0
Rotating outer ring load	Normal load Outer ring location in axial direction can not be adjusted	–	152.4	+13	0	–13	–25	38T	13T
		152.4	304.8	+13	0	–13	–38	51T	13T
		304.8	609.6	+25	0	–13	–39	64T	13T
		609.6	914.4	+38	0	–13	–51	89T	13T

Note: (1) T: Tight fit L: Loose fit
(2) This tables is not applied to the bearing with tolerance class 0 whose bore diameter is over 304.8 mm.

Table 8.10 Amounts of Fits: Radial Bearings with Tolerance JIS Class 0 (ISO Normal Class)

Table 8.10.1 Inner Ring with Shaft (1/2)

Unit: μm

Nominal diameter (mm)		Single plane mean bore diameter deviation of bearing Δdmp		Shaft with tolerance grade IT5											
				m5		k5		j5		h5		g5			
				Tight		Tight		Tight	Loose	Tight	Loose	Tight	Loose		
Over	Incl.	High	Low	Max	Min	Max	Min	Max	Max	Max	Max	Max	Max		
3	6	0	– 8	–	–	–	–	11	2	8	5	4	9		
6	10	0	– 8	–	–	–	–	12	2	8	6	3	11		
10	18	0	– 8	–	–	17	1	13	3	8	8	2	14		
18	30	0	–10	–	–	21	2	15	4	10	9	3	16		
30	50	0	–12	32	9	25	2	18	5	12	11	3	20		
50	80	0	–15	39	11	30	2	21	7	15	13	5	23		
80	120	0	–20	48	13	38	3	26	9	20	15	8	27		
120	140														
140	160	0	–25	58	15	46	3	–	–	25	18	11	32		
160	180														
180	200														
200	225	0	–30	67	17	54	4	–	–	30	20	15	35		
225	250														
250	280	0	–35	–	–	–	–	–	–	35	23	18	40		
280	315														
315	355	0	–40	–	–	–	–	–	–	40	25	22	43		
355	400														
400	450	0	–45	–	–	–	–	–	–	45	27	25	47		
450	500														

Table 8.10 Amounts of Fits: Radial Bearings with Tolerance JIS Class 0 (ISO Normal Class)

Table 8.10.1 Inner Ring with Shaft

Unit: μm

Nominal diameter (mm)		Shaft with tolerance grade IT6															
		r6		p6		n6		m6		k6		j6		h6		g6	
		Tight		Tight		Tight		Tight		Tight		Tight	Loose	Tight	Loose	Tight	Loose
Over	Incl.	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Max	Max	Max	Max	Max
3	6	–	–	–	–	–	–	–	–	–	–	14	2	8	8	4	12
6	10	–	–	–	–	–	–	–	–	–	–	15	2	8	9	3	14
10	18	–	–	–	–	–	–	–	–	20	1	16	3	8	11	2	17
18	30	–	–	–	–	–	–	–	–	25	2	19	4	10	13	3	20
30	50	–	–	–	–	45	17	37	9	30	2	23	5	12	16	3	25
50	80	–	–	–	–	54	20	45	11	36	2	27	7	15	19	5	29
80	120	–	–	76	37	65	23	55	13	45	3	33	9	20	22	8	34
120	140	113	63	93	43	77	27	65	15	53	3	39	11	25	25	11	39
140	160	115	65														
160	180	118	68														
180	200	136	77	109	50	90	31	76	17	63	4	46	13	30	29	15	44
200	225	139	80														
225	250	143	84														
250	280	161	94	123	56	–	–	–	–	–	–	51	16	35	32	18	49
280	315	165	98														
315	355	184	108	138	62	–	–	–	–	–	–	58	18	40	36	22	54
355	400	190	114														
400	450	211	126	–	–	–	–	–	–	–	–	65	20	45	40	26	60
450	500	217	132														

Table 8.10 Amounts of Fits: Radial Bearings with Tolerance JIS Class 0 (ISO Normal Class)

Table 8.10.2 Outer Ring with Housing (1/2)

Unit: μm

Nominal diameter (mm)		Single plane mean outside diameter deviation of bearing ΔDmp		Housing with tolerance grade IT6					
				K6		J6		H6	
				Tight	Loose	Tight	Loose	Tight	Loose
Over	Incl.	High	Low	Max	Max	Max	Max	Max	Max
6	10	0	− 8	7	10	4	13	0	17
10	18	0	− 8	9	10	5	14	0	19
18	30	0	− 8	11	11	5	17	0	22
30	50	0	−11	13	14	6	21	0	27
50	80	0	−13	15	17	6	26	0	32
80	120	0	−15	18	19	6	31	0	37
120	150	0	−18	21	22	7	36	0	43
150	180	0	−25	21	29	7	43	0	50
180	250	0	−30	24	35	7	52	0	59
250	315	0	−35	27	40	7	60	0	67
315	400	0	−40	29	47	7	69	0	76
400	500	0	−45	32	53	7	78	0	85

Table 8.10 Amounts of Fits: Radial Bearings with Tolerance JIS Class 0 (ISO Normal Class)

Table 8.10.2 Outer Ring with Housing

Unit: μm

Nominal diameter (mm)		Housing with tolerance grade IT7													
		P7		N7		M7		K7		J7		H7		G7	
		Tight		Tight	Loose	Tight	Loose	Tight	Loose	Tight	Loose	Tight	Loose	Loose	
Over	Incl.	Max	Min	Max	Max	Max	Max	Max	Max	Max	Max	Max	Max	Min	Max
6	10	24	1	19	4	15	8	10	13	7	16	0	23	5	28
10	18	29	3	23	3	18	8	12	14	8	18	0	26	6	32
18	30	35	5	28	2	21	9	15	15	9	21	0	30	7	37
30	50	42	6	33	3	25	11	18	18	11	25	0	36	9	45
50	80	51	8	39	4	30	13	21	22	12	31	0	43	10	53
80	120	59	9	45	5	35	15	25	25	13	37	0	50	12	62
120	150	68	10	52	6	40	18	28	30	14	44	0	58	14	72
150	180	68	3	60	13	40	25	28	37	14	51	0	65	14	79
180	250	79	3	60	16	46	30	33	43	16	60	0	76	15	91
250	315	88	1	66	21	52	35	36	51	16	71	0	87	17	104
315	400	98	1	73	24	57	40	40	57	18	79	0	97	18	115
400	500	108	0	80	28	63	45	45	63	20	88	0	108	20	128

Table 8.11 Amounts of Fits: Radial Bearings with Tolerance JIS (ISO) Class 6

Table 8.11.1 Inner Ring with Shaft

(1/2)

Unit: μm

Nominal diameter (mm)		Single plane mean bore diameter deviation of bearing Δdmp		Shaft with tolerance grade IT5											
				m5		k5		j5		h5		g5			
				Tight		Tight		Tight	Loose	Tight	Loose	Tight	Loose		
Over	Incl.	High	Low	Max	Min	Max	Min	Max	Max	Max	Max	Max	Max		
3	6	0	− 7	–	–	–	–	10	2	7	5	3	9		
6	10	0	− 7	–	–	–	–	11	2	7	6	2	11		
10	18	0	− 7	–	–	16	1	12	3	7	8	1	14		
18	30	0	− 8	–	–	19	2	13	4	8	9	1	16		
30	50	0	−10	30	9	23	2	16	5	10	11	1	20		
50	80	0	−12	36	11	27	2	18	7	12	13	2	23		
80	120	0	−15	43	13	33	3	21	9	15	15	3	27		
120	140	0	−18	51	15	39	3	–	–	18	18	4	32		
140	160														
160	180														
180	200	0	−22	59	17	46	4	–	–	22	20	7	35		
200	225														
225	250														

Table 8.11 Amounts of Fits: Radial Bearings with Tolerance JIS (ISO) Class 6

Table 8.11.1 Inner Ring with Shaft

Unit: μm

Nominal diameter (mm)		Shaft with tolerance grade IT6															
		r6		p6		n6		m6		k6		j6		h6		g6	
		Tight		Tight		Tight		Tight		Tight		Tight	Loose	Tight	Loose	Tight	Loose
Over	Incl.	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Max	Max	Max	Max	Max
3	6	–	–	–	–	–	–	–	–	–	–	13	2	7	8	3	12
6	10	–	–	–	–	–	–	–	–	–	–	14	2	7	9	2	14
10	18	–	–	–	–	–	–	–	–	19	1	15	3	7	11	1	17
18	30	–	–	–	–	–	–	–	–	23	2	17	4	8	13	1	20
30	50	–	–	–	–	43	17	35	9	28	2	21	5	10	16	1	25
50	80	–	–	–	–	51	20	42	11	33	2	24	7	12	19	2	29
80	120	–	–	74	37	60	23	50	13	40	3	28	9	15	22	3	34
120	140	106	63														
140	160	108	65	86	43	70	27	58	15	46	3	32	11	18	25	4	39
160	180	111	68														
180	200	128	77														
200	225	131	80	101	50	82	31	68	17	55	4	38	13	22	29	7	44
225	250	135	84														

Table 8.11 Amounts of Fits: Radial Bearings with Tolerance JIS (ISO) Class 6

Table 8.11.2 Outer Ring with Housing

Unit: μm

Nominal diameter (mm)		Single plane mean outside diameter deviation of bearing ΔDmp		Housing with tolerance grade IT6					
				K6		J6		H6	
				Tight	Loose	Tight	Loose	Tight	Loose
Over	Incl.	High	Low	Max	Max	Max	Max	Max	Max
6	10	0	− 7	7	9	4	12	0	16
10	18	0	− 7	9	9	5	13	0	18
18	30	0	− 8	11	10	5	16	0	21
30	50	0	− 9	13	12	6	19	0	25
50	80	0	−11	15	15	6	24	0	30
80	120	0	−13	18	17	6	29	0	35
120	150	0	−15	21	19	7	33	0	40
150	180	0	−18	21	22	7	36	0	43
180	250	0	−20	24	25	7	42	0	49
250	315	0	−25	27	30	7	50	0	57
315	400	0	−28	29	35	7	57	0	64

Table 8.11 Amounts of Fits: Radial Bearings with Tolerance JIS (ISO) Class 6

Table 8.11.2 Outer Ring with Housing **(2/2)** Unit: μm

Nominal diameter (mm)		Housing with tolerance grade IT7														
		P7		N7		M7		K7		J7		H7		G7	
		Tight		Tight	Loose	Tight	Loose	Tight	Loose	Tight	Loose	Tight	Loose	Loose	
Over	Incl.	Max	Min	Max	Max	Max	Max	Max	Max	Max	Max	Max	Max	Min	Max
6	10	24	2	19	3	15	7	10	12	7	15	0	22	5	27
10	18	29	4	23	2	18	7	12	13	8	17	0	25	6	31
18	30	35	6	28	1	21	8	15	14	9	20	0	29	7	36
30	50	42	8	33	1	25	9	18	16	11	23	0	34	9	43
50	80	51	10	39	2	30	11	21	20	12	29	0	41	10	51
80	120	59	11	45	3	35	13	25	23	13	35	0	48	12	60
120	150	68	13	52	3	40	15	28	27	14	41	0	55	14	69
150	180	68	10	60	6	40	18	28	30	14	44	0	58	14	72
180	250	79	13	60	6	46	20	33	33	16	50	0	66	15	81
250	315	88	11	66	11	52	25	36	41	16	61	0	77	17	94
315	400	98	13	73	12	57	28	40	45	18	67	0	85	18	103

Table 8.12 Amounts of Fits: Radial Bearings with Tolerance JIS (ISO) Class 5

Table 8.12.1 Inner Ring with Shaft

Unit: μm

Nominal diameter (mm)		Single plane mean bore diameter deviation of bearing Δdmp		Shaft with tolerance grade IT4								Shaft with tolerance grade IT5			
				m4		k4		js4		h4		m5		h5	
				Tight		Tight		Tight	Loose	Tight	Loose	Tight		Tight	Loose
Over	Incl.	High	Low	Max	Min	Max	Min	Max	Max	Max	Max	Max	Min	Max	Max
3	6	0	– 5	13	4	10	1	7	2	5	4	14	4	5	5
6	10	0	– 5	15	6	10	1	7	2	5	4	17	6	5	6
10	18	0	– 5	17	7	11	1	7.5	2.5	5	5	20	7	5	8
18	30	0	– 6	20	8	14	2	9	3	6	6	23	8	6	9
30	50	0	– 8	24	9	17	2	11.5	3.5	8	7	28	9	8	11
50	80	0	– 9	28	11	19	2	13	4	9	8	33	11	9	13
80	120	0	–10	33	13	23	3	15	5	10	10	38	13	10	15
120	180	0	–13	40	15	28	3	19	6	13	12	46	15	13	18
180	250	0	–15	46	17	33	4	22	7	15	14	52	17	15	20

Table 8.12 Amounts of Fits: Radial Bearings with Tolerance JIS (ISO) Class 5

Table 8.12.2 Outer Ring with Housing

Unit: μm

Nominal diameter (mm)		Single plane mean outside diameter deviation of bearing ΔDmp		Housing with tolerance grade IT5							
				M5		K5		Js5		H5	
				Tight	Loose	Tight	Loose	Tight	Loose	Tight	Loose
Over	Incl.	High	Low	Max	Max	Max	Max	Max	Max	Max	Max
6	10	0	− 5	10	1	5	6	3	8	0	11
10	18	0	− 5	12	1	6	7	4	9	0	13
18	30	0	− 6	14	1	8	7	4.5	10.5	0	15
30	50	0	− 7	16	2	9	9	5.5	12.5	0	18
50	80	0	− 9	19	3	10	12	6.5	15.5	0	22
80	120	0	−10	23	2	13	12	7.5	17.5	0	25
120	150	0	−11	27	2	15	14	9	20	0	29
150	180	0	−13	27	4	15	16	9	22	0	31
180	250	0	−15	31	4	18	17	10	25	0	35
250	315	0	−18	36	5	20	21	11.5	29.5	0	41
315	400	0	−20	39	6	22	23	12.5	32.5	0	45

Table 8.13 Amounts of Fits: Radial Bearings with Tolerance JIS (ISO) Class 4

Table 8.13.1 Inner Ring with Shaft

Unit: μm

Nominal diameter (mm)		Single plane mean bore diameter deviation of bearing Δdmp		Shaft with tolerance grade IT4								Shaft with tolerance grade IT5			
				m4		k4		js4		h4		m5		h5	
				Tight		Tight		Tight	Loose	Tight	Loose	Tight		Tight	Loose
Over	Incl.	High	Low	Max	Min	Max	Min	Max	Max	Max	Max	Max	Min	Max	Max
3	6	0	− 4	12	4	9	1	6	2	4	4	13	4	4	5
6	10	0	− 4	14	6	9	1	6	2	4	4	16	6	4	6
10	18	0	− 4	16	7	10	1	6.5	2.5	4	5	19	7	4	8
18	30	0	− 5	19	8	13	2	8	3	5	6	22	8	5	9
30	50	0	− 6	22	9	15	2	9.5	3.5	6	7	26	9	6	11
50	80	0	− 7	26	11	17	2	11	4	7	8	31	11	7	13
80	120	0	− 8	31	13	21	3	13	5	8	10	36	13	8	15
120	180	0	10	37	15	25	3	16	6	10	12	43	15	10	18
180	250	0	−12	43	17	30	4	19	7	12	14	49	17	12	20

Table 8.13 Amounts of Fits: Radial Bearings with Tolerance JIS (ISO) Class 4

Table 8.13.2 Outer Ring with Housing

Unit: μm

Nominal diameter (mm)		Single plane mean outside diameter deviation of bearing ΔDmp		Housing with tolerance grade IT5							
				M5		K5		Js5		H5	
				Tight	Loose	Tight	Loose	Tight	Loose	Tight	Loose
Over	Incl.	High	Low	Max	Max	Max	Max	Max	Max	Max	Max
6	10	0	− 4	10	0	5	5	3	7	0	10
10	18	0	− 4	12	0	6	6	4	8	0	12
18	30	0	− 5	14	0	8	6	4.5	9.5	0	14
30	50	0	− 6	16	1	9	8	5.5	11.5	0	17
50	80	0	− 7	19	1	10	10	6.5	13.5	0	20
80	120	0	− 8	23	0	13	10	7.5	15.5	0	23
120	150	0	− 9	27	0	15	12	9	18	0	27
150	180	0	−10	27	1	15	13	9	19	0	28
180	250	0	−11	31	0	18	13	10	21	0	31
250	315	0	−13	36	0	20	16	11.5	24.5	0	36
315	400	0	−15	39	1	22	18	12.5	27.5	0	40

Table 8.14 Amounts of Fits: Thrust Bearings with Tolerance JIS (ISO) Class 0

Table 8.14.1 Shaft Washer or Center Washer with Shaft

Unit: μm

Nominal diameter (mm)		Single plane mean bore diameter deviation of bearing Δdmp		Shaft with tolerance grade IT6									
				n6		m6		k6		j6			
				Tight		Tight		Tight		Tight	Loose		
Over	Incl.	High	Low	Max	Min	Max	Min	Max	Min	Max	Max		
6	10	0	− 8	–	–	–	–	18	1	15	2		
10	18	0	− 8	–	–	–	–	20	1	16	3		
18	30	0	−10	–	–	–	–	25	2	19	4		
30	50	0	−12	–	–	–	–	30	2	23	5		
50	80	0	−15	–	–	–	–	36	2	27	7		
80	120	0	−20	–	–	–	–	45	3	33	9		
120	180	0	−25	–	–	–	–	53	3	39	11		
180	250	0	−30	–	–	76	17	63	4	46	13		
250	315	0	−35	–	–	87	20	–	–	51	16		
315	400	0	−40	–	–	97	21	–	–	58	18		
400	500	0	−45	125	40	–	–	–	–	65	20		

Table 8.14 Amounts of Fits: Thrust Bearings with Tolerance JIS (ISO) Class 0

Table 8.14.2 Housing Washer with Housing

Unit: μm

Nominal diameter (mm)		Single plane mean outside diameter deviation of bearing ΔDmp		Housing with tolerance grade IT7			
				M7		H7	
				Tight	Loose	Tight	Loose
Over	Incl.	High	Low	Max	Max	Max	Max
10	18	0	–11	18	11	0	29
18	30	0	–13	21	13	0	34
30	50	0	–16	25	16	0	41
50	80	0	–19	30	19	0	49
80	120	0	–22	35	22	0	57
120	180	0	–25	40	25	0	65
180	250	0	–30	46	30	0	76
250	315	0	–35	52	35	0	87
315	400	0	–40	57	40	0	97
400	500	0	–45	63	45	0	108

8.1.3 Calculating Fits

The fits for bearings are often determined empirically according to Table 8.1 through Table 8.14.These tables are NOT to be used for the following cases:

- If special materials are used for interfaces.
- If a hollow shaft is used.
- For high-precision applications.

(1) Reduction of Interference due to Bearing Load

When load is applied through a rotating inner ring, the ring will deform slightly and a gap will occur between the ring and the shaft at a position 180°from the point of load. This gap and "arc-of-no-contact" will increase as the load becomes heavier. A gearing effect will also occur due to the difference in diameters of rotation of the interfacing parts.

Formula (8.1) and Fig. 8.1 define the reduction (millimeters) in interference fit of the inner ring due to bearing load.

$$\Delta dF = 0.08 \times 10^{-3} \sqrt{\frac{d}{B} Fr} \quad \cdots\cdots (8.1)$$

where:

ΔdF : Reduction in interference of inner ring fit due to bearing load (mm)
d : Bearing bore (shaft diameter) (mm)
B : Bearing inner ring width (mm)
Fr : Radial load on the bearing (N)

If the radial load is greater than 20% of the basic static load rating Cor, Formula (8.2) is to be used.

$$\Delta dF \geq 0.02 \times 10^{-3} \frac{Fr}{B} \quad \cdots\cdots (8.2)$$

Calculation example: 6

Object: to obtain the amount of reduction in interference from bearing load where Fr on a single-row, Deep-groove ball bearing number 6320 is 21000N.

From the dimensional tables, d=100 mm, B=47 mm. From Fig. 8.1;

(a) Find 21000 on the line of Fr. Move vertically and intersect the line of d=100 (at point X).

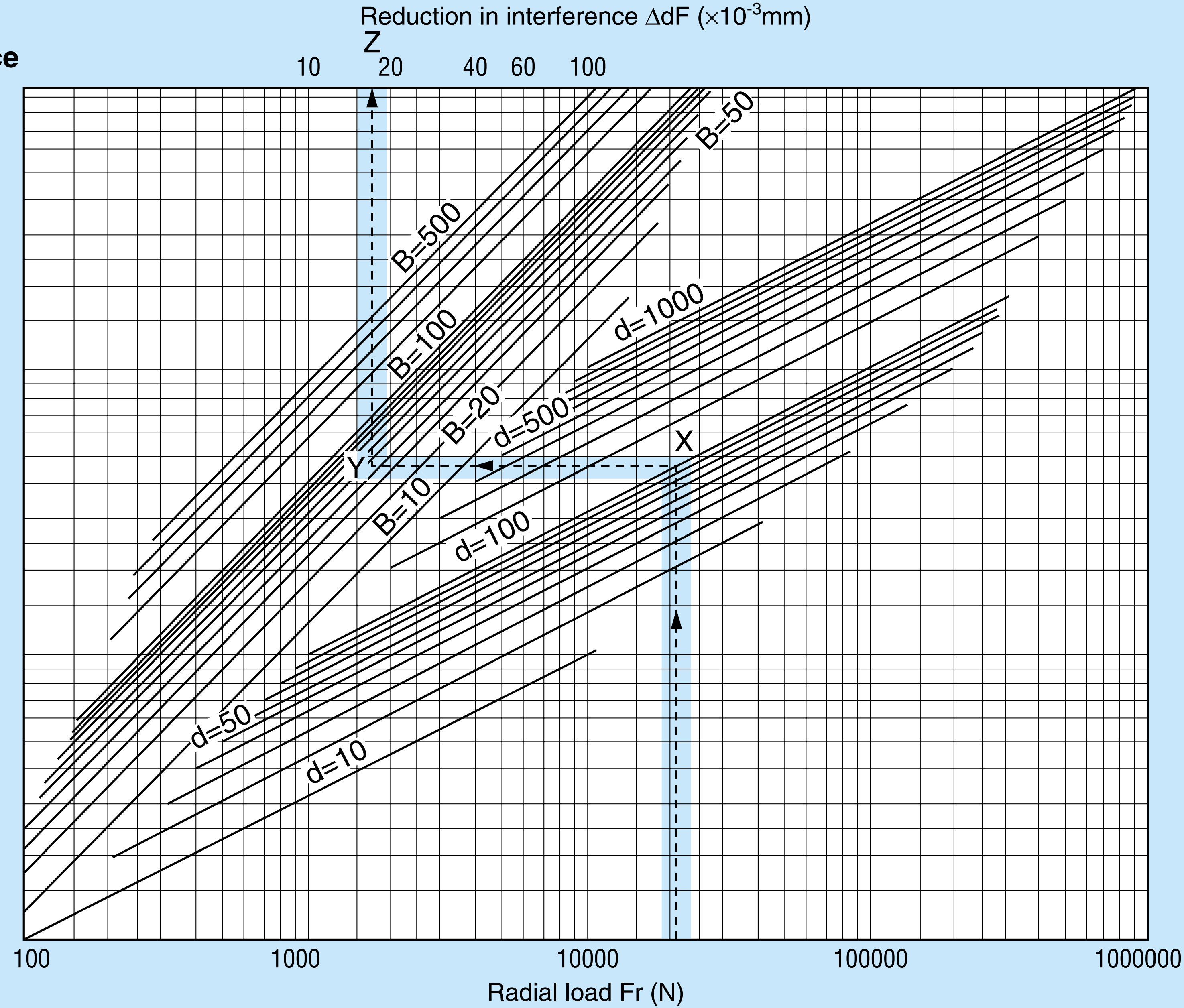
(b) From the point X, move parallel with line Fr and intersect the line of B=47 (at point Y).

(c) Extend vertically from point Y. The intercept with the chart upper limit at point Z indicates the reduction dF (mm) of interference. In this case, ΔdF loss=0.017 (mm).

Fig. 8.1 Change in interference due to load

[→Continue]

Fig. 8.1
Change in interference due to load



[Continue→]

(2) Reduction in Interference due to temperature difference

Operating temperature differences will generally exist between the inner ring and shaft or the outer ring and bearing housing. Fits must be adjusted for differences of thermal expansion coefficients in the mating materials.

- If the bearing temperature is higher than that of the shaft, increase the fit.
- If heat is transferred through the shaft, the fit becomes tighter due to thermal expansion of the shaft. In such cases, increase the radial internal clearance of the bearing.
- When the outer ring temperature is higher than the housing, reduce the fit with the housing and the radial internal clearance of the bearing.
- If the housing temperature is hotter than the bearing outer ring, check the rates of thermal expansion. It will probably be necessary to increase the fit due to larger growth of the housing bore.

Reduction of interference fit of the inner ring due to temperature differentials can be calculated using Formula (8.3) and Fig. 8.2.

$$\Delta dT = 0.0015 \Delta T \cdot d \cdot 10^{-3} \quad \bullet\bullet\bullet\bullet\bullet\bullet \ (8.3)$$

where:

ΔdT: Reduction in interference of inner ring fit due to temperature difference (mm)
ΔT: Temperature difference between bearing and housing ambient (°C)
d : Bearing bore (shaft diameter) (mm)

Calculation example: 7

Obtain the reduction in interference for a temperature difference of 20°C existing between housing ambient temperature and internal temperature of a bearing with a bore diameter of 100 mm. From Fig. 8.2.

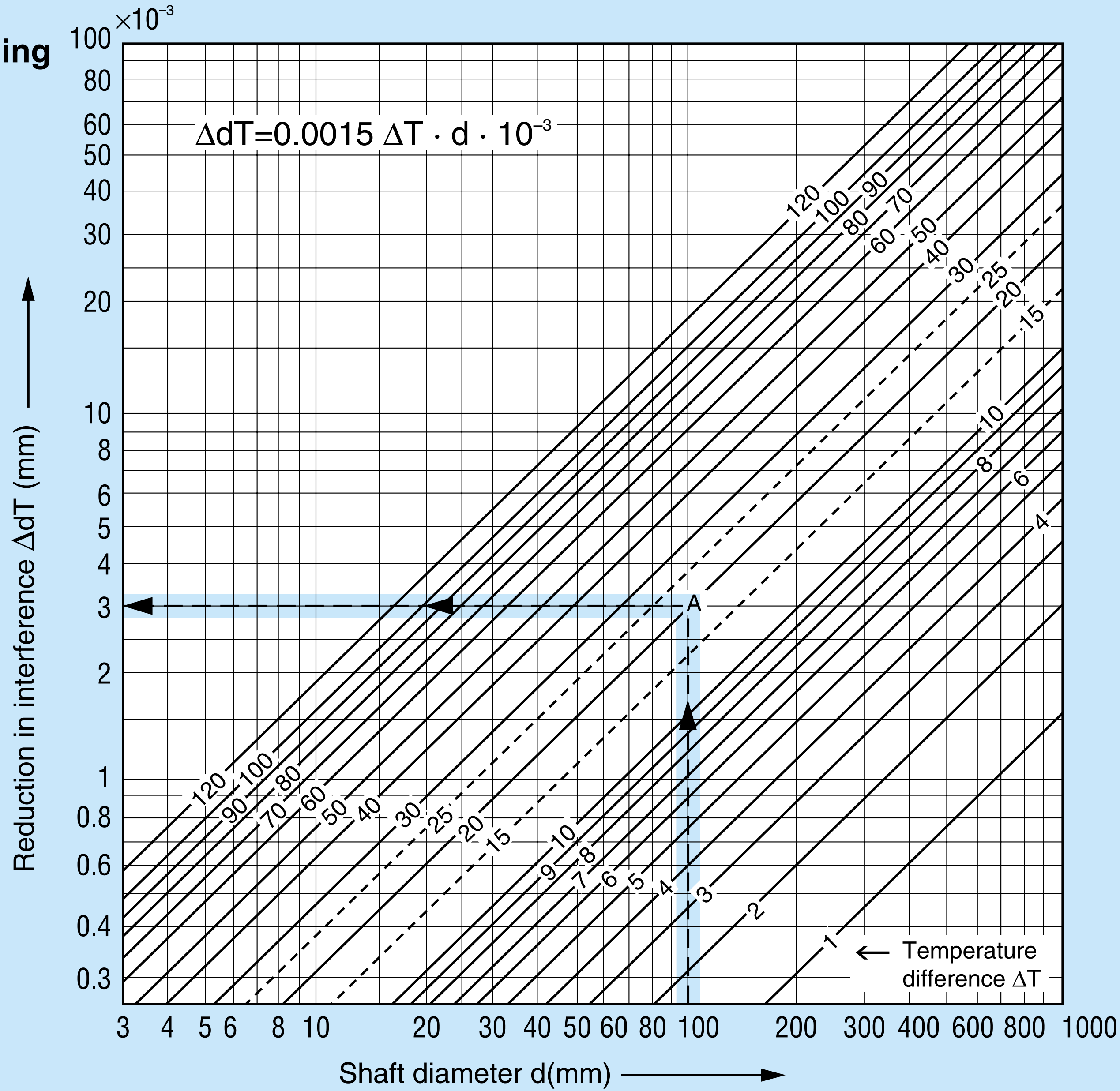
(a) Find the bore diameter d=100 on the horizontal axis. Draw a vertical line from the point until it intersects the line of temperature difference of 20 °C at point A.

(b) Extend a line horizontally from point A left to the Y-axis. The reduction in interference can be read from the intersection with the vertical axis as ΔdT=0.003mm.

Fig. 8.2 Reduction in Interference of Inner Ring Due to Temperature Difference

[→Continue]

Fig. 8.2
Reduction in Interference of Inner Ring Due to Temperature Difference



[Continue→]

(3) Surface Finish Effects on Interference

Since surface asperities are subjected to smoothing when bearings are press-fit, the effective fit becomes smaller than the calculated fit. The amount of reduction in fit is dependent on the surface finish of the interfacing materials.
Effective fit of the inner ring to a solid shaft is calculated using Formulas (8.4.1), and (8.4.2).
For ground and polished shafts,

$$\Delta de = \frac{d}{d+2}\Delta da \quad \cdots\cdots (8.4.1)$$

For machined shafts,

$$\Delta de = \frac{d}{d+3}\Delta da \quad \cdots\cdots (8.4.2)$$

where:

Δde : Effective interference (mm)
Δda : Calculated interference (mm)
d : Bearing bore diameter (mm)

(4) Necessary Interference for Inner Rings

Formulas (8.1), (8.2), (8.3), (8.4.1) and (8.4.2) have been used to calculate the effects of Load, Temperature, and Surface Finish in interference. To summarize the effects to a total required interference for the inner ring and shaft (where inner ring rotates against load), refer to Formulas (8.5.1) and (8.5.2).
For ground and polished shafts,

$$\Delta da \geq (\Delta dF + \Delta dT)\left(\frac{d+2}{d}\right) \quad \cdots\cdots (8.5.1)$$

For machined shafts,

$$\Delta da \geq (\Delta dF + \Delta dT)\left(\frac{d+3}{d}\right) \quad \cdots\cdots (8.5.2)$$

[→Continue]

(5) Expansion Stress from Fits

When interference is provided, the bearing ring undergoes tensile stress. If the stress is excessive, the bearing ring will be damaged. When an inner ring is fitted to a solid steel shaft, stress,σi, should be limited to 100MPa or smaller using Formula (8.6). Empirically, the criterion of interference 0.001 of shaft diameter,

$$\sigma_i = \frac{E}{2} \cdot \frac{\Delta de}{d} \left\{ 1 + \left(\frac{d}{d_i} \right)^2 \right\} \quad \cdots\cdots (8.6)$$

where:

σi	: Maximum bore diameter surface stress (MPa)
E	: Vertical elastic coefficient for steel: 2.07×10^5(MPa)
Δde	: Effective interference (mm)
d	: Bearing bore diameter (mm)
di	: Mean outside diameter of inner ring (mm)

Cylindrical roller bearings; and Self-aligning ball bearings of series 22 and 23:

$di \fallingdotseq 0.25(D+3d)$

where:

D : Bearing outside diameter (mm)

All other bearings:

$di \fallingdotseq 0.1(3D+7d)$

(6) Fits for Inner Rings with Hollow Shafts

Equivalent effective fit for a hollow shaft.

(a) Calculate the interference, Δda for a solid shaft of the identical diameter inner ring with either Table 8.4 or Formulas (8.5.1) and (8.5.2).

(b) Calculate interference Δdha for a hollow shaft and inner ring with Formula (8.7).

$$\Delta dha = \frac{1 - \left(\frac{d_h}{d_i} \right)^2}{1 - \left(\frac{d_h}{d} \right)^2} \Delta da \quad \cdots\cdots (8.7)$$

where:

Δdha	: Calculated interference of hollow shaft (mm)
dh	: Bore diameter of hollow shaft (mm). For solid shaft, dh=0
d	: bearing bore diameter (mm)
Δda	: Calculated interference of solid shaft and inner ring (mm)

(c) Expansion stress force from fits for hollow steel shaft is calculated using Formula (8.8).

$$\sigma_i = \frac{E}{2} \cdot \frac{\Delta de}{d} \cdot \frac{\left\{ 1 - \left(\frac{d_h}{d} \right)^2 \right\} \left\{ 1 + \left(\frac{d}{d_i} \right)^2 \right\}}{\left\{ 1 - \left(\frac{d_h}{d_i} \right)^2 \right\}} \quad \cdots\cdots (8.8)$$

(7) Outer Ring to Housing Fits

Interference fit must be provided between the outer ring and housing where there is rotating outer ring load or indeterminate load. Fits for outer ring and steel housing can be obtained by using Table 8.6 and maximum stress of the outer ring can be calculated with Formula (8.9).

$$\sigma_o = \frac{E}{2} \cdot \frac{\Delta De}{D} \cdot \frac{1-\left(\frac{D}{Dh}\right)^2}{1-\left(\frac{De}{Dh}\right)^2} \quad \cdots\cdots\cdots (8.9)$$

where:

σo	: Maximum outer ring bore surface stress (MPa)
E	: Vertical elastic coefficient for steel: 2.07×10^5(MPa)
ΔDe	: Effective interference (mm)
D	: Bearing outside diameter (mm)
Dh	: Housing outside diameter (mm)

(Note): If the housing is rigid body;

$Dh = \infty$

De = Mean bore diameter of outer ring (mm)

Cylindrical roller bearings and Self-aligning ball bearings of series 22 and 23:

$De \fallingdotseq 0.25(3D+d)$

All other bearings:

$De \fallingdotseq 0.1(7D+3d)$

8.1.4 Selection of Bearing Clearance

The internal clearance of rolling contact bearings during operation (the operating clearance) is a factor which can affect bearing life, vibration, heat, sound, etc.
Theoretically, bearing life is maximum if bearings operate with a slight preload (a slight negative operating clearance). If a bearing is to operate with a slight preload, great care must be taken in the analysis and design of the application to be sure that preloads do not begin to rise during the bearing operation to a level which will lead to an upward spiraling of heat=greater preload=more heat=early bearing failure. And also a bearing with an excessive operating clearance will not perform its maximum load capability.
To prevent clearance problems, unmounted bearing clearance should be selected so that operating clearance will be slightly positive. (Note that bearings chosen for precision location functions are preloaded, but the amount of preload must be precisely controlled at assembly).
For non-separable, radial bearings, and for radial Cylindrical roller bearings, which are assembled in clearance groups with a set amount of “unmounted” internal clearance; the initial internal clearance will be the unmounted clearance minus clearance losses from mounting fits.
Typical clearance groups for the above types of bearings are:

- C2 : less than Normal clearance
- CN : Normal clearance
- C3 : more than Normal clearance

CN (Normal) internal clearance is determined so that appropriate clearance will remain after the bearing is mounted to the shaft with an interference fit, but with no fit (no interference) between the outer ring and housing and the temperature difference between inner and outer ring is 10 °C or less.
Table 8.15 indicates examples of selection for clearance groups other than CN (Normal) internal clearance.
Bearing clearance varies during operation with respect to the temperature rise and the type and magnitude of load. For example, if large reduction of clearance is expected, more initial clearance is required.
Fig. 8.3 illustrates radial clearance of a single-row Deep-groove ball bearing.

Table 8.15 Examples of Selection of Clearance Other Than CN (Normal) Clearance

Fig. 8.3 Radial Clearance

[→Continue]

Table 8.15 Examples of Selection of Clearance Other Than CN (Normal) Clearance

Service Conditions	Clearance	Application Examples (reference)
Large interference for heavy or impact load	C3 clearance or larger	Railroad car axle
Interference in required for both inner and outer rings due to indeterminate heavy impact load		Traction motor
Inner ring is exposed to high temperature. Outer ring exposed to low temperature.		Pulp and paper machine dryer For outdoor use in cold area
When shaft has a large deflection. For increasing axial load capacity by increasing contact angle.		Semi-floating axle of automobile Bearing of rail road car axle for carring axial load. Thrust bearing of axles of rolling stock
When both inner and outer rings are clearance-fitted.	C2 clearance or smaller	Roll neck of rolling machine
For controlling vibration and sound.		Small, special electric motors
For post-assembly adjustment of clearance such as controlling deviation of shaft, etc.	C9na , C1na	Cylindrical roller bearing for lathe main shaft

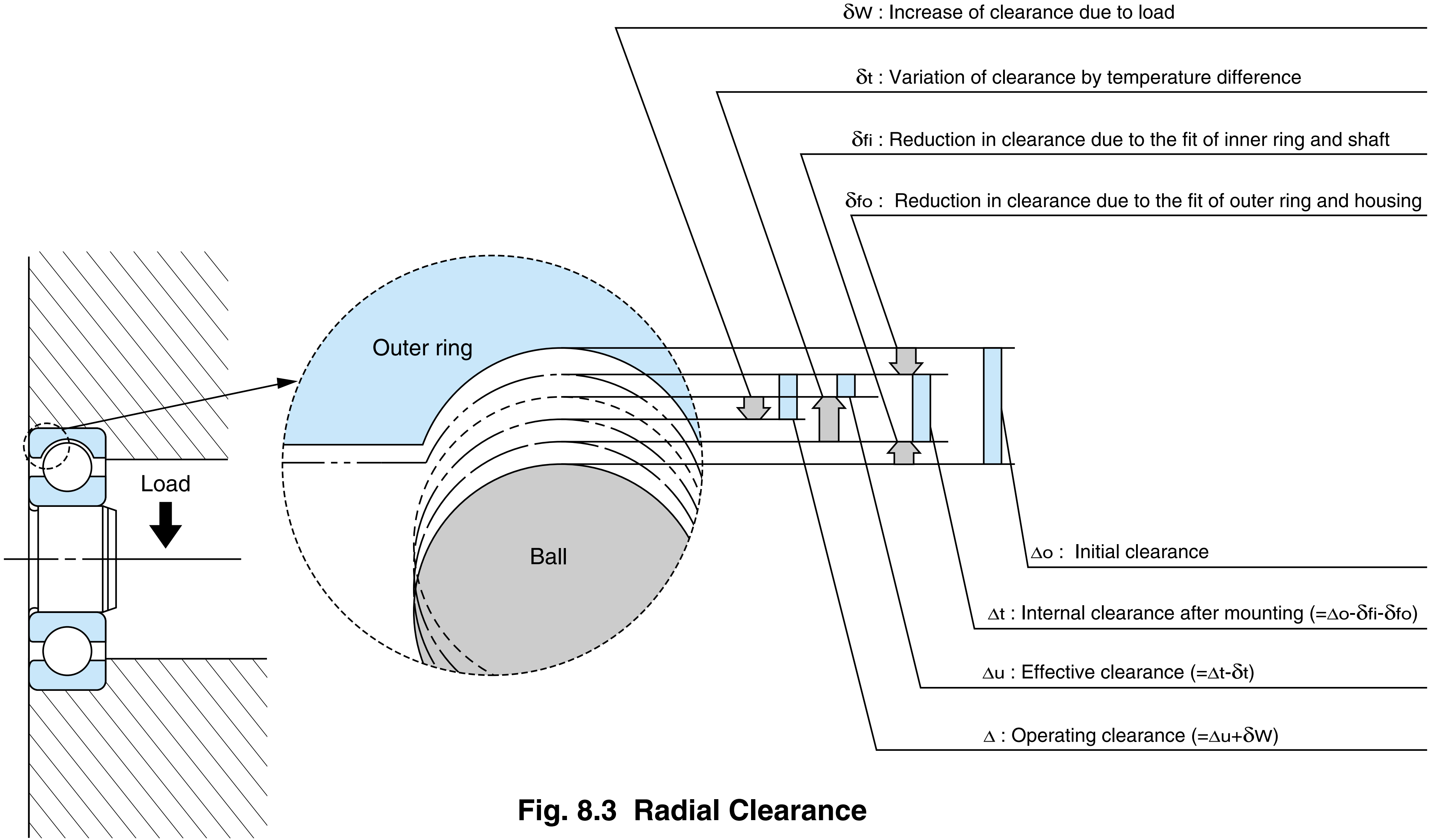


Fig. 8.3 Radial Clearance

[Continue→]

(1) Operating Clearance

Operating clearance is defined as the clearance of a bearing operating in a machine at the operating temperature and load.

$$\Delta = \Delta o - (\delta t + \delta f) + \delta w \quad \cdots\cdots (8.10)$$

where:

Δ : Operating clearance (mm)
Δo : Unmounted bearing clearance
δt : Variation of clearance from temperature difference between inner and outer rings (mm)
δf : Reduction in clearance due to the fit of inner and outer rings (mm)
δw : Increase of clearance due to load (mm)

(2) Internal clearance reduction due to temperature difference between inner and outer rings

Under normal operating conditions, the temperature of the rolling contact bearing components is, in ascending order from the lowest to the highest; the outer ring, the inner ring, and the rolling elements.
Since it is extremely difficult to measure the temperature of the rolling elements, operating temperature is calculated under the assumption that the temperature of the rolling element is equal to that of the inner ring. Therefore, the reduction in clearance due to temperature difference between the inner and outer rings can be obtained by the following formula:

$$\delta t = \alpha \cdot \Delta T \cdot Do \quad \cdots\cdots (8.11)$$

where:

δt : Reduction in clearance due to temperature difference between inner and outer rings (mm)
α : Linear expansion coefficient of bearing steel: $1.12 \times 10\text{-}5$ (1/ °C) for operating temperature 300°C or less
ΔT : Temperature difference between the inner and outer rings (°C)
Do : Outer ring raceway diameter (mm)

$Do \fallingdotseq 0.2(4D+d)$ for Deep-groove ball bearings and Spherical roller bearings.
$Do \fallingdotseq 0.25(3D+d)$ for Cylindrical roller bearings.

[→Continue]

[Continue→]

(3) Reduction in clearance due to fit

When a bearing is mounted to a shaft or housing with an interference fit, the inner ring will expand or the outer ring will contract (due to the fit), causing reduction in the bearing internal clearance. Reduction in clearance due to fit can be calculated from the following formula:

$$\delta f = \delta fi + \delta fo \quad \cdots\cdots (8.12)$$

where:

- δf : Reduction in clearance due to fit (mm)
- δfi : Reduction in clearance due to expansion of the inner ring (mm)
- δfo : Reduction in clearance due to the contraction of the outer ring (mm)

$$\delta fi = \Delta de \cdot \frac{d}{di} \cdot \frac{1-\left(\frac{dh}{d}\right)^2}{1-\left(\frac{dh}{di}\right)^2} \quad \cdots\cdots (8.13)$$

$$\delta fo = \Delta De \cdot \frac{De}{D} \cdot \frac{1-\left(\frac{D}{Dh}\right)^2}{1-\left(\frac{De}{Dh}\right)^2} \quad \cdots\cdots (8.14)$$

where:

- Δde : Effective interference of the inner ring (mm)
- d : Bearing bore diameter (mm)
- di : Mean outside diameter of inner ring (mm)
- dh : Inside diameter of hollow shaft (mm)
  (Note): For solid shaft, dh=0
- ΔDe : Effective interference of outer ring (mm)
- D : Bearing outside diameter (mm)
- De : Mean inside diameter of outer ring (mm)
- Dh : Housing outside diameter (mm)
  Note: If the housing is a rigid body, $Dh=\infty$.
- di $\fallingdotseq 0.25(D+3d)$ for Cylindrical roller bearings and Self-aligning Ball bearings of bearing series 22 and 23
- di $\fallingdotseq 0.1(3D+7d)$ for other bearings
- De $\fallingdotseq 0.25(3D+d)$ for Cylindrical roller bearings and Self-aligning Ball bearings of bearing series 22 and 23
- De $\fallingdotseq 0.1(7D+3d)$ for other bearings

For estimating δf, the following may be used:

$\delta f=0.7\ (\Delta de+\Delta De)$ to

$0.9\ (\Delta de+\Delta De)$,

with smaller values for heavy-section bearings (e.g. bearings of diameter series 4) and larger values for light-section bearing rings. (e.g. bearings of diameter series 9)

[→Continue]

[Continue→]

(4) Increase of clearance due to load

When a bearing is subjected to a load, elastic deformation will occur and this deformation will cause an increase in internal clearance. Table 8.16 outlines elastic deformation δr and δa.

Table 8.16 Load and Elastic Deformation

Table 8.16 Load and Elastic Deformation

Bearing type	Approximation of deformation from radial load δr (mm)	Approximation of deformation from axial load δa (mm)
Self-aligning Ball bearings	$\delta r = \dfrac{0.00070}{\cos\alpha} \sqrt[3]{\dfrac{Po^2}{Dw}}$	$\delta a = \dfrac{0.00070}{\sin\alpha} \sqrt[3]{\dfrac{P^2}{Dw}}$
Deep groove ball bearings Angular Contact ball bearings	$\delta r = \dfrac{0.00044}{\cos\alpha} \sqrt[3]{\dfrac{Po^2}{Dw}}$	$\delta a = \dfrac{0.00044}{\sin\alpha} \sqrt[3]{\dfrac{P^2}{Dw}}$
Spherical roller bearings	$\delta r = \dfrac{0.00018}{\cos\alpha} \sqrt[4]{\dfrac{Po^3}{Lwe^2}}$	$\delta a = \dfrac{0.00018}{\sin\alpha} \sqrt[4]{\dfrac{P^3}{Lwe^2}}$
Cylindrical roller bearings Tapered roller bearings	$\delta r = \dfrac{0.000077}{\cos\alpha} \cdot \dfrac{Po^{0.9}}{Lwe^{0.8}}$	$\delta a = \dfrac{0.000077}{\sin\alpha} \cdot \dfrac{P^{0.9}}{Lwe^{0.8}}$
Thrust ball bearings	–	$\delta a = \dfrac{0.00052}{\sin\alpha} \sqrt[3]{\dfrac{P^2}{Dw}}$
Po and P	$Po = \dfrac{5Fr}{iz\cos\alpha}$	$P = \dfrac{Fa}{z\sin\alpha}$

where:
- Fr = Radial load (N)
- Fa = Axial load (N)
- α = Contact angle (°)
- Dw = Diameter of ball or roller (mm)
- Lwe = Effective roller length (mm)
- i = Number of row of ball or roller
- z = Number of ball or roller per row

8.2 Preload and Rigidity

Generally, rolling contact bearings are mounted so that in operation, there will be a small amount of internal clearance. Applications may sometimes require that the bearings be provided with appropriate negative clearance called "preload" when assembled.
Preload has various purposes and effects. Since an incorrect amount of preload may adversely affect the rolling resistance, life, temperature rise, sound, etc. of bearings; extreme care must be taken when applying preload.

8.2.1 Purposes of Preload

(1) Increases rigidity of a shaft (that is, preloading can help to decrease the deflection of shafting).
(2) Enhances rotating accuracy of shaft. Minimizes axial movements and helps to prevent vibration and decrease noise.
(3) Prevents fretting caused by external vibration.
Item 1 and 2 are pertinent with respect to proper gear engagement, rotating accuracy of precision machinery and resonance of electric motor rotors.

8.2.2 Preloading Method and Measurement

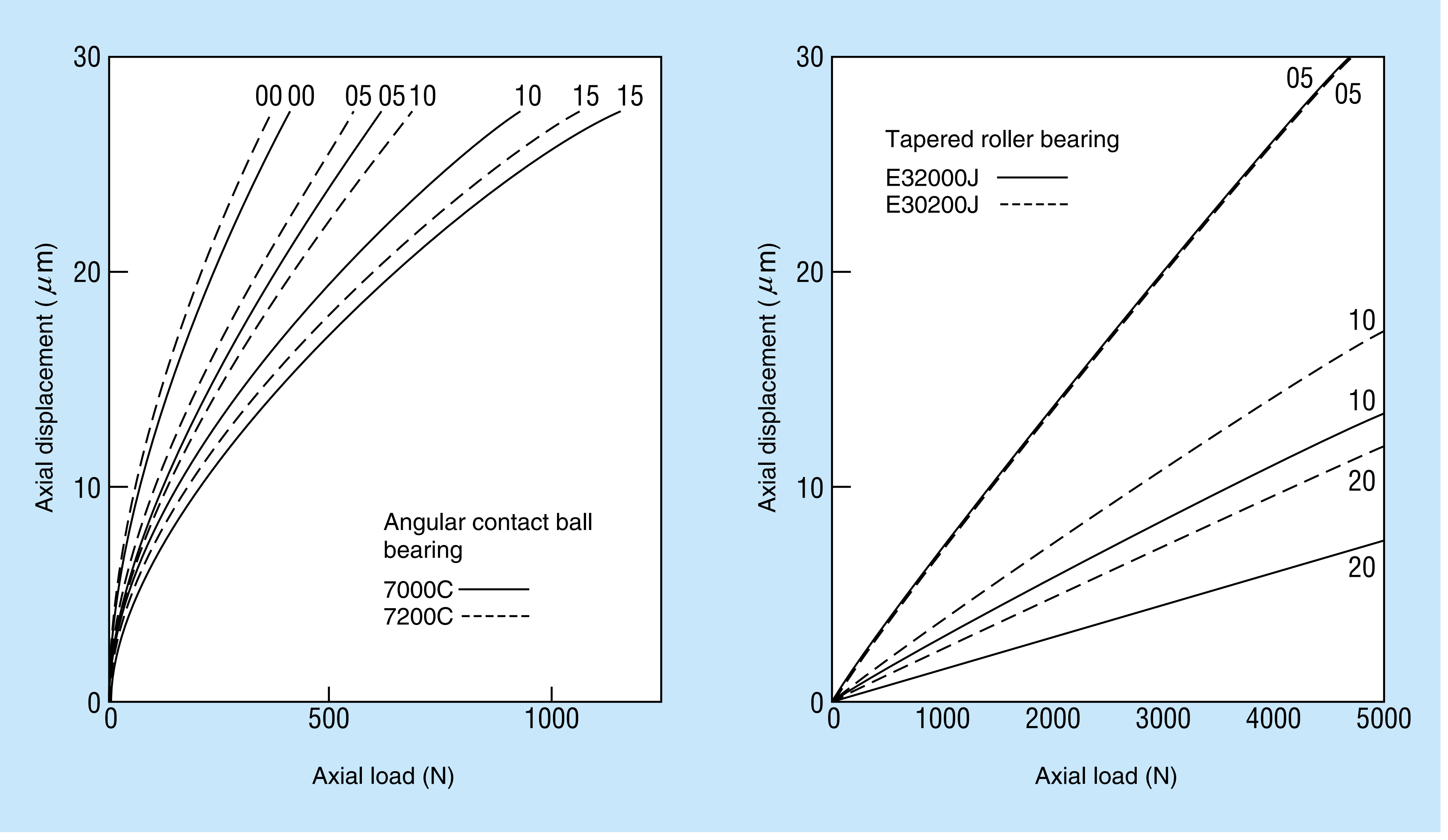
(1) Preloading method
Preloading can be accomplished using one or more of the following methods:
a) Use of springs (disc and coil springs) "Constant-pressure" preloading.
b) Use of clamping nut "Fixed-position" preloading.
c) Use of spacer (spacer and shim) "Fixed-position" preloading.

(2) Measurement of preloading amount
a) Measuring method using axial load.
- If preloading is done using springs, the preloading amount is determined by the amount of spring displacement.
- If preloading is done using a clamping nut, the preload amount is determined by the relationship of the fastening torque of the nut and clamping force.

b) Measuring method using the bearing axial displacement (Fig. 8.4).
Preload amount is determined by relationship of axial load on the bearing and resulting axial displacement.
c) Measuring method using start-up friction torque of the bearing. Relationship between axial load and friction torque should be known for this method.

Fig 8.4 Axial Load and Axial Displacement

Fig 8.4 Axial Load and Axial Displacement



8.2.3 Effect of Preloading

To illustrate the effects of preloading on a duplex Tapered roller bearing set, apply the formula from Table 8.16 to calculate a set of curves for bearing A and bearing B. The example bearing set (see Fig. 8.5) is preloaded (fixed-position), and external load, Tw, is applied.

Load distribution to the two units of bearing in terms of the axial displacement will be calculated using the graphical solution procedures described as follows:

(1) Draw T-δa curve of bearing A.
(2) Take preload Tp on axis T, determine intersection P with the curve of bearing A, and draw T-δa curve of bearing B through point P.
(3) Connect the two curves with a length equivalent to the value of external load Tw.
(4) Load Ta and Tb equivalent to this point will become the load of the individual bearings under external load Tw.
(5) Disposition of bearing is obtained by the disposition δw of bearing B.

The disposition of bearing B will be obtained by subtracting disposition to Tp from the counterpart to Tb. The reason for this is that if the bearings are preloaded, the disposition of both bearings becomes constant within a range where preload is not offset to zero by an external load (O - O' in Fig. 8.5 is constant). In other words, bearing A becomes loosened by the amount displaced by the external load on bearing B. If the external load increases and preload is eliminated, load Tb on bearing B will be equal to the external load Tw and the load on bearing A becomes zero. Magnitude of the external load causing loss of preload is represented by Tpo in Fig. 8.5.

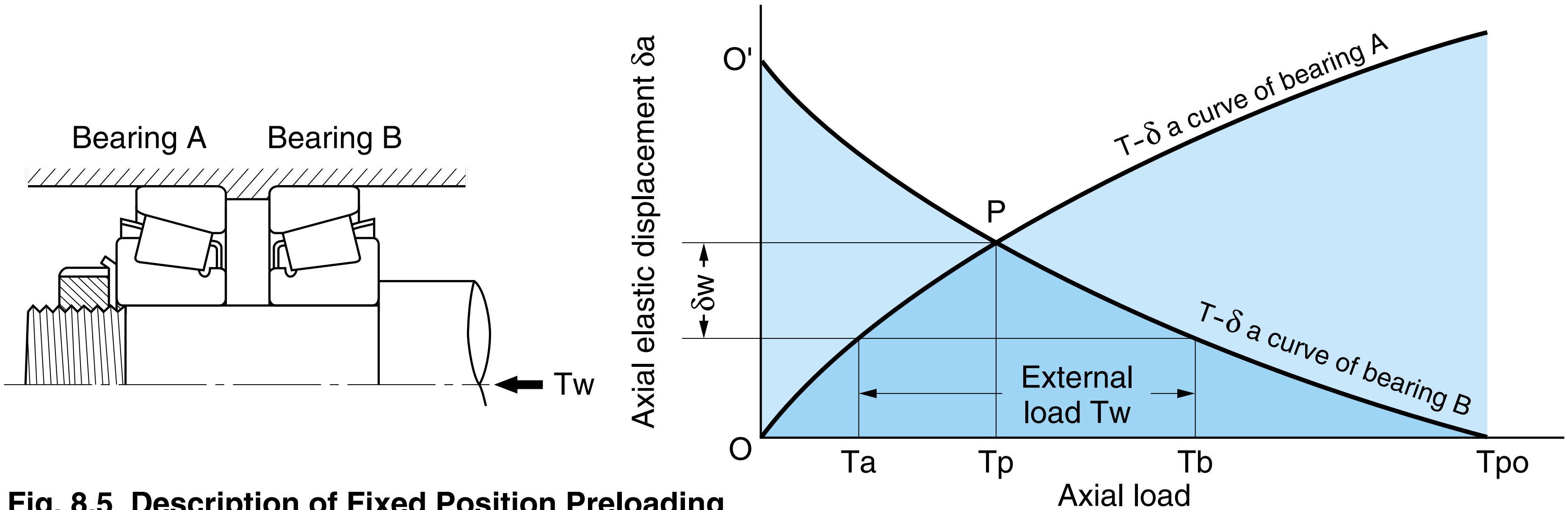


Fig. 8.5 Description of Fixed Position Preloading

8.2.4 Duplex Bearing Preload, Clearance

The preload of duplex bearings can be defined as the clearance, 2A as shown in Fig. 8.6.

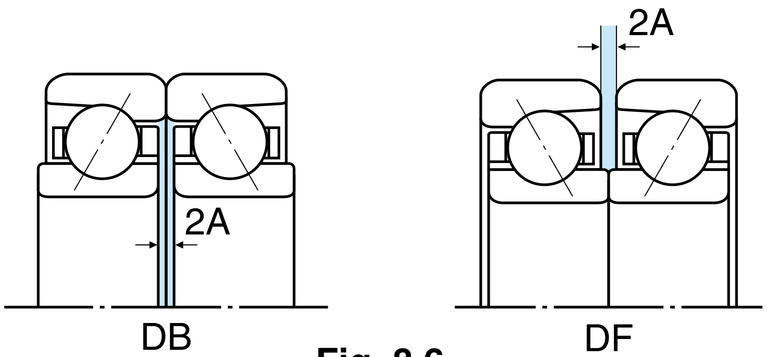


Fig. 8.6

If preloading is an application necessity, it is very important that a very thorough application analysis is made, since, if an excessive amount of preload is applied, there can be abnormal heating, increase in rotating torque and / or a sharp drop in bearing life. Table 8.17 shows standard preload and Table 8.18 outlines target amount of fits for precision (tolerance class 5 or 4), Angular Contact ball bearings.

Table 8.17 Standard Preload Amounts for Precision (tolerance class 5 or 4) Angular Contact Ball Bearings

Table 8.18 Target Interference Values for Precision (tolerance class 5 or 4) Angular Contact Ball Bearings

Table 8.17 Standard Preload Amounts for Precision (tolerance class 5 or 4) Angular Contact Ball Bearings

Unit: N

Preload (N) / Bore diameter number	7000C (DB , DF)				7200C (DB , DF)				7300C (DB , DF)			
	E	L	M	H	E	L	M	H	E	L	M	H
00	20	50	100	145	30	70	145	195	50	100	195	295
01	20	50	100	145	30	70	145	195	50	100	195	295
02	20	50	100	145	30	70	145	195	50	100	195	295
03	20	50	100	145	30	70	145	195	50	100	195	295
04	50	100	195	295	70	145	295	490	100	195	390	590
05	50	100	195	295	70	145	295	490	100	195	390	590
06	50	100	195	390	70	145	295	590	100	195	390	685
07	70	145	295	390	100	195	490	590	145	295	590	685
08	70	145	295	590	100	195	490	785	145	295	590	980
09	70	145	295	590	100	195	490	785	145	295	590	980
10	70	145	295	590	100	195	490	785	145	295	590	980
11	100	195	390	785	145	295	590	980	195	390	785	1470
12	100	195	390	785	145	295	590	980	195	390	785	1470
13	100	195	390	785	145	295	590	980	195	390	785	1470
14	145	295	590	1170	195	390	785	1470	295	590	980	1960
15	145	295	590	1170	195	390	785	1470	295	590	980	1960
16	145	295	590	1170	195	390	785	1470	295	590	980	1960
17	195	390	785	1470	295	490	980	1960	390	785	1470	2940
18	195	390	785	1470	295	490	980	1960	390	785	1470	2940
19	195	390	785	1470	295	490	980	1960	390	785	1470	2940
20	195	390	785	1470	295	490	980	1960	390	785	1470	2940

Table 8.18 Target Interference Values for Precision (tolerance class 5 or 4) Angular Contact Ball Bearings

Unit: μm

Bearing bore diameter Nominal d (mm)		Shaft to inner ring	Bearing outside diameter Nominal D (mm)		Housing to outer ring
Over	Incl.	Interference	Over	Incl.	Clearance
–	18	0 ~ 2	–	18	–
18	30	0 ~ 3	18	30	2 ~ 6
30	50	0 ~ 3	30	50	2 ~ 6
50	80	0 ~ 4	50	80	3 ~ 9
80	120	0 ~ 4	80	120	3 ~ 9
120	150	–	120	150	4 ~ 12
150	180	–	150	180	4 ~ 12
180	250	–	180	250	5 ~ 15

Remarks: Regarding the fit of housing and outer ring, take the samller values of target clearance for the clamping side bearing and the larger values for the floating side.

8.2.5 Thrust Bearing Minimum Axial Loads

When rotated at relatively high speeds, the contact angle between rolling elements and raceways of a thrust bearing changes due to centrifugal force. This can cause a skidding (sliding) action between the rolling elements and the raceways. This skidding action may cause smearing and scuffing on the rolling elements and raceway surfaces.
To prevent sliding action, thrust bearings must always be loaded with a minimum axial load. The minimum axial load is derived form Formulas (8.15), (8.16) and (8.17).
Thrust bearings can sustain axial load in only one direction. When a bi-directional axial load exists, preload must be provided by either using double bearings or springs (or load washers) to maintain the minimum axial load.
For vertical shafts, the axial load due to dead weight of the shaft (etc.), will often exceed the minimum axial load. Even in such cases, reversing axial loads may occur during operation causing the initial axial load to fall below the minimum load.

(1) Thrust ball bearing (adopt larger of values below)

$$Fa\,min = K \cdot n^2 \qquad \cdots\cdots (8.15)$$

$$Fa\,min = \frac{Coa}{1000} \qquad \cdots\cdots (8.16)$$

where:

Fa min : Minimum axial load (N)
K : Minimum axial load factor
n : Rotating speed (rpm)
Coa : Basic static load rating (N)

(2) Spherical Roller Thrust Bearing

$$Fa\,min = \frac{Coa}{1000} \qquad \cdots\cdots (8.17)$$

Minimum axial factor K $(\times 10^{-6})$

(1/2)

Series / Bore No.	511	512, 522	513, 523	514, 524
00	1.03	1.55	–	–
01	1.26	1.92	–	–
02	1.56	3.36	–	–
03	1.84	4.09	–	–
04	3.42	7.33	–	–
05	7.19	13.1	20.4	43.8
06	9.36	17.2	33.1	81.4
07	11.2	32.8	58.3	128
08	20.4	49.7	97.2	221
09	24.6	57.9	138	316
10	29.3	66.8	211	440
11	44.6	133	326	656
12	64.7	160	375	956
13	72.0	179	428	1240
14	82.8	200	596	1580
15	94.3	222	808	1800
16	103	245	907	2230
17	116	359	1240	2740
18	187	528	1390	4320
20	363	850	1850	4790
22	423	1010	2740	8220
24	488	1130	4130	9980
26	648	1940	5140	16100
28	782	2150	6330	16900
30	886	2490	7140	25800
32	997	2880	9960	30000
34	1420	3940	11100	40100
36	1540	4330	15800	46330
38	2340	6290	23100	–
40	2520	6880	29700	–
44	3000	8130	–	–
48	4900	15900	–	–
52	5580	18400	–	–
56	9800	20400	–	–
60	14600	38000	–	–
64	16400	41800	–	–
68	18300	45700	–	–
72	20300	75600	–	–

Minimum axial factor K ($\times 10^{-6}$)

Bore No. \ Series	29	39
00	1.55	–
01	1.92	–
02	2.64	–
03	3.30	–
04	3.82	–
04 1/2	6.41	–
05	7.51	14.2
06	9.72	28.9
07	20.1	52.3
08	25.1	81.0
09	31.6	140
10	46.1	209
11	54.4	284
12	60.7	350
13	86.0	426

Bore No. \ Series	29	39
14	99.5	556
15	114	704
16	152	927
17	172	1210
18	187	1580
19	286	2010
20	321	2090
21	346	2390
22	361	3220
23	350	3940
24	538	4500
25	498	–
26	–	–
27	–	–
28	794	–

Bore No. \ Series	O –
3	1.34
4	3.62
5	4.65
6	6.40
7	7.76
8	9.24
9	11.6
10	16.5
11	19.0
12	23.0
13	21.0
14	31.3
15	42.1
16	46.9
17	75.0

Bore No. \ Series	O –
18	82.8
19	110
20	121
21	132
22	176
23	204
24	223
26	350
28	395
30	431
32	580
36	1100
40	1730
44	2840
48	3690

8.3 Shaft and Housing Selection

Care must be taken in the design and manufacture of shafts and housings since inaccuracies in these components will probably result in poor bearing performance.

8.3.1 Accuracy and Surface Finish; Shafts and Housings

For general service conditions, the fit surfaces for shafts and housing bores for rolling contact bearings can be made using lathes or fine boring machines.
For applications requiring high-running accuracy, or for very quiet operation, or where high loads exist, a ground finish will be necessary.
Table 8.19 indicates the shaft and housing accuracy and surface roughness for normal service condition.

Table 8. 19 Shaft and Housing Accuracy and Surface Roughness

Table 8. 19 Shaft and Housing Accuracy and Surface Roughness

Item	Shaft	Housing Bore
Roundness	≤0.5 times shaft diametral deviation	≤0.5 times housing bore diametral deviation
Cylindricity	≤0.5 times shaft diametral deviation within range of bearing width	≤0.5 times housing bore diametral deviation within range of bearing width
Shoulder Squareness	≤0.0003 (small bearing) ≤0.0004 (medium bearing) ≤0.0005 (large bearing)	
Fit Surface Rounghness	Ra<0.8 μm (small & medium bearing) Ra<1.6 μm (large bearing)	Ra<0.8 μm (small & medium bearing) Ra<3.2 μm (large bearing)

8.3.2 Shaft and Housing Design; Recommendations

- Design shafts as short as possible and of sufficient diameter to prevent bending. Design the housing and supports for appropriate rigidity.
- Use care in specifying the roundness, cylindricity, and surface finish of shaft and housing fit surfaces. See Table 8.19.
- Use care in specifying the squareness of the shaft shoulder to the shaft center and squareness of the housing shoulder to the housing. See Table 8.19.
- Make sure that the radius, ra, of the corner roundness is smaller than the bearing chamfer dimension, r, (minimum) or, r1 (minimum) to prevent the shaft or housing from interfering with proper bearing seating. See Fig. 8.7.

For radial bearings in general, determine the maximum value of radius ra of the corner roundness and the minimum value of the shoulder height according to Table 8.20.

When using a ground finish, provide and undercut as shown in Fig. 8.8. See Table 8.21 for undercut dimensions.

- When using a radius, (ra2) of corner roundness larger than the bearing chamfer dimension (for enhancing the strength of the shaft or when shoulder height must be lower than specified in the dimension tables), install a spacer between the bearing and the shaft shoulder as shown in Fig. 8.9 and Fig. 8.10.
- For ease of dismounting, make the height of the shaft shoulder smaller than the inner ring outside (or land) diameter. If a higher shoulder is required for applying heavy axial load, install an undercut in the shaft as shown in Fig. 8.11.
- Finish bearing mounting screws, or clamping nuts as right-angled to the shaft as possible and thread screws reverse to the rotating direction of the shaft.
- For split-type housings, carefully finish the matching faces of the split housing and install a relief on both sides of the bore diameter of the cap to prevent excessive force from being applied to the bearing when the housing cap is tightened.
- For light-alloy housings (having less rigidity), insert a steel bushing to provide additional rigidity. In general the interference fit is not enough to locate a bearing axially. In principle it is necessary to fix a bearing axially by same method.
- Generally, an interference fit is not adequate to axially locate a bearing. A shaft or housing backing shoulder should be used.

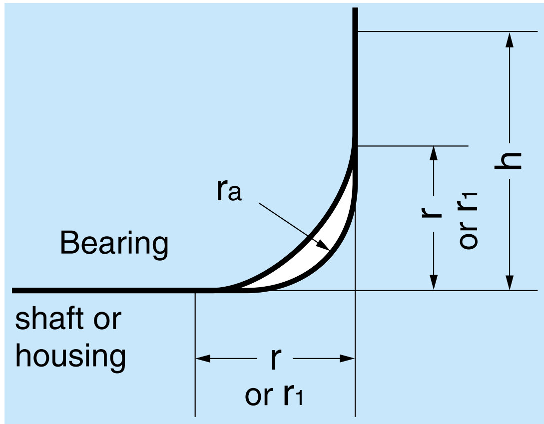


Fig. 8.7 Chamfer Dimension, Radius of Corner Roundness, and Shoulder Height

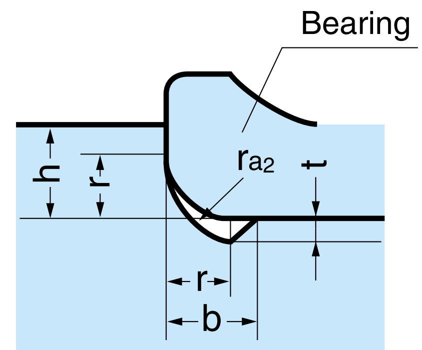


Fig. 8.8 Chamfer Dimension and Radius of Corner Roundness

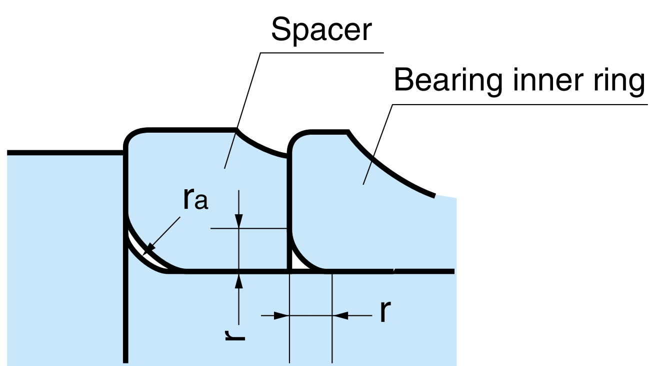


Fig. 8.9 Chamfer Dimension and Radius of Corner Roundness when Using a Spacer

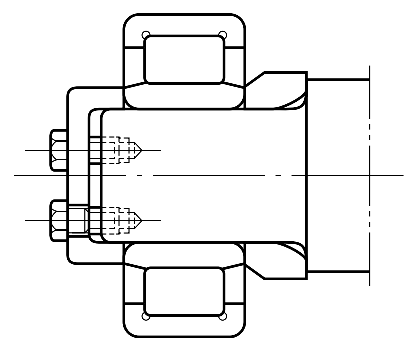


Fig. 8.10

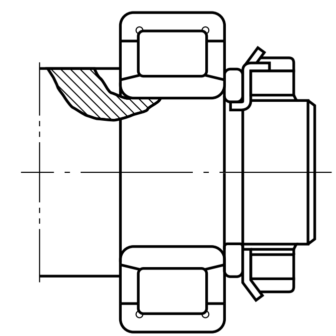


Fig. 8.11

Table 8.20 Maximum Corner Radius and Minimum Shoulder Heights

Unit: mm

Minimum tolerance chamfer dimension r (min) or r_1 (min)	Shaft or housing		
	Radius r_a (max) of corner roundness	Shoulder height h (min)	
		General cases	Special cases (1)
0.1	0.1	0.4	0.4
0.15	0.15	0.6	0.6
0.2	0.2	0.8	0.8
0.3	0.3	1.25	1
0.6	0.6	2.25	2
1	1	2.75	2.5
1.1	1	3.5	3.25
1.5	1.5	4.25	4
2	2	5	4.5
2.1	2	6	5.5
2.5	2	6	5.5
3	2.5	7	6.5
4	3	9	8
5	4	11	10
6	5	14	12
7.5	6	18	16
9.5	8	22	20
12	10	27	–
15	12	32	–
19	15	–	–

Note (1) Data in the columns for special cases should be used when axial load is extremely small.The values Table 8. 21 do not apply to tapered roller bearings, spherical roller bearingsand angular contact ball bearings.
Remarks: Symbols are based on Fig. 8.7.

Table 8.21 Undercut dimensions for Ground Shaft Finish

Unit: mm

Minimum tolerance chamfer dimension r (min) or r_1 (min)	Notch dimensions		
	t	r_{a2}	b
1	0.2	1.3	2
1.1	0.3	1.5	2.4
1.5	0.4	2	3.2
2	0.5	2.5	4
2.1	0.5	2.5	4
2.5	0.5	2.5	4
3	0.5	3	4.7
4	0.5	4	5.9
5	0.6	5	7.4
6	0.6	6	8.6
7.5	0.6	7	10

Remarks: Symbols are based on Fig. 8. 8.

8.3.3 Examples of Shaft Designs

(1) Cylindrical-bore Bearing Shaft Design

- If axial load is applied away from the shaft shoulder, the inner ring can be locked into position using; a) nuts and washers (Fig. 8.12a); b) nuts and lock washers (Fig. 8.12b); or end plates and bolts (Fig. 8.12c). When using a lock washer WITHOUT a shaft keyway or slot, it is recommended that the direction of the nut thread be made reverse to that of the shaft rotation. Note: Careful analysis of bearing load, shaft fits, and finishes, and bearing clearance may show that the shaft fit may be more than adequate to support the axial loading on the bearing.
- When not supporting axial load on the shaft-end on the side opposite the shaft shoulder, you may elect to insert a snap ring in a shaft groove to prevent the inner ring from moving axially. To remove clearance between the snap ring and bearing ring, shims or spacers can be inserted. See Fig. 8.13.
- Snap rings can be applied when using spacers between gears, or pulleys instead of using a shaft shoulder. If axial load will act on the snap ring, insert a shim or spacer between the bearing ring and the snap ring to prevent the axial load from applying bending stress to the snap ring, and to eliminate any axial clearance from between the snap ring and the ring groove. See Fig. 8.13.

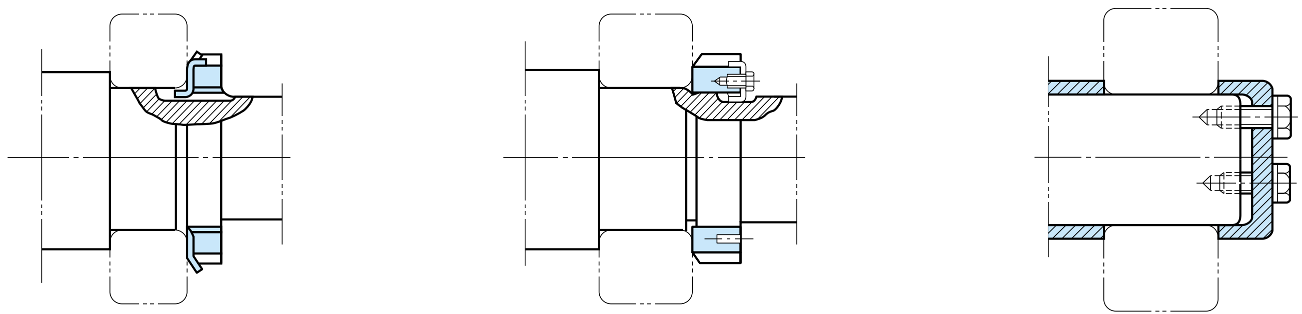


Fig. 8.12 (a) Shaft Nut and Washer (b) Shaft Nut and Lock Washer (c) End Plate and Bolts

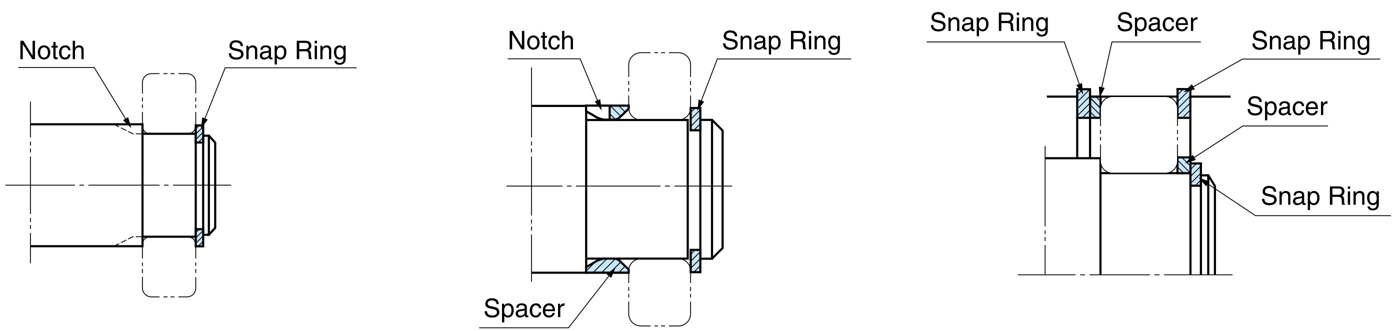


Fig. 8.13 (a) Snap Ring and Notched Shoulder (b) Snap Ring and Notched Spacer (c) Snap Ring and Spacer

(2) Tapered-bore Bearing Shaft Designs

Two methods of mounting tapered-bore bearings to a shaft are; direct mounting to a tapered shaft, or mounting to a cylindrical shaft using adapter or withdrawal sleeves.

Use of adapter or withdrawal sleeves may allow use of less expensive shaft seats (no tapering cost), permits use of a larger shaft tolerance and allows variable location of a bearing on a shaft. See Figs. 8.14 to 8.16. Since the dimensional accuracy of sleeves is not as high as that of bearings, sleeves are not appropriate for applications requiring high accuracy or high rotational speed.

- Normally, tapered-bore bearings used with adapters do not employ shaft shoulders.
  To prevent nuts from loosening, use washers for shafts of diameters 200 mm or less, and lock plates for shafts of diameters 200 mm or more.
  Nut thread direction to be made reverse to direction of rotation.
- For shafts with shoulders, mount the tapered-bore bearing with withdrawal sleeves with nuts and washer or end plates and bolts. See Fig. 8.17.
  Nut thread direction should be reverse to direction of rotation.
- When accuracy is of primary importance, use the direct mount method using tapered-bore bearings mounted directly to tapered shafts. See Fig. 8.18.

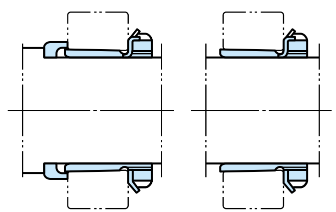


Fig. 8.14 Adapter Sleeve Mounting

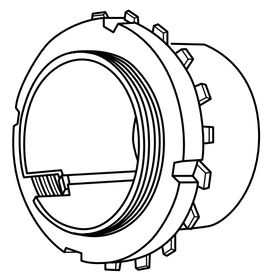


Fig. 8.15 Adapter Using Washer (Bearing Bore ≤ 200 mm)

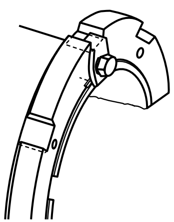


Fig. 8.16 Adapter Using Lock plate (Bearing Bore > 200 mm)

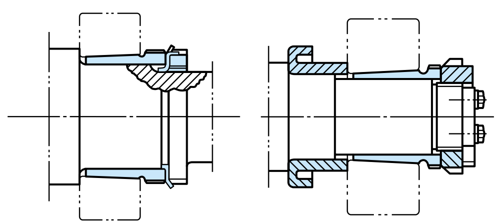


Fig. 8.17 Withdrawal Sleeve Mounting

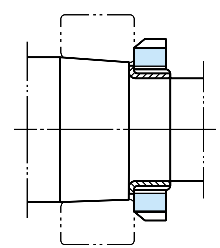


Fig. 8.18 Tapered Shaft Mounting Using Split Ring, Nut and Washer

8.3.4 Housing Designs

- When mounting two bearings to a common shaft, it is necessary to design a structure that allows linear expansion of the shaft due to temperature rise, and for mounting interval errors made during assembly. To accomplish this, mount one of the bearings to support both radial and axial loads. Fix the inner and outer rings to the shaft and housing so that neither ring will move axially. Mount the other bearing so it can move axially as the "free" side bearing capable of supporting only radial load.
  If a bearing configuration is selected for the free side bearing which will not accommodate the linear movement of the shaft created by thermal expansion, select a housing fit which will permit axial movement of the outer ring in the housing.

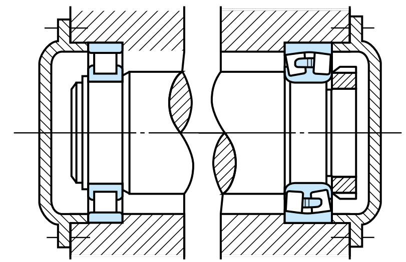


Fig. 8.19 N-Configuration Cylindrical Roller Bearing as Free Bearing

- If a Cylindrical roller bearing with an N, NU, or RNU configuration is used for the free side bearing, then shaft expansion due to temperature rise can be relieved by axial movement of the inner ring of the bearing. See Fig. 8.19. Use of Cylindrical roller bearings may also facilitate assembly if an interference fit is required for both inner and outer rings (due to the load relationship).

- If Cylindrical roller bearings with an NF or NJ configuration are used at both ends of a shaft, axial clearance must be prevented from becoming too small. Referring to Fig. 8.20, make width B (inner ring spacer) larger than the distance A between the outer rings.

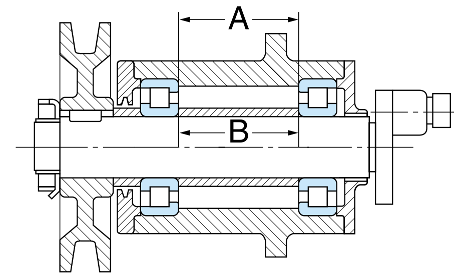


Fig. 8.20

- If the amount of shaft expansion is small (due to small temperature rise or short shaft), and precise axial location is not needed then two units of non-separable configuration bearings may be used with both units having floating axial movement. In such cases, assemble the two units with axial clearance on both ends of the assembly. See Fig. 8.21. For mounting of two Deep-groove ball bearing pillow blocks with spherical outer ring bearing surfaces, lock and bolt the first pillow block into position, then lock the second block to the shaft. Pull the second block away from the first block while tightening the mounting bolts. Where axial expansion can not be handled by the clearance within the bearings, please consult NACHI.

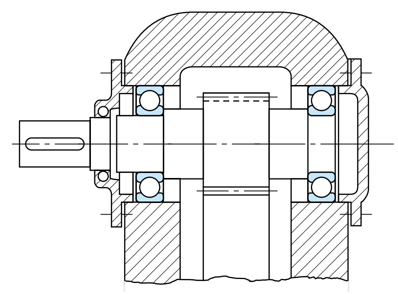


Fig. 8.21

• Pairs of single-row Angular Contact ball, or Tapered roller bearings are often used for axial positioning. When bearing spacing is large, axial expansion from temperature rise is best handled using an assembly as shown in Fig. 8.22, where the paired bearings take axial and radial loads and another bearing (in the Figure, an NU-configured Cylindrical roller bearing), permits linear shaft expansion.

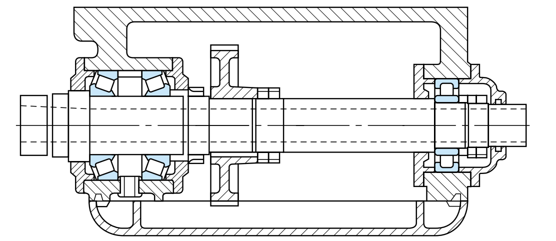


Fig. 8.22

• When using horizontally-split pillow blocks as the fixed side bearing, the outer ring is located by using one or two positioning rings. When one ring is used, place it to the side of the adapter nut as shown in Fig. 8.23. When two positioning rings are used, place one on each side of the bearing (also see Fig. 8.23). To use a horizontally-split pillow block as the floating side bearing, mount the bearing without positioning rings.

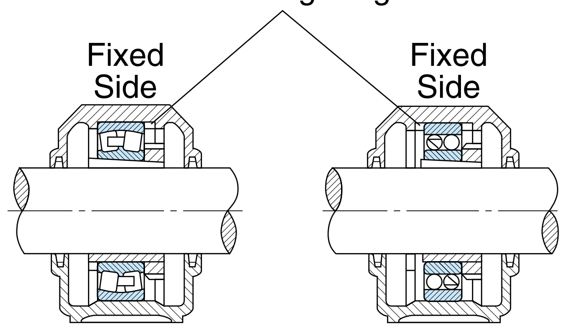


Fig. 8.23

• Determine the position of the fixed bearing by considering the machinery application and the balance of rated life of the individual bearings. For example, when a bevel gear is used (see Fig. 8.24), set the bevel gear side as the fixed side to maintain the accuracy of the gear engagement. For electric motors, the fixed side bearing is often positioned on the non-driving side where a lower amount of radial load is applied, in order to equalize the bearing equivalent load and rated life between the two bearings.

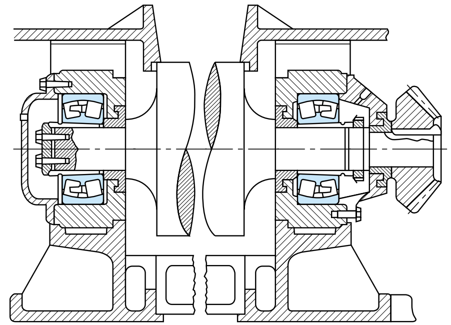


Fig. 8.24

8.4 Sealing Devices

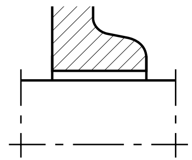
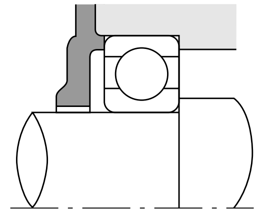
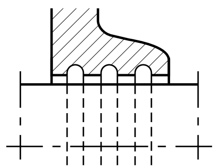
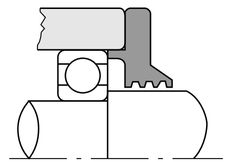
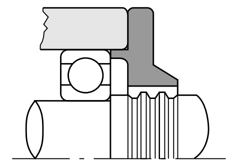
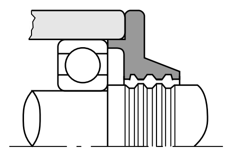
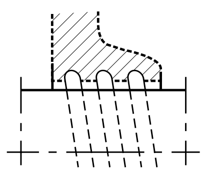
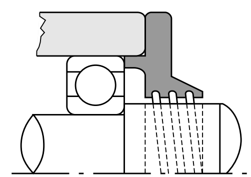
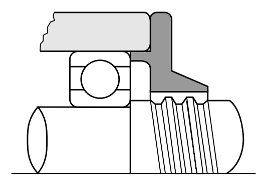
8.4.1 Sealing Device Requirements

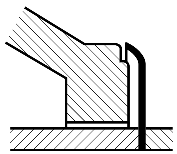
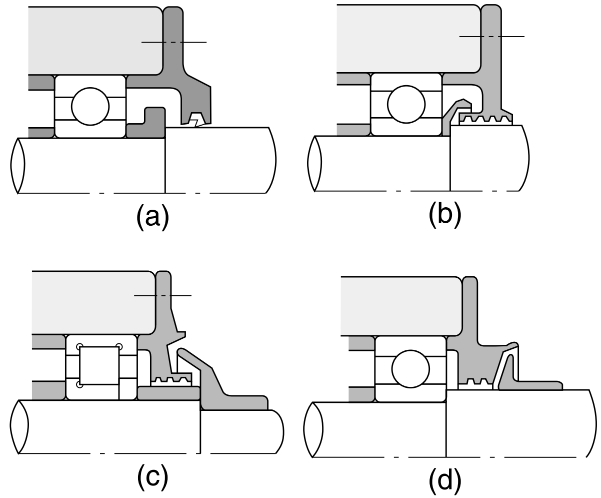
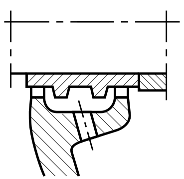
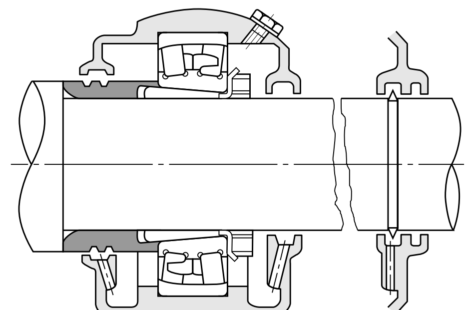
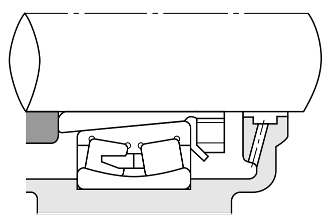
- Must effectively stop foreign material intrusion.
- Must not create excessive frictional loss or heat.
- Must be easy to mount, dismount, and maintain.
- Must be inexpensive.

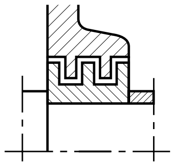
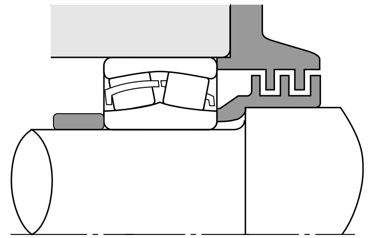
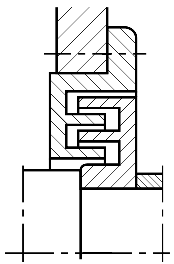
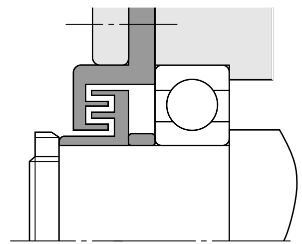
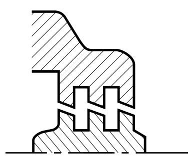
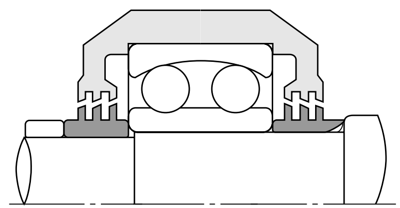
The lubrication method and sealing devices used must be compatible and appropriate for the application.

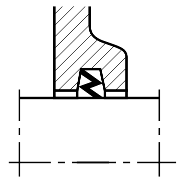
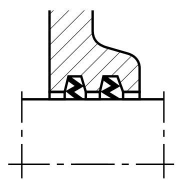
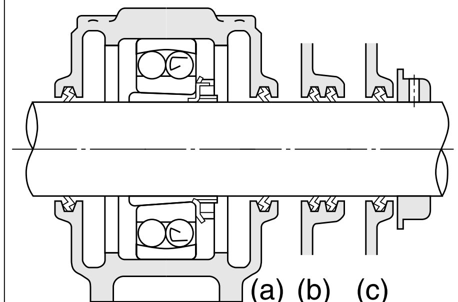
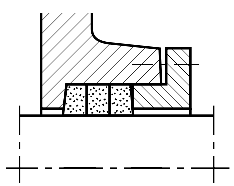
Integrally-sealed or shielded bearings may need separate, additional sealing devices if they are to be operated in an adverse atmosphere.

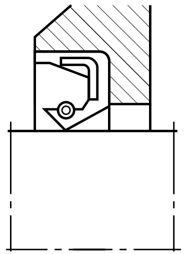
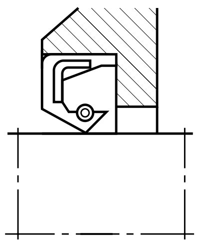
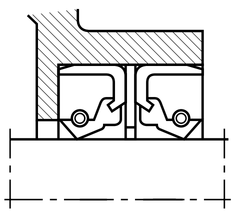
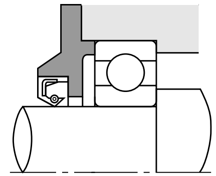
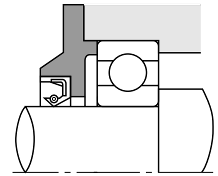
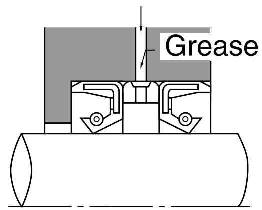
- Linear gap (simple gap type)
- Coaxial groove (oil groove type)
- Threaded groove
- Slinger type
- Slinger type (for oil lubrication)
- Radial labyrinth type
- Axial labyrinth type
- Self-aligning type labyrinth
- Seal ring type (felt, leather, rubber, plastic)
- Adjustable Seal type (includes metal packing O-ring, etc)
- Oil seal Type

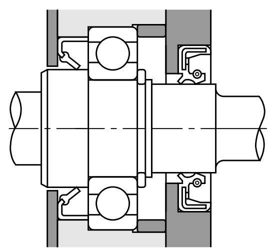
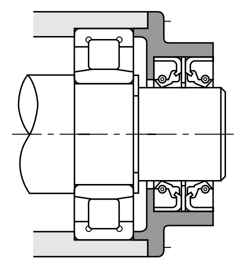
<table>
<tr><th colspan="2">Type of Sealing Device</th><th>Design Example</th><th>Design Precautions</th></tr>
<tr><td>Linear gap (simple gap type)</td><td></td><td></td><td rowspan="3">1) Clearance between shaft and bearing housing
<table><tr><th>Shaft dia. (mm)</th><th>Radial clearance (mm)</th></tr><tr><td>50 or less</td><td>0.25 ~ 0.4</td></tr><tr><td>50 Over 200 Incl.</td><td>0.5 ~ 1.5</td></tr></table>2) Groove dimensions
Width: 3 to 5 mm
Depth: 4 to 5 mm
3) Where possible, provide three grooves or more.
4) Fill grooves with grease to aid in sealing out foreign material.
5) The threaded grooves type is applicable to oil lubricated, applications where the shaft is horizontal and operates in a constant rotational direction. Thread grooves must be reverse to the rotation direction.
6) Oil grooves are used alone only where preventional oil relatively clean.
Oil grooves are for preventing oil leaks and are generally used in combination with other sealing devices.</td></tr>
<tr><td>Coaxial groove (oil groove type)</td><td></td><td>
(a)

(b)

(c)</td></tr>
<tr><td>Threaded groove</td><td></td><td> </td></tr>
</table>

Type of Sealing Device		Design Example	Design Precautions
Slinger type			1) Seal types that sling oil, prevent oil leakage and dust entry through the centrifugal force generated by a rotor attached to the shaft. 2) (a) and (b) are good for preventing oil leakage. 3) (c) and (d) are good for preventing dust and water intrusion.
Slinger type (for oil lubrication)		 A type of slinger 	1) Oil deposited in the grooves returns to the housing.

<table>
<tr><th colspan="2">Type of Sealing Device</th><th>Design Example</th><th>Design Precautions</th></tr>
<tr><td>Radial labyrinth type</td><td></td><td></td><td rowspan="3">1) Labyrinth Clearance
<table>
<tr><th rowspan="2">Shaft dia. (mm)</th><th colspan="2">Clearance (mm)</th></tr>
<tr><th>Radial</th><th>Axial</th></tr>
<tr><td>50 or less</td><td>0.25 ~ 0.4</td><td>1 ~ 2</td></tr>
<tr><td>50 Over 200 Incl.</td><td>0.5 ~ 1.5</td><td>2 ~ 5</td></tr>
</table>
2) Radial and axial labyrinth seals. The radial groove type requires a split housing.
3) These seals are very suitable for the prevention of oil leakage of high speed shafts.
4) For low speed rotation, apply grease to the grooves for better sealing.
5) If angular misalignment exists, use self-aligning type labyrinth.</td></tr>
<tr><td>Axial labyrinth type</td><td></td><td></td></tr>
<tr><td>Self-aligning type labyrinth</td><td></td><td></td></tr>
</table>

<table>
<tr><th colspan="2">Type of Sealing Device</th><th>Design Example</th><th>Design Precautions</th></tr>
<tr>
<td>Seal ring type
(felt, leather,
rubber, plastic)</td>
<td>
</td>
<td>
</td>
<td rowspan="2">
1) Sealing Material Temperature Range
<table>
<tr><th>Sealing material</th><th>Operating temperature range °C</th></tr>
<tr><td>Nitrile</td><td>–25 ~ 100</td></tr>
<tr><td>Acrylic</td><td>–15 ~ 130</td></tr>
<tr><td>Silicon</td><td>–70 ~ 200</td></tr>
<tr><td>Flourine</td><td>–30 ~ 200</td></tr>
<tr><td>Ethylene tetrafluoride</td><td>–50 ~ 220</td></tr>
<tr><td>Felt</td><td>–40 ~ 120</td></tr>
</table>
2) Sealing Material Speed Limits (m/s)
<table>
<tr><th rowspan="2">Seal Material</th><th colspan="3">Shaft diameter (mm)</th></tr>
<tr><th>to 20</th><th>20 to 40</th><th>40 and up</th></tr>
<tr><td>Nitrile</td><td>4 ~ 8</td><td>8 ~ 12</td><td>12 ~ 16</td></tr>
<tr><td>Acrylic</td><td>4 ~ 12</td><td>12 ~ 18</td><td>18 ~ 25</td></tr>
<tr><td>Silicon</td><td>4 ~ 18</td><td>18 ~ 25</td><td>25 ~ 32</td></tr>
<tr><td>Flourine</td><td>4 ~ 18</td><td>18 ~ 25</td><td>25 ~ 32</td></tr>
<tr><td>Ethylene tetrafluoride</td><td colspan="3">15</td></tr>
<tr><td>Felt</td><td colspan="3">3.5 ~ 4.5</td></tr>
</table>
These values apply when shafts have good surface finish, roundness, and run-out.
3) Lubricate the sliding surfaces of seal and shaft.
4) These seal types are mainly applicable to grease lubricated bearings.
5) Install one to three pieces of felt ring.
6) For high speed applications, use hard seal material. Coat with mineral oil before mounting and insert tightly.
7) Felt will harden and lose elasticity under high temperature or speed.
8) Felt rings are good for relatively. clean, dust-free conditions. For application in excessively dusty conditions, synthetic rubber rings or additional seal made of synthetic rubber should be used.
</td>
</tr>
<tr>
<td>Adjustable Seal type
(includes metal packing
O-ring, etc)</td>
<td></td>
<td></td>
</tr>
</table>

<table>
<tr><th colspan="2">Type of Sealing Device</th><th>Design Example</th><th>Design Precautions</th></tr>
<tr>
<td>Oil seal Type</td>
<td>

</td>
<td>
(a)

(b)



Example 1

Example 2</td>
<td>1) Speed and shaft Surface Roughness
<table>
<tr><th>Speed (m/s)</th><th>Surface Finish</th><th>Finish method</th></tr>
<tr><td>to 5</td><td>Ra < 0.8μm</td><td>Paper finish after grinding</td></tr>
<tr><td>5 ~ 10</td><td>Ra < 0.4μm</td><td>Paper finish after grinding</td></tr>
<tr><td>10-up</td><td>Ra < 0.2μm</td><td>Lapping, or superfinishing, or electro polishing after quenching and grinding</td></tr>
</table>
2) Seal contact to shaft section should be minimum hardness of HRC40. HRC55 or even higher is desirable.
3) Dimensional tolerance of seal contact to shaft section should be h9 and the counterpart for seal housing should be H8 or H7.
4) Since there are various shapes and materials for seal, select those that meet purposes.
5) Control the shaft eccentricity to under 0.02 to 0.05 mm where possible.
6) Coat the contact surfaces of seal and shaft with lubricant at initial installation.</td>
</tr>
</table>

8.5 Lubrication

8.5.1 Functions of Lubrication

The main purpose of lubricants in rolling contact bearings is to reduce friction and wear of each element. Lubricants perform this function by separating rolling and sliding surfaces with a very thin film of oil. Bearing performance and service life is largely dependent on the suitability of the lubricating system and lubricant to the application. Functions of lubrication in rolling contact bearings are:

① Lubrication of friction surfaces:
Reduction in;
1) Rolling friction between the rolling elements and raceways.
2) Sliding friction between roller end and guide faces of roller bearings.
3) Sliding friction between the rolling elements and retainer.
4) Sliding friction between the retainer and raceway guide surface.

② Removal of the heat from system produced by friction and external sources. An example of the heat removal function would be use of a circulating oil lubrication system for a high-speed application.

③ Dust-proofing and rust prevention:
1) Prevention of foreign material from entering the bearing.
2) Protection of bearing components from corrosion.

④ Relief of stress concentration:
1) Uniform distribution of stress to the rolling contact surface.
2) Relief of impact loads.

8.5.2 Lubrication Cautions

① Adequate lubricant film separation should be maintained between friction surfaces.
② Since the oil film required on contact surfaces is thermally feeble, adequate oil viscosity must be maintained.
③ Since lubricants tend to deteriorate with increase in temperature, bearing applications should be designed to keep the operating temperature as low as possible.
④ The lubricating system (method) must be suitable for the application and the lubricant must have appropriate properties.
⑤ The lubricant must be kept free from contamination.

8.5.3 Lubricating Methods

(1) Oil Lubrication

(1.1) Oil Bath Lubrication

- Oil bath lubrication is generally used for low-to-medium-speed operation.
- Excessive oil causes churning which can cause excessive temperature rise. Insufficient oil will probably lead to early bearing failure.
- Oil level gauges are recommended to check (and maintain) the proper oil level.
- Separation ribs may be installed at the bottom of the housing to reduce churning and or to dissipate heat. See Fig. 8.25.
- Static oil level should be at slightly below the center of the lowest rolling element of a bearing applied to a horizontal shaft. For vertical shafts, static oil level should cover 50% to 80% of the rolling element.
- When two or more bearings are used on a vertical shaft in the same housing, the lower bearing may create excessive temperature rise if an oil bath system is used (unless operated at very low speed). If excessive heat occurs, use a drip, splash, or circulating oil system.

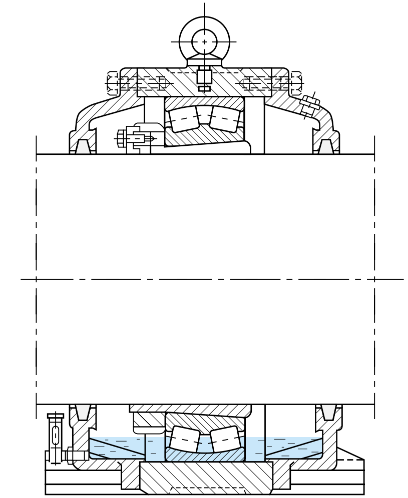


Fig. 8.25

(1.2) Splash Lubrication

- In splash lubrication, oil is splashed on the bearing by a rotating element (an impeller or "slinger") mounted on the shaft. The bearing is not immersed in the oil.
- In a gear box, the gears and bearings are often lubricated from a common oil reservoir with the gears serving as a slinger. Since oil viscosity for the gears may differ from that required for the bearings and the oil may contain particles worn from the gears, a separate lubrication system or method may provide improved bearing life. Sealed or shielded bearings and "magnetic" plugs are often used in conjunction with gear drives.
- A bearing on a vertical shaft can be provided with a conical rotary element under the bearing so that the oil rises on the conical surface and is atomized before entering the bearing. See Fig. 8.26.

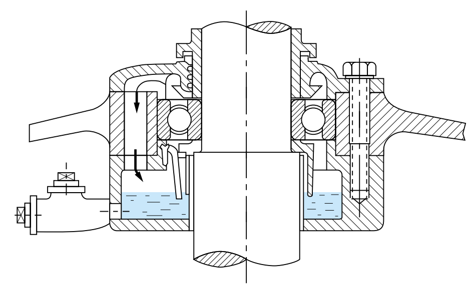


Fig. 8.26

(1.3) Drip Lubrication

- Drip lubrication is used for bearings operated at relatively high speeds under low-to-medium loads.
- Drip lubrication is generally used for the radial bearing on a vertical or inclined shaft and oil is fed directly to the bearing.
- The lubricating oil is contained in a lubricator, and is fed to the bearing through a wick which also serves as a filter. A sight window is provided to allow checking the oil level.
  Fig. 8.27 shows a drip lubricating system provided with a lubricator on top of the housing. Oil is dripped onto the shaft nut in the bearing box, and is atomized before entering the bearing.
  Fig. 8.28 shows an oil metering system designed to feed several oil drops per minute to the bearing.

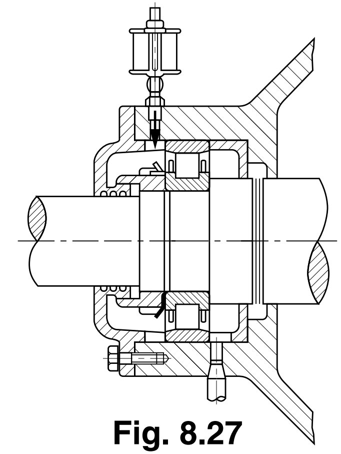


Fig. 8.27

(1.4) Circulating Oil Lubrication

- Circulating oil lubrication is used for two purposes:
  1) To cool the bearing.
  2) To automatically feed oil to a specific area from a central system.
- A circulating oil system consists of an oil pump, cooling device, filter and delivery piping. Circulating oil systems utilize the pumping action of the bearings and augment the cooling effects of slingers.
- Circulating oil lubrication includes: drip, forced, and spray-mist lubrication.
- In the circulating oil lubricating system, the bearing is provided with an oil inlet located on one side of the bearing, and an oil outlet on the other side of the bearing.
- The oil outlet should be larger than the oil inlet so that excess oil does not remain in the bearing housing.
  Fig. 8.29 shows a circulating system with an oil passage in the area of the housing which carries no load. This system is for steam-heated calender rolls in a paper mill. Cooled oil is circulated through the inner wall of the housing and passes through both bearings.

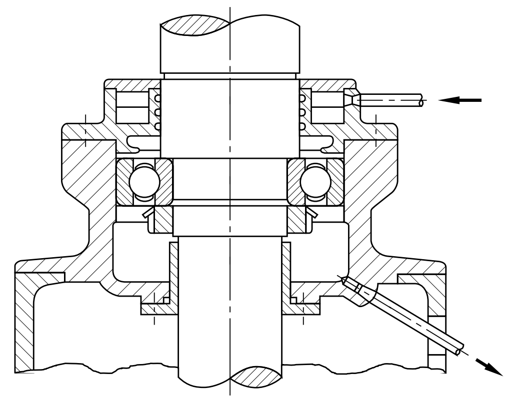


Fig. 8.28

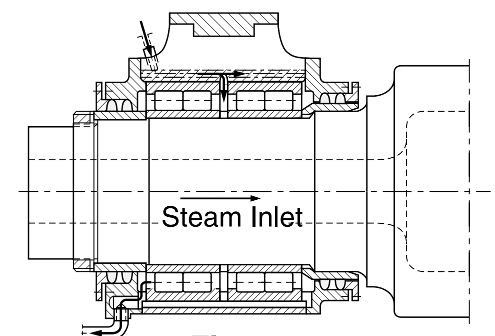


Fig. 8.29

(1.5) Forced Lubrication

Forced lubrication is used to feed oil under pressure to overcome internal housing pressure in high-speed operation.

- The oil outlet should have a cross section twice that of the oil inlet.
- A "jet" lube system is sometimes used in high-speed applications to target oil directly to the rolling and sliding components of the bearing. See Fig. 8.30. Excessive oil should be discharged with a pump.

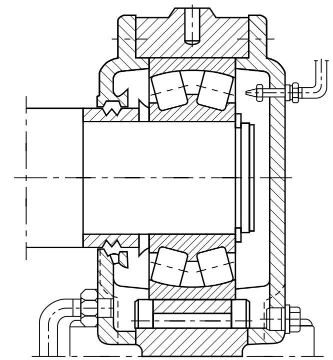


Fig. 8.30

(1.6) Disk Lubrication

Disk lubrication utilizes a disk on the shaft which rotates at high speed. The disk is partially submerged in oil, and splashes oil to an upper oil sump, which in turn delivers the oil to the bearing by gravity. Disk lubrication is used on the bearings of superchargers and blowers. See Fig. 8.31.

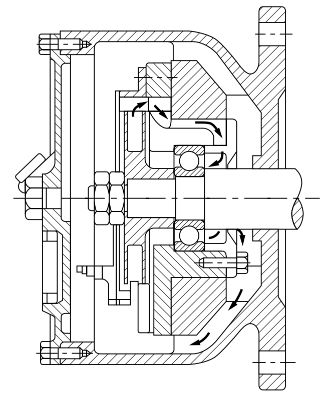


Fig. 8.31

(1.7) Spray Mist Lubrication

- Fig. 8.32 shows an example of spray lubrication, which uses a turbo-compressor impeller to force oil into the bearing.
- Fig. 8.33 shows an example of oil mist applied to an oil atomizer (0.5 to 5.0 cc/h).

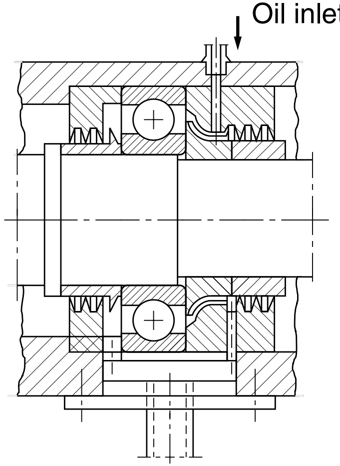


Fig. 8.32

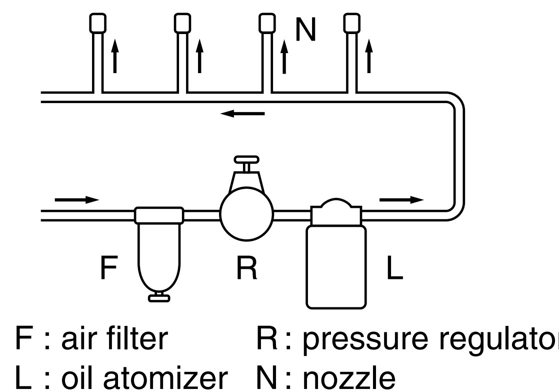


Fig. 8.33

(1.8) Oil/Air Lubrication

Using the oil/air lubrication, a very small amount of oil is mixed with a certain amount of compressed air with a constant-quantity piston and mixing valve. This mixture is supplied to rolling part of bearings.

Because oil/air lubrication can remove heat generation from bearings, this method is suited for high speed application such as machine tool.

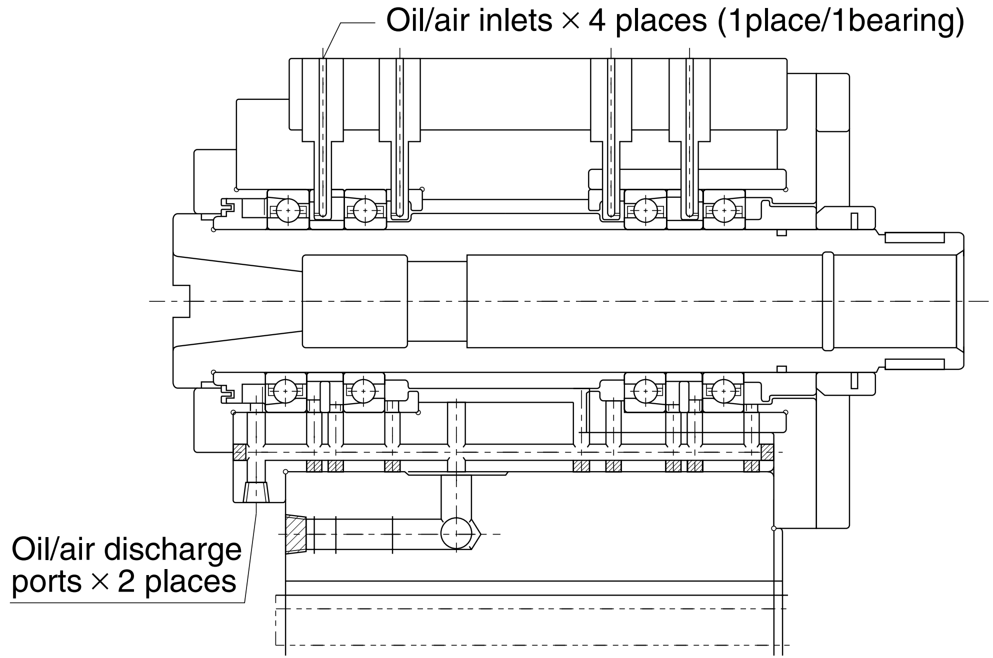


Fig. 8.34

(2) Grease Lubrication

In using grease lubrication, the following items should be considered:

- Select grease having correct properties.
- Grease must be delivered in the right amount to the correct bearing area.
- Determine method of relubrication. Different greases should not be mixed because it can cause a poor lubrication performance.
- Consider centralized lubrication for large-size machinery such as rolling mill equipment. See Fig. 8.35.1.

Fig. 8.35.2 shows a design utilizing a grease supply plate. Symbols S, R, and Z refer to the nozzle, oil groove, and supply plate, respectively.

Fig. 8.35.1

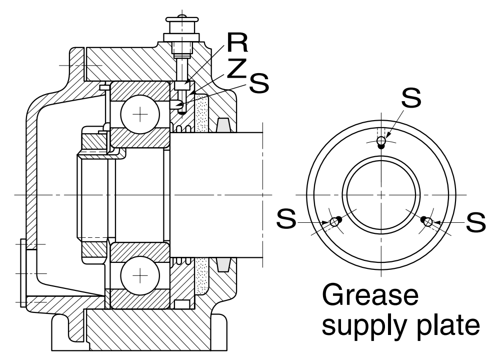


Fig. 8.35.2

8.5.4 Lubricants

Rolling contact bearings use two forms of lubricants; lubricating oil and grease. In some special applications, solid lubricant such as molybdenum-disulfide, graphite, or PTFE are used. The lubricant should have the following properties:

- Low impurity and moisture content
- Temperature stability
- Non-corrosiveness
- Load pressure resistance
- Anti-wear action
- Anti-friction action
- High mechanical stability

See Table 8.22 for a guide to selection of lubricating oil and grease.

Table 8.22 Guide to selecting Oil and Grease

Operating Condition	Grease	Oil
Temperature	Available for range of -30° to +150°	Applicable for high temperatures (with circulating cooling)
Speed	Low to medium speeds	Applicable for high speed operation (depending on lube method)
Load	Low to medium loads	Suitable for high loads
Housing Design	Simple	Complicated by sealing requirements
Maintenance	Easy	Easy to difficult
Centralized Lubrication	Possible	Possible
Dust Filtration	Dependent on seal devices.	Possible (Circulating lubrication provides a filter to trap dust)
Rolling Resistance	Relatively high	Small (Correct oil quantity must be maintained)

A wide variety of lubricating oils and greases are commercially available for rolling contact bearings. It is important to select oils or greases with base oils having a viscosity which is appropriate for the operating condition.
Table 8.23.1 and 8.23.2 give generally recommended viscosities for bearings under normal operating conditions.

Table 8.23.1 Bearing types and Proper Viscosity of Lubricating Oils

Bearing Type	Viscosity at Operating Temperature
Deep Groove Ball Cylindrical Roller	Over 13 mm^2/s
Tapered Roller Spherical Roller	Over 20 mm^2/s
Spherical Roller thrust	Over 32 mm^2/s

Remarks: 1mm^2/s = 1cSt (centistokes)

Table 8.23.2 General Oil Selection Guide

(1) Lubricating Oil
Oils with a viscosity too low for the application may allow a partial loss of raceway to rolling element separation, leading to early bearing failure. Oils with too high a viscosity will cause an increase in torque, resulting in power loss and abnormal temperature rise. In general, as the load increases, increase the oil viscosity. As speed of rotation increases, decrease the oil viscosity.
For Extra-small or Miniature ball bearings, low-viscosity lubricating oil will often be selected for low-torque requirements.
Table 8.23.2, and Fig. 8.36 on following pages can be used to aid in selection of appropriate oil viscosity.

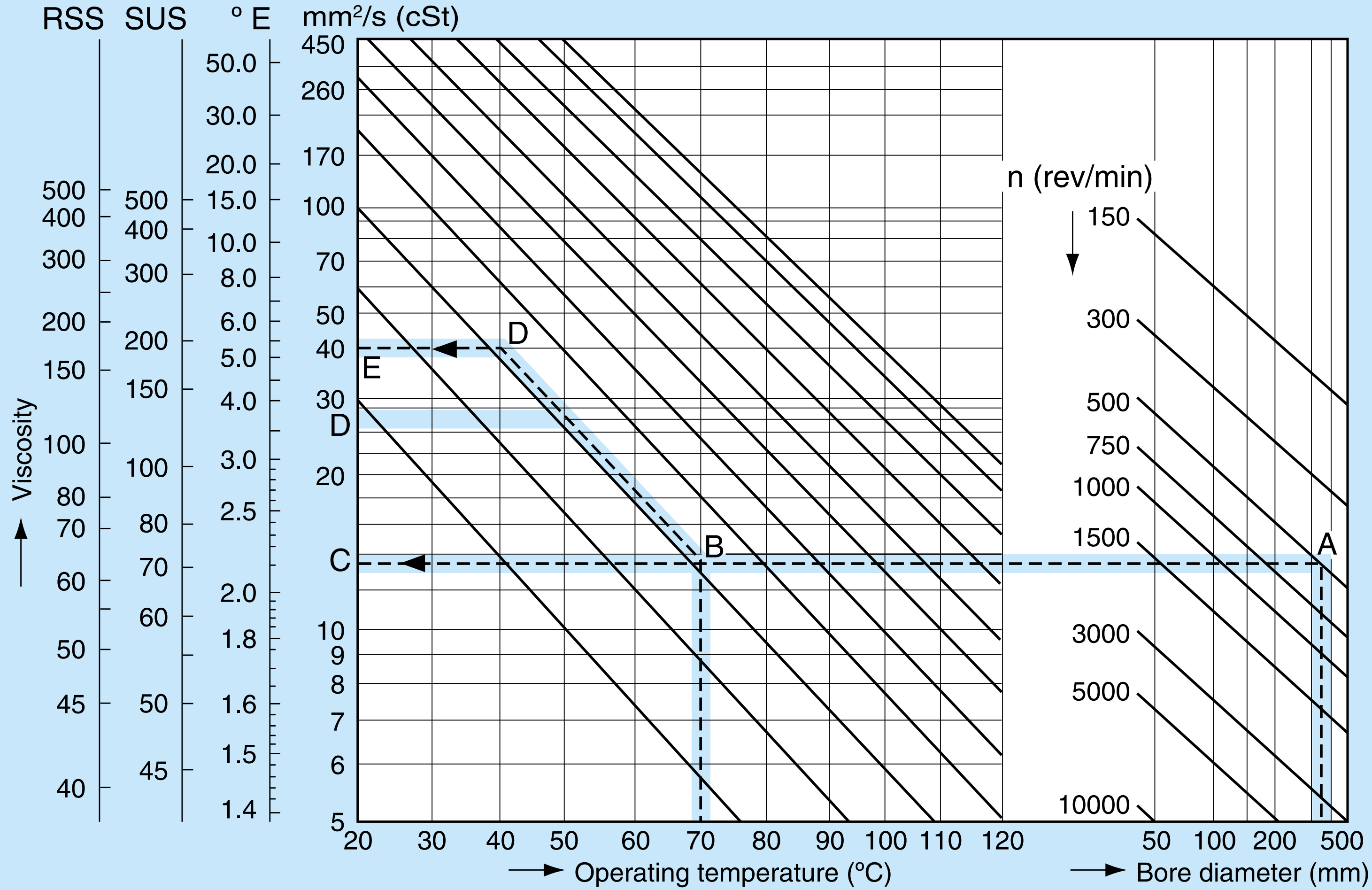
Fig 8.36 Viscosity-Temperture Line Diagram

Table 8.23.2 General Oil Selection Guide

Bearing Operating Temperature (°C)	dn value	ISO Viscosity grade (VG) of lubricating oil		Suitable bearing type(s)
		Normal load	Heavy or impact load	
–30 ~ 0	Up to speed limit	22 32	46	All
0 ~ 60	Up to 15000	46 68	100	All
	15000 ~ 80000	32 46	68	All
	80000 ~ 150000	22 32	32	Except thrust ball bearing
	150000 ~ 500000	10	22 32	Single row deep groove ball and cylindrical roller bearing
60 ~ 100	Up to 15000	150	220	All
	15000 ~ 80000	100	150	All
	80000 ~ 150000	68	100 150	Except thrust ball bearing
	150000 ~ 500000	32 46	68	Single row deep groove ball and cylindrical roller bearing
100 ~ 150	Up to speed limit	320		All
0 ~ 60	Up to speed limit	46 68		Spherical roller bearings
60 ~ 100	Up to speed limit	150		

Remarks:
1. This Table shows the guide for selecting oil, based on JIS K 2001 classification of Industrial Lubricating Oil Viscosity.
2. Generally as load increases or speed decreases, viscosity is increased.
3. This Table is applicable for oil bath lubrication and circulating oil lubrication.
4. For information on operating conditions beyond those of This Table, contact NACHI.

Fig 8.36
Viscosity-Temperture Line Diagram



Example:

Bearing Type : Cylindrical roller bearing
Bearing Bore : 340mm
Rotating Speed : 500rpm
Operating temp : 70 °C

Fig. 8.36 can be used to select both the correct minimum viscosity at operating temperature and to establish the required oil viscosity rating (at 40°C) which will meet the specified minimum viscosity .
Find the intersection of 340 and 500 (see point A) and an horizontal line from point A to point C (intersection of Y-axis)
To find the minimum viscosity required AT THE OPERATING temperature: read the minimum viscosity required ($13mm^2/s$) at point C.

To establish the required oil viscosity rating ;
1) At the intersection of A - C and a vertical line from 70 (point B), draw a line toward the 40° -axis line parallel to the closest viscosity-temperature line.
2) Draw a horizontal line from the point (point D) of intersection of the above line with the 40° -axis to the Y-axis (see point E).
3) Read the viscosity $40mm^2/s$ at point E. As a result, ISO viscosity grade VG46 should be selected.

(2) Lubricating Grease

Lubricating grease is composed of a base oil, a thickener, and additives.

• Base Oil

Base oil refers to the liquid lubricant carried by a thickener. Mineral oils are widely used as the base oils for grease. Synthetic oils such as diester or silicone oil are also used for improving the heat resistance and stability of grease. In general, grease with a low-viscosity base oil is suitable for low temperatures and or low loads, while grease with a high viscosity base oil is suitable for high temperatures and or high loads.

Since lubricating performance is dependent on the thickener, additives, and viscosity, these components must be carefully selected to meet operating conditions.

• Thickener

The thickener has a sponge-like structure composed of a loose combination of fine fibers or particles. Thickeners are roughly divided into metal soap, and non-soap types as shown below.

Sodium (Na) soap grease may react with water to form an emulsion, and should not be used for bearings operating in a high-moisture atmosphere.

THICKENER
- Metal soap: Ca, Li, Na
- Non-soap
  - Heat resistant organic base: Polyurea & Fluorides
  - Inorganic base: Silica gel and organic bentonite

• Additives

An additive is an agent that provides extreme pressure and rust resistance, anti-oxidation performance, and other properties to grease.

Various additives are added to grease to provide specific properties to the grease. Additives such as anti-oxidants, extreme pressure enhancers, and rust preventatives are often added to lubricating greases.

Anti-oxidant additives protect grease from oxidation and deterioration under thermal influence over a long period. Extreme pressure additives improve load resistance and impact resistance. Rust preventive additives protect the bearing and other surrounding components against rusting.

• Penetration

Penetration is a measure which indicates the solidity of grease. A measurment device has a cone with a specified weight and shape. The cone is penetrated into the sample grease for a specified time. Penetration is the depth to which the cone penetrates (in units of 1/10 mm).

• Dropping Point

Dropping point is the temperature at which a grease sample drops through a specified hole size after being heated and fluidized.

Table 8. 24 Grease Number and Penetration

<table>
<tr><th>NLGI No.</th><th>ASTM Worked penetration</th><td rowspan="8">Grease is numbered differently by the grease manufacturers. Numbers 250 and 300 of cup and fiber grease generally use penetration (at 25ºC), while most versatile greases employ NLGI penetration numbers such as 0, 1, and 2.</td></tr>
<tr><td>0</td><td>355 ~ 385</td></tr>
<tr><td>1</td><td>310 ~ 340</td></tr>
<tr><td>2</td><td>265 ~ 295</td></tr>
<tr><td>3</td><td>220 ~ 250</td></tr>
<tr><td>4</td><td>175 ~ 205</td></tr>
<tr><td>5</td><td>130 ~ 160</td></tr>
<tr><td>6</td><td>85 ~ 115</td></tr>
</table>

(3) Lubrication Amount

① **Oil**

When oil bath lubrication is being used and a bearing is mounted with its axis horizontal, oil should be added until the static oil level is at the center of the lowest bearing rolling element. For vertical shafts, add oil to cover 50% to 80% of the rolling element.

② **Grease**

The rolling bearing and bearing housing should be filled until the grease occupies about 33 to 50 % of the respective volumes. Temperatures will tend to rise as speed increases (due to churning). Higher-speed operation will be more sensitive to excess grease fill, so it follows that at higher dmn values, the grease-fill quantity must be reduced.

a) Amount of Initial Grease Fill

The amount of initial grease-fill required is calculated from the following equations:

Ball bearing:

$$Q = \frac{d^{2.5}}{900} \qquad \cdots\cdots (8.18)$$

Roller bearing:

$$Q = \frac{d^{2.5}}{350} \qquad \cdots\cdots (8.19)$$

where:
- Q =Amount of filling grease (g) (specific gravity of grease=0.9)
- d =Bore diameter of bearing (mm)

b) Relubrication Amount Added at Service

$$Q = 0.005 \times D \cdot B \qquad \cdots\cdots (8.20)$$

where:
- Q =Amount of grease to add (g) (specific gravity of grease=0.9)
- D =Outside diameter of bearing (mm)
- B =Inner ring width (mm)

③ Lubrication Interval

For a typical bearing, which operates at about 50ºC, lubricant should be replaced once a year. If operating temperature is 100ºC or more, the lubricant should be replaced more than once every three months even if it has good heat stability.
If oil bath lubricant becomes contaminated by water or foreign particles, it must be replaced immediately.
The grease relubrication interval can be estimated from Fig 8.37.

[Fig. 8.37 Grease lubrication interval]

④ Grease Service Life

For applications where relubrication is not possible or practical, grease service life may be estimated using Formula (8.21).
The following formula was derived using a grease with Lithium thickener and mineral oil base.

$$\log L = (0.018f - 0.025)T - 2.77f + 6.3 \quad \cdots\cdots\cdots (8.20)$$

where:
- L =Grease life (h)
- f =(Operating speed) (rpm)/(Bearing grease speed limit) (rpm)
  If f is less than 0.25, f is set = 0.25
- T =Operating temperature (ºC)

If T is less than 30ºC, T is set = 30.

[Table 8.25 Grease Properties]

Fig. 8.37
Grease lubrication interval

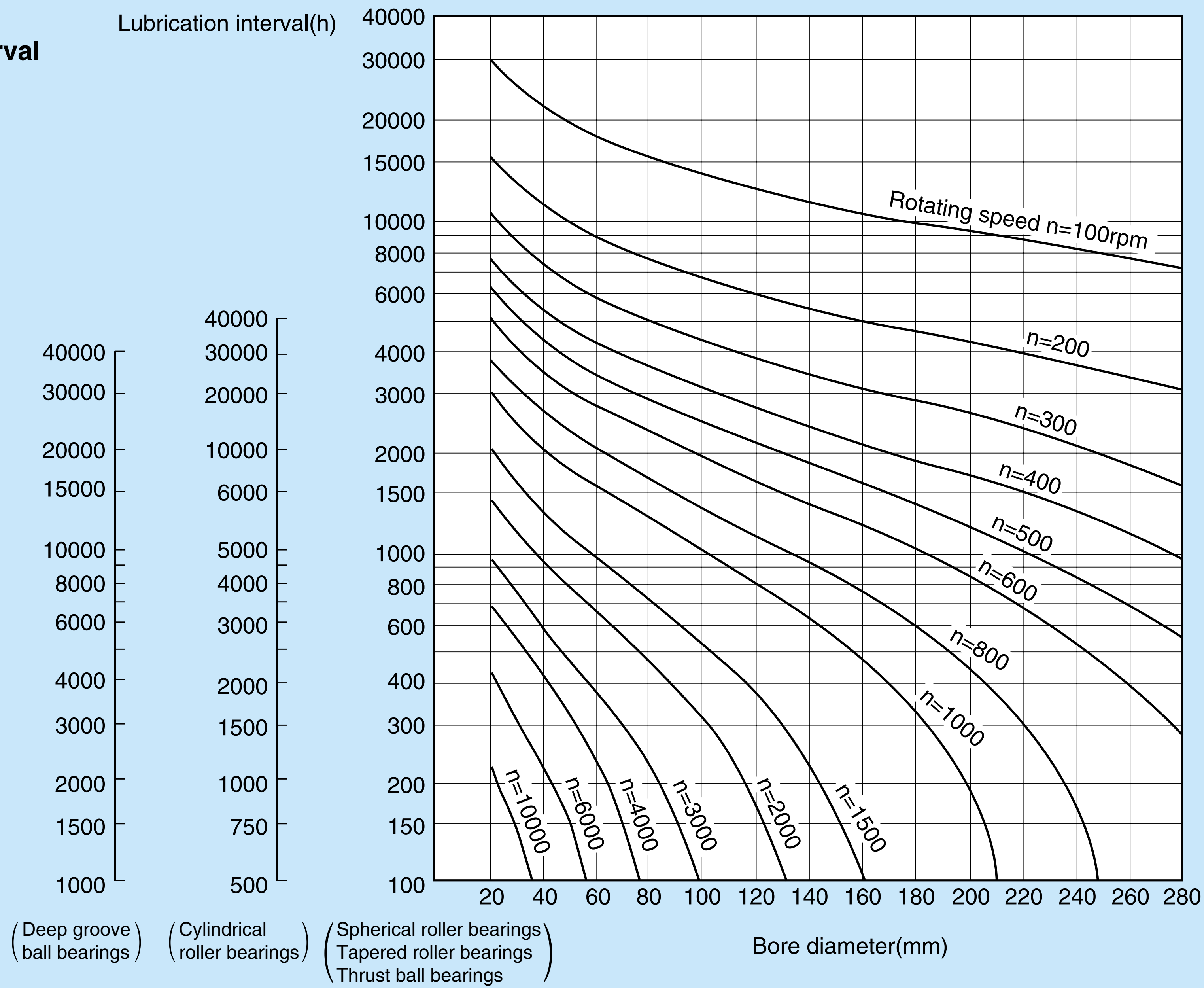


Table 8.25 Grease Properties

Popular Name	Cup Grease	Fiber Grease	Aluminum Grease	General -purpose Grease	Diester Grease	Silicone Grease	Mixed Base Grease	Complex Grease	Non-Soap Base Grease	
Thickener	Ca Soap	Na Soap	Al Soap	Li Soap			Ca + Na Soap, etc.	Li Complex Soap, etc.	Bentonite, Urea Fluoric, etc.	
Base Oil / Properties	Mineral Oil	Mineral Oil	Mineral Oil	Mineral Oil	Diester Oil	Silicone Oil	Mineral Oil	Mineral Oil	Mineral Oil	Synthetic Oil
Dropping Point (°C)	85	160 or Higher	85	170 or Higher		200 or Higher	150 or Higher	200 or Higher	250 or Higher	
Working Temperature (°C)	–20 ~ +70	–10 ~ +120	–10 ~ +80	–30 ~ +120	–50 ~ +130	–50 ~ +170	–30 ~ +120	–30 ~ +140	–10 ~ +130	–50 ~ +200
Water Resistance	Good	Poor	Good	Good			Poor for Na Soap	Good	Good	
Mechanical Stability	Fair	Good	Fair	Good			Good	Good	Good	
Remarks	Contains small amount of moisture for structure stability Not suitable for use at high temperature	Can not be used with water or moisture due to emulsification with water Used at relatively high temperature	Used in vibrating condition due to good tackiness	General purpose grease widely used small or medium size ball bearings	Suitable for low temperature operation	Wide working temperature range Mainly for light load conditions.	Used in large size bearings	Suitable for high temperature and heavy load conditions	Wide working temperature range Depending on combination of thickener and base oil used, good high temperature, low temperature or chemical stability can be obtained	

Note: 1. Greases with sodium (Na) soap thickener can not be used in applications there is a risk of water or high humidity because they became soft and flow out if they mix with water.
2. In case of mixing different brands of grease (not recommendable), please consult grease manufacturer to determine if there are any detrimental effects.
3. In case operating temperature are beyond what is shown in the table, please consult NACHI.

8.6 Speed Limit

- Bearings exceeding a certain operating speed will begin to create internal heat which may not be controllable.
- Speed limits vary with bearing types, dimensions, lubrication system, internal design of the bearing, and working loads. In addition, speed limits will vary according to the type of integral bearing seal which may be used (dependent on the speed of the seal contact area).
- The term "speed limit" refers to the estimated speed, in revolutions per minute, at which bearings will remain serviceable.

The dimension tables show speed limits for both grease and oil lubrication. Please note that the published speed limits are based on operation of properly lubricated, lightly-loaded bearings, installed on a horizontal shaft.

8.6.1 Speed Limit Correction for Load

As noted above, bearing speed limits will vary with respect to load. Figs. 8.38.1 and 8.38.2 allow calculation of a speed limit correction factor which is applied to the speed limit tables.

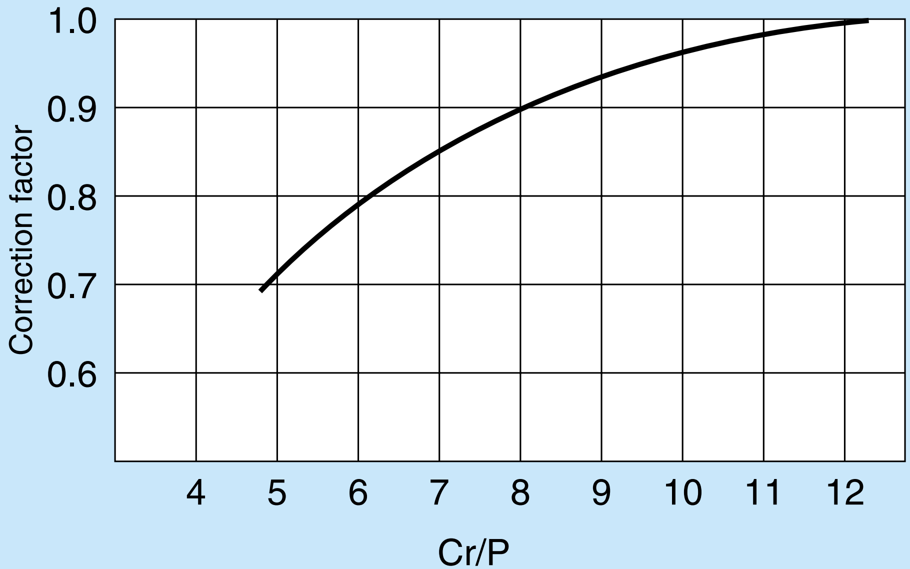
- In Fig. 8.38.1, Cr is the basic dynamic load rating and P is the equivalent dynamic load. If Cr/P is < 13, then the table speed limit is multiplied by the correction factor from the curve shown in Fig. 8.38.1.
- In addition, if the ratio of the axial load (Fa) to the radial load (Fr) is larger than 0.3, that is, if Fa/Fr > 0.3, then the speed limit must be FURTHER multiplied by a correction factor as shown in Fig. 8.38.2
- Where the bearing is used at 75% or more of the speed limit, lubrication becomes a more sensitive operating consideration. If grease is to be used, then selection of the correct type and amount of grease is of paramount importance. If oil is used, then the correct selection of the feeding method and rate, and oil specification is of extreme importance.
- Please contact NACHI for help in cases where application rotating speed exceeds the corrected bearing speed limit.
- If the bearing is used in excess of the corrected speed limit, consideration must be given to the accuracy and clearance of the bearing; and to the material and shape of the retainer. Table 8.26 provides a guideline for maximum speed for bearings using special cages and internal design.

Table 8.26
Correction of Allowable Speed Limit in High-speed Operation

Bearing type	Correction Factor
Deep groove ball	2.5
Angular Contact ball	2
Cylindrical roller (single-row)	2.5
Tapered roller	2
Spherical roller	1.5

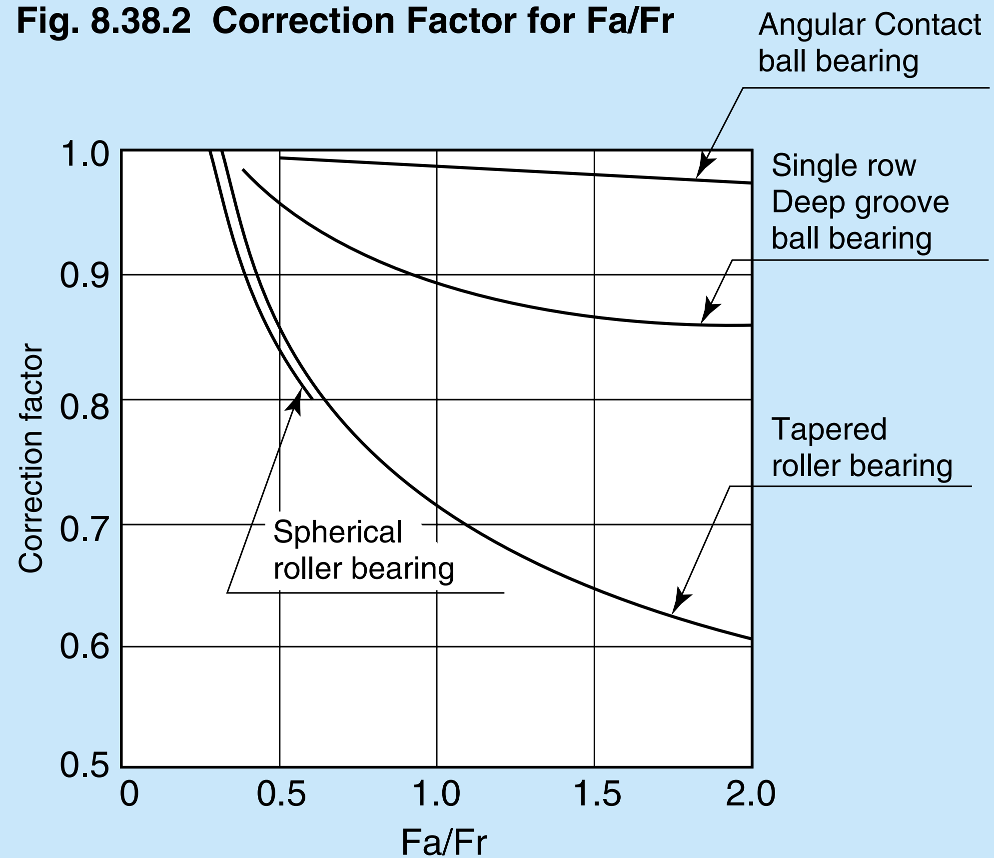
Fig. 8.38.1 Correction Factor for Bearing Load
Fig. 8.38.2 Correction Factor for Fa/Fr

Fig. 8.38.1 Correction Factor for Bearing Load



Cr: Dynamic load rating (N)
P : Dynamic equivalent load (N)

Fig. 8.38.2 Correction Factor for Fa/Fr



Fa : Axial load (N)
Fr : Radial load (N)

8.7 Friction and Temperature Rise

8.7.1 Friction Torque

Friction torque in rolling bearings will vary with the bearing load and the condition of the lubricant.
Where the bearing load is light-to-normal ($P \leq 0.12C$) and the lubricant provides good separation between the rolling contact surfaces, bearing friction torque may be calculated using the following formula:

$$M = \mu \cdot F \cdot \frac{d}{2} \quad \cdots\cdots\cdot (8.22)$$

where:
M = friction torque (N·mm)
μ = coefficient of friction
F = bearing load (N)
d = shaft diameter (mm)

The coefficient of friction for various bearing types is shown in Table 8.27.

Table 8.27 Coefficient of Friction

Bearing type	Coefficient of friction (μ)	Load condition
Ball Bearings: Single row deep groove	0.0010 ~ 0.0015	Radial load
Single row angular contact	0.0012 ~ 0.0018	Radial load
Self-aligning	0.0008 ~ 0.0012	Radial load
Thrust	0.0010 ~ 0.0015	Axial load
Roller Bearings: Cylindrical	0.0008 ~ 0.0012	Radial load
Spherical	0.0020 ~ 0.0025	Radial load
Spherical thrust	0.0020 ~ 0.0025	Axial load
Tapered	0.0018 ~ 0.0025	Radial load

8.7.2 Temperature Rise

- Temperature rise in bearings is caused by the conversion of friction energy into heat.
- Bearing temperature will generally rise quite abruptly during the initial stage of operation and then gradually climb until a steady state is reached. The steady state condition will exist if temperature rise from frictional energy is removed by the cooling "heat-sink" effect from the shaft and housing, and from heat conductance via the shaft, housing and lubricant.
- The time until equilibrium is attained depends on the difference between heating volume generated by the bearing and the heating volume removed by the cooling effect.
- If the equilibrium temperature is excessively high, then review of the bearing application should be done. The bearing internal clearance or preload, fits, bearing support structure, seal contact area surface finish, rotating speed, load, and lubrication type, amount, and delivery system are subjects for investigation where excessive temperature occurs.
- An abnormal temperature rise can cause a spiraling condition where no equilibrium will occur, thus leading to a break-down in the lubricant and lubricant film, with catastrophic results.

8.8 Mounting and Dismounting

Rolling bearings have higher accuracy than other parts in most equipment and are often considered to be the most important rotating component. Improper handling of bearings reduces machine accuracy and can cause early bearing failures. To attain predicted bearing performance, utmost care should be taken in handling bearings from the point-of-receipt through the mounting operation.

8.8.1 Storage and Handling

The major problems encountered during the bearing storage and retrieval operations are in rusting and impact damage to the parts.

- To protect bearings against rusting during storage, parts should be placed in a dry, clean, cool area. Bearings should not be subjected to extremes of humidity during storage.
- Impacts to bearings can create damage to the raceways, rolling elements, and cages. Do not drop bearings. Bearings which are dropped should not be used for service.

8.8.2 Mounting

Proper bearing mounting governs the life, accuracy, and performance of a bearing. Before mounting the bearing, carefully check the following points.

Check to see if:

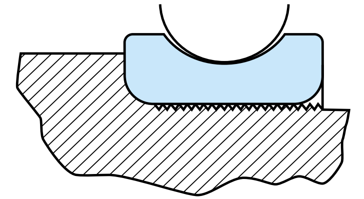
- the job standards are established and the necessary jigs are prepared.
- the shaft and housing size, tolerance, and finish are defined and met.
- lubricant type and amount specified is at hand.
- inspection standards are established.
- the method of cleaning the bearing and relevant parts is clear.

(1) Mounting Precautions

- Select a clean, dry place to handle the bearing, and keep necessary tools and workbench clean.
- Do not unpack the bearing until it is to be mounted.
- If the bearing is unpacked before mounting for acceptance inspection or for any other reason, follow these directions:

a) If the bearing is to be mounted within a short time period, coat it with rust preventive oil and place it in a clean container.
b) If the bearing will not be mounted in a short time, coat it with rust preventative oil and repack it in the original container.

- Check to see that the lubricant drums, cans, tubes, or applicators are clean and or closed. Check to be sure that the bearing housing is clean and free from flaws, impressions, burrs, or any other defects.
- For grease lubrication, you may fill the new bearing with grease without cleaning the bearing. If the bearing is small or is used for high-speed operation, whether it is lubricated with oil or grease, wash the bearing with clean kerosene or warm, light oil to remove the rust preventative. However bearings with seals or shields must not be washed and heated.
  If gear oil is used for lubrication, clean the bearing to remove any rust preventive oil.

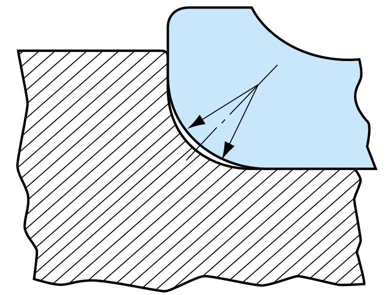


Surface finish

Fig. 8.39

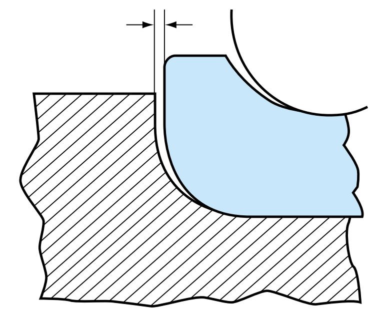
(2) Shaft

- Before mounting the bearing on the shaft, check to see that the shaft is finished to the specified size and accuracy.
- Check the shaft for surface finish. If the shaft fit surface has a poor surface finish (see Fig. 8.39), the surface may be smoothed during mounting, possibly resulting in bearing ring creep, shaft wear, and early bearing failure.
- Be sure that the shaft shoulders are finished at a right angle to the shaft axis, otherwise the bearing will be misaligned resulting in early bearing failure.
- Finish the corner radius of the shaft to the specified dimensions. Make sure the corner radius of the shaft is slightly smaller than that of the bearing as shown in Fig. 8.40. Never have the corner radius of the shaft larger than that of the bearing (see Fig. 8.41), otherwise, the bearing ring may be misaligned and early bearing failure will occur.
- Out-of-roundness of shaft
  Make sure that the shaft is accurate to out-of-roundness and cylindricity specifications. The inner ring of the bearing is an elastic body, having a relatively thin wall, so if the inner ring is fitted to a shaft having poor roundness, the inner ring raceway will be deformed accordingly.
- Contact surface of oil seals
  When using an oil seal, finish the seal contact surface to Ra < 0.8 μm. If the finish is rougher than Ra < 0.8 μm, the seal will gradually wear until it has no sealing effect. Also make sure that the contact surface is within the runout tolerance, otherwise oil leaks may occur since the seal lip may not stay in contact with the rotating shaft.



Corner radius of shaft (Good)

Fig. 8.40



Corner radius of shaft (Poor)

Fig. 8.41

(3) Bearing Housing

- Purposes of the bearing housing are:

  a) to maintain the bearing position for load support.
  b) to protect the bearing from the intrusion of foreign material.
  c) to provide a structure that will keep the bearing well lubricated.

- Verify that the housing bore diameter is to design specifications. If a loose fit class of H or looser is specified, check to make certain that the bearing will move freely in the bearing housing during installation. On horizontally-split bearing housings such as used on pillow blocks, do not mix the caps and bases during a reassembly procedure since these parts are mated during manufacture. In the latter case, mixing may cause either pinching or looseness of the bearing.
- Allowance must be made for linear expansion of the shaft due to temperature rise. When two or more bearings are mounted on a single shaft, comply with the following directions:
  Fix one bearing in the axial direction in the housing, and make sure that the other bearing(s) are free to move in an axial direction.

(4) Accessory Mounting Parts

Prior to bearing mounting, gather a set of the parts required for the mounting job. These accessory parts may include washers, adapters, withdrawal sleeves, spacer rings, slingers, oil seals, O-rings, shaft nuts, and snap rings for the shaft and or housing bore. Thoroughly clean these accessory parts and check them for appearance and size.

Other Precautions

- Be sure that the side of the shaft nut is at a right angle to the thread, otherwise, when tightened, the side of the shaft nut will make uneven contact with the side of the bearing causing early bearing failure. Use particular care when the bearing is used for high-accuracy applications such as lathes.
- Check the washer and spacer ring for parallelism of both sides.
- The oil seal and O-ring may create a temperature rise because the contact force is too great or because they are initially dry. Apply oil or grease to the contact surfaces to help prevent premature wear and reduce torque.

8.8.3 Bearing Mounting Considerations

When pressing a bearing into position, press against the ring with interference fit. Pressing through the rolling elements will cause damage, such as brinell marks or cracks to the elements and rings and the bearing will be unusable.
For inner ring rotating loads, the bearing is generally interference-fit to the shaft and either expansion fitting or press fitting can be used. expansion fitting may be the more appropriate method for mounting larger bore bearings.
A tapered-bore bearing can be mounted directly to a tapered shaft or with an adapter or withdrawal sleeve. When a withdrawal sleeve is used for larger bore bearings, the hydraulic mounting procedure will facilitate the process. Note that the use of hydraulic mounting of bearings to tapered journals is also very useful for larger bearing sizes.
For an outer ring rotating load, the bearing is usually interference-fitted with the housing. Either press fitting or shrink fitting may be used. In the case of the latter process, the bearing or bearing outer ring may be cooled to attain the fit.

(1) Mounting Cylindrical-bore Bearings

• Press fitting

Many cylindrical-bore bearing applications use press fitting with the shaft. Use a jig which matches the inner ring as shown in Fig. 8.42. Press fit the inner ring using a press or jack.
To press fit the inner and outer rings simultaneously, use a jig as shown in Fig. 8.43.
Apply high-viscosity oil to the shaft and the contact faces of the bearing before press fitting.

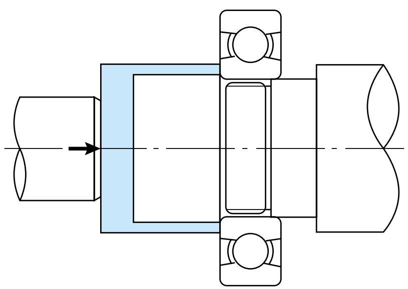


Fig. 8.42 Press Fitting of Inner Ring

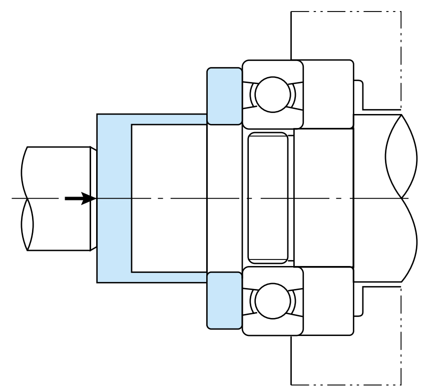


Fig. 8.43 Simultaneous Press Fitting of Inner Ring

• Expansion Fitting

Expansion fitting is an appropriate procedure for mounting larger bore bearings. This fitting procedure can be completed quickly without applying undue stress to the ring being fit. The ring may be heated using a heating tank or an induction heater. Bearing rings must not be heated to a temperature exceeding 120 °C.
Fig. 8.44 provides the amount of heat rise required, vs. bore size, to expand inner rings to net 6 interference fit classes.
After mounting a heated bearing, secure it in the required position otherwise the bearing will tend to move axially as it cools.

Caution: When expansion fitting rings onto a shaft or into a housing, be sure that the procedure can be completed smoothly and quickly. If the ring should misalign or stop movement before it has reached the desired fit position, it may be very difficult to reposition the ring to the correct fit position.
As noted above, adapter or withdrawal sleeves are used for mounting tapered-bore bearings to cylindrical shafts. Tapered-bore bearings are also mounted directly to tapered shafts. While these methods are technically not press or expansion fitting procedures, the resulting shaft fits are similar to those obtained using the press fitting procedure, and, in certain cases, these methods are far more convenient.

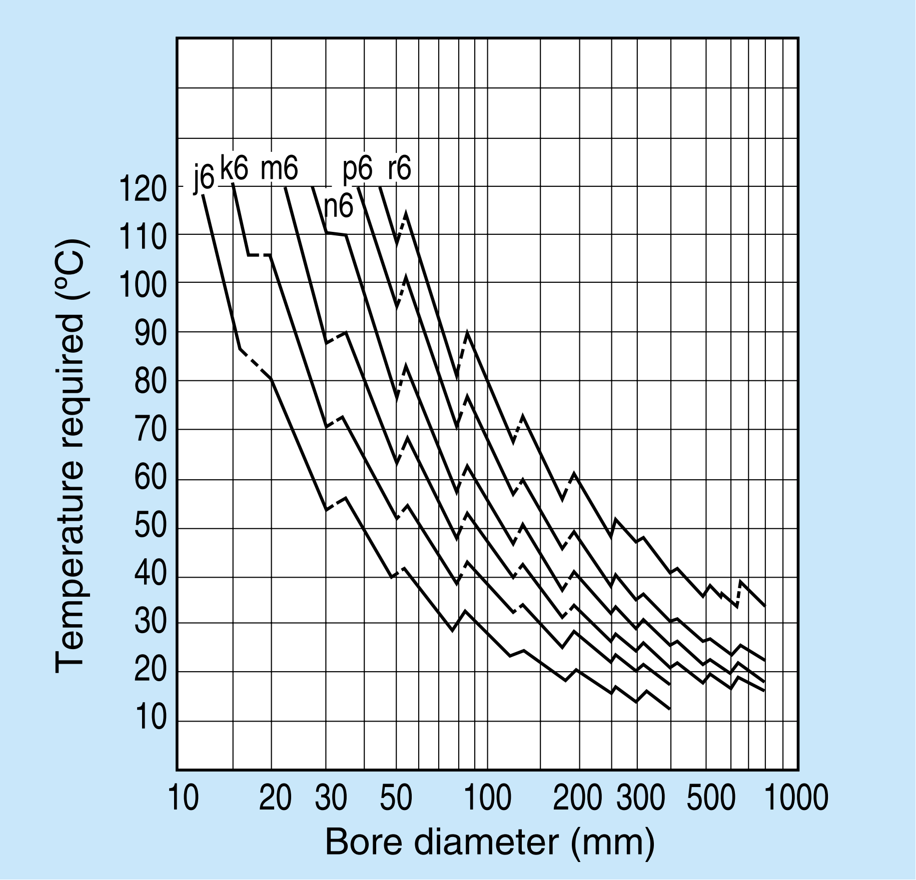


Fig. 8.44 Expension Heat Required

(2) Mounting Tapered-bore Bearings

Using a split-sleeve adapter permits the mounting of tapered-bore bearings in any axial position on shaft but care must be take to ensure that the bearing will be located at the correct position.
To mount a tapered-bore bearing using an adapter sleeve, first mount the bearing which is to be the stationary (fixed) bearing. Define and record the distance which the free bearing is expected to move in an axial direction in the housing.
Mount the free bearing so that the axial clearance provided for axial travel of the outer ring of the free bearing is on the outboard side (side farthest from the stationary bearing).
The required interference fit for tapered-bore, Spherical roller bearings can be attained using one of two methods:

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a) by driving the bearing onto the sleeve by a predetermined distance; or,
b) by measurement of residual bearing internal clearance as the sleeve is pushed into the bearing inner ring (see Table 8.28).
Since exact measurement of the axial drive-up distance is extremely difficult, the residual method is usually the method of choice.

The residual method involves measuring the bearing unmounted internal clearance and then pulling up the adapter sleeve until the measured clearance (the residual) = the unmounted (original) clearance - the reduction amount required to attain the correct interference fit (see Table 8.28 for the reduction amount). Clearance measurements are made using a thickness (feeler) gauge. (Note that the thickness gauge should be inserted over two or three unloaded rollers on each row of rollers and that the bearing bore must be in a horizontal position with respect to the shaft axis, with the outer ring centered over the rolling elements).
Table 8.28 shows axial movement and radial clearance reduction for the mounting of Spherical roller bearings.
Heating of larger tapered-bore bearings may be used in conjunction with measurement of travel distance but be sure to check the results using the residual method (taking the unmounted clearance measurements and the final, residual clearance, when the bearing is cool). Also be sure that the bearing is not heated to over 120 ºC.
When using a withdrawal sleeve for large-bore bearings, use of a hydraulic assist procedure is recommended. See Fig. 8.45 which shows use of a hydraulic nut.

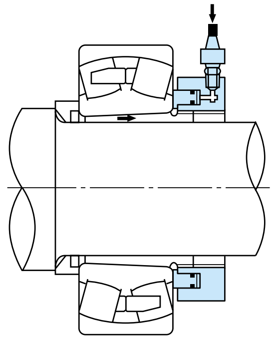

Fig. 8.45 Hydraulic Nut

Table 8.28 Tapered-bore Spherical Roller Bearings: Axial Movement and Radial Clearance Reduction

(3) Other Mounting Precautions

- For paired Tapered roller bearings, be sure to adjust the axial clearance to the specified value using shims where necessary.
- For bearing types with separable inner and outer members such as Cylindrical or Tapered roller bearings, mount the inner and outer ring separately and carefully assemble the shaft into the housing while making sure that no damage occurs to the inner or outer rings or rolling elements.

Table 8.28 Tapered-bore Spherical Roller Bearings: Axial Movement and Radial Clearance Reduction

Nominal bore diameter d (mm)		Radial clearance reduction (mm)		Axial movement (mm)			
				Taper: 1/12		Taper: 1/30	
Over	Incl.	min	max	min	max	min	max
30	40	0.020	0.025	0.35	0.4	–	–
40	50	0.025	0.030	0.4	0.45	–	–
50	65	0.030	0.040	0.45	0.6	–	–
65	80	0.040	0.050	0.6	0.75	–	–
80	100	0.045	0.060	0.7	0.9	1.75	2.25
100	120	0.050	0.070	0.75	1.1	1.9	2.75
120	140	0.065	0.090	1.1	1.4	2.75	3.5
140	160	0.075	0.100	1.2	1.6	3.0	4.0
160	180	0.080	0.110	1.3	1.7	3.25	4.25
180	200	0.090	0.120	1.4	1.9	3.5	5.0
200	225	0.100	0.140	1.6	2.2	4.0	5.5
225	250	0.110	0.150	1.7	2.4	4.25	6.0
250	280	0.120	0.170	1.9	2.7	4.75	6.75
280	315	0.130	0.190	2.0	3.0	5.0	7.5
315	355	0.150	0.210	2.4	3.3	6.0	8.25
355	400	0.170	0.230	2.6	3.6	6.5	9.0
400	450	0.200	0.260	3.1	4.0	7.75	10.0
450	500	0.210	0.280	3.3	4.4	8.25	11.0

8.8.4 Mounting and Dismounting Force

An approximate force necessary to install or remove an inner ring from a shaft may be calculated using the following equation.

$$Ka = fk \cdot fe \cdot \Delta de \qquad \cdots\cdots (8.23)$$

where:

Ka	= press fit or dismount force (KN)
Δde	= effective interference (mm)
fk	= factor from Table 8.29
fe	= from following equation

Table. 8.29 Value fk (Average)

Condition	fk
Inner ring pressed to cylindrical shaft*	39
Inner ring pulled from cylindrical shaft	59
Inner ring press fit to tapered shaft or sleeve*	54
Inner ring pulled from tapered shaft	44
Tapered sleeve press fit between shaft & bearing*	98
Tapered sleeve pulled from between shaft & bearing	108

* Shaft and bearing bore thinly coated with oil.

$$fe = B \cdot \left[1 - \left(\frac{d}{di}\right)^2\right]$$

where:

B	= inner ring width (mm)
d	= inner ring bore diameter (mm)
di	= mean inner ring outside diameter (mm)
di	≒ 0.25 (D+3d)···for Cylindrical roller bearings and Self-aligning Ball bearing series 22 and 23
di	≒ 0.1 (3D+7d)···for other bearings

where:

D	= Bearing outside diameter

Fig. 8.46 ~ 8.49 show dismount and press fit force by diameter series.

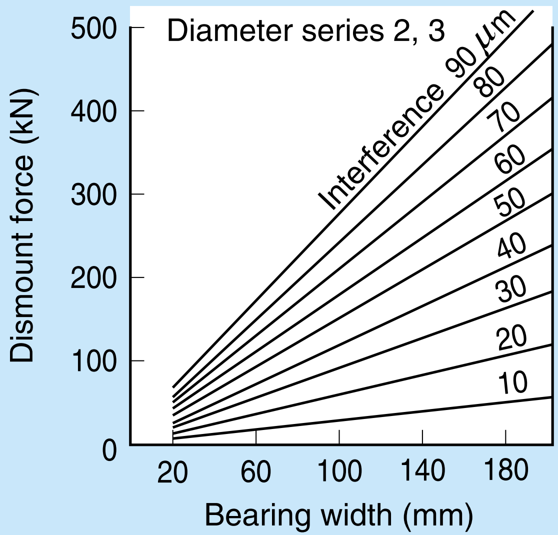

Fig. 8.46 Dismount Force

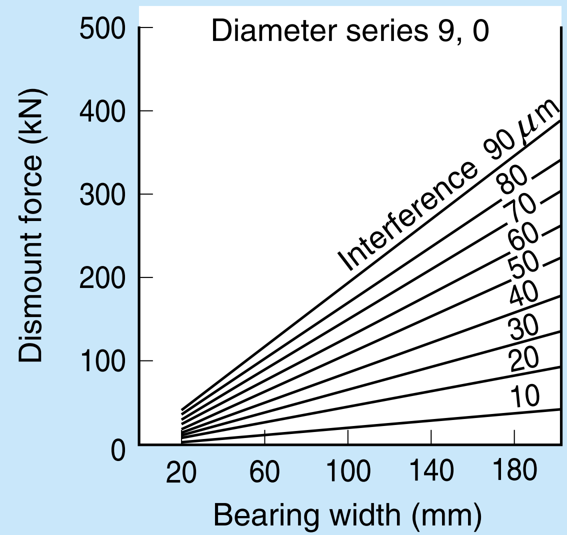

Fig. 8.47 Dismount Force

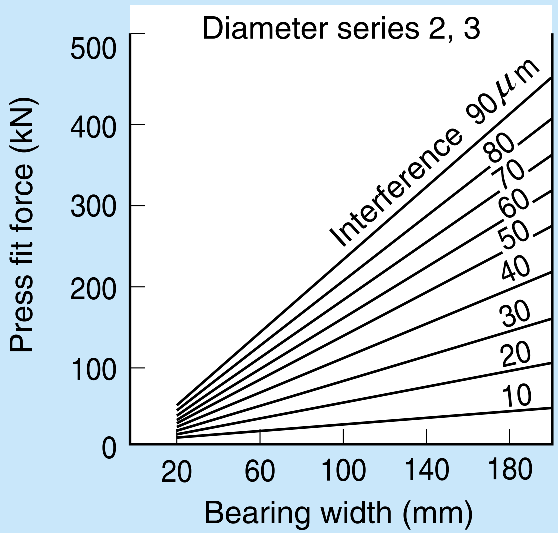

Fig. 8.48 Press-fit Force

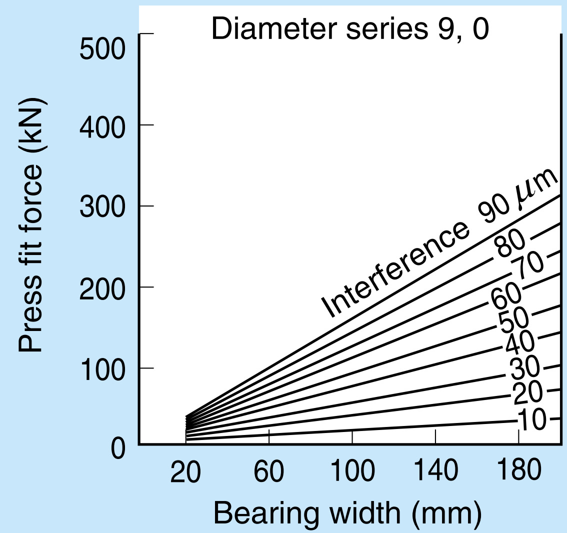

Fig. 8.49 Press-fit Force

8.8.5 Operation Inspection

Verify satisfactory service with a test run. General precautions for a test run are:

- Make sure that all drive covers are in place, all bolts and nuts are tight, and appropriate clearance is provided between the shaft and all stationary parts.
- If possible, manually turn the shaft to see if there is rubbing or abnormal noise.
- If the machine is large and the shaft cannot be turned by hand, start the machine at as low speed as possible and check for rubbing or abnormal noise while coasting the machine.
- If no trouble is found during the above checks, run the machine at the design speed until attaining a steady-state temperature.
- Recheck bolt and nut tightness. Check for oil leaks, and abnormal noise. If possible, extract a sample of the oil and check it for foreign matter.
- Begin regular operation.

If trouble is encountered during machine operation, refer to Section 9, "Trouble-shooting Bearing Problems".

8.8.6 Dismounting

Bearings may be dismounted for periodic machine inspection, or when machine break down has occurred. The condition of all rotating parts and interfaces should be checked and recorded to collect data for operating improvements. The recording of data is essential where a parts failure has occurred to enable a solution to any existing trouble.
In dismounting the bearing, check to see:

- If the bearing is satisfactorily mounted. (Bolts, and nuts tightened, interference of slinger with bearing housing, etc.)
- If there is (was) an adequate lube supply. Check for lubricant contamination and sample for residues.
- That the inner and outer ring have retained the fits as mounted.
- If the bearing clearance is as specified. If possible, measure the clearance of the mounted bearing.
- The condition of the bearing.

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Before starting to dismount a bearing, review the following points:

- Dismount method
- Fit conditions
- Jigs required for dismounting
  - Press (Fig. 8.50)
  - Spanner wrench (Fig. 8.51)
  - Puller (Fig. 8.52)
  - Special puller (Fig. 8.53)
  - Holder (Fig. 8.54)

To dismount a Cylindrical roller bearing, the inner ring may be locally heated with an induction heater to facilitate removal from the shaft. (See Fig. 8.55.)
For large-bore bearings, which are often difficult to dismount, a hydraulic nut or oil injector system is recommended. See Fig. 8.45 and Fig. 8.56 respectively.

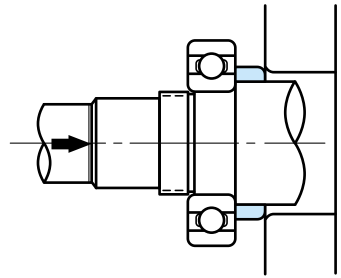

Fig. 8.50 Dismounting Bearing Using Press

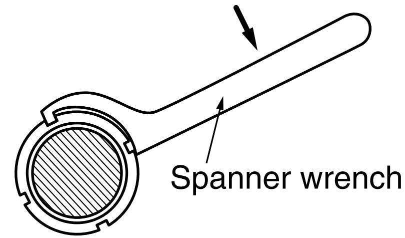

Fig. 8.51 Dismounting Bearing with Spanner Wrench

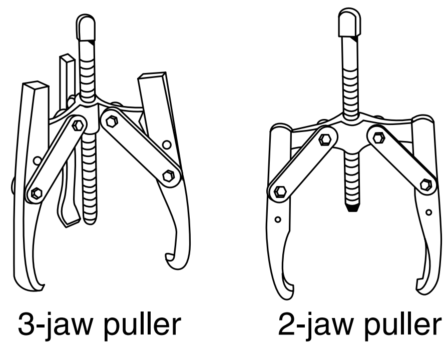

Fig. 8.52

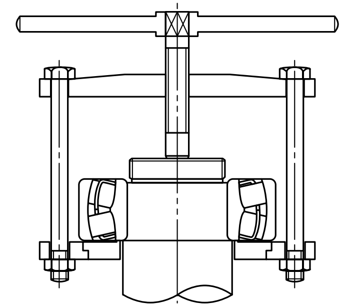

Fig. 8.53 Dismounting Bearing with Special Puller

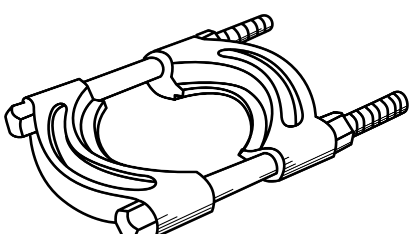

Fig. 8.54 Puller Attachment

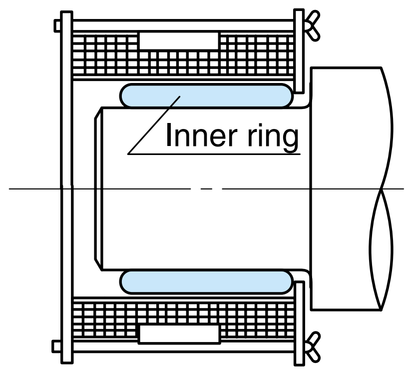

Fig. 8.55 Inner Ring Removal with Induction Heater

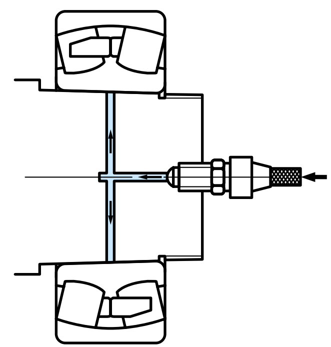

Fig. 8.56 Oil Injector