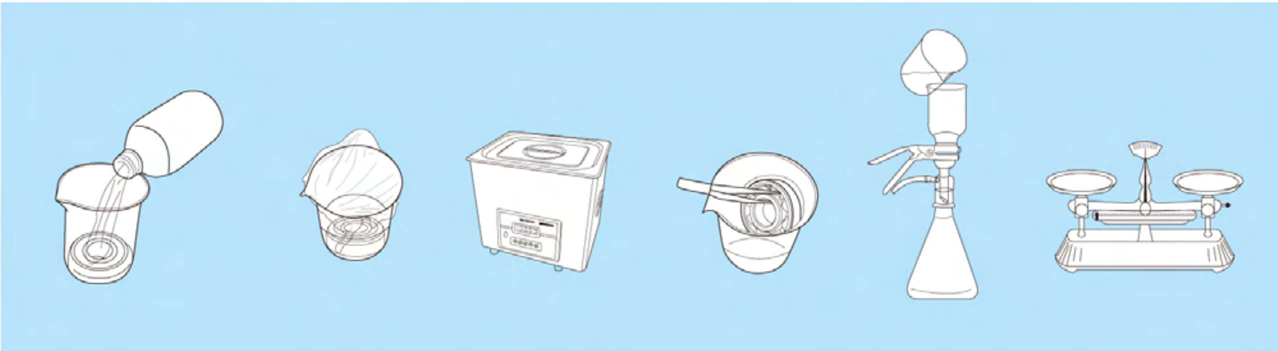
BEARING CLEANLINESS

Bearing cleanliness is essentially important since the contamination will directly influence the bearing life. The influence of contamination on bearing depends on a number of parameters including bearing size, relative lubricant film thickness, size and distribution of solid contaminant particles, types of contamination (soft, hard etc). The influence of contamination on bearing is complex and many of the parameters are difficult to quantify. It is therefore not possible to allocate precise values that would have general validity. HCH has introduced one method for checking the contamination level is the blank experiment shown as following:



Guideline values are provided according to JB/T 7050-2005 as following table:

Outer Diameter *D* mm		Deep groove ball bearing without lubrication			Deep groove ball bearing with lubrication type			Tapered roller bearing		
		Dimension Series								
Over	Including	7, 8, 9, 0, 00	2	3, 4	0, 7, 8, 9	2	3	02, 03, 13	20, 22, 29, 23	32, 31, 30
		Average maximum contamination level w mg/PC								
-	10	0.3	0.3	0.5	0.5	0.7	0.8	-	-	-
10	20	0.5	0.7	0.8	0.8	1.0	1.2	-	-	-
20	28	0.8	1.0	1.2	1.2	1.4	1.6	-	2.9	-
28	40	1.2	1.3	1.7	1.3	1.6	1.8	-	3.7	-
40	50	1.5	1.7	1.8	1.7	1.8	2.2	-	4.9	5.4
50	60	1.8	2.0	2.2	2.0	2.2	2.7	4.9	5.7	6.4
60	70	2.0	2.2	2.7	2.2	2.7	3.0	5.7	6.4	7.6
70	80	2.2	2.7	3.0	2.3	2.8	3.3	6.4	7.6	8.7
80	90	3.8	4.3	5.2	4.0	4.7	5.3	7.1	8.4	9.7
90	100	4.2	5.2	5.8	4.7	5.3	6.2	7.9	9.5	10.8
100	110	5.0	5.8	6.7	5.2	6.0	7.0	8.7	10.3	11.6
110	120	5.7	6.3	7.3	5.8	6.8	7.8	9.7	11.4	12.9
120	130	6.2	7.2	8.0	6.3	7.3	8.3	10.6	12.3	14.0
130	140	6.7	8.0	9.0	7.0	8.2	9.3	11.4	13.3	15.1
140	150	7.2	9.0	9.8	7.7	9.0	10.2	12.1	14.3	16.3
150	160	7.8	9.2	10.7	8.0	9.3	10.8	12.9	15.1	17.4
160	170	8.3	10.0	11.3	8.5	9.7	11.4	13.9	16.3	19.1
170	180	9.0	11.0	12..2	9.5	10.8	12.2	14.6	17.1	20.0

1) 包括 15 mm

TOLERANCE

Bearing tolerances include dimensional tolerance and rotary tolerance or dimensional accuracy and running accuracy, are regulated by ISO or JIS standards (rolling bearing tolerances) etc. For dimensional accuracy, these standards prescribe the tolerances necessary when installing bearings on shafts or in housings. Running accuracy is defined as the allowable limits for bearing run-out during operation.

According to dimension tolerances and runout accuracy, HCH deep groove ball bearings are ranged from ABEC-1 to ABEC-7. ABEC-1 is standard grade, while ABEC-3 is higher, ABEC-5 is much higher and ABEC-7 is the highest.

● Comparison of tolerance classifications of national standards

Standard	Applicable standard	Tolerance class				
ANSI	ANSI/ABMA Std.20	ABEC-1	ABEC-3	ABEC-5	ABEC-7	ABEC-9
JIS	JIS B 1514	Class0,6X	Class 6	Class 5	Class 4	Class 2
ISO	ISO 492	Class0,6X	Class 6	Class 5	Class 4	Class 2
DIN	DIN 620	P0	P6	P5	P4	P2

Symbols: The following symbols are used to identify boundary of dimensions, size and size variations, and runout errors.

● Symbols for bearing dimensions and accuracy

Basic Dimensions

d = Basic bore diameter

D = Basic outside diameter

B = Basic inner ring width

C = Basic outer ring width

r = Chamfer of inner ring and outer ring

Dimentional Variaties

V_{dmp} = Mean bore diameter variation

V_{Dmp} = Mean outside diameter variation

V_{Bs} = Inner ring width variation

V_{Cs} = Outer ring width variation

Dimentional Accuracy

Δ_{dmp} = Single plane mean bore diameter deviation from basic

Δ_{Dmp} = Single plane mean outside diameter deviation from basic

Δ_{Bs} = Single inner ring width deviation from basic

Δ_{Cs} = Single outer ring width deviation from basic

Running Accuracy

K_{ia} = Radial runout of assembled bearing inner ring

K_{ea} = Radial runout of assembled bearing outer ring

S_{ia} = Axial runout of assembled bearing inner ring

S_{ea} = Axial runout of assembled bearing outer ring

S_d = Inner ring reference face runout with bore

S_D = Outside cylindrical surface runout with outer ring reference face

DEEP GROOVE BALL BEARING TOLERANCE CLASS ABEC-1 (P0)

INNER RING

Tolerance in µm.

d mm		Δ_{dmp}		V_{dp} [2] Diameter Series			V_{dmp}	K_{ia}	Δ_{Bs}		V_{Bs}
				9	0,1	2,3,4					
over	incl.	high	low	max.			max.	max.	high	low	max.
[1] 0.6	2.5	0	-8	10	8	6	6	10	0	-40	12
2.5	10	0	-8	10	8	6	6	10	0	-120	15
10	18	0	-8	10	8	6	6	10	0	-120	20
18	30	0	-10	13	10	8	8	13	0	-120	20
30	50	0	-12	15	12	9	9	15	0	-120	20
50	80	0	-15	19	19	11	11	20	0	-150	25
80	120	0	-20	25	25	12	12	25	0	-200	25
120	180	0	-25	31	31	19	19	30	0	-250	30

Notes:

1) 0.6 is included.

2) No values for diameter series 7,8.

OUTER RING

Tolerance in µm.

D mm		Δ_{Dmp}		V_{Dp} [2][4] Open Bearings			Capped Bearings [2][3]	V_{Dmp} [4]	K_{ea}	Δ_{Cs}		V_{Cs}
				Diameter Series								
				9	0,1	2,3,4	2,3,4					
over	incl.	high	low	max.				max.	max.	high	low	max.
[1] 2.5	6	0	-8	10	8	6	10	6	15	Identical to Δ_{Bs} and V_{Bs} of inner ring of same bearing		
6	18	0	-8	10	8	6	10	6	15			
18	30	0	-9	12	9	7	12	7	15			
30	50	0	-11	14	11	8	16	8	20			
50	80	0	-13	16	13	10	20	10	25			
80	120	0	-15	19	19	11	26	11	35			
120	150	0	-18	23	23	14	30	14	40			
150	180	0	-25	31	31	19	38	19	45			
180	250	0	-30	38	38	23	–	23	50			

Notes:

1) 2.5 is included.

2) No values for diameter series 7,8.

3) No values for diameter series 9,0,1.

4) Fit for before assembling and after disestablishing of snap rings.

Inch-dimension equivalent formula	1 inch = 25.4 mm
	1 mm = 0.0393700787 inch

DEEP GROOVE BALL BEARING TOLERANCE CLASS ABEC-3 (P6)

INNER RING

Tolerance in µm.

d mm		Δ_{dmp}		V_{dp} [2] Diameter Series			V_{dmp}	K_{ia}	Δ_{Bs}		V_{Bs}
				9	0,1	2,3,4					
over	incl.	high	low	max.			max.	max.	high	low	max.
[1] 0.6	2.5	0	-7	9	7	5	5	5	0	-40	12
2.5	10	0	-7	9	7	5	5	6	0	-120	15
10	18	0	-7	9	7	5	5	7	0	-120	20
18	30	0	-8	10	8	6	6	8	0	-120	20
30	50	0	-10	13	10	8	8	10	0	-120	20
50	80	0	-12	15	15	9	9	10	0	-150	25
80	120	0	-15	19	19	11	11	13	0	-200	25
120	180	0	-18	23	23	14	14	18	0	-250	30

Notes:

1) 0.6 is included.

2) No values for diameter series 7,8.

OUTER RING

Tolerance in µm.

D mm		Δ_{Dmp}		V_{Dp} [2][4] Open Bearings			Capped Bearings [3]	V_{Dmp} [4]	K_{ea}	Δ_{Cs}		V_{Cs}
				Diameter Series								
				9	0,1	2,3,4	0,1,2,3,4					
over	incl.	high	low	max.				max.	max.	high	low	max.
[1] 2.5	6	0	-7	9	7	5	9	5	8	Identical to Δ_{Bs} and V_{Bs} of inner ring of same bearing		
6	18	0	-7	9	7	5	9	5	8			
18	30	0	-8	10	8	6	10	6	9			
30	50	0	-9	11	9	7	13	7	10			
50	80	0	-11	14	11	8	16	8	13			
80	120	0	-13	16	16	10	20	10	18			
120	150	0	-15	19	19	11	25	11	20			
150	180	0	-18	23	23	14	30	14	23			
180	250	0	-20	25	25	15	–	15	25			

Notes:

1) 2.5 is included.

2) No values for diameter series 7,8.

3) No values for diameter series 9.

4) Fit for before assembling and after disestablishing of snap rings.

Inch-dimension equivalent formula	1 inch = 25.4 mm
	1 mm = 0.0393700787 inch

DEEP GROOVE BALL BEARING TOLERANCE CLASS ABEC-5 (P5)

INNER RING

Tolerance in μm.

d mm		Δdmp		V_{dp} 2) Diameter Series 9	V_{dp} 2) Diameter Series 0,1,2,3,4	V_{dmp}	K_{ia}	S_d	S_{ia} 3)	Δ_{Bs}		V_{Bs}
over	incl.	high	low	max.		max.	max.	max.	max.	high	low	max.
1) 0.6	2.5	0	-5	5	4	3	4	7	7	0	-40	5
2.5	10	0	-5	5	4	3	4	7	7	0	-40	5
10	18	0	-5	5	4	3	4	7	7	0	-80	5
18	30	0	-6	6	5	3	4	8	8	0	-120	5
30	50	0	-8	8	6	4	5	8	8	0	-120	5
50	80	0	-9	9	7	5	5	8	8	0	-150	6
80	120	0	-10	10	8	5	6	9	9	0	-200	7
120	180	0	-13	13	10	7	8	10	10	0	-250	8

Notes:

1) 0.6 is included.

2) No values for diameter series 7,8.

3) Fit for groove ball bearing only.

OUTER RING

Tolerance in μm.

D mm		ΔDmp		V_{Dp} 2),3) Diameter Series 9	V_{Dp} 2),3) Diameter Series 0,1,2,3,4	V_{Dmp}	K_{ea}	S_D 4)	S_{ea} 4)5)	Δ_{Cs}		V_{Cs}
over	incl.	high	low	max.		max.	max.	max.	max.	high	low	max.
1) 2.5	6	0	-5	5	4	3	5	8	8	Identical to Δ_{Bs} of inner ring of same bearing		5
6	18	0	-5	5	4	3	5	8	8			5
18	30	0	-6	6	5	3	6	8	8			5
30	50	0	-7	7	5	4	7	8	8			5
50	80	0	-9	9	7	5	8	8	10			6
80	120	0	-10	10	8	5	10	9	11			8
120	150	0	-11	11	8	6	11	10	13			8
150	180	0	-13	13	10	7	13	10	14			8
180	250	0	-15	15	11	8	15	11	15			10

Notes:

1) 2.5 is included.

2) No values for diameter series 7,8.

3) No values for shielded and sealed bearings.

4) No values for flanged bearings.

5) Fit for groove ball bearings.

Inch-dimension equivalent formula	1 inch = 25.4 mm
	1 mm = 0.0393700787 inch

DEEP GROOVE BALL BEARING TOLERANCE CLASS ABEC-7 (P4)

INNER RING

Tolerance in μm.

d mm		Δ_{dmp}		Δ_{ds}		V_{dp} 2) Diameter Series 7,8,9	V_{dp} 2) Diameter Series 1,7,2,3,4	V_{dmp}	K_{ia}	S_d	S_{ia}	Δ_{Bs}		V_{Bs}
over	incl.	high	low	high	low	max.		max.	max.	max.	max.	high	low	max.
1) 0.6	2.5	0	-4	0	-4	4	3	2	2.5	3	3	0	-40	2.5
2.5	10	0	-4	0	-4	4	3	2	2.5	3	3	0	-40	2.5
10	18	0	-4	0	-4	4	3	2	2.5	3	3	0	-80	2.5
18	30	0	-5	0	-5	5	4	2.5	3	4	4	0	-120	2.5
30	50	0	-6	0	-6	6	5	3	4	4	4	0	-120	3
50	80	0	-7	0	-7	7	5	3.5	4	5	5	0	-150	4
80	120	0	-8	0	-8	8	6	4	5	5	5	0	-200	4
120	180	0	-10	0	-10	10	8	5	6	6	7	0	-250	5

Notes:

1) 0.6 is included.

2) Diameter Series 7 refers to those defined by GB273.3 as the miniature bearing series 7.

OUTER RING

Tolerance in μm.

D mm		Δ_{Dmp}		Δ_{Ds}		V_{Dp} 2) Diameter Series 7,8,9	V_{Dp} 2) Diameter Series 1,7,2,3,4	V_{Dmp}	K_{Ca}	S_D	S_{ea}	Δ_{Cs}	V_{Cs}
over	incl.	high	low	high	low	max.	max.	max.	max.	max.	high	low	max.
1) 2.5	6	0	-4	0	-4	4	3	2	3	4	5	Identical to Δ_{Bs} of inner ring of same bearing	2.5
6	18	0	-4	0	-4	4	3	2	3	4	5		2.5
18	30	0	-5	0	-5	5	4	2.5	4	4	5		2.5
30	50	0	-6	0	-6	6	5	3	5	4	5		2.5
50	80	0	-7	0	-7	7	5	3.5	5	4	5		3
80	120	0	-8	0	-8	8	6	4	6	5	6		4
120	150	0	-9	0	-9	9	7	5	7	5	7		5
150	180	0	-10	0	-10	10	8	5	8	5	8		5
180	250	0	-11	0	-11	11	8	6	10	7	10		7

Notes:

1) 2.5 is included.

2) These windage is applicable for diameter series 1. 7. 2. 3. and 4.

3) Not applicable for closed type bearings.

4) Diameter Series 7 refers to those defined by GB273.3 as the miniature bearing series 7.

Inch-dimension equivalent formula	1 inch = 25.4 mm
	1 mm = 0.0393700787 inch

METRIC TAPERED ROLLER BEARINGS

TOLERANCE CLASS P0 (Normal)

INNER RING

Tolerance in μm.

d (mm)		Δdmp		Vdp	$Vdmp$	Sia	Kia	ΔBs	
over	incl.	high	low	max.	max.	max.	max.	high	low
10	18	0	-12	12	9	24	15	0	-120
18	30	0	-12	12	9	24	18	0	-120
30	50	0	-12	12	9	24	20	0	-120
50	80	0	-15	15	11	30	25	0	-150
80	120	0	-20	20	15	30	30	0	-200

OUTER RING

Tolerance in μm.

D (mm)		ΔDmp		VDp	$VDmp$	Sea	Kea	ΔCs	
over	incl.	high	low	max.	max.	max.	max.	high	low
18	30	0	-12	12	9	40	18	Identical to ΔBs of inner ring of same bearing	
30	50	0	-14	14	11	40	20		
50	80	0	-16	16	12	40	25		
80	120	0	-18	18	14	45	35		
120	150	0	-20	20	15	50	40		
150	180	0	-25	25	19	60	45		
180	250	0	-30	30	23	70	50		

BEARING WIDTHS

d (mm)		ΔTs		$\Delta T1s$		$\Delta T2s$	
over	incl.	high	low	high	low	high	low
10	18	200	0	100	0	100	0
18	30	200	0	100	0	100	0
30	50	200	0	100	0	100	0
50	80	200	0	100	0	100	0
80	120	200	-200	100	-100	100	-100

METRIC TAPERED ROLLER BEARINGS

TOLERANCE CLASS P6 (CLN)

INNER RING

Tolerance in μm.

d (mm)		Δdmp		Vdp	$Vdmp$	Sia	Kia	ΔBs	
over	incl.	high	low	max.	max.	max.	max.	high	low
10	18	0	-12	12	9	24	7	0	-120
18	30	0	-12	12	9	24	8	0	-120
30	50	0	-12	12	9	24	10	0	-120
50	80	0	-15	15	11	30	13	0	-150
80	120	0	-20	20	15	30	–	0	-200

OUTER RING

Tolerance in μm.

D (mm)		ΔDmp		VDp	$VDmp$	Sea	Kea	ΔCs	
over	incl.	high	low	max.	max.	max.	max.	high	low
18	30	0	-12	12	9	40	9	Identical to ΔBs of inner ring of same bearing	
30	50	0	-14	14	11	40	10		
50	80	0	-16	16	12	40	13		
80	120	0	-18	18	14	45	18		
120	150	0	-20	20	15	50	20		
150	180	0	-25	25	19	60	23		
180	250	0	-30	30	23	70	–		

BEARING WIDTHS

d (mm)		ΔTs		$\Delta T1s$		$\Delta T2s$	
over	incl.	high	low	high	low	high	low
10	18	200	0	100	0	100	0
18	30	200	0	100	0	100	0
30	50	200	0	100	0	100	0
50	80	200	0	100	0	100	0
80	120	200	-200	100	-100	100	-100

METRIC TAPERED ROLLER BEARINGS

TOLERANCE CLASS P6x (CL7C)

INNER RING

Tolerance in µm.

d (mm)		Δ_{dmp}		V_{dp}	V_{dmp}	S_{ia}	K_{ia}	Δ_{Bs}	
over	incl.	high	low	max.	max.	max.	max.	high	low
10	18	0	-12	12	9	24	15	0	-50
18	30	0	-12	12	9	24	18	0	-50
30	50	0	-12	12	9	24	20	0	-50
50	80	0	-15	15	11	30	25	0	-50
80	120	0	-20	20	15	30	30	0	-50

OUTER RING

Tolerance in µm.

D (mm)		Δ_{Dmp}		V_{Dp}	V_{Dmp}	S_{ea}	K_{ea}	Δ_{Cs}	
over	incl.	high	low	max.	max.	max.	max.	high	low
18	30	0	-12	12	9	40	18	0	-100
30	50	0	-14	14	11	40	20	0	-100
50	80	0	-16	16	12	40	25	0	-100
80	120	0	-18	18	14	45	35	0	-100
120	150	0	-20	20	15	50	40	0	-100
150	180	0	-25	25	19	60	45	0	-100
180	250	0	-30	30	23	70	50	0	-100

BEARING WIDTHS

d (mm)		Δ_{Ts}		Δ_{T1s}		Δ_{T2s}	
over	incl.	high	low	high	low	high	low
10	18	100	0	50	0	50	0
18	30	100	0	50	0	50	0
30	50	100	0	50	0	50	0
50	80	100	0	50	0	50	0
80	120	100	0	50	0	50	0

METRIC TAPERED ROLLER BEARINGS

TOLERANCE CLASS P5

INNER RING

Tolerance in µm.

d (mm)		Δ_{dmp}		V_{dp}	V_{dmp}	S_{ia}	K_{ia}	S_{d}	Δ_{Bs}	
over	incl.	high	low	max.	max.	max.	max.	max.	high	low
10	18	0	-7	5	5	8	7	7	0	-200
18	30	0	-8	6	5	8	8	8	0	-200
30	50	0	-10	8	5	8	10	8	0	-240
50	80	0	-12	9	6	10	10	8	0	-300
80	120	0	-15	11	8	10	13	9	0	-400

OUTER RING

Tolerance in µm.

D (mm)		Δ_{Dmp}		V_{Dp}	V_{Dmp}	S_{ea}	K_{ea}	S_{D}	Δ_{Cs}	
over	incl.	high	low	max.	max.	max.	max.	max.	high	low
18	30	0	-8	6	5	20	6	8	Identical to Δ_{Bs} of inner ring of same bearing	
30	50	0	-9	7	5	20	7	8		
50	80	0	-11	8	6	20	8	8		
80	120	0	-13	10	7	22	10	9		
120	150	0	-15	11	8	25	11	10		
150	180	0	-18	14	9	30	13	10		
180	250	0	-20	15	10	35	15	11		

BEARING WIDTHS

d (mm)		Δ_{Ts}	
over	incl.	high	low
10	18	200	-200
18	30	200	-200
30	50	200	-200
50	80	200	-200
80	120	200	-200

INCH TAPERED ROLLER BEARINGS

TOLERANCE

INNER RING

µm.

Tolerance class	d (mm)		Δds		Vdp	$Vdmp$	Kia	Sia	ΔBs		ΔTs	
	over	incl.	high	low	max.	max.	max.	max.	high	low	high	low
4	—	76.200	+13	0	11	9	51	51	+76	-254	+203	0
4	76.2	101.600	+25	0	15	13	51	51	+76	-254	+203	0
4	101.600	152.400	+25	0	18	15	51	51	+76	-254	+356	-254
2	—	76.200	+13	0	11	9	38	38	+76	_254	+203	0
2	76.200	152.400	+25	0	15	13	38	38	+76	-254	+203	0
3	—	152.400	+13	0	11	9	8	8	+76	-254	+203	-203
0	—	152.400	+8	0	6	5	4	4	+76	-254	+203	-203
00	—	152.400	+8	0	6	5	2	2	+76	-254	+203	-203

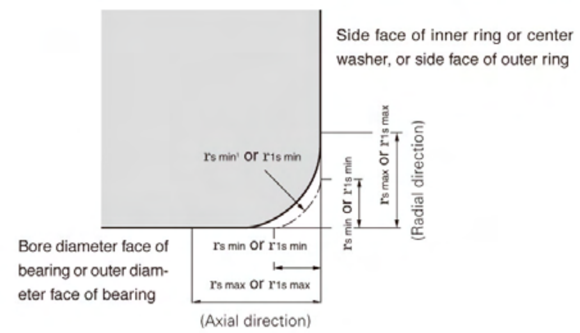
OUTER RING

µm.

Tolerance class	D (mm)		ΔDs		VDp	$VDmp$	Kea	Sea	ΔCs	
	over	incl.	high	low	max.	max.	max.	max.	high	low
4	—	266.700	+25	0	15	13	51	51	+51	-254
4	266.700	304.800	+25	0	18	15	51	51	+51	-254
2	—	266.700	+25	0	15	13	38	38	+51	-254
2	266.700	304.800	+25	0	15	13	38	38	+51	-254
3	—	266.700	+13	0	11	9	8	8	+51	-254
3	266.700	304.800	+13	0	11	9	8	8	+51	-254
0	—	266.800	+13	0	11	9	4	4	+51	-254
0	266.700	304.800	+8	0	6	5	4	4	+51	-254
00	—	266.800	+8	0	6	5	2	2	+51	-254

Inch-dimension equivalent formula	1 inch = 25.4 mm
	1 mm = 0.0393700787 inch

Chamfer measurements and tolerance



Allowable critical-value of bearing chamfer
Deep groove ball bearings

Unit mm

$r_{s\ min}$[1] or $r_{1s\ min}$	Nominal bore diameter d		$r_{s\ max}$ or $r_{1s\ max}$	
	over	incl	Radial direction	Axial direction
0.1	—	—	0.2	0.4
0.15	—	—	0.3	0.6
0.2	—	—	0.5	0.8
0.3	—	40	0.6	1
	40	—	0.8	1
0.6	—	40	1	2
	40	—	1.3	2
1	—	50	1.5	3
	50	—	1.9	3
1.1	—	120	2	3.5
	120	—	2.5	4
1.5	—	120	2.3	4
	120	—	3	5
2	—	80	3	4.5
	80	220	3.5	5
	220	—	3.8	6
2.1	—	280	4	6.5
	280	—	4.5	7
2.5	—	100	3.8	6
	100	280	4.5	6
	280	—	5	7
3	—	280	5	8
	280	—	5.5	8

Notes:
These are the allowable minimum dimensions of the chamfer dimension "r" and are described in the dimensional table.

Tapered roller bearings of metric system

Unit mm

$r_{s\ min}$[2] or $r_{1s\ min}$	Nominal bore diameter of bearing "d" or nominal outside diameter "D"		$r_{s\ max}$ or $r_{1s\ max}$	
	over	incl	Radial direction	Axial direction
0.3	—	40	0.7	1.4
	40	—	0.9	1.6
0.6	—	40	1.1	1.7
	40	—	1.3	2
1	—	50	1.6	2.5
	50	—	1.9	3
1.5	—	120	2.3	3
	120	250	2.8	3.5
	250		3.5	4
2	—	120	2.8	4
	120	250	3.5	4.5
	250	—	4	5
2.5	—	120	3.5	5
	120	250	4	5.5
	250	—	4.5	6
3	—	120	4	5.5
	120	250	4.5	6.5
	250	400	5	7
	400	—	5.5	7.5
4	—	120	5	7
	120	250	5.5	7.5
	250	400	6	8
	400	—	6.5	8.5

Notes:
These are the allowable minimum dimensions of the chamfer dimension "r" or "r1" and are described in the dimensional table.

LIFE & LOADS

When in service, even a bearing that is properly lubricated, properly installed, and adequately protected from abrasives, moisture, and corrosive reagents, can fail from material fatigue. Material fatigue is manifested as a flaking off of metallic particles from the surface of a raceway or rolling element. This flaking will eventually cause the bearings to fail. The effective life of a bearing is usually defined in terms of the total number of revolutions a bearing can undergo before flaking of either the raceway surface or the rolling element surfaces occurs.

● Machine applications and requisite life

The requisite life of the bearing is usually determined by the type of machine in which the bearing will be used, duration of service and operational reliability requirements. A general guide to these requisite life criteria is shown in the following table.

● Machine applications and requisite life

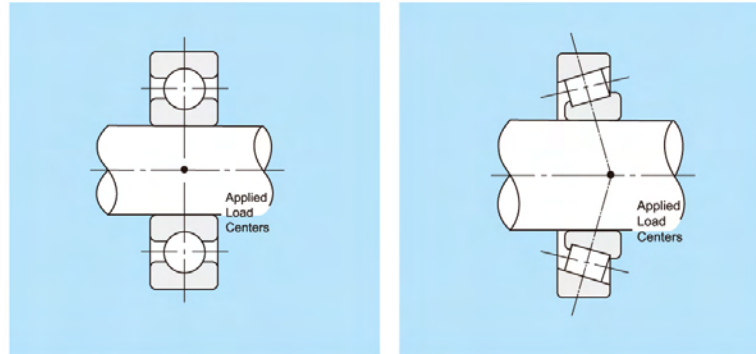
Operating condition	Application	Recommended service life
Short or intermittent operation	Household electric appliance, electric tools, agricultural equipment, heavy cargo hoisting equipment	4000 – 8000
Not extended duration, but stable operation required	Household air conditioner motors, construction equipment, conveyers, elevators	8000 – 12000
Intermittent but extended operation	Rolling mill roll necks, small motors, cranes	8000 – 12000
	Motors used in factories, general gears	12000 – 20000
	Machine tools, shaker screens, crushers	20000 – 30000
	Compressors, pumps, gears for essential use	40000 – 60000
Daily operation more than 8 hr.or continuous extended operation	Escalators	12000 – 20000
	Centrifugal separators, air conditioners, air blowers, woodworking equipment, passenger coach axle journals	20000 – 30000
	Large motors, mine hoists, locomotive axle journals, railway rolling stock traction motors	40000 – 60000
	Paper manufacturing equipment	100000 – 200000
24 hr.operation (no failure allowed)	Water supply facilities, power stations, mine water discharge facilities	100000 – 200000

● Service life of bearing system comprising two or more bearings

Even for systems which comprise two or more bearings, if one bearing is damaged, the entire system malfunctions. Where all bearings used in an application are regarded as one system, the service life of the bearing system can be calculated using the following equation.

$$\frac{1}{L^e} = \frac{1}{L_1{}^e} + \frac{1}{L_2{}^e} + \frac{1}{L_3{}^e}$$

where:
L:rating life of system
L_1, L_2, L_3……: rating life of each bearing
e :constant
e=10/9……ball bearing
e=9/8……roller bearing
The mean value is for a system using both ball and roller bearings.

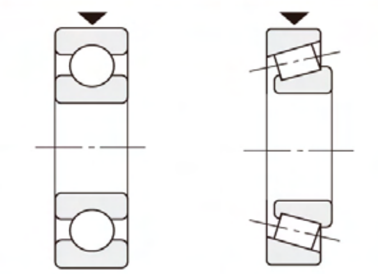
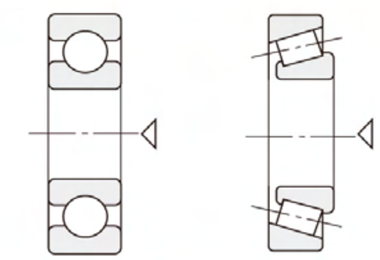
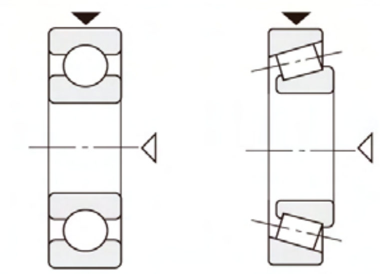


● Bearing loads

When calculating bearing load using the loads on a position on the shaft, it is necessary to calculate center distance between the load application points of the bearings. Deep groove ball bearings have load center points at the center line of the width. Single-row tapered roller bearings, have load center points off-center to the center line of the bearing width.

● Magnitude of load

The magnitude of the load is one of the factors that usually determine the size of the bearing. Generally, roller bearings are able to support heavier loads than similar sized ball bearings and bearings having a full complement of rolling elements can accommodate heavier loads than the corresponding caged bearings. Ball bearings are mostly used where loads are light or moderate. For heavy loads and where shaft diameters are large, roller bearings are usually the more appropriate choice.

Radial load	Axial load	Combined load
		
Bearings can only support pure radial loads.	Bearings are suitable for light or moderate loads that are purely axial.	A combined load comprises a radial and axial load acting simultaneously.

● Load acting on shafts

It is possible to calculate theoretical values for these loads. However, there are many instances where the actual operational shaft load is much greater than the theoretically calculated load, due to machine vibration and/or shock. This actual shaft load can be found by using formula

$$K = f_w \cdot K_c$$

where,
K : Actual shaft load {kgf}
f_w : Load factor
K_c: Theoretically calculated value {kgf}

● Load factor f_w

Amount of shock	f_w	Application
Heavy shock	1.0-1.2	Electric machines, machine tools, measuring instruments.
Light shock	1.2-1.5	Railway vehicles, automobiles, rolling mills, metal working machines, paper making machines, rubber mixing machines, printing machines, aircraft, textile machines, electrical units, office machines.
Very little or no shock	1.5-3.0	Crushers, agricultural equipment, construction equipment, cranes.

Basic rating life and basic dynamic load rating

The basic dynamic load rating is an expression of the load capacity of a bearing based on a constant load which the bearing can sustain for 1,000,000 revolutions (the basic rating life). For radial bearings this rating applies to pure radial loads. The common measurement is "L_{10}" life, defined as the number of revolutions before metal fatigue first appears on 10% of a large group of like bearings. This is referred to as basic rating life of fatigue life.The relationship between the basic rating life, the basic dynamic load rating and the bearing load is given in following formula:

$$L_{10}=\left(\frac{C_r}{P}\right)^p$$

where,
$p = 3$ for ball bearings
$p = 10/3$ for roller bearings
L_{10} : Basic rating life (at 90% reliability), millions of revolutions
C_r: Basic dynamic load rating, N or kgf
P: Equivalent dynamic load, N or kgf

If the speed is constant, the basic rating life can also be expressed in terms of hours of operation (revolution), and is calculated as shown in the formula below:

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

where,
n: Rotational speed, rpm

The relationship between rotational speed "n" and speed factor as well as the relation between the basic rating life and the life factor is shown as following:

$$L_{10h}=500\,f_h^{\,p}$$

$$f_h=f_n\frac{C}{P}$$

$$f_n=\left(\frac{33.3}{n}\right)^{1/P}$$

where,
L_{10h} : Basic rating life
f_h : Life factor
f_n : Speed factor

Ball bearing	
L_{10h}	$\frac{10^6}{60n}\left(\frac{C_r}{P}\right)^3=500f_h^{\,3}$
f_h	$f_n\frac{C}{P}$
f_n	$\left(\frac{33.3}{n}\right)^{1/3}$

Roller bearing	
L_{10h}	$\frac{10^6}{60n}\left(\frac{C_r}{P}\right)^{10/3}=500f_h^{\,10/3}$
f_h	$f_n\frac{C}{P}$
f_n	$\left(\frac{33.3}{n}\right)^{3/10}$

Example 1: What is the rating life in hours of operation (L10h) for deep groove ball bearing 6208 operating at 650 r/min, with a radial load Fr of 3.2 kN ?

$P_r = F_r = 3.2\text{kN}\ \{326\text{kgf}\}$

$f_h = f_n\frac{C_r}{P_r} = 0.37 \times \frac{29.52}{3.2} = 3.41$

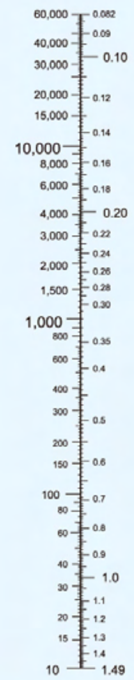
The basic dynamic rated load for bearing 6208 (from bearing table) is 29.52 kN, and the speed factor (fn) for ball bearings at 650 r/min (n) from the table in the following page is 0.37. The life factor, fh, from formula is:

Therefore, with fh = 3.41 from the table in the following page, the rated life, L10h, is approximately 20,000 hours.

● Correlation of bearing basic rating life, life factor and speed factor (Ball bearing)

Ball bearings

n r/min f_n



n	f_n	n	f_n
10	1.49	600	0.382
12	1.41	700	0.362
14	1.34	800	0.347
16	1.28	900	0.333
18	1.23	1000	0.322
20	1.19	1200	0.303
30	1.04	1400	0.288
40	0.941	1600	0.275
50	0.874	1800	0.265
60	0.822	2000	0.255
70	0.781	3000	0.223
80	0.747	4000	0.203
90	0.718	5000	0.188
100	0.693	6000	0.177
120	0.652	7000	0.168
140	0.62	8000	0.161
160	0.593	9000	0.155
180	0.57	10000	0.149
200	0.55	12000	0.141
220	0.533	14000	0.134
240	0.518	16000	0.128
260	0.504	18000	0.123
280	0.492	20000	0.119
300	0.481	30000	0.104
400	0.437	40000	0.0941
500	0.405	50000	0.0874

Ball bearing

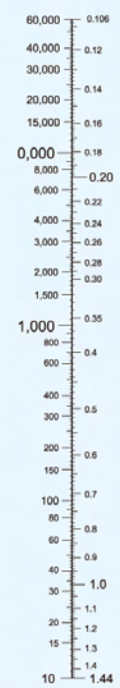
L_{10h} n f_h

L_{10h}	f_h	L_{10h}	f_h
100	0.585	2800	1.78
120	0.621	3000	1.82
140	0.654	4000	2
160	0.684	5000	2.15
180	0.711	6000	2.29
200	0.737	7000	2.41
220	0.761	8000	2.52
240	0.783	9000	2.62
260	0.804	10000	2.71
280	0.824	12000	2.88
300	0.843	14000	3.04
400	0.928	16000	3.17
500	1	18000	3.3
600	1.06	20000	3.42
700	1.12	22000	3.53
800	1.17	24000	3.63
900	1.22	26000	3.73
1000	1.26	28000	3.83
1200	1.34	30000	3.91
1400	1.41	40000	4.31
1600	1.47	50000	4.64
1800	1.53	60000	4.93
2000	1.59	70000	5.19
2200	1.64	80000	5.43
2400	1.69	90000	5.65
2600	1.73	100000	5.85

● Correlation of bearing basic rating life, life factor and speed factor (Roller bearing)

Roller bearings

n r/min f_n



n	f_n	n	f_n
10	1.44	600	0.42
12	1.36	700	0.401
14	1.3	800	0.385
16	1.25	900	0.372
18	1.2	1000	0.36
20	1.17	1200	0.341
30	1.03	1400	0.326
40	0.947	1600	0.313
50	0.885	1800	0.302
60	0.838	2000	0.293
70	0.8	3000	0.259
80	0.769	4000	0.238
90	0.742	5000	0.222
100	0.719	6000	0.211
120	0.681	7000	0.201
140	0.65	8000	0.193
160	0.625	9000	0.186
180	0.603	10000	0.181
200	0.584	12000	0.171
220	0.568	14000	0.163
240	0.553	16000	0.157
260	0.54	18000	0.151
280	0.528	20000	0.147
300	0.517	30000	0.13
400	0.475	40000	0.119
500	0.444	50000	0.111

Roller bearings

L_{10h} n f_h

L_{10h}	f_h	L_{10h}	f_h
100	0.617	2800	1.68
120	0.652	3000	1.71
140	0.683	4000	1.87
160	0.71	5000	2
180	0.736	6000	2.11
200	0.76	7000	2.21
220	0.782	8000	2.3
240	0.802	9000	2.38
260	0.822	10000	2.46
280	0.84	12000	2.59
300	0.858	14000	2.72
400	0.935	16000	2.83
500	1	18000	2.93
600	1.06	20000	3.02
700	1.11	22000	3.11
800	1.15	24000	3.19
900	1.19	26000	3.27
1000	1.23	28000	3.35
1200	1.3	30000	3.42
1400	1.36	40000	3.72
1600	1.42	50000	3.98
1800	1.47	60000	4.2
2000	1.52	70000	4.4
2200	1.56	80000	4.58
2400	1.6	90000	4.75
2600	1.64	100000	4.9

Correction of basic load rating

● Correction due to temperature

The operating temperature will significantly affect the fatigue life by altering the hardness of the bearing. Consequently, the basic load rating, which depends on the physical properties of the bearing material, will decrease with higher temperatures. Thus, the basic load rating must be corrected for higher temperatures using the equation:

$$C_t = f_t * C$$

where,
C_t: Basic load rating after temperature correction
f_t: Temperature factor (see following table)
C: Basic load rating, before application of temperature correction.

	Bearing Temperature (℃)			
	≤150℃	175℃	200℃	250℃
Temperature Factor f_t	1.00	0.95	0.90	0.75

● Adjustment to fatigue life rating

Some common applications require a bearing that can handle misalignment, loads in both directions, high speeds, etc., or a combination of two or more. These operating conditions will alter the bearing life and are accounted for by using correction factors for temperature, reliability, bearing material, and other operating conditions. The formula for adjusting life based on reliability, material, and operating conditions is:

$$L_{na} = a_1 * a_2 * a_3 * L_{10}$$

Where,
L_{na}: Adjusted life rating
a_1: Life correction factor for reliability. This is determined from the reliability required of the bearing for its application (see table right).
a_2: Life correction factor for bearing material
a_3: Life correction factor for operating conditions

(1) Reliability coeffcient a1

The table on the right side describes reliability coefficient, a1, which is necessary to obtain the corrected rating life of reliability greater than 90%.

L_{10}: Life rating, adjusted for fatigue life for 90% reliability. This may not satisfy all applications. For higher reliability requirements, L_{10} must be adjusted.

Reliability coefficient

	Reliability		
	90%	95%	96%
a_1	1.00	0.62	0.53
	97%	98%	99%
a_1	0.44	0.33	0.21

(2) Bearing characteristic coefficient a2

The bearing characteristic in relation to bearing life may differ according to bearing materials(steel types and their quality), and may be altered by production process, design, etc. In such cases, the bearing life calculation can be corrected using the bearing characteristic coefficient a2.

HCH has employed vacuum-degassed bearing steel as HCH standard bearing material. It has a significant effect on bearing life extension which was verified through studies at HCH laboratory. The basic dynamic load rating of bearings made of vacuum-degassed bearing steel is specified in the bearing specification table, taking the bearing characteristic coefficient as a2=1. For bearings made of special materials to extend fatigue life, the bearing characteristic coefficient is treated as a2>1

(3) Operating condition coefficient a3

When bearings are used under operating conditions which directly affect their service life including improper lubrication, the service life calculation can be corrected by using a3. Under normal lubrication, the calculation can be performed with a3=1; and, under favorable lubrication,with a3>1. In the following cases, the operating condition coefficient is treated as a3<1

Operating lubricant of low kinematic viscosity (Ball bearing……13mm²/s or less, Roller bearing……20 mm2²/s or less)
Operation at very slow rotational speed (Product of rolling element pitch diameter and rotational speed is 10000 or less)
Contamination of lubricant is expected
Greater misalignment of inner and outer rings is present

Note: Since a2 and a3 are inter-dependent, some calculations treat them as one coefficient,a23. Due to difficult of value determination, for most applications, a23=1

● Factor η_c for contamination level

This factor was introduced to consider the contamination level of the lubricant in the bearing life calculation. The influence of contamination on bearing fatigue depends on a number of parameters including bearing size, relative lubricant film thickness, size and distribution of solid contaminant particles, types of contamination (soft, hard etc). The influence of these parameters on bearing life is complex and many of the parameters are difficult to quantify. It is therefore not possible to allocate precise values to η_c that would have general validity. However, some guideline values are provided in the following table.

Guideline values of factor η_c for different levels of contamination Condition	Facor η_c 1) for bearings with diameter d_m<100mm	 d_m ≥ 100mm
Extreme cleanliness Particle size of the order of the lubricant film thickness Laboratory conditions	1	1
High cleanliness Oil filtered through an extremely fine filter Conditions typical of bearings greased for life and sealed	0,8 ... 0,6	0,9 ... 0,8
Normal cleanliness Oil filtered through a fine filter Conditions typical of bearings greased for life and shielded	0,6 ... 0,5	0,8 ... 0,6
Slight contamination Slight contamination of the lubricant	0,5 ... 0,3	0,6 ... 0,4
Typical contamination Conditions typical of bearings without integral seals, coarse filtering, wear particles and ingress from surroundings	0,3 ... 0,1	0,4 ... 0,2
Severe contamination Bearing environment heavily contaminated Arrangement with inadequate sealing	0,1 ... 0	0,1 ... 0
Very severe contamination Under extreme contamination, values of η_c can be outside the scale resulting in a more severe reduction of life than predicte	0	0

ISO classification - allocation of scale number Number of particles per millilitre oil over	 incl.	 Scale number
2 500 000		>28
1 300 000	2 500 000	28
640 000	1 300 000	27
320 000	640 000	26
160 000	320 000	25
80 000	160 000	24
40 000	80 000	23
20 000	40 000	22
10 000	20 000	21
5 000	10 000	20
2 500	5 000	19
1 300	2 500	18
640	1 300	17
320	640	16
160	32	15
80	160	14
40	80	13
20	40	12
10	20	11
5	10	10
2,5	5	9
1,3	2,5	8
0,64	1,3	7
0,32	0,64	6
0,16	0,32	5
0,08	0,16	4
0,04	0,08	3
0,02	0,04	2
0,01	0,02	1
0,00	0,01	0

Note:
1) This approach will probably indicate only an approximate value of the effective factor η_c for the contamination level of the application.
2) The scale for η_c refers only to typical solid contaminants, Contamination by water or other fluids detrimental to bearing life in not included. In case of very heavy contamination (η_c=0), failure will be caused by wear, the useful life of the bearing can be shorter than the rated life.

Dynamic equivalent load

When both dynamic radial loads and dynamic axial loads act on a bearing at the same time, the hypothetical load acting on the center of the bearing which gives the bearings the same life as if they had only a radial load or only an axial load is called the dynamic equivalent load. For radial bearings, this load is expressed as pure radial load and is called the dynamic equivalent radial load. The dynamic equivalent radial load is expressed by formula below:

$$P = XF_r + YF_a$$

where,
P: Equivalent dynamic radial load, N or kgf
F_r: Actual radial load, N or kgf
F_a: Actual axial load, N or kgf
X: Radial load factor
Y: Axial load factor

The values for X and Y are listed in the bearing tables

Equivalent dynamic P=XFr+YFa

$\frac{f_0 F_a}{C_{0r}}$	e	$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$	
		X	Y	X	Y
0.172	0.19	1	0	0.56	2.30
0.345	0.22	1	0	0.56	1.99
0.689	0.26	1	0	0.56	1.71
1.03	0.28	1	0	0.56	1.55
1.38	0.30	1	0	0.56	1.45
2.07	0.37	1	0	0.56	1.31
3.45	0.38	1	0	0.56	1.15
5.14	0.42	1	0	0.56	1.04
6.89	0.44	1	1	0.56	1.00

In the table, the "e" value is definded as axial load influence factor. When $F_a/F_r > e$, it means that the axial load influences a lot, and the equivalent radial load P should be calculated by the formula $P=XFr+YFa$. Otherwise, when $F_a/F_r \le e$, the influence of the axial load can be ignored, here $P=Fr$. The "e" value is determined by multiplying the axial load applied to the bearing by the bearing coefficient factor "f_0", which is obtained from the bearing table above.

Example 2: What is the life rating L10h for the same bearing and conditions as in Example 1, but with an additional axial load Fa of 1.8 kN ?

To find the dynamic equivalent radial load value for Pr, the radial load factor X and axial load factor Y are used.The basic static load rating, Cor, for bearing 6208 is 18.14 from the bearing table.

$$\frac{f_0 F_a}{C_{or}} = \frac{14\text{x}1.8}{18.14} = 1.37$$

Therefore, from the bearing tables e= 0.29. For the operating radial load and axial load:

$$\frac{F_a}{F_r} = \frac{1.8}{3.2} = 0.56 > e = 0.29$$

From the bearing tables X = 0.56 and Y = 1.48, and from formula the equivalent radial load, Pr, is:

$$P_r = XF_r + YF_a = 0.56\text{x}3.2 + 1.48\text{x}1.8 = 4.46 \text{ kN } \{455\text{kgf}\}$$

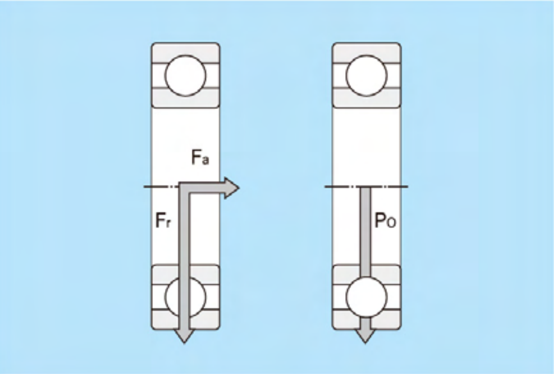
Therefore, with life factor fh = 2.45, from the table, the rated life, L10h, is approximately 7,500 hours.

$$F_h = F_h \frac{C_r}{P_r} = 0.37\text{x}\frac{29.52}{4.46} = 2.45$$

Static bearing load

The basic static load rating "C_o" is used in calculations when the bearings are to:
– rotate at very slow speeds (n < 10 r/min).
– perform very slow oscillating movements.
– be stationary under load for certain extended periods.

It is also most important to check the safety factor of short duration loads such as shock or heavy peak loads that act on a bearing, whether it is rotating (dynamically stressed) or at rest.



● Static equivalent load

The static equivalent load is a hypothetical load which would cause the same total permanent deformation at the most heavily stressed contact point between the rolling elements and the raceway under the actual load conditions where both static radial load and static axial load are simultaneously applied to a bearing. For a radial bearing, this hypothetical load refers to pure radial load. For radial bearings the static equivalent radial load can be found by using formula below. The greater of the two resultant values is always taken for "P_{or}".

$$P_{or} = X_o F_r + Y_o F_a$$

$$P_{or} = F_r$$

Where,
P_{or}: Static equivalent radial load, N or kgf
F_r : Actual radial load, N or kgf
F_a : Actual axial load, N or kgf
X_o : Static radial load factor
Y_o : Static axial load factor

● Required basic static load rating

The basic static load rating is considered as the limiting load for general applications. In terms of a safety factor, this means that, by definition, a safety factor "S_o" is set as a base of 1. An application may require a larger or allow a smaller safety factor provides a guide for selection of the safety factor "S_o", to be used with formula for calculation of the maximum (weighted) static equivalent load.

$$C_o = S_o \cdot P_o\text{max}$$

where,
C_o: Basic static load rating, KN
P_o: Equivalent static bearing load, KN
S_o: Static safety factor

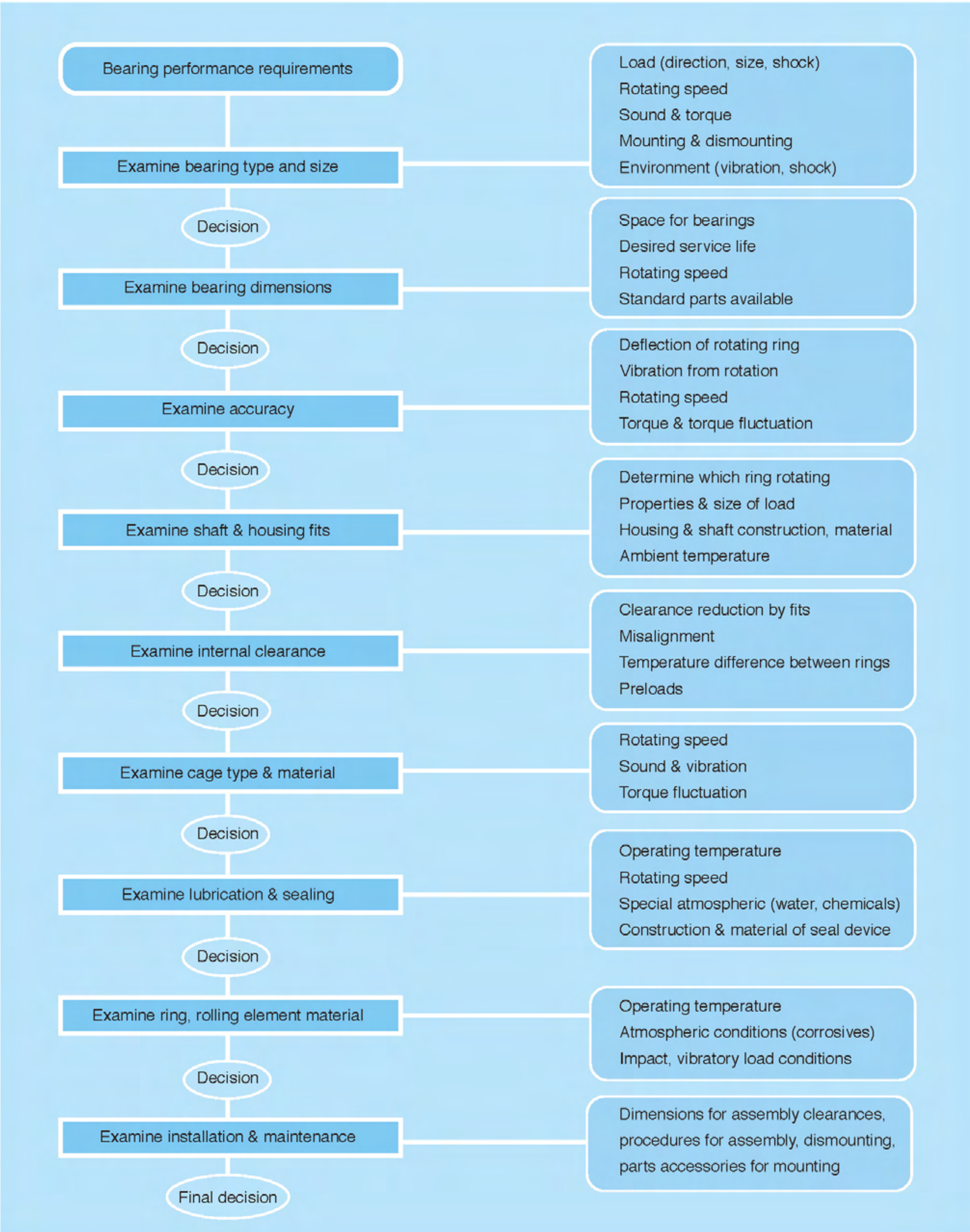
Guideline values for the static safety factor S_o

Type of operation	Rotating bearing Requirements regarding quiet running						Non-rotating bearing	
	unimportant		normal		high			
	Ball bearing	Roller bearing	Ball bearing	Roller bearing	Ball bearing	Roller bearing	Ball bearing	Roller bearing
Smooth, vibration-free	0,5	1	1	1,5	2	3	0,4	0,8
Normal	0,5	1	1	1,5	2	3,5	0,5	1
Pronounced shock loads	≥1,5	≥2,5	≥1,5	≥3	≥2	≥4	≥1	≥2

Where the magnitude of the shock load is not known, values of "S_o" at least as large as those quoted above should be used. If the magnitude of the shock loads is exactly known, smaller values of "S_o" can be applied.

BEARING SELECTION SERVICES

Machine performance and service life largely depend on which bearings are selected, it is often difficult to select the optimal bearing from among the many available variations. The diagram below provides an example of a procedure based on the establishment of priorities for the required bearing characteristics.



Bearing replacement

Ordering the correct replacement bearing is a critical task - but it is not difficult if you take time to gather the right information. Please follow these steps:

First - IDENTIFY the type of bearing you need to replace.

Ball Bearing - Deep groove ball bearing
Roller Bearing - Tapered roller bearing
Split Pillow Block - Pillow Blocks

Second - LOCATE the identification number on the bearing.

Bearing identification numbers are usually located on the inner ring face, outer ring face, bearing O.D. or seals. Mounted units are identified by a number tag fastened to the unit or by a housing number cast into the housing cap.

Third - MEASURE if you need to.

If a bearing identification number is not legible, you will need to determine the following:

1. Inner ring bore (inside diameter)
2. Outer ring outside diameter
3. Inner width and outer width (these may be different)
4. Shape of the bore and/or outside diameter of bearing

Fourth - RECORD additional relevant information.

The more information available, the easier it will be to identify the replacement bearing needed. Record:

1. Unique features such as machined shoulders, etc.
2. Application/equipment data

Fifth - LOOK in the appropriate section of the catalog.

1. Ball bearings
2. Tapered roller bearings
3. Engineering information, for all bearing types

Notes: If you are still unable to identify the bearing you need, call your HCH distribution center.

Deep groove ball bearing number formulation

Bearings are identified by numbers and letters which designate bearing type, boundary dimensions, tolerance class, internal clearance and other specifications.

S	F	608	.3122	2RS	NR	C3	EMQ	A3	SRL	C
1	2	3	4	5	6	7	8	9	10	11

Material Codes

Blank	-GCr 15 Chrome Steel
S	-440 ℃ Stainless Steel

Ring/OD Modification Codes

F	- Flanged Outer Ring
W	- Extended Inner Ring
FW	- Flanged Outer/Extended Inner Rings

Basic Bearing Number Codes

ID Modification Codes

.3122	- .3122 inch Bore
13	- 13 mm Bore
5/8	- 5/8 inch Bore
16	- 16 mm Bore
3/4	- 3/4 inch Bore
7/8	- 7/8 inch Bore

Shield/Closure Codes

Blank	- Open Bearing
Z	- Single Shield
ZZ	- Double Shields
RS	- Single Seal
2RS	- Double Seals

Snap Ring Codes

N	- Groove without Ring
NR	- Groove with Ring

Grease fill volume

K	- Fill volume 20%
S	- Fill volume 30%
M	- Fill volume 50%
F	- Fill volume 60%
N	- Fill volume 80%

Grease Type

Kyodo yushi SRL2
Shell AV2
Cosmo ENMQ2
Exxon Mobil Polyrex EM
Chevron SRI2
Kluber Asonic GLY 32

ABEC Tolerance Codes

Blank	- ABEC 1
A3	- ABEC 3
A5	- ABEC 5
A7	- ABEC 7

Vibration Level Codes

Blank	- Standard
EMQ	- Electric Motor Quality

Internal Clearance Codes

Blank	- Standard
C2	- Tight
C3	- Loose
C4	- Looser than C3
C5	- Looser than C4

Note: The boundary dirmension of HCH deep groove ball bearing are in accordance with ISO 15:1998.

Tapered roller bearing number formulation

The basic number indicates general information about a bearing such as its fundamental type, boundary dimensions, series number, bore diameter code and contact angle. The supplementary code derive from profixes and suffixes which indicate a bearing's related specifications.

● Nomenclature-Metric Tapered Roller Bearings

series

30200	32200
30300	32300
31300	32900
32000	33000
	33100
	33200

Bore Size
(04 and up：multiply last two numbers by 5 to get bore in mm)

02:12mm	08: 40mm
03:17mm	10: 50mm
04:20mm	20: 100mm

Special bore sizes are as follows：

/22:22mm	/32: 32mm
/28:28mm	

3 2 2 10

Note: Metric tapered roller bearings follow ISO0355-1977 and ANSI/ABMA standard 19.1-1987, which is similar to the system used for inch tapered roller bearings ANSI/ABMA standard 19.2-1994.

● Nomenclature - Inch Tapered Roller Bearings

Prefix

EH:	Extra heavy
EL:	Extra light
H:	Heavy
HH:	Heavier than heavy
HM:	Heavy medium
J:	Metric designed
L:	Light
LL:	Lighter than light
LM:	Light medium
M:	Medium

Code	Included Cup Angle	
1	0	to 23° 59'59.99"
2	24°	to 25° 29'59.99"
3	25° 30'	to 26° 59'59.99"
4	27°	to 28° 29'59.99"
5	28°30'	to 30° 29'59.99"
6	30°30'	to 32° 29'59.99"
7	32°30'	to 35° 59'59.99"
8	36°	to 44° 59'59.99"
9	45° up, but not thrust only	

Design Number
2 digits which identify the cup or cone

LM 1 19 49

Basic Series
1-3 digit number that identifies maximum bore range

Please refer to the bearing tables for exact part number options.

Interchange

Description		Bearing Interchange					
		HCH	SKF	FAG	TORR	SNR	Russia
Part Number Suffix	Open	BLANK	BLANK	BLANK	BLANK	BLANK	BLANK
	One shield	Z	Z(Z)	ZR(Z)	D	Z	60
	Two shields	ZZ	ZZ(ZZ)	2ZR(2Z)	DD	ZZ	160
	One seal(contact)	RS (RS$_1$)	RS1(RSH)	RSR	P	E	80
	Two seals(contact)	2RS(2RS$_1$)	2RS1(2RSH)	2RSR	PP	EE	180
	One seal(non contact)	RZ	RZ(RSL)	RSD	--	--	--
	Two seals(non contact)	2RZ	2RZ(2RSL)	2RSD	--	--	--
	Tight clearance	C2	C2	C2	H	J 20	C2
	Normal clearance	BLANK/C0/CN	BLANK/C0/CN	BLANK/C0/CN	R	BLANK	BLANK/C0/CN
	Larger than normal clearance	C3	C3	C3	P	J 30	C3
	Large clearance	C4	C4	C4	J	J40	C4
	Miniature bearing clearance classification	MC1	--	--	--	--	--
		MC2	--	--	--	--	--
		MC3	--	--	--	--	--
		MC4	--	--	--	--	--
	MC5	--	--	--	--	--	
	Electric Motor Quality	EMQ	QE6	--	--	--	--
	Clearance for electric motor	CM	--	--	--	--	--
	Snap groove	N	N	N	N	N	N
	Snap ring	NR	NR	NR	NR	NR	NR
	Steel Cage	BLANK	BLANK	BLANK	BLANK	BLANK	BLANK
	Brass Cage	M	M	M	M	M	M
	Deep Groove ball Bearing Type	Rxx	Rxx	Rxx	Sxx	--	--
		6xx	6xx	6xx	3x	6xx	--
		60xx	60xx	60xx	91xxK	60xx	1xx
		62xx	62xx	62xx	2xx	62xx	2xx
		63xx	63xx	63xx	3xx	63xx	3xx
		64xx	64xx	64xx	--	--	--
		67xx	--	--	--	--	--
		68xx	618xx	68xx	--	--	--
		69xx	69xx	619xx	93xxK	--	--
		160xx	160xx	160xx	--	--	--
		622xx	622xx	622xx	--	--	--
		623xx	623xx	623xx	--	--	--
		630xx	630xx	630xx	--	--	--

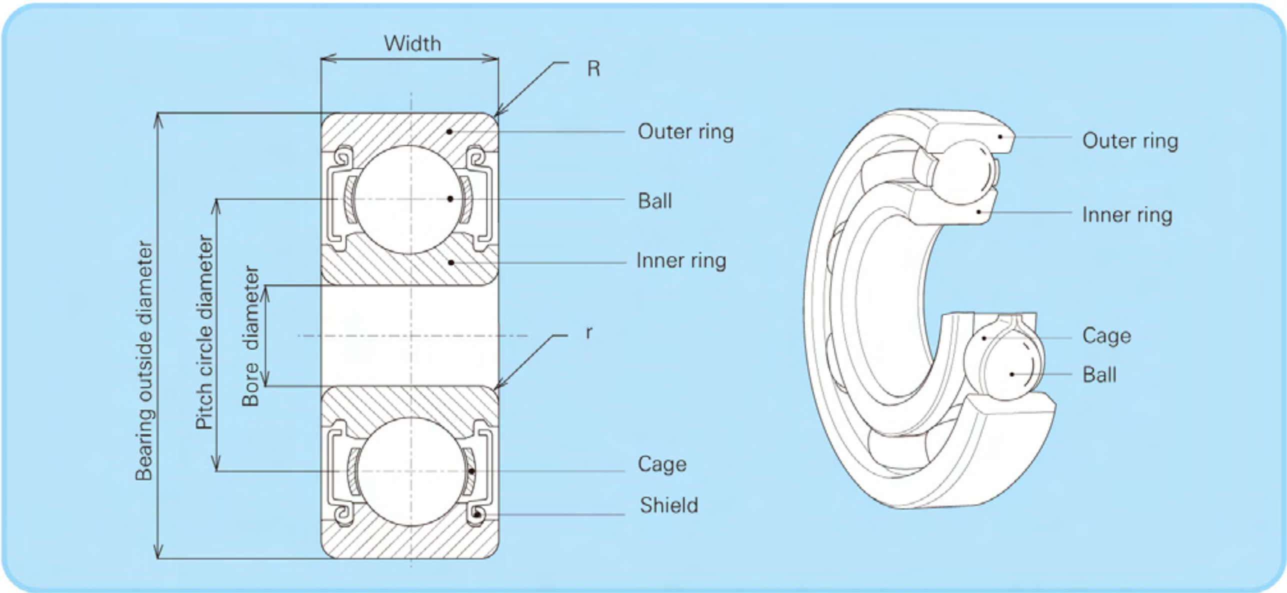
Note: 1. Above interchange information is for reference purpose only.
2. SKF contact seal with RSH means inner ring with groove.
3. SKF contact seal with RS1 means inner ring without groove.
4. SKF non contact seal with RSL means inner ring with groove.
5. SKF non contact seal with RZ means inner ring without groove.

Description		Bearing Interchange					
		HCH	NSK	NTN	KOYO	NACHI	NMB
Part Number Suffix	Open	BLANK	BLANK	BLANK	BLANK	BLANK(BLANK)	BLANK
	One shield	Z	Z	Z	Z	ZE(Z)	Z
	Two shields	ZZ	ZZ	ZZ	ZZ	2ZE(ZZ)	ZZ
	One seal(contact)	RS(RS$_1$)	DU	LU	RS(RK,RD)	NSE(NSL)	D
	Two seals(contact)	2RS(2RS$_1$)	DDU	LLU	2RS(2RK,2RD)	2NSE(2NSL)	DD
	One seal(non contact)	RZ	V	LB	RU	NKE(NK)	S
	Two seals(non contact)	2RZ	VV	LLB	2RU	2NKE(2NK)	SS
	Tight clearance	C2	C2	C2	C2	C2	C2
	Normal clearance	BLANK/C0/CN	BLANK/C0/CN	BLANK/C0/CN	BLANK/C0/CN	BLANK/C0/CN	BLANK/C0/CN
	Larger than normal clearance	C3	C3	C3	C3	C3	C3
	Large clearance	C4	C4	C4	C4	C4	C4
	Miniature bearing clearance classification	MC1	MC1	--	MC1	C1P	MC1
		MC2	MC2	--	MC2	C2P	MC2
		MC3	MC3	--	MC3	C3P	MC3
		MC4	MC4	--	MC4	C4P	MC4
	MC5	MC5	--	MC5	C5P	MC5	
	Electric Motor Quality	EMQ	E	E	E	--	E
	Clearance for electric motor	CM	CM	CM	CM	CM	CM
	Snap groove	N	N	N	N	N	N
	Snap ring	NR	NR	NR	NR	NR	NR
	Steel Cage	BLANK	BLANK	BLANK	BLANK	BLANK	BLANK
	Brass Cage	M	M	M	M	M	M
	Deep Groove ball Bearing Type	Rxx	Rxx	--	Rxx	--	--
		6xx	6xx	--	6xx	--	6xx
		60xx	60xx	60xx	60xx	60xx	60xx
		62xx	62xx	62xx	62xx	62xx	62xx
		63xx	63xx	63xx	63xx	63xx	63xx
		64xx	64xx	64xx	64xx	--	--
		67xx	--	67xx	67xx	--	--
		68xx	68xx	68xx	68xx	68xx	--
		69xx	69xx	69xx	69xx	69xx	--
		160xx	160xx	160xx	160xx	160xx	160xx
		622xx	--	--	--	--	--
		623xx	--	--	--	--	--
		630xx	--	--	--	--	--

Note: 1. Above interchange information is for reference purpose only.
2. NKS contact seal with DU means rubber seal with three lips.
3. NTN contact seal with LU means normal contact seal, LH means light contact seal.
4. Koyo Contact seal with RS means normal contact seal, with RK means heavy contact seal and with RD means light contact seal.

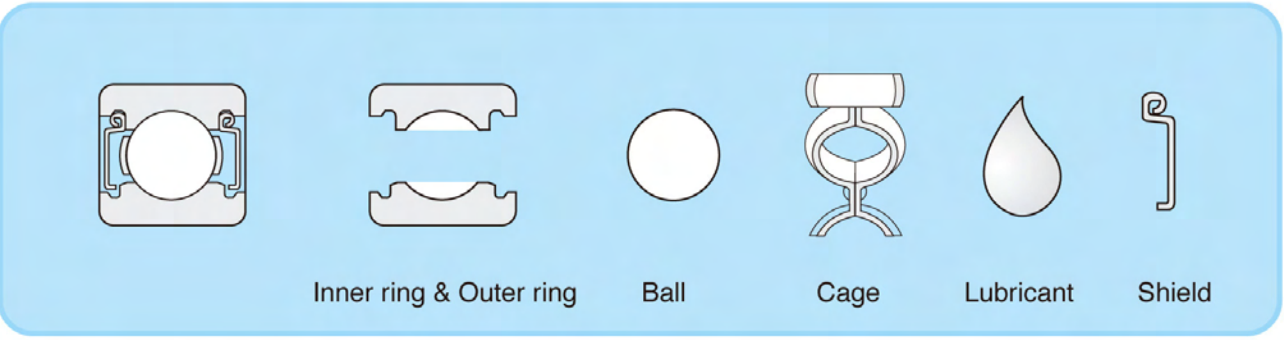
CONSTRUCTION & COMPONENTS

Deep groove ball bearing



HCH rolling bearing basically consists of two rings, rolling elements and a cage, which keeps the rolling elements at equal intervals. Seals and shields are applied to prevent the bearing from outside affect such as dust or oil invasion. The main purpose of lubricants in rolling bearing is to reduce friction and wear of each element.

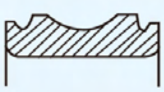
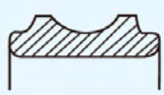
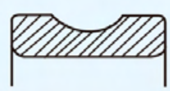
● Deep groove ball bearing components (ZZ type)



Notes:

The above bearing construction drawing only lists one type of inner ring construction (V-groove), which is applied usually in sealed bearings. But there are other two types of inner ring constructions applied as below chart.

Inner ring seal groove types

V-groove	L-groove	Plain
		

Notes:

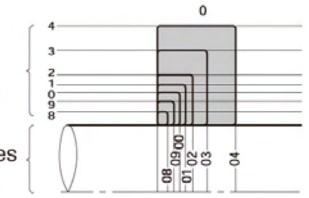
1) V-groove type are usually applied for small and medium size HCH deep groove ball bearings with seals or shieds.
2) L-groove type are usually applied for miniature size HCH deep groove ball bearings with seals or shieds.
3) Plain type are usually applied for HCH open type deep groove ball bearings.

● Dimensions

A rolling bearing's major dimensions, is known as "boundary dimensions". To facilitate international bearing interchangeability and economical bearing production, bearing boundary dimensions have been standardized by the International Standards Organization (ISO). Those boundary dimensions which have been standardized include: bearing bore diameter, outside diameter, width/height, and chamfer dimensions-all important dimensions.

Outer diameter dimensions (D) for radial bearings with standardized bore diameter dimensions (d) are covered in the "diameter series"; their corresponding width dimensions (B) are covered in the "width series". The ISO General Plans for boundary dimensions of deep groove ball bearings contain a progressive series of standardized outside diameters for every standard bore diameter arranged in Diameter series 8, 9, 0, 2, 3, 4 (in order of increasing outside diameter).

Diameter series numbers

Metric deep groove ball bearing	Diameter series (outer diameter dimensions)	
	number 8, 9, 0, 2, 3, 4	Diameter series Diamension series 
	dimensions small ←——→ large	

● Cage

Cages are mechanically stressed by frictional, strain and inertia forces and they may also be subjected to the chemical action of certain lubricants, lubricant additives or products of their ageing, organic solvents or coolants. Therefore the design and material are of paramount importance for the performance of the cage as well as for the operational reliability of the bearing itself.

Following three types of cage are widely applied in HCH deep groove ball bearings.

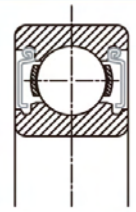
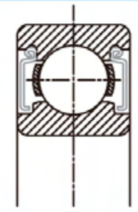
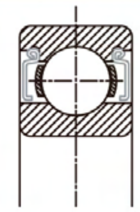
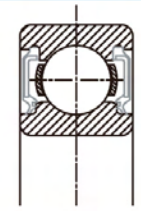
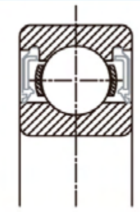
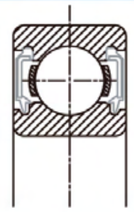
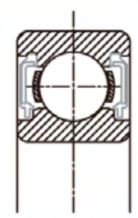
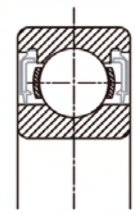
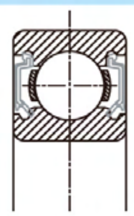
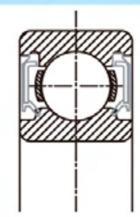
Rivet-type cage	Tongue-type cage	Polyamide cage
		
Made from high precision strip steel, pressed and formed with spherical ball pockets. The retainer halves are fixed together with rivets. HCH small size deep groove ball bearings widely use steel Rivet-type cages.	Made from high precision strip steel, pressed and formed with spherical ball pocket and tongue shape. The male and female retainer halves are caulked together for HCH miniature ball bearings applicatioin.	Polyamide cage are offered a range of materials, such as Nylon 6/6 with glass fiber, and has an advantage in self- lubricating property and lower acoustic noise. They are usually used in HCH ultra-low noise bearings.

● Internal sealing arrangements

The performance of sealing is vital to the cleanliness of the lubricant and the overall service life of the bearing arrangement. Integral bearing seals must be able to keep contaminants out and lubricant in the bearing cavity. Sealed bearings are generally used for arrangements where a sufficiently effective external seal cannot be provided because there is inadequate space.

When selecting a seal, the following factors need to be taken into consideration: the type of lubricant (oil or grease), seal peripheral speed, shaft fitting errors, space limitations, seal friction, resultant heat increase, as well as the cost.

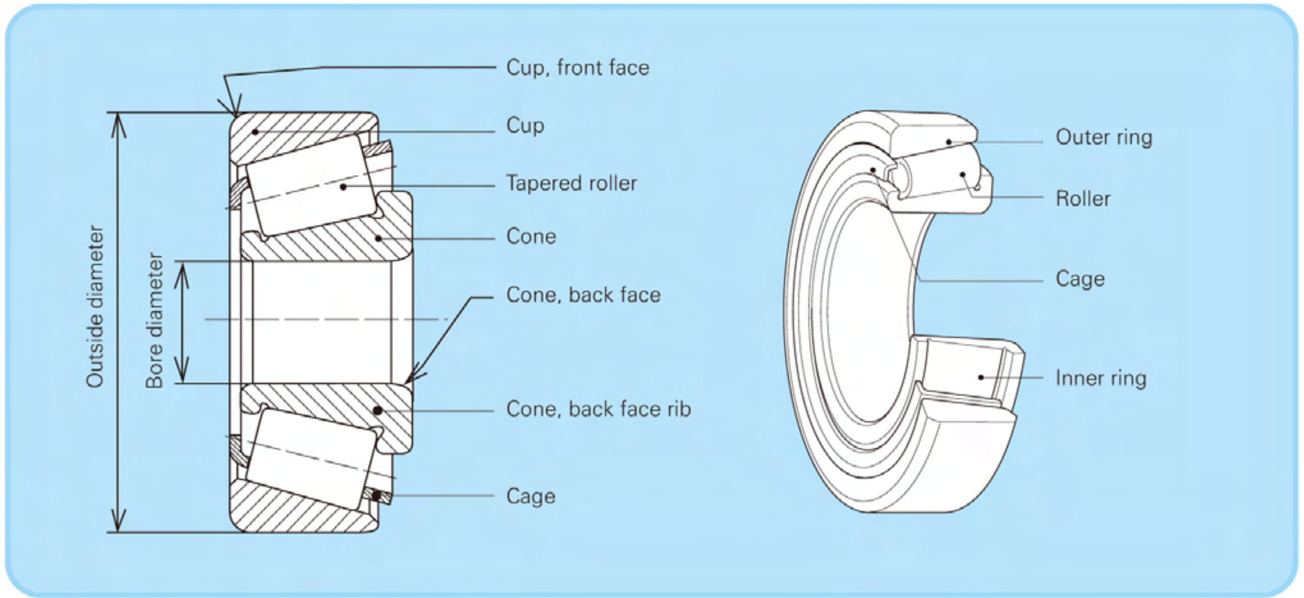
Shields and seals

"V" sealing groove (inner ring)	"L" sealing groove (inner ring)	No sealing groove (inner ring)
 Metal shields (ZZ) Non-contact High temperature Very high speed For small and medium size bearings	 Metal shields (ZZ) Non-contact High temperature Very high speed For miniature size bearings	 Metal shields (ZZ) Non-contact High temperature Very high speed Applicable due to assembly requirement
"V" sealing groove (inner ring)	**"L" sealing groove (inner ring)**	**No sealing groove (inner ring)**
 Rubber seals (2RS) Contact Very good dust and water proofing For small and medium size bearings	 Rubber seals (2RS) Contact Very good dust and water proofing For miniature size bearings	 Rubber seals (2RS) Contact Very good dust and water proofing Applicable due to assembly requirement
"V" sealing groove (inner ring)	**"L" sealing groove (inner ring)**	**No sealing groove (inner ring)**
 Rubber seals (2RZ) Non-contact Very high speed Good dust proofing For small and medium size ball bearings	 Rubber seals (2RZ) Non-contact Very high speed Good dust proofing For miniature size bearings	 Rubber seals (2RZ) Non-contact Very high speed Good dust proofing Applicable due to assembly requirement
"V" sealing groove (inner ring)	**"L" sealing groove (inner ring)**	
 Rubber seals (2RS1) Contact High speed Excellent dust and water proofing For small and medium size bearings	 Rubber seals (2RS1) Contact High speed Excellent dust and water proofing For miniazture size bearings	

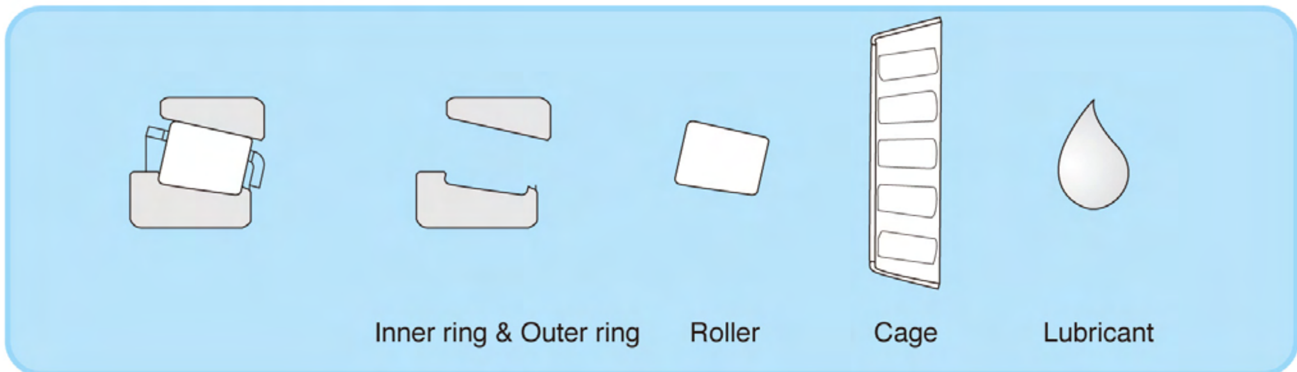
Notes:

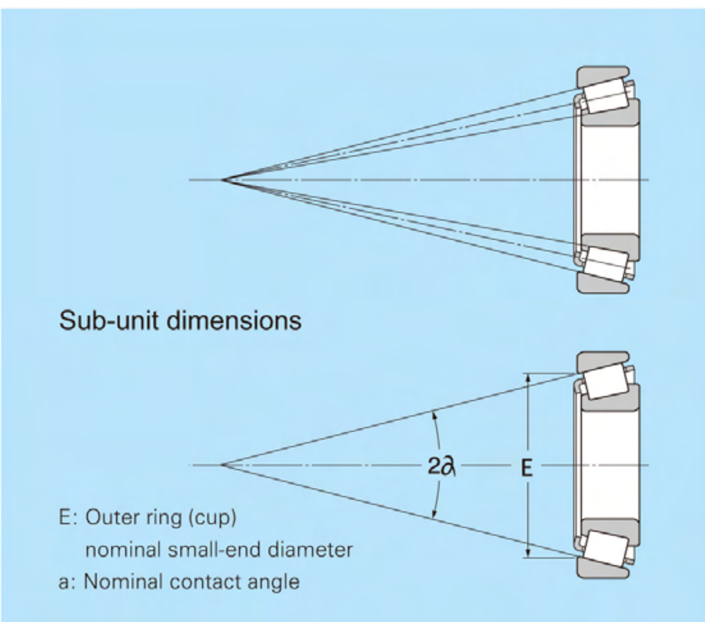
1) This chart lists double shielded and double sealed bearings, but single shielded (Z) and single sealed (RZ, RS, RS1) are also available.
2) The above sealing structures are only for reference. We reserve the right to change specifications and other information included in this catalogue without notice.
3) HCH also could provide bearings with other sealing designs including customized sealing design. Please consult HCH engineers for more information.

Tapered roller bearing



● Tapered roller bearing components





Notes:

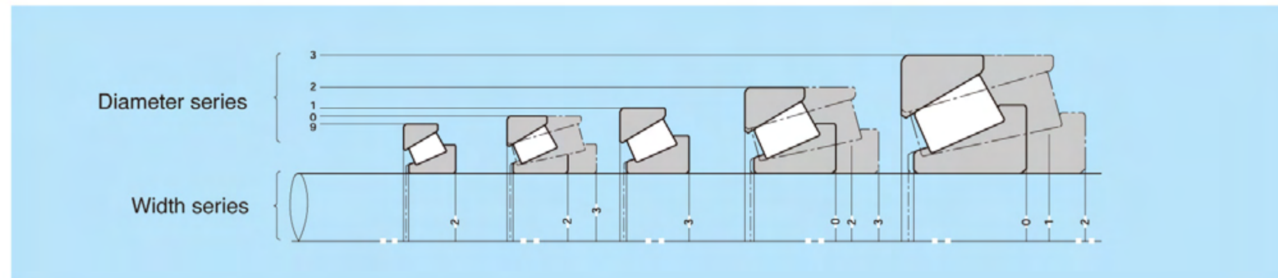
Tapered roller bearings are designed so that the center lines of the raceways and rollers all converge at a single point as shown in the right diagram. Due to this design feature, rollers move along the center of the raceway surfaces.

Aside from any cautionary notes that may appear, the single row tapered roller bearings have sub-units standardized for both metric and inch systems (including J series).

● Dimension series for metric tapered roller bearings

In the ISO general plan for single row metric tapered roller bearings, the boundary dimensions are grouped for certain ranges of the contact angle, known as the angle series (angle series 2, 3, 4, 5, 6 and 7 in order of increasing angle). Based on the relationship between the outside and bore diameters, and between the total bearing width and the cross-sectional height, diameter and width series have also been established. Here, a dimension series is obtained by combining the angle series with a diameter and a width series. These dimension series consist of one figure for the angle series and two letters, where the first letter identifies the diameter series and the second the width series.

Metric tapered roller bearings		Dimension series	
		Diameter series (outer diameter dimensions)	Width series (width dimensions)
	number	9, 0, 1, 2, 3	0, 1, 2, 3
	dimensions	small ←→ large	small ←→ large



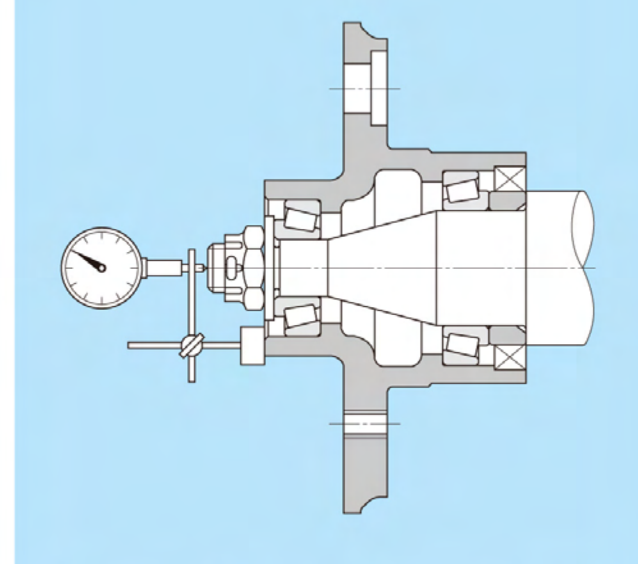
● Dimensions for inch tapered roller bearings

Experience has shown that the requirements of the vast majority of bearing applications can be met using bearings with these standardized dimensions. However, inch sized bearings are still popular nowadays. A large group of bearings in inch sizes are inch tapered roller bearings. The dimensions of these bearings conform to AFBMA Standard 19-1974 (ANSI B3.19-1975). ANSI/ABIVIA Standard 19.2-1994 has subsequently replaced this standard, but this later standard no longer includes dimensions.

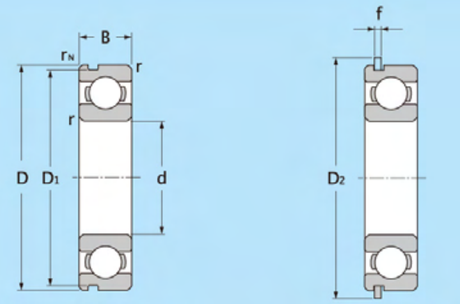
● Design of bearing arrangements

When designing bearing arrangements incorporating single row tapered roller bearings, it is necessary to consider the special characteristics of these bearings. Because of their internal design, they cannot be used singly and a second bearing is required; alternatively a paired set may be used. When the arrangement comprises two single row bearings they must be adjusted against each other as described under "Internal clearance and preload".

A correctly dimensioned operational clearance or preload is vital to the correct performance of single row tapered roller bearings and also to the operational reliability of the arrangement. If the operational clearance is too large, the full load carrying capacity of the bearing will not be exploited. If the preload is too great, then frictional losses will increase, as will operating temperature. In both cases the bearing service life could be substantially reduced.


Measurement of axial internal clearance adjustment

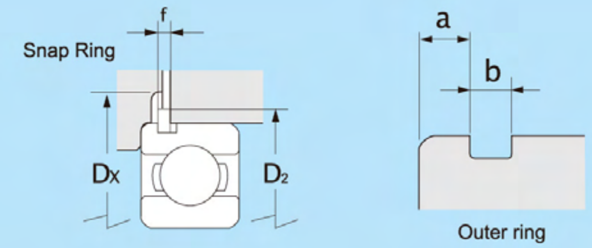
Snap ring and groove dimensions



With snap ring groove   With snap ring

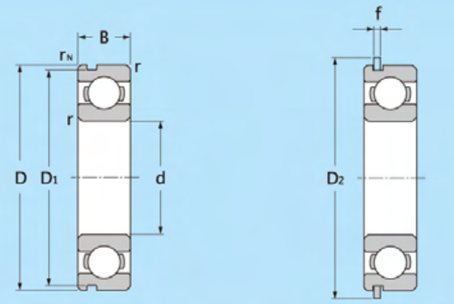
Applicable Bearings			Snap Ring Groove							
d		D	Snap Ring Groove Diameter D_1		Snap Ring Groove Position a				Snap Ring Groove Width b	
Diameter Series					Bearing Diameter Series					
					18		19			
18	19		max.	min.	max.	min.	max.	min.	max.	min.
–	10	**22**	20.8	20.5	–	–	1.05	0.9	1.05	0.8
–	12	**24**	22.8	22.5	–	–	1.05	0.9	1.05	0.8
–	15	**28**	26.7	26.4	–	–	1.3	1.15	1.2	0.95
–	17	**30**	28.7	28.4	–	–	1.3	1.15	1.2	0.95
20	–	**32**	30.7	30.4	1.3	1.15	–	–	1.2	0.95
22	–	**34**	32.7	32.4	1.3	1.15	–	–	1.2	0.95
25	20	**37**	37.5	35.4	1.3	1.15	1.7	1.55	1.2	0.95
–	22	39	37.7	37.4	–	–	1.7	1.55	1.2	0.95
28	–	40	38.7	38.4	1.3	1.15	–	–	1.2	0.95
30	25	42	40.7	40.4	1.3	1.15	1.7	1.55	1.2	0.95
32	–	44	42.7	42.4	1.3	1.15	–	–	1.2	0.95
–	28	45	43.7	43.4	–	–	1.7	1.55	1.2	0.95
35	30	47	45.5	45.4	1.3	1.15	1.7	1.55	1.2	0.95
40	32	52	50.7	50.4	1.3	1.15	1.7	1.55	1.2	0.95
–	35	55	53.7	53.4	–	–	1.7	1.55	1.2	0.95
45	–	58	56.7	56.4	1.3	1.15	–	–	1.2	0.95
–	40	62	60.7	60.3	–	–	1.7	1.55	1.2	0.95
50	–	65	63.7	63.3	1.3	1.15	–	–	1.2	0.95
–	45	68	66.7	66.3	–	–	1.7	1.55	1.2	0.95
55	50	72	70.7	70.3	1.7	1.55	1.7	1.55	1.2	0.95
60	–	78	76.2	75.8	1.7	1.55	–	–	1.6	1.3
–	55	80	77.9	77.5	–	–	2.1	1.9	1.6	1.3
65	60	85	82.9	82.5	1.7	1.55	2.1	1.9	1.6	1.3
70	65	90	87.9	87.5	1.7	1.55	2.1	1.9	1.6	1.3
75	–	95	92.9	92.5	1.7	1.55	–	–	1.6	1.3
80	70	100	97.9	97.5	1.7	1.55	2.5	2.3	1.6	1.3
–	75	105	102.6	102.1	–	–	2.5	2.3	1.6	1.3
85	80	110	107.6	107.1	2.1	1.9	2.5	2.3	1.6	1.3
90	–	115	112.6	112.1	2.1	1.9	–	–	1.6	1.3
95	85	120	117.6	117.1	2.1	1.9	3.3	3.1	1.6	1.3
100	90	125	122.6	122.1	2.1	1.9	3.3	3.1	1.6	1.3
105	95	130	127.6	127.1	2.1	1.9	3.3	3.1	1.6	1.3
110	100	140	137.6	137.1	2.5	2.3	3.3	3.1	2.2	1.9
–	105	145	142.6	142.1	–	–	3.3	3.1	2.2	1.9
120	110	150	147.6	147.1	2.5	2.3	3.3	3.1	2.2	1.9
130	120	165	161.8	161.3	3.3	3.1	3.7	3.5	2.2	1.9
140	–	175	171.8	171.3	3.3	3.1	–	–	2.2	1.9
–	130	180	176.8	176.3	–	–	3.7	3.5	2.2	1.9
150	140	190	186.8	186.3	3.3	3.1	3.7	3.5	2.2	1.9
160	–	200	196.8	196.3	3.3	3.1	–	–	2.2	1.9

Remarks The minimum permissible chamfer dimensions *r*N on the snap-ring-groove side of the outer rings are as follows:
Dimension series 18 : For outside diameters of 78mm and less, use 0.3mm chamfer. For all others exceeding 78mm, use 0.5mm chamfer.
Dimension series 19 : For outside diameters of 24mm and less, use 0.2mm chamfer. For 47mm and less, use 0.3mm chamfer. For all others exceeding 47mm, use 0.5mm chamfer.

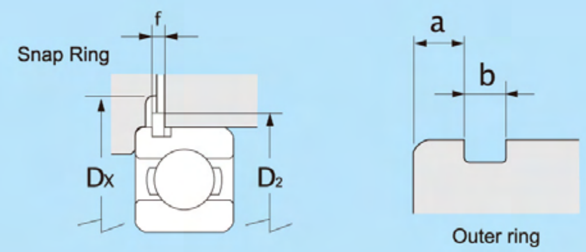


Snap Ring Groove	Locating Snap Ring							Side Cover
Radius of Bottom Corners r_0	Locationg Snap Ring Number	Cross Sectional Height e		Thickness f		Geometry of Snap Ring fitted in groove (Reference)		Stepped Bore Diameter (Reference) D_x
						Slit Width g	Snap Ring Outside Diameter D_2	
max.		max.	min.	max.	min.	approx.	max.	min.
0.2	**NR 1022**	2.0	1.85	0.7	0.6	2	24.8	25.5
0.2	**NR 1024**	2.0	1.85	0.7	0.6	2	26.8	27.5
0.25	**NR 1028**	2.05	1.9	0.85	0.75	3	30.8	31.5
0.25	**NR 1030**	2.05	1.9	0.85	0.75	3	32.8	33.5
0.25	**NR 1032**	2.05	1.9	0.85	0.75	3	34.8	35.5
0.25	**NR 1034**	2.05	1.9	0.85	0.75	3	36.8	37.5
0.25	**NR 1037**	2.05	1.9	0.85	0.75	3	39.8	40.5
0.25	**NR 1039**	2.05	1.9	0.85	0.75	3	41.8	42.5
0.25	**NR 1040**	2.05	1.9	0.85	0.75	3	42.8	43.5
0.25	**NR 1042**	2.05	1.9	0.85	0.75	3	44.8	45.5
0.25	**NR 1044**	2.05	1.9	0.85	0.75	4	46.8	47.5
0.25	**NR 1045**	2.05	1.9	0.85	0.75	4	47.8	48.5
0.25	**NR 1047**	2.05	1.9	0.85	0.75	4	49.8	50.5
0.25	**NR 1052**	2.05	1.9	0.85	0.75	4	54.8	55.5
0.25	**NR 1055**	2.05	1.9	0.85	0.75	4	57.8	58.5
0.25	**NR 1058**	2.05	1.9	0.85	0.75	4	60.8	61.5
0.25	**NR 1062**	2.05	1.9	0.85	0.75	4	64.8	65.5
0.25	**NR 1065**	2.05	1.9	0.85	0.75	4	67.8	68.5
0.25	**NR 1068**	2.05	1.9	0.85	0.75	5	70.8	72
0.25	**NR 1072**	2.05	1.9	0.85	0.75	5	74.8	76
0.4	**NR 1078**	3.25	3.1	1.12	1.02	5	82.7	84
0.4	**NR 1080**	3.25	3.1	1.12	1.02	5	84.4	86
0.4	**NR 1085**	3.25	3.1	1.12	1.02	5	89.4	91
0.4	**NR 1090**	3.25	3.1	1.12	1.02	5	94.4	96
0.4	**NR 1095**	3.25	3.1	1.12	1.02	5	99.4	101
0.4	**NR 1100**	3.25	3.1	1.12	1.02	5	104.4	106
0.4	**NR 1105**	4.04	3.89	1.12	1.02	5	110.7	112
0.4	**NR 1110**	4.04	3.89	1.12	1.02	5	115.7	117
0.4	**NR 1115**	4.04	3.89	1.12	1.02	5	120.7	122
0.4	**NR 1120**	4.04	3.89	1.12	1.02	7	125.7	127
0.4	**NR 1125**	4.04	3.89	1.12	1.02	7	130.7	132
0.4	**NR 1130**	4.04	3.89	1.12	1.02	7	135.7	137
0.6	**NR 1140**	4.04	3.89	1.7	1.6	7	145.7	147
0.6	**NR 1145**	4.04	3.89	1.7	1.6	7	150.7	152
0.6	**NR 1150**	4.04	3.89	1.7	1.6	7	155.7	157
0.6	**NR 1165**	4.85	4.7	1.7	1.6	7	171.5	173
0.6	**NR 1175**	4.85	4.7	1.7	1.6	10	181.5	183
0.6	**NR 1180**	4.85	4.7	1.7	1.6	10	186.5	188
0.6	**NR 1190**	4.85	4.7	1.7	1.6	10	196.5	198
0.6	**NR 1200**	4.85	4.7	1.7	1.6	10	206.5	208

Snap ring and groove dimensions



With snap ring groove With snap ring



Applicable Bearings					Snap Ring Groove							
d				D	Snap Ring Groove Diameter D_1		Snap Ring Groove Position a				Snap Ring Groove Width b	
Diameter Series							Bearing Diameter Series					
							0		2, 3, 4			
0	2	3	4		max.	min.	max.	min.	max.	min.	max.	min.
10	–	–	–	**26**	24.5	24.25	1.35	1.19	–	–	1.17	0.87
12	–	–	–	**28**	26.5	26.25	1.35	1.19	–	–	1.17	0.87
–	10	9	8	**30**	28.17	27.91	–	–	2.06	1.9	1.65	1.35
15	12	–	9	**32**	30.15	29.9	2.06	1.9	2.06	1.9	1.65	1.35
17	15	10	–	**35**	33.17	32.92	2.06	1.9	2.06	1.9	1.65	1.35
–	–	12	10	**37**	34.77	34.52	–	–	2.06	1.9	1.65	1.35
–	17	–	–	**40**	38.1	37.85	–	–	2.06	1.9	1.65	1.35
20	–	15	12	**42**	39.75	39.5	2.06	1.9	2.06	1.9	1.65	1.35
22	–	–	–	**44**	41.75	41.5	2.06	1.9	–	–	1.65	1.35
25	20	17	–	**47**	44.6	44.35	2.06	1.9	2.46	2.31	1.65	1.35
–	22	–	–	**50**	47.6	47.35	–	–	2.46	2.31	1.65	1.35
28	25	20	15	**52**	49.73	49.48	2.06	1.9	2.46	2.31	1.65	1.35
30	–	–	–	**55**	52.6	52.35	2.08	1.88	–	–	1.65	1.35
–	–	22	–	**56**	53.6	53.35	–	–	2.46	2.31	1.65	1.35
32	28	–	–	**58**	55.6	55.35	2.08	1.88	2.46	2.31	1.65	1.35
35	30	25	17	**62**	59.61	59.11	2.08	1.88	3.28	3.07	2.2	1.9
–	32	–	–	**65**	62.6	62.1	–	–	3.28	3.07	2.2	1.9
40	–	28	–	**68**	64.82	64.31	2.49	2.29	3.28	3.07	2.2	1.9
–	35	30	20	**72**	68.81	68.3	–	–	3.28	3.07	2.2	1.9
45	–	32	–	**75**	71.83	71.32	2.49	2.29	3.28	3.07	2.2	1.9
50	40	35	25	**80**	76.81	76.3	2.49	2.29	3.28	3.07	2.2	1.9
–	45	–	–	**85**	81.81	81.31	–	–	3.28	3.07	2.2	1.9
55	50	40	30	**90**	86.79	86.28	2.87	2.67	3.28	3.07	3	2.7
60	–	–	–	**95**	91.82	91.31	2.87	2.67	–	–	3	2.7
65	55	45	35	**100**	96.8	96.29	2.87	2.67	3.28	3.07	3	2.7
70	60	50	40	**110**	106.81	106.3	2.87	2.67	3.28	3.07	3	2.7
75	–	–	–	**115**	111.81	111.3	2.87	2.67	–	–	3	2.7
–	65	55	45	**120**	115.21	114.71	–	–	4.06	3.86	3.4	3.1
80	70	–	–	**125**	120.22	119.71	2.87	2.67	4.06	3.86	3.4	3.1
85	75	60	50	**130**	125.22	124.71	2.87	2.67	4.06	3.86	3.4	3.1
90	80	65	55	**140**	135.23	134.72	3.71	3.45	4.9	4.65	3.4	3.1
95	–	–	–	**145**	140.23	139.73	3.71	3.45	–	–	3.4	3.1
100	85	70	60	**150**	145.24	144.73	3.71	3.45	4.9	4.65	3.4	3.1
105	90	75	65	**160**	155.22	154.71	3.71	3.45	4.9	4.65	3.4	3.1
110	95	80	–	**170**	163.65	163.14	3.71	3.45	5.69	5.44	3.8	3.5
120	100	85	70	**180**	173.66	173.15	3.71	3.45	5.69	5.44	3.8	3.5
–	105	90	75	**190**	183.64	183.13	–	–	5.69	5.44	3.8	3.5
130	110	95	80	**200**	193.65	193.14	5.69	5.44	5.69	5.44	3.8	3.5

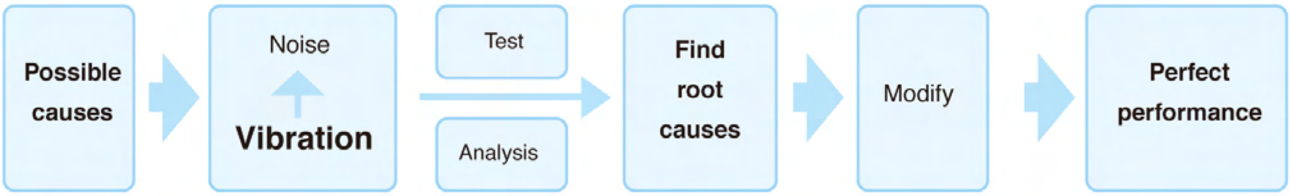
Note: The locating snap rings and snap ring grooves of these bearings are not specified by ISO.

Remarks: The minimum permissible chamfer dimension rN on the snap-ring side of outer rings is 0.5mm. However, for bearings of diameter series 0 having outside diameters 35mm and below, it is 0.3mm.

Snap Ring Groove	Locating Snap Ring							Side Cover
Radius of Bottom Corners r_0	Locationg Snap Ring Number	Cross Sectional Height e		Thickness f		Geometry of Snap Ring fitted in groove (Reference)		Stepped Bore Diameter (Reference) D_x
						Slit Width g	Snap Ring Outside Diameter D_2	
max.		max.	min.	max.	min.	approx.	max.	min.
0.2	**NR 26** (1)	2.06	1.91	0.84	0.74	3	28.7	29.4
0.4	**NR 28**(1)	2.06	1.91	0.84	0.74	3	30.7	31.4
0.4	**NR 30**	3.25	3.1	1.12	1.02	3	34.7	35.5
0.4	**NR 32**	3.25	3.1	1.12	1.02	3	36.7	37.5
0.4	**NR 35**	3.25	3.1	1.12	1.02	3	39.7	40.5
0.4	**NR 37**	3.25	3.1	1.12	1.02 0	3	41.3	42
0.4	**NR 40**	3.25	3.1	1.12	1.02	03	44.6	45.5
0.4	**NR 42**	3.25	3.1	1.12	1.02	3	46.3	47
0.4	**NR 44**	3.25	3.1	1.12	1.02	3	48.3	49
0.4	**NR 47**	4.04	3.89	1.12	1.02	4	52.7	53.5
0.4	**NR 50**	4.04	3.89	1.12	1.02	4	55.7	56.5
0.4	**NR 52**	4.04	3.89	1.12	1.02	4	57.9	58.5
0.4	**NR 55**	4.04	3.89	1.12	1.02	4	60.7	61.5
0.4	**NR 56**	4.04	3.89	1.12	1.02	4	61.7	62.5
0.4	**NR 58**	4.04	3.89	1.12	1.02	4	63.7	64.5
0.6	**NR 62**	4.04	3.89	1.7	1.6	4	67.7	68.5
0.6	**NR 65**	4.04	3.89	1.7	1.6	4	70.7	71.5
0.6	**NR 68**	4.85	4.7	1.7	1.6	5	74.6	76
0.6	**NR 72**	4.85	4.7	1.7	1.6	5	78.6	80
0.6	**NR 75**	4.85	4.7	1.7	1.6	5	81.6	83
0.6	**NR 80**	4.85	4.7	1.7	1.6	5	86.6	88
0.6	**NR 85**	4.85	4.7	1.7	1.6	5	91.6	93
0.6	**NR 90**	4.85	4.7	2.46	2.36	5	96.5	98
0.6	**NR 95**	4.85	4.7	2.46	2.36	5	101.6	103
0.6	**NR 100**	4.85	4.7	2.46	2.36	5	106.5	108
0.6	**NR 110**	4.85	4.7	2.46	2.36	5	116.6	118
0.6	**NR 115**	4.85	4.7	2.46	2.36	5	121.6	123
0.6	**NR 120**	7.21	7.06	2.82	2.72	7	129.7	131.5
0.6	**NR 125**	7.21	7.06	2.82	2.72	7	134.7	136.5
0.6	**NR 130**	7.21	7.06	2.82	2.72	7	139.7	141.5
0.6	**NR 140**	7.21	7.06	2.82	2.72	7	149.7	152
0.6	**NR 145**	7.21	7.06	2.82	2.72	7	154.7	157
0.6	**NR 150**	7.21	7.06	2.82	2.72	7	159.7	162
0.6	**NR 160**	7.21	7.06	2.82	2.72	7	169.7	172
0.6	**NR 170**	9.6	9.45	3.1	3	10	182.9	185
0.6	**NR 180**	9.6	9.45	3.1	3	10	192.9	195
0.6	**NR 190**	9.6	9.45	3.1	3	10	202.9	205
0.6	**NR 200**	9.6	9.45	3.1	3	10	212.9	215

NOISE AND VIBRATION

In addition to the basic requirements on a bearing like load capacity, speed limit and life time; low noise and vibration is becoming more and more important in most applicatons. Vibrations in bearings are caused by time varying forces in bearings. The contact forces move around the bearing, giving rise to perfect bearing vibrations in the outer ring. It is well-known that excecssive vibrations can cause premature failure and costly maintenance, often including unplanned down-time and loss of production. High vibration levels also increase energy consumption. High noise levels, in turn, result in a poorer life environment for personnel and family. Therefore, to find out the roof causes of noise & vibration and prevent potential from the beginning is critical to perfect performance of the bearings.



Vibration Rising and Countermeasures

HCH is making 100% such noise and vibration testing before every single bearing leaves the factory. Also, HCH has recently significantly improved design of deep groove ball bearings, to further reduce noise and vibration levels.

Customers need to pay attention when coming across the following conditions.

Types	Description	Causes	Countermeasures
Self-Generated Vibration	Vibration generated from the bearing itself when it is in the rotating condition.	Variations of circular form in the bearing balls and raceway.	Can not be avoided, but could reduce the vibration level by selecting the proper clearance due to the application.
Vibration Arising from Exposure to External	Disturbed noises occur with the performance degrades of bearings in modes known as wear oxidation or fretting corrosion.	The contaminated surrounding environment affects bearing. Loaded bearings operate without sufficient lubrication.	These conditions can be relieved by properly designed isolation supports and adequate lubrication.
Vibration from Misalignment	Not well-aligned bearings make noise when they are rotating.	Bearings are not well aligned on the shafts or houses during installation. The shafts and houses are not accurate.	Good alignment methods and special alignment tools to reduce vibration. Applying high accuracy shafts and houses.
Local Damage Vibration	The small damaged sections on the raceways and rolling elements generate a specific vibration frequency.	Mishandling or incorrect mounting.	Applying correct mounting methods and mechanical tools such as fitting tools. Applying induction heaters with time control and preset temperature mode.

Noise & vibration testing

The vibration and noise of bearings are classified as four classes measured by BVT-1 and classified as V_1, V_2, V_3 and V_4.

Data for vibration of single deep groove ball bearing

Tolerance in µm/s.

d	V			V_1			V_2			V_3			V_4		
mm	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band
3,4	80	44	44	60	35	32	48	26	22	31	16	15	28	10	10
5,6	110	72	60	74	48	40	58	36	30	35	21	18	32	11	11
7,8,9	130	96	80	92	66	54	72	48	40	44	28	24	38	12	12
10,12	160	120	100	120	80	70	90	60	50	55	35	30	45	14	15
15	210	150	120	150	100	85	110	78	60	65	46	35	52	18	18
17	210	150	120	150	100	85	110	78	60	65	46	35	52	25	25
20	260	190	150	180	125	100	130	100	75	80	60	45	60	25	25
22,25	260	190	150	180	125	100	130	100	75	80	60	45	60	30	32
28	260	190	150	180	125	100	130	100	75	80	60	45	60	35	40
30,32	300	240	190	200	150	130	150	120	100	90	75	60	70	35	40
35	300	240	190	200	150	130	150	120	100	90	75	60	70	42	45
40	360	300	260	240	180	160	180	150	130	110	90	80	82	50	50
45	360	300	260	240	180	160	180	150	130	110	90	80	82	60	60
50	420	320	320	280	200	200	210	160	160	125	100	100	95	70	70
55	420	360	360	280	220	200	210	180	180	125	110	110	95	70	70
60	480	360	440	320	220	240	240	180	200	145	110	130	100	80	80

Tolerance in µm/s.

d	V			V_1			V_2			V_3			V_4		
mm	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band
65	300	260	420	180	160	240	130	100	150	105	80	105	50	50	75
70	360	310	460	200	180	280	150	120	200	110	90	135	58	58	88
75	360	310	460	200	180	280	150	120	200	110	90	135	58	58	88
80	420	360	540	240	210	320	180	120	240	130	110	160	65	65	100
85	420	360	540	240	210	320	180	120	240	130	110	160	65	65	100
90	480	420	600	290	250	370	210	180	270	145	125	180	75	75	115
95	480	420	600	290	250	370	210	180	270	145	125	180	75	75	115
100	560	490	670	340	300	420	250	215	310	170	145	200	88	88	135
105	560	490	670	340	300	420	250	215	310	170	145	200	88	88	135
110	640	570	750	400	350	480	290	260	350	190	175	225	100	100	160
115	640	570	750	400	350	480	290	260	350	190	175	225	100	100	160

Data for vibration of single tapered roller bearing

Tolerance in µm/s.

d	V			V_1			V_2			V_3		
mm	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band	Low Band	Medium Band	High Band
15	310	500	500	220	360	360	150	220	220	100	100	100
17	330	550	550	240	400	400	170	240	240	110	110	110
20	330	550	550	240	400	400	170	240	240	110	110	110
25	360	590	600	280	440	450	210	280	280	120	140	130
30	360	590	600	280	440	450	210	280	280	120	140	130
35	400	640	670	320	480	500	250	320	300	150	180	160
40	440	690	740	360	530	560	280	350	320	170	210	190
45	440	690	740	360	530	560	280	350	320	170	210	190
50	480	750	810	400	600	620	320	400	360	220	260	240
55	480	750	810	400	600	680	320	400	360	220	260	240
60	530	850	1000	450	680	760	370	460	420	300	330	300

dB noise testing

The noise of bearings by dB is classified as four classes as Z1, Z2, Z3 and Z4. It is measured by the instrument of S0910-1.

Maximum vibration acceleration of single bearing

Tolerance in dB

d	Diameter Series (0)					Diameter Series (2)					Diameter Series (3)				
mm	Z	Z_1	Z_2	Z_3	Z_4	Z	Z_1	Z_2	Z_3	Z_4	Z	Z_1	Z_2	Z_3	Z_4
3	35	34	32	28	24	36	35	32	30	26	37	36	33	31	27
4	35	34	32	28	24	36	35	32	30	26	37	36	33	31	27
5	37	36	34	30	26	38	37	34	32	28	39	37	35	33	29
6	37	36	34	30	26	38	37	34	32	28	39	37	35	33	29
7	39	38	35	31	27	40	38	36	34	29	41	39	37	35	30
8	39	38	35	31	27	40	38	36	34	29	41	39	37	35	30
9	41	40	36	32	28	42	40	37	35	30	43	41	39	37	32
10	43	42	38	33	28	44	42	39	35	30	46	44	40	37	32
12	44	43	39	34	29	45	43	39	35	30	47	45	40	37	32
15	45	44	40	35	30	46	44	41	36	31	48	46	42	38	33
17	46	44	40	35	30	47	45	41	36	31	49	47	42	38	33
20	47	45	41	36	31	48	46	42	38	33	50	48	43	39	34
22	47	45	41	36	31	48	46	42	38	33	50	48	43	39	34
25	48	46	42	38	34	49	47	43	40	36	51	49	44	41	37
28	49	47	43	39	35	50	48	44	41	37	52	50	45	42	38
30	49	47	43	39	35	50	48	44	41	37	52	50	45	42	38
32	50	48	44	40	36	51	49	45	42	38	53	51	46	43	39
35	51	49	45	41	37	52	50	46	43	39	54	52	47	44	40
40	53	51	46	42	38	54	52	47	44	40	56	54	49	45	41
45	55	53	48	45	42	56	54	49	46	43	58	56	51	47	44
50	57	54	50	47	44	58	55	51	48	45	60	57	53	49	46
55	59	56	52	49	46	60	57	53	50	47	62	59	54	51	48
60	61	58	54	51	48	62	59	54	51	48	64	61	56	53	50

Maximum vibration acceleration of single bearing

Tolerance in dB

d	Diameter Series (0)				Diameter Series (2)				Diameter Series (3)			
mm	Z	Z_1	Z_2	Z_3	Z	Z_1	Z_2	Z_3	Z	Z_1	Z_2	Z_3
65	49	48	46	41	50	49	47	42	51	50	48	43
70	50	49	47	42	51	50	48	43	52	51	49	44
75	51	50	48	43	52	51	49	44	53	52	50	45
80	52	51	49	44	53	52	50	45	54	53	51	46
85	53	52	50	45	54	53	51	46	56	55	52	47
90	54	53	52	47	56	55	53	48	58	57	54	49
95	56	55	54	49	58	57	55	50	60	59	56	51
100	58	57	56	51	60	59	57	52	62	51	58	53
105	60	59	58	53	62	61	59	54	64	63	60	55
110	62	61	60	55	64	63	61	56	66	65	62	57
120	64	63	62	57	66	65	63	58	68	67	64	59

d	30200, 32200 Series			30300, 32300 Series		
mm	Z[1)]	Z_1	Z_2	Z	Z_1	Z_2
15	-	-	-	56	54	50
17	56	54	50	58	56	52
20	57	55	51	61	58	53
25	58	56	52	64	61	56
30	59	56	52	67	64	59
35	61	58	53	68	65	60
40	63	60	55	69	66	61
45	65	62	57	69	66	61
50	67	64	59	71	68	63
55	69	66	61	74	71	66
60	71	68	63	77	74	69

Discord sound testing

dB noise testing is a traditional noise measurement which can only give a general idea of bearing quality by their noise level. Vibration measurements are of great importance for high-quality bearing production which is applied widespread over the world famous bearing manufacturers. HCH's bearings are 100% tested by noise and vibration measurements.

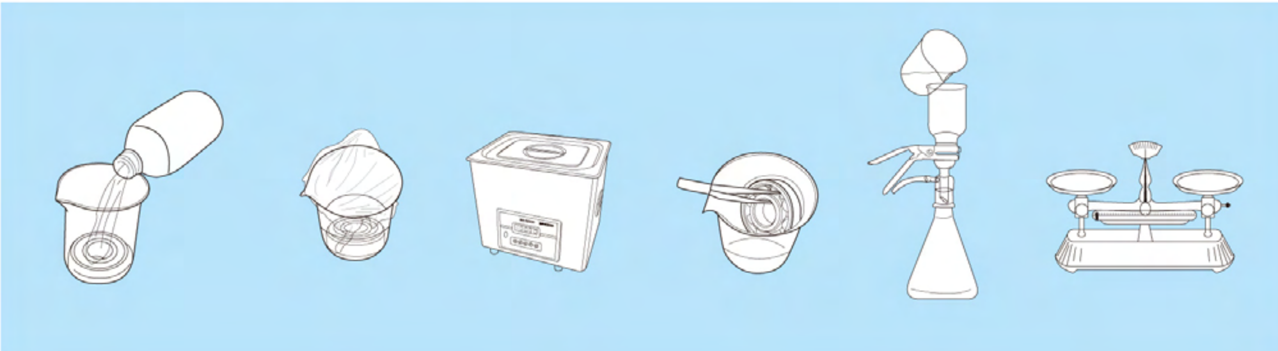
However, besides waviness, roundness and non-adequate lubrication etc., a noise application can lead the unreliable bearing performance which can also be caused by local defects, dirt particles and cage problems. In addition, HCH is also implementing 100% discord sound testing of bearings, which can test and detect all of these bearing quality issues. Following table lists common types of discord sound, their causes as well as countermeasures.

Causes of discord sound and countermeasures

Familiar discord sound	Cause	Countermeasure
Squeak	This is a dense and strident squeak just like noise when steel is sawed by saw blade and the wave crest occurs rhythmically. It will be specific reflected by BVT-1 equipment via high band: the pointer will rise and fluctuate. The amplitude is changing according to the intensity of noise. This kind of noise is the most deleterious. It is mainly caused by surface bumping (knocking) between the ball and raceway.	1. Choose the ball with surface strengthened. 2. Strictly control the quality of ball's surface (flaw, stripe, black spot and surface scuffing). 3. Strictly control the bumping (knocking) hurt of inner and outer raceway.
Chatter	This kind of noise is arrhythmic noise when grease or impurity is rolled. Sometimes it occurs and sometime it's gone. When this noise appears, the high band's pointer of BVT-1 equipment rises suddenly. The amplitude is changing according to the intensity of noise. It is closely related to the cleanliness of bearings, anti-rust oil and impurity of grease.	1. Do the inspection and control of bearing cleanliness. 2. Choose the anti-rust oil and grease with low impurity(namely high cleanliness anti-rust oil and grease).
Hum	When hum appears, normally the read of the low band of BVT-1 equipment is high. Louder the noise is, higher the read will be. This sound is closely related to the noise rising of the machine. It is related to the roundness and accuracy of bearing's inner and outer raceway.	Strictly control roundness of raceway.
Ripple	A kind of high-frequent noise which is symmetrical, ringing and continuous. The read of high band of BVT-1 equipment is a little bit higher which will increase the noise of the finished products. It is related to the raceway's chatter mark. reflecting the defect of the raceway waviness.	Strict control inner and outer race's undulation variation (monitor by Roundness & Waviness meter).
Grease stirring	Sounds like the clop.	Check the characteristic of grease. If the grease used is with high viscocity, it is normal agitating sound. But if the grease used is with low viscocity, please check the quality of the grease.

BEARING CLEANLINESS

Bearing cleanliness is essentially important since the contamination will directly influence the bearing life. The influence of contamination on bearing depends on a number of parameters including bearing size, relative lubricant film thickness, size and distribution of solid contaminant particles, types of contamination (soft, hard etc). The influence of contamination on bearing is complex and many of the parameters are difficult to quantify. It is therefore not possible to allocate precise values that would have general validity. HCH has introduced one method for checking the contamination level is the blank experiment shown as following:



Guideline values are provided according to JB/T 7050-2005 as following table:

Outer Diameter D mm		Deep groove ball bearing without lubrication			Deep groove ball bearing with lubrication type			Tapered roller bearing		
		Dimension Series								
Over	Including	7, 8, 9, 0, 00	2	3, 4	0, 7, 8, 9	2	3	02, 03, 13	20, 22, 29, 23	32, 31, 30
		Average maximum contamination level w mg/PC								
-	10	0.3	0.3	0.5	0.5	0.7	0.8	-	-	-
10	20	0.5	0.7	0.8	0.8	1.0	1.2	-	-	-
20	28	0.8	1.0	1.2	1.2	1.4	1.6	-	2.9	-
28	40	1.2	1.3	1.7	1.3	1.6	1.8	-	3.7	-
40	50	1.5	1.7	1.8	1.7	1.8	2.2	-	4.9	5.4
50	60	1.8	2.0	2.2	2.0	2.2	2.7	4.9	5.7	6.4
60	70	2.0	2.2	2.7	2.2	2.7	3.0	5.7	6.4	7.6
70	80	2.2	2.7	3.0	2.3	2.8	3.3	6.4	7.6	8.7
80	90	3.8	4.3	5.2	4.0	4.7	5.3	7.1	8.4	9.7
90	100	4.2	5.2	5.8	4.7	5.3	6.2	7.9	9.5	10.8
100	110	5.0	5.8	6.7	5.2	6.0	7.0	8.7	10.3	11.6
110	120	5.7	6.3	7.3	5.8	6.8	7.8	9.7	11.4	12.9
120	130	6.2	7.2	8.0	6.3	7.3	8.3	10.6	12.3	14.0
130	140	6.7	8.0	9.0	7.0	8.2	9.3	11.4	13.3	15.1
140	150	7.2	9.0	9.8	7.7	9.0	10.2	12.1	14.3	16.3
150	160	7.8	9.2	10.7	8.0	9.3	10.8	12.9	15.1	17.4
160	170	8.3	10.0	11.3	8.5	9.7	11.4	13.9	16.3	19.1
170	180	9.0	11.0	12..2	9.5	10.8	12.2	14.6	17.1	20.0

1) 包括 15 mm

STANDARD BEARING MATERIAL

The materials from which the bearing components are made determine to a large extent the performance and reliability of rolling bearings.

- For the bearing rings and rolling elements, typical considerations include hardness for load carrying capacity, fatigue resistance under rolling contact conditions, under clean or contaminated lubrication conditions, and the dimensional stability of the bearing components.
- For the cage, considerations include friction, strain, inertia forces, and in some cases, the chemical action of certain lubricants, solvents, coolants and refrigerants.
- Contact seals integrated in rolling bearings can also have a considerable impact on the performance and reliability of the bearings. The materials they are made of have to offer excellent oxidation, thermal or chemical resistance.

The relative importance of these considerations can be affected by other operational parameters such as corrosion, elevated temperatures, shock loads or combinations of these and other conditions.

All HCH bearings conform to RoHS and Reach Directive and have passed SGS testing which is further garanteed by HCH environment management system ISO 14001:2004, so you can use them with full confidence.

RoHS COMPLIANT 环保物料

REACH

ISO14001:2004

Rings & rolling elements

Because of high, repetitive stress to the rolling contact areas, fatigue phenomenon will occur to the bearing material after a duration of operation. Loading stress ultimately dislodges a surface section and the bearing fails. To delay the advent of material fatigue, bearing ring and rolling element materials should have the following properties:

- High level of hardness
- High rolling contact fatigue resistance
- Good wear resistance
- Dimensional stability
- Good mechanical strength

Today, carbon chromium steel is one of the oldest and most intensively investigated steels; due to the continuously increasing demands for extended bearing service life. The composition of this rolling bearing steel provides an optimum balance between manufacturing and application performance. This steel is normally given a martensitic or bainitic heat treatment during which it is hardened to the range of 58 to 65 HRC. Vacuum degassed, chromium-bearing steel GCr15 is the standard material for precision bearing rings and rolling elements. The material has uniform specification as SAE52100 (America), DIN100 Cr6 (German), JISSUJ2 (Japan).

Chemical composition of representative carbon chrome bearing steels

Steel No.	Chemical Composition %								
	C	Si	Mn	P	S	Cr	Mo	Cu	Ni
GCr 15 SAE52100 JISSUJ2 DIN 100 Gr6	0.95-1.05	0.15-0.35	0.25-0.45	≤0.02	≤0.015	1.4-1.6	0-0.08	0.06-0.02	≤0.2

Because HCH has the competence and facilities to provide a variety of materials, processes and coatings, HCH application engineers can assist in selecting those bearings that will provide superior performance for particular applications.

Cage material

Material for cages is required to have properties of good wear resistance, dimensional stability and good mechanical strength for perfect bearing running purpose. Therefore, for selection of cage material, it is important to consider the operation conditions.

● Sheet steel cages

These light weight cages have relatively high strength and can be surface treated to further reduce friction and wear. Cold-rolled steel is used for pressed cages.(Specification see the below table)

Sheet Steel No.	Chemical Composition %						
	C	Si	Mn	P	S	Ni	Cr
JISG 3141 SPCC	<0.12	—	<0.5	<0.04	<0.045	—	—

● Sheet brass cages

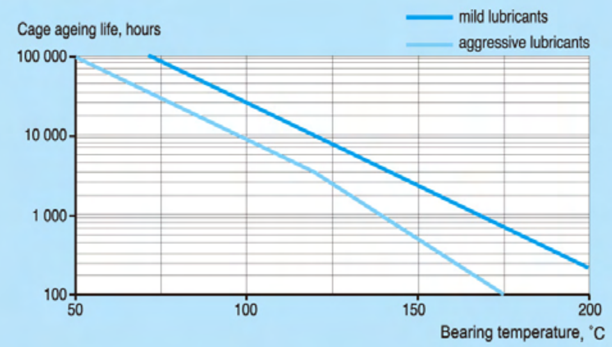
Pressed sheet brass cages are used for some small and medium-sized bearings. In applications such as compressors for refriger-action using ammonia, season cracking in sheet brass might occur, therefore steel cages should be used instead.

Standard	Symbols	Chemical Composition %								
		Cu	Zn	Mn	Fe	Al	Sn	Ni	Impurities	
									Pb	Si
JIS H 5120	CAC301 (HBsC 1)	55.0 - 60.0	33.0 - 42.0	0.1 - 1.5	0.5 - 1.5	0.5 -1.5	<1.0	<1.0	<0.4	<1.0
JIS H 3250	C 6782	56.0 - 60.5	Residual	0.1 - 1.5	0.5 -1.5	0.5 -1.5	—	—	<0.5	—

● Polymer cages

Polyamide resin is applied depending on the type of bearing and the application, but should not be used at temperatures above 120 ℃ or below -40℃. For the majority of injection moulded cages, polyamide 6.6 is used. This material, with glass fibre reinforcement or without, is characterized by a favourable combination of strength and elasticity.

The mechanical properties like strength and elasticity of polymeric materials are temperature dependent and subject to permanent changes under operating conditions, called ageing. The most important factors that play a role in this ageing behaviour are temperature, time and the medium (lubricant) to which the polymer below. The relationship between these factors for glass fibre reinforced polyamide 6.6 is illustrated in diagram. It appears that the cage life decreases with increasing temperature and the aggressiveness of the lubricant.Therefore, whether polyamide cages are suitable for a specific application depends on the operating conditions and life requirements.



Cage ageing life for glass fibre reinforced polyamide 6.6

Notes: Please consult the HCH application engineering service for cage availability for specific bearing executions.

Shields & seals material

● Shields material

HCH bearings employ carbon steel as standard, and the option of AISI-300 stainless steel is available when needed.

● Seals material

HCH bearings use a variety of sealing materials to meet the requirements of high temperature operation and compatibility with greases. Buna Nitrile is the standard material used, while fluorocarbon, silicone, and teflon seals are commonly specified for high temperatures.

Type	ASTM D1418 Designation	Temperature Range	Hardness (Shore A)	Features	Limitation
Nitrile (Buna)	NBR	(-40~250F)	40~90	Low compression set, high tensile strength, and high abrasion resistance. Excellent resistance to petroleum-based oils and fuels.	Not recommended for very high temperature. Not recommended for exposure to sunlight and chemicals as acid, ether and esters.
Silicone	MQ/PMQ/ VMQ/PVMP	(-94~400F)	25~80	High temperature and dry heat resistance. High resistance to aging effects of both suglight and ozone attach.	Low abrasion and tear resistance. High friction characteristics.
Nitrile (Hydrogenated)	HNBR/NEM	(-30~330F)	50~90	The hydrogenation of Nirtrile, heat resistance, tensile strength. Enhanced physical strength and chemical resistance.	Not recommended for very low temperature. Not recommended for exposure to sunlight and chemicals as acid, ether and esters.
Fluororubber	FKM/FPM	(-20~400F)	50~95	Combining high temperature resistance with outstanding chemical resistance. Highly resistance to petroleum products,acids and silicone fluids.	Not recommended for situations requiring low temperature flexibility.
Polyacrylate	ACM Rubber	(0~350F)	40~90	Excellent resistance to hot oil, sunlight and ozone degradation. Also features an enhanced ability to resist flex cracking.	Water resistance is lower. Not recommended for very low temperature.

WARNING!

Safety precautions for fluoro rubber is very stable and harmless in normal operating conditions up to +200 ℃. However, if exposed to extreme temperatures above 300 ℃, e.g. fire or the flame of a cutting torch, fluoro rubber seals give off hazardous fumes. These fumes can be harmful if inhaled, as well as to the eyes. In addition, should not be in contact with the skin either.

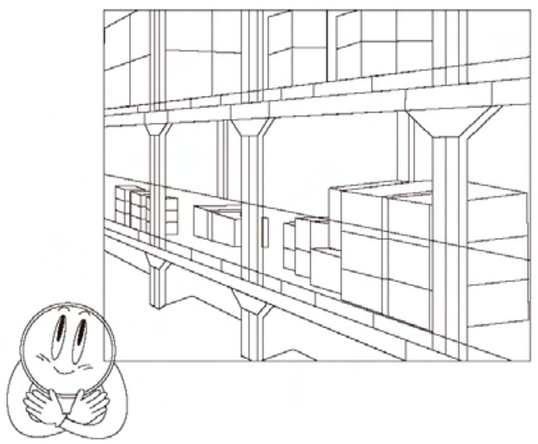
BEARING HANDLING SERVICE

The best bearing in the world will not reach its maximum life unless it is properly installed and maintained. With decades of experience in bearing technology, HCH understands the importance of proper maintenance procedures in maximizing product and equipment life. The value-added knowledge is grounded in our long time experience of motion, lubrication, friction and metallurgy. We are sharing this information to help you extend bearing life in your applications through proper installation, removal and service.

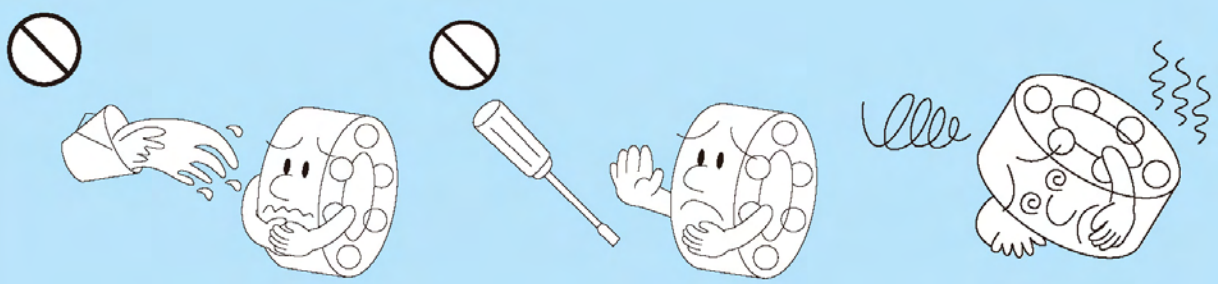
As a general precaution to ensure proper installation, it is important to keep bearings and related components as clean as possible. This means they should be handled in a environment free from debris or high humidity, using clean rinsing oil, with due consideration for guarding against corrosion or rust. Be sure to check each part before mounting. Inspect the sealed areas as well as dimensions, shape, appearance and accuracy of shafts and housing. While checking, use care to prevent perspiration from the hands, or debris present at the site, from coming into contact with the bearing.

● Bearing storage

Bearings are coated with a rust preventative compound and securely packaged before delivery. When handling and storing bearings, care must be taken to ensure that they will not rust or corrode. Even a small amount of moisture or chemical left on an unprotected bearing by a glove or hand can result in a small etched area, which may initiate bearing fatigue.Please observe the following guidelines when storing bearings.



- The wrapping paper cannot completely provide protection from the circulation of ambient air. Ideally, bearings should be stored in a location with low humidity, i.e. less than 60% relative humidity.
- Bearings should be stored in clean, well ventilated spaces with no direct exposure to sunlight. Bearings should never be stored on the ground, but should be stored on shelves or pallets at least 20 cm above the ground.
- Boxes of bearings should not be stacked too high or the rust preventative compound may be squeezed out of bearings on the bottom.
- When bearings are unwrapped for inspection prior to acceptance, they must stored with due attention to the application of anti-corrosion agent and then re-packaged.



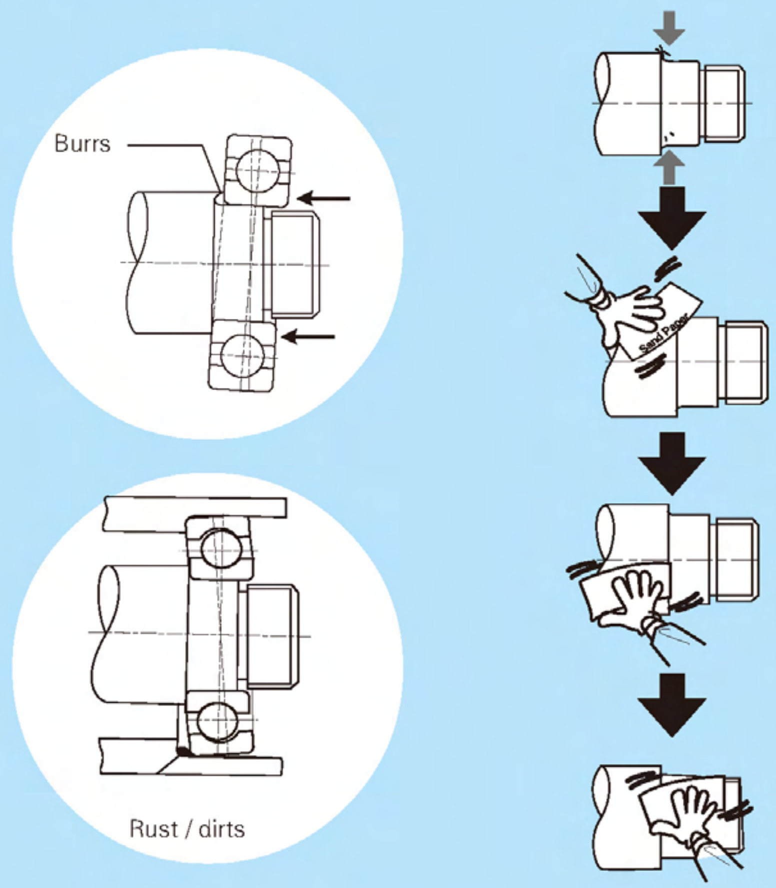
Installation prepare

Bearings are precision parts and in order to preserve their accuracy and reliability, special care must be taken before their installation:

① Do not unwrap the packing before using, and do not lay it aside after unwrapping.
② Keep the working area clean.
③ Use proper fitting tools, and keep them clean.
④ Do not use cloth which may produce scraps, and do not use dirty cloth.
⑤ Clean the components all around before installing.
⑥ Do not touch the bearing with bare hand directly.

Cleanliness during the bearing mounting operation is essential for a rolling bearing to operate for maximum service life. Any burrs, cutting chips, rust or dirt that infiltrate the bearing before installation should first be removed from the bearing mounting surfaces. Mounting can then be simplified if the clean surfaces are lubricated with spindle oil.

● Preparation procedure



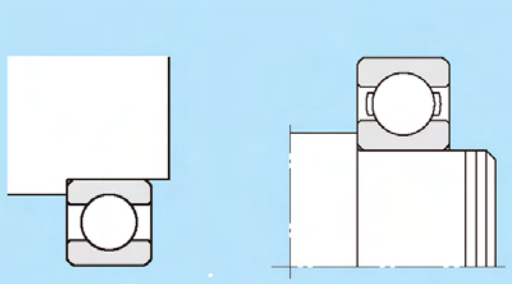
Note: Bearings should also be washed with benzene or petroleum solvent and dried before installation if the package has been damaged or there are other chances that the bearings have been contaminated. Double shielded bearings and sealed bearings should never be washed.

Mounting design

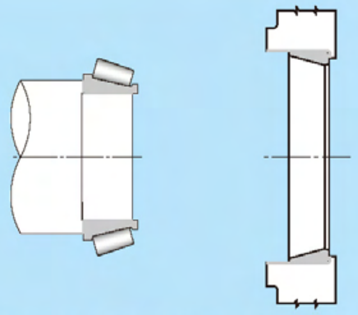
When fixing a bearing in position on a shaft or housing, there are many instances where the interference fit alone is not enough to hold the bearing in place. Bearings must be fixed in place by various methods so that they do not move axially when placed under load.

● Backing shoulder design

The primary function of either the inner ring or outer ring backing shoulders is to positively establish the axial location and alignment of the bearing and its adjacent parts under all loading and operating conditions. The conventional and most widely accepted method used to provide bearing backing is to machine a shoulder on a shaft or in the housing.

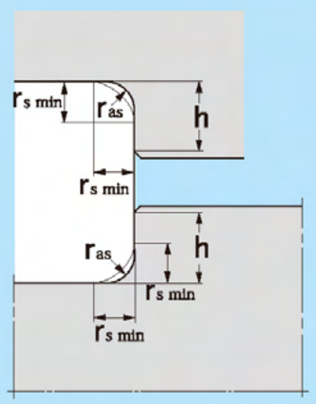


Deep groove ball bearing shaft and housing shoulders.



Tapered roller bearing shaft and housing shoulders.

Since shaft shoulder height and the outer diameters of spacers or the housing shoulder height are closely related to the dismounting of bearings, their standard dimensions are described in our catalogues for reference purposes. Along with these shoulder heights, the fillet radius in the corners of shafts and housings is also important. The shaft and housing abutment height (h) should be larger than the bearings' maximum allowable chamfer dimensions (rs max), and the abutment should be designed so that it directly contacts the flat part of the bearing end face. The fillet radius must be smaller than the bearing's minimum allowable chamfer dimension (rs min) so that it does not interfere with bearing seating. Following talbe lists abutment height (h) and fillet radius (ras).



1. It is necessary to be larger abutment height than the above value under larger thrust load.

2. The values in this "Special Case" column should be adopted in cases where the thrust load is extremely small except for tapered roller bearings angular contact bearings, spherical roller bearings.

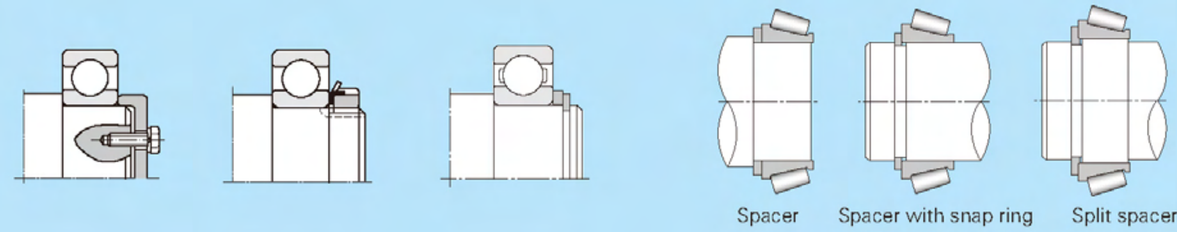
Note: ras max maximum allowable fillet radius.

Fillet radius and abutment height — Unit mm

rs min	ras max	h(min) Normal use[1]	h(min) Special use[2]
0.05	0.05	0.3	
0.08	0.08	0.3	
0.1	0.1	0.4	
0.15	0.15	0.6	
0.2	0.2	0.8	
0.3	0.3	1.25	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

● Spacer & snap ring design for bearing inner ring

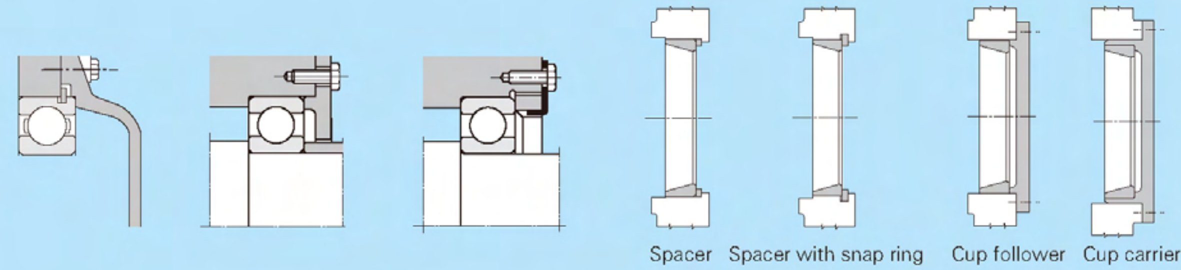
For a bearing to operate for maximum service, it must be of sufficient section and design to resist axial movement due to loading or distortion and must be wear-resistant at the interface with the bearing. In some applications a spacer is used between a cone and shaft shoulder or a snap ring. As a further alternative, a split spacer can be used.



Separate member used to provide adequate shaft shoulder diameter.

● Spacer & snap ring design for beairng outer ring

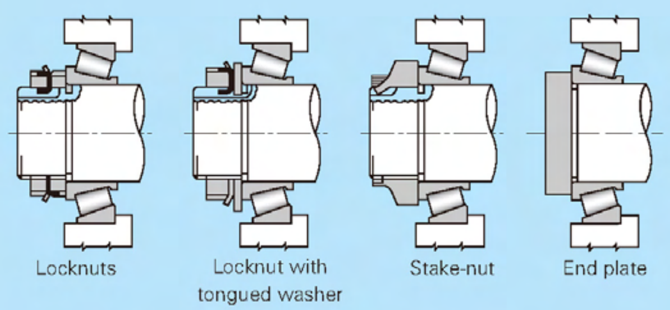
A spacer or snap ring can also be used for outer ring backing. If a snap ring is used for bearing backing, it is suggested that an interference cup fit be used. For a tapered roller bearing, the cup used for bearing setting in a direct mounting (roller small ends pointing outwards) is usually set in position by a cup follower or by mounting in a carrier.



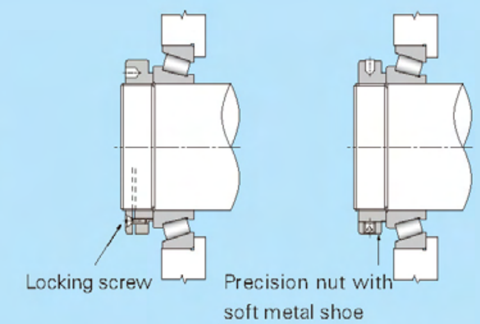
Separate member used to provide adequate housing backing diameter.

● Indirect mounting design for tapered roller bearings

With an indirect mounting (roller small ends pointing inwards), bearing setting can be achieved by a wide variety of devices. In applications requiring precision class bearings, a special precision nut can be used. This has a soft metal shoe that is clamped against the threads with a locking screw. Other solutions can use split nut and/or ground spacers where setting cannot be altered



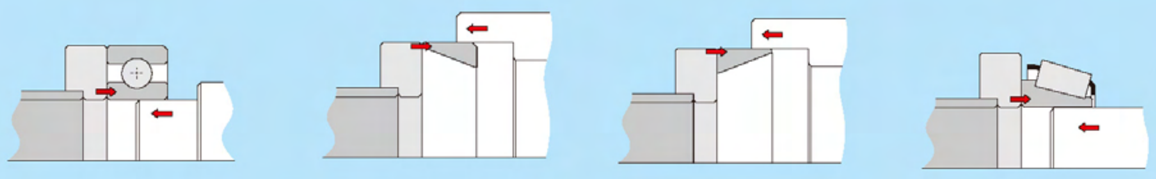
Bearing setting devices - indirect mounting.



Setting devices using split nut and precision nut with soft metal shoe.

Bearing mounting

Proper mounting allows the load to be transmitted to the ring experiencing the interference fit. Mounting forces are not transmitted via the rolling elements, helping to prevent damage to the raceways. To fix a radial bearing to a shaft, generally bring the bearing into close contact with the shaft shoulders and spacer and fix it in position by tightening the shaft nut. The ends of the shaft shoulders and the spacer must be perpendicular, bearing rotation accuracy and roller contact performance will be adversely affected, resulting in heat generation and premature fatigue. The same care must be taken to ensure the proper contact between the housing shoulders and the lateral face of the outer rings.



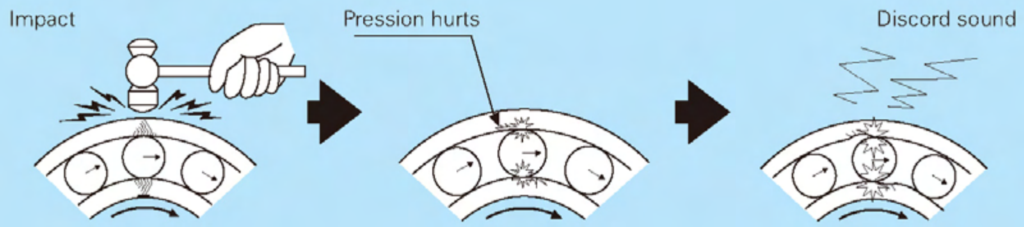
● Prevent 16% of premature bearing failures

Around 16% of all premature bearing failures are a result of poor fitting or using incorrect mounting techniques. Individual applications may require mechanical, heat or hydraulic mounting methods for correct and efficient bearing mounting. Selecting the mounting technique appropriate for your application will help you extend your bearing's service life and reduce costs resulting from premature bearing failure as well as potential damage to the application.

● Warning in installation

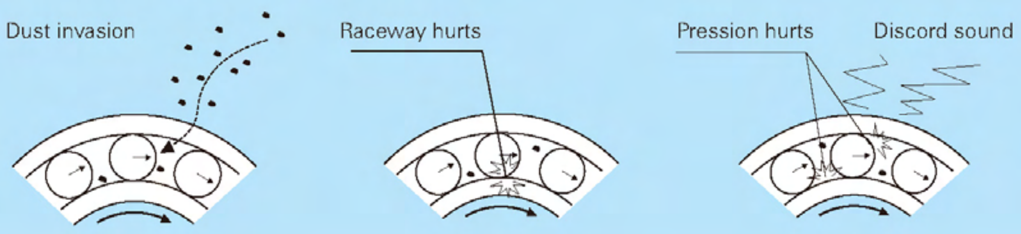
① To avoid severe impact

The gap between raceway surface and balls is approximate to zero contact, so excessive impact may cause dents on the raceway surface. Hammering or dropping must be avoided.



② To avoid any dust entry

When using bearings, if the interior suffers from dust invasion, raceway surface and rolling elements surface can be hurt, which will cause discord sound or bad rotation.



● Fitting consideration

Bearing fits are selected by reviewing the application details, general guidelines and with experience. The fit, or amount of interference that exists between mating components, can be devised into three resultant categories: press fit, transition fit and loose fit. Typically, an inner ring rotating application will use a tight or interference fit on the shaft, and the housing will use a loose or clearance fit. Likewise, with an outer ring rotating application, the opposite is true. To determine the shaft and housing fits required for a particular application, one must consider such variables as load, ambient temperature and the type of bearing. Moreover, the following questions should be answered:

What is the bearing size and type?
Is the application inner or outer ring rotating?
What is the load direction and condition?
What kind of shaft and housing are being used? i.e. -Is the shaft solid or hollow?
-Are the shaft and housing of steel or aluminum construction?

For the details, please check the next section: bearing fitting service.

● Personal Safety- From You to Everyone Around

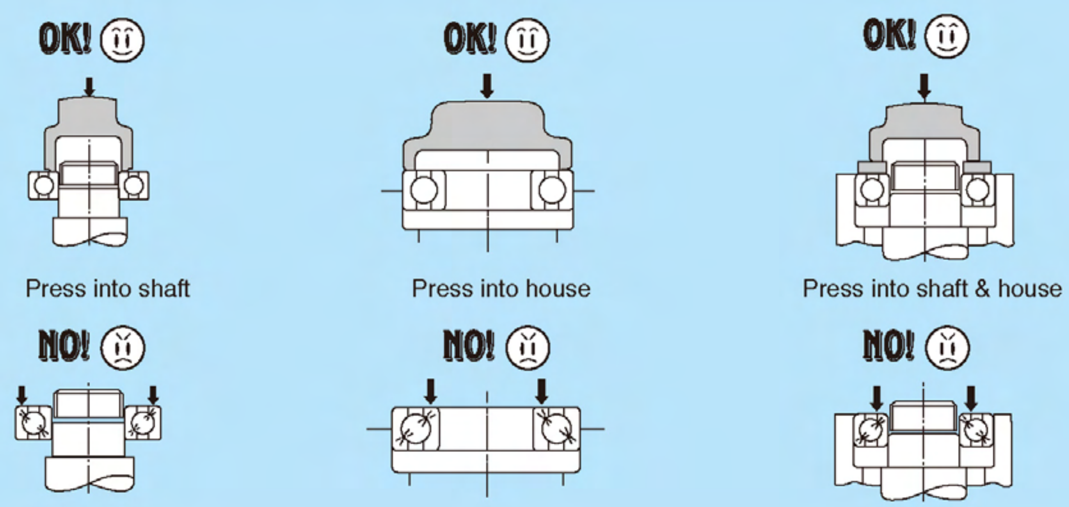
To keep you and those around you safe always read the Instruction manuals before operating any of our products and pay particular attention to the safety warnings.

1. Always wear protective clothing and goggles when mounting and dismounting rolling element bearings.
2. Always use heat resistant gloves when mounting heated bearings and when dismounting bearings using heat.
3. For your own safety, please do not strike the bearing directly with any hard object such as a hammer or chisel.
4. Select a suitable puller for the application with sufficient pulling force in order to help prevent puller overload. Overloading a puller can result in puller arm or beam breakage, causing personal injury.
5. A safety blanket fitted around the puller and bearing helps reduce the risk on injury in case the puller's arm breaks.
6. Remember that corrosion on the interference fit may require significantly higher dismounting force. If the interference fit is corroded, use penetrating oil to dissolve/loosen the rust.
7. Powerful forces may be involved when dismounting bearings and care must always be taken to avoid injury.

Deep groove ball bearing mounting

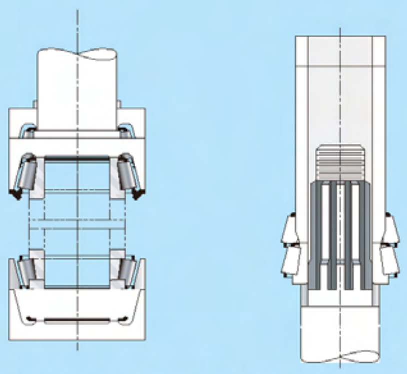
Adequate tools and mounting methods must be provided to properly fit the inner and outer races on shafts or in housings to avoid damage. Direct shock on the races must be avoided.

● Deep groove ball bearing installing

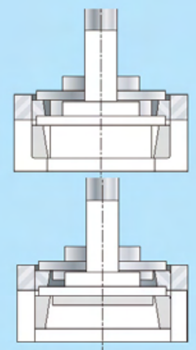


Tapered roller bearing mounting

During the mounting of bearings where the faces lie in the same plane, the collets enable the load to be transmitted to the ring experiencing the interference fit. If the impact mounting tool is used, mounting forces are not transmitted via the rolling elements and damage to the raceways is avoided.



Inner ring assemblies should be pressed on the shaft using the proper drivers

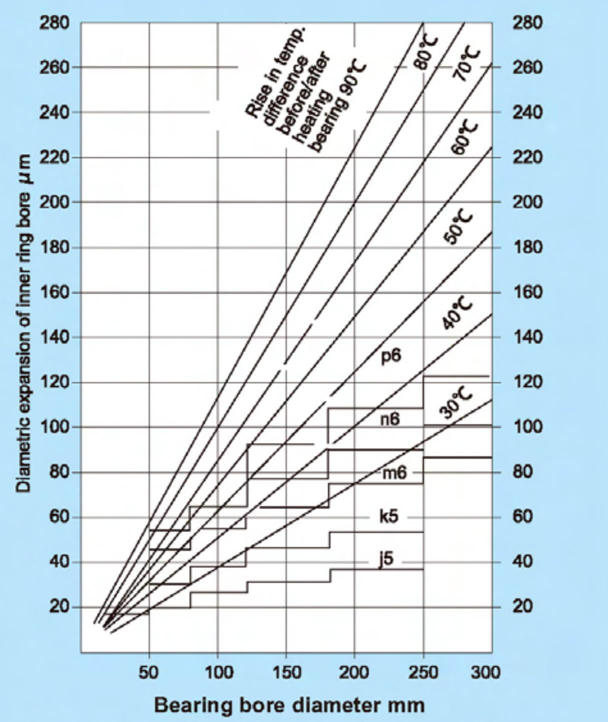


Outer rings can be pressed into the housing with a simple driver

Care should be taken when mounting tapered roller bearings. The cup can be mounted in either direction, but the cone can only be mounted from the back face. This ensures that the cage does not overhang. Never mount a cup and cone together or mount a cone from the front face.

Mounting Bearings with heaters

Heat mounting is recommended as an easy mounting method for bearings with tighter interference. Heating temperature can be determined from the following chart according to the specific bearing dimension and the intended interference.



Temperature differential required for shrinkage fit of inner ring

The required relative temperature difference between the inner ring and the fitting surface depends on the amount of interference and the shaft fitting surface diameter. The chart on the left shows the relation between the bearing inner bore diameter temperature differential and the amount of thermal expansion. For applications that require a tight fit of the outer ring in housing, it may also be possible to heat the housing to expand it, allowing the bearing to install more easily. In actual mounting work, as the bearing cools, it cannot be easily mounted on the shaft. Therefore, heat the bearing to 20˚C to 30˚C higher than the lowest temperature required for mounting. For example, when a bearing with a bore diameter of 120 mm is mounted with fit n6, the maximum interference is 65um. In this case, the required heating temperature may be room temperature +50˚C as shown in the left chart, whereas the temperature must be raised an additional 20˚C to 30˚C in order to easily press it onto the shaft. Consequently, the required heating temperature can be seen to be room temperature +70˚C to 80˚C.

Bearings can be heated in a pan or metal container filled with oil

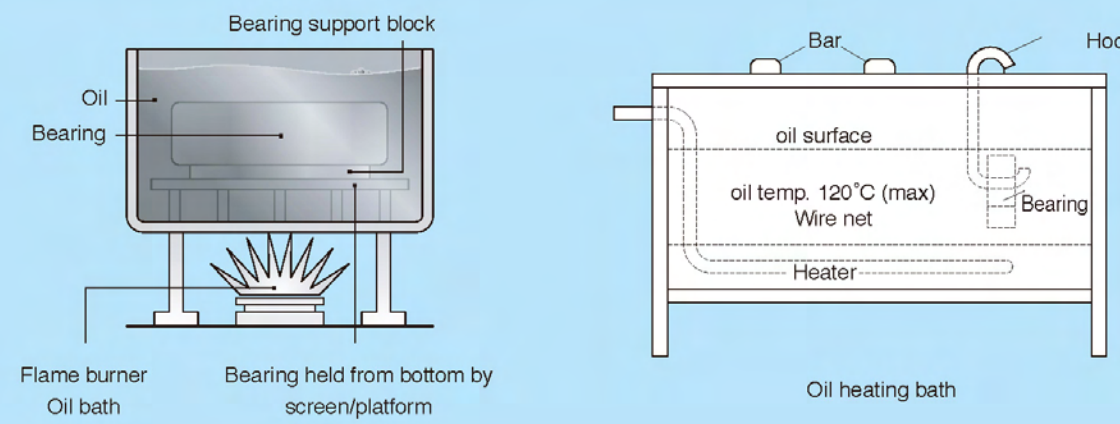
● Small bearing heating

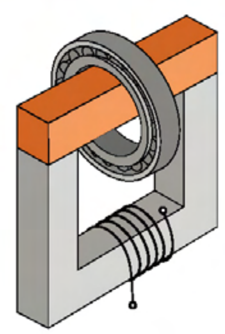
Small bearings can be heated using several methods. They can be heated in a pan or metal container filled with oil as in the left picture. A screen or platform should be used to keep the bearing from resting on the bottom of the pan where heat is applied. A high quality mineral oil should be used for the heating oil. However, this method should not be used for prelubricated shielded and sealed bearings.

A heat lamp can also be used to heat rings, and the temperature regulated by adjusting the distance from the light to the ring.

● Larger bearing heating

For larger bearings, you may need to use an oil bath to heat the bearing The bearing should be positioned in the center of the tank, and allowed to heat long enough to fully expand. Do not allow the bearing to come in direct contact with the heat source. The oil bath should be large enough to accommodate two to five bearings, with a sufficient amount of oil to completely cover the bearings. Precautions for use of the oil bath are shown in the follow picture. Be sure to use a wire net or equivalent device in the bath to support the bearings in the oil without allowing them to directly contact either the heater or the bottom of the bath. For easy handling, place a long bar across the top of the oil bath with an attached hook from which to suspend the bearings.





● Induction heaters

Induction heating is a superior, fast and controlled heating method. It is safer and more environmentally alternative to traditional heating methods such as oven, oil baths or blow torches. We advise our customers to heat the work piece in a horizontal position around the pole if possible. This will bring more energy into the work piece since it is closer to the coil. Hanging the work piece on the yoke will create more distance between it and the coil which means less energy and slower heating time. If possible, always place the work piece around the coil to achieve the fastest heating results.

Warning:

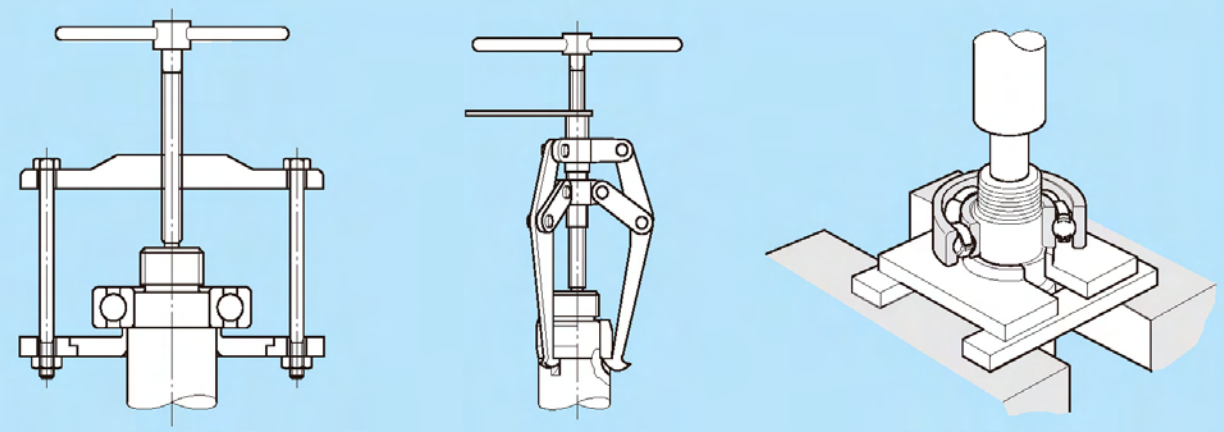
1. Do not heat standard bearings above 150˚C or freeze outer races below -55˚C. For precision bearings, do not heat above 65˚C or freeze below -30˚C.
2. When heated bearings are installed on shafts, the inner rings must be held against the shaft abutment until the bearing has been cooled in order to prevent gaps from occurring between the ring and the abutment face.
3. A bearing attached to a shaft cools rapidly, and after heating, an expanded bearing shrinks in a crosswise direction. In some cases, in order to avoid a clearance between the inner ring and shoulder, a shaft nut or other appropriate tool is applied.

Bearing disassembly

At some point, the bearing will reach the end of its service life and will have to be replaced. It is extremely important to dismount it correctly so that the service life of the replacement bearing is not compromised. When dismounting bearings, care must be taken not to damage other machine components, such as the shaft or housing, as damage can result in compromising the machine's efficiency and lifetime. Bearings are sometimes dismounted to maintain or replace other components of the machine. These bearings are often reused. Dismounting bearings can be a hazardous and demanding task. Selecting the correct dismounting methods and tools is therefore of utmost importance for reducing the risk of personal injuries. Individual applications may require mechanical, heat or hydraulic dismounting methods and tools to allow safe, correct and efficient bearing dismounting.

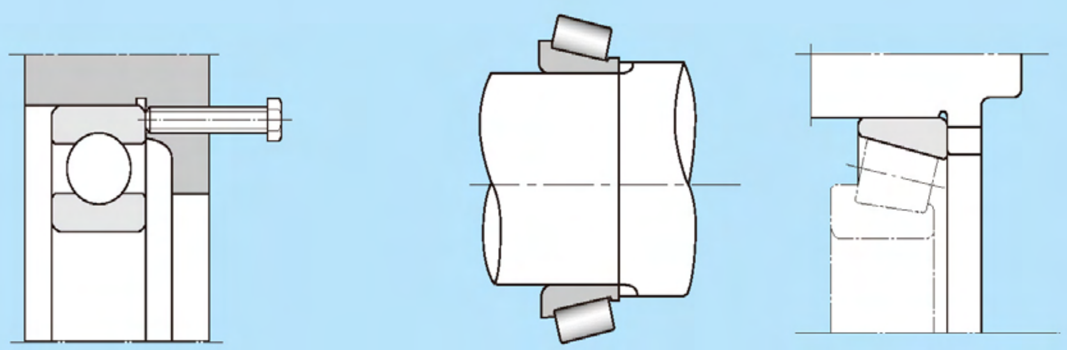
Mechanical dismounting

When removing inner and outer rings which have been installed with interference fits, proper disassembly pullers must be employed to prevent the internal damage of bearings' raceway or rolling elements. People should consider both puller type and maximum withdrawl capacity since it is crucial for completing any dismounting job safely and easily. Puller overload can result in breakage of the puller's arms or beam and therefore shuld be avoided. In general, it is recommended to use a three arm puller as shown following rather than two arm puller as three arm puller is more stable and has stronger withdrawal capacity. .



a b Press disassembly

Although a claw tool is used and then dismount the bearing, it may still spoil the bearing if shaft or housing design did not take disassembly into consideration. HCH suggest our customers can provide knockout slots, puller grooves or axial holes on adjacent bearing parts for easy bearing removal.

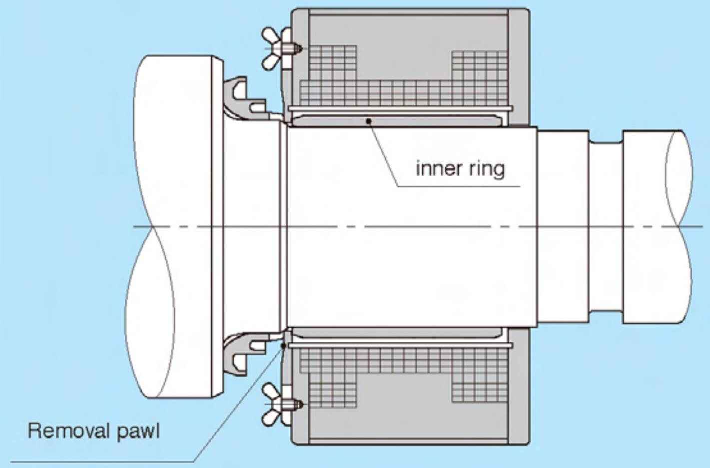


Outer ring disassembly bolt

Removal slots or puller grooves to ease removal.

Dismounting using heat

The inner rings of cylindrical roller bearings generally have a tight interference fit, which requires high forces to dismount. In such cases, using a puller can cause damage to the shaft and ring, and can be hazardous to the operator. Using heating equipment facilitates easy and quick dismounting while reducing the risk of damage to the ring and shaft. HCH offers a range of heating equipment, which includes aluminium heating rings as well as adjustable and fixed induction heaters, for dismounting cylindrical roller bearing inner rings.

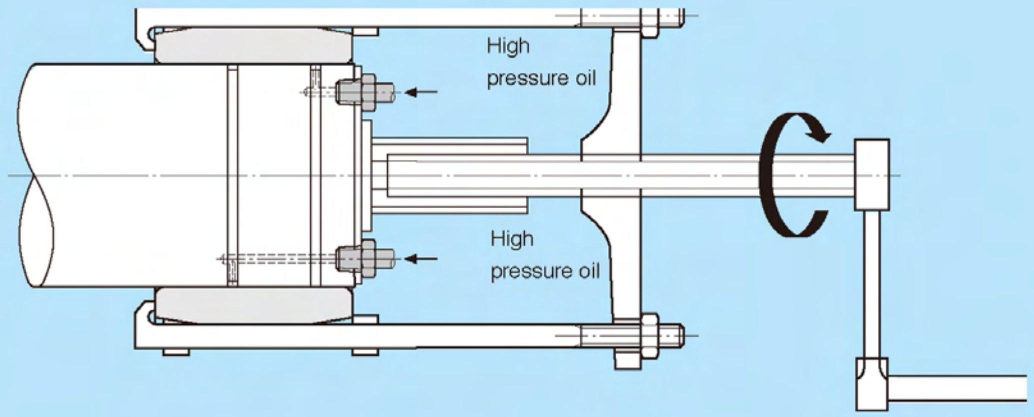


As shown in left picture, a removal pawl, or tool, can also be used to dismount the inner ring when using the induction heating method described above.

Removal of inner ring using an induction heater

Dismounting bearings using hydraulic techniques

The HCH hydraulic techniques are often the preferred method for dismounting larger bearings as well as other components. These techniques, which employ hydraulic pumps, nuts and oil injectors, allow the application of substantial forces to dismount bearings or other components.



Disassembly using high pressure oil (hydraulic)

Warning: large bearings, installed with tight fits, and having been in service for a long period of time, will likely have developed fretting corrosion on fitted surfaces and will require considerable dismounting force.

Inspection after mounting

To confirm that the bearing has been mounted correctly, perform a rotation check.

① First, rotate the shaft or housing to see whether anything unusual can be detected.
② Next, engage the bearing without applying a load, observing its rotational condition at low speed.
③ Then, slowly increase the rotational speed and load while checking for any rise in operational noise levels, vibrations and temperature.

● Noise inspection

Unusual noise or noise levels checking should be performed by well-trained staff who is familiar with the sound of properly mounted bearings operated under standard conditions. A clear, smooth and continuous running sound is normal. A high, metallic or irregular sound indicates some error in function. Vibration can be accurately checked with a vibration measuring instrument, and the amplitude and frequency characteristics measured against a fixed standard.

Irregularities		Possible Causes	Countermeasures
Noise	Loud Metallic Sound	Abnormal Load	Correction of fit, internal clearance, preload, position of housing shoulder, etc.
		Incorrect mounting	Correction of alignment of shaft and housing, accuracy of mounting method.
		Insufficient or improper lubricant	Replenish lubricant or select proper lubricant.
		Squeaking noise	Replacement by low-noise bearings, selection of small clearance bearings.
		Sliding of balls	Adjustment of preload, selection of small clearance bearings, or adoption of softer grease.
	Loud Regular Sound	Contact of rotating parts	Correction of labyrinth seal, etc.
		Flaws, corrosion, or scratches on the raceways	Replacement of bearing, cleaning, improvement of sals, and usage of clean lubricant.
		Brinelling	Replacement of bearing and careful handling.
		Flaking on the raceways	Replacement of bearing
	Irregular Sound	Excessive clearance	Correction of fit and clearance and correction of preload
		Penetration by foreign particles	Replacement of bearing, cleaning, improvement of seals, and relubrication using clean lubricant.
		Flaws or flaking on the ball surfaces	Replacement of bearing
		Excessive amount of lubricant	Reduce amount of lubricant, select stiffer grease.
Abnormal Temperature Rise		Insufficient or improper lubricant	Replenish lubricant or select proper lubricant.
		Abnormal load	Correction of fit, internal clearance, preload, position of housing shoulder.
		Incorrect mounting	Correction of alignment of shaft and housing, accuracy of mounting, or mounting method.
		Creep of fitted surfaces, excessive seal friction.	Correction of seals, replacement of bearing, correction of fit or mounting.
Vibration		Brinelling	Replacement of bearings and careful handling.
		Flaking	Replacement of bearing
		Incorrect mounting	Correction of squareness between shaft and housing shoulder or side of spacer
		Penetration by foreign particles	Replacement of bearing, cleaning, correction of seals.
Leakage or Discoloration of Lubricant		To much lubrication. Penetration by foreign particles or abrasion chips.	Reduce amount of lubricant, select stiffer grease. Replace bearing or lubricant. Clean housing and adjacent parts.

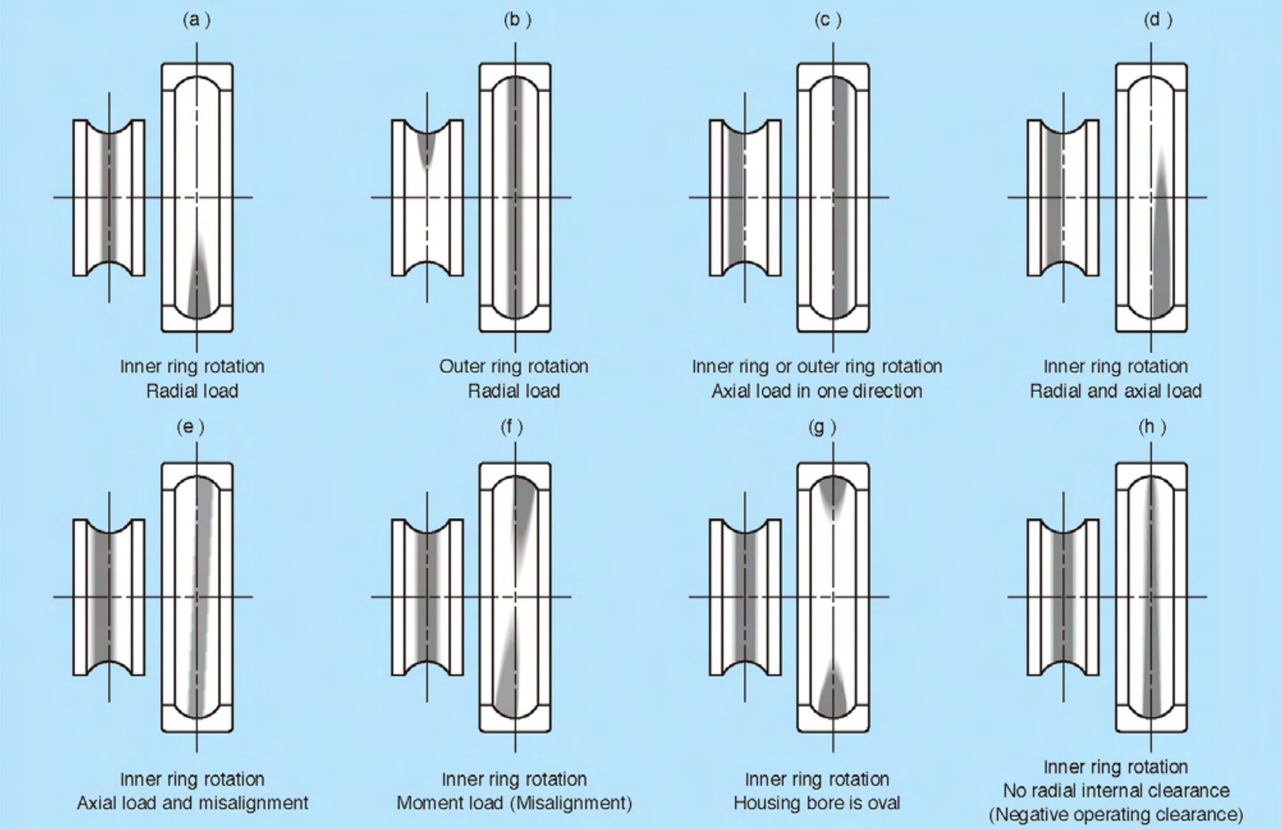
● Temperature inspection

Another method for for post installation running test is by temperature variation. Usually the bearing temperature can be estimated from the housing surface temperature. However, if the bearing outer ring is accessible through oil inlets, etc., the temperature can be more accurately measured. Under normal conditions, bearing temperature rises with rotation time and then reaches a stable operating temperature after a certain period of time. If the temperature does not level off and continues to rise, or if there is a sudden temperature rise, or if the temperature is unusually high, the bearing should be inspected.

● Running Traces inspection

As the bearing rotates, the raceways of the inner ring and the outer ring make contact with the rolling elements. This results in a wear path on both the rolling elements and raceways. It is normal for the running trace to be marked on the raceway, and the extent and shape of this running trace provides a useful indication of loading conditions.

It is possible to determine from careful observation of the running traces whether the bearing is carrying a radial load, a large axial load, or a moment load, or if there is extreme rigidity variations of the housing. Unexpected load applied on the bearing or excessive mounting error or the like can also be determined, providing a clue to the investigation of causes for bearing failure.



Typical Running Traces of Deep Groove Ball Bearings

Representative running traces of deep groove ball bearings are shown in above chart. Above chart (a) to (d), show general running traces under radial load or axial load. The running traces vary according to whether the load is fixed to the inner ring or the outer ring, and according to load conditions. Running traces (e) to (h) often cause bearing failure, and must be carefully observed.

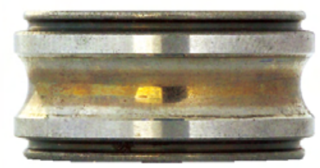
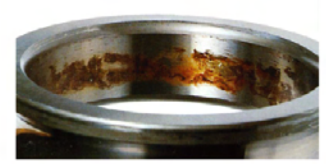
- (e) shows running traces with a shaft inclined due to misalignment.
- (f) is a running trace under a moment load.
- (g) is a running trace in a housing which is elliptically shaped and has poor inner diameter accuracy.
- (h) is a running trace in a bearing with insufficient internal clearance.

Trouble-shooting bearing problems

The bearing normally can be used up to the end of the rolling fatigue life if handled properly. If it fails earlier, it may caused by improper mounting, mishandling, poor lubrication, entry of foreign matter or abnormal heat generation. Considering possible causes of bearing failure and damages according to the condition of the machine on which the bearings failed and taking countermeasures is very important to prevent the recurrence of similar problems.

Following types of damage typically encountered are presented in the table below.

Damage type	Illustration of damage conditions	Possible Causes	Solutions
Flaking		• Over and excessive load • Improper handling and mounting • Poor shaft or housing accuracy etc. • Unsuitable bearing clearance • Installation error etc.	• Find out the cause of the heavy load • Examine operating conditions • Improve the mounting method • Check the precision of shaft and housing • Check the bearing internal clearance
Speckles		• Ingress of foreign particles	• Improvement in sealing • Improvement of the operating environment if possible • Filtration of the oil or grease
Cracking		• Excessive load • Excessive impacts • Rapid cooling • Excessive interference • Overheating by creeping • Very loose fit • Large flaking etc.	• Find out the cause of very large load • Improve the installation process • Correct the interference • Prevent the creep
Rust and Corrosion		• Ingress of water or corrosive material (such as acid) • Condensation of moisture contained in the air • Poor packaging and storing conditions • Handling with bare hands etc.	• Do not use bad quality varnish • Dry the varnish properly with sufficient time • Improvement in sealing effect • Careful handling of bearing
Mounting Flaws		• Inclination of inner and outer rings during mounting or dismounting. • Shock load during mounting or dismounting.	• Use appropriate jig and tool • Avoid a shock load by use of a press machine • Center the relative mating parts during mounting

Damage type	Illustration of damage conditions	Possible Causes	Solutions
Eletrical Corrosion		• Electrical potential difference between inner and outer rings • Electrical potential difference of a high frequency that is generated by instruments or substrates when used near a bearing	• Design electric circuits which prevent current flow through the bearings • Insulation of the bearing
Seizure		• The grease's maximum working temperature is lower than the actual temperature and the grease get failure. • Excessive rotational speed • Dissipation of heat generated by bearing is not enough. • Clearance too small. • Excessive load (or preload) • Installation error etc.	• Selection of suitable lubricant grease • Improve dissipation of heat from the bearing • Improvement in clearance and preload • Improvement in operating conditions
Fracture		• Impact during mounting • Excessive load • Poor handling such as dropping	• Improve the mounting method (shrink fit, use proper tools) • Reconsider the load conditions • Provide enough back-up and support for the bearing rib.
Fretting		• Poor lubrication • Vibration with a small amplitude • Insufficient interference	• Use a proper lubricant • Apply a preload • Check the interference fit • Apply a film of lubricant to the fitting surface
Cage Damage		• Poor mounting (Bearing misalignment) • Large moment load • Shock and large vibration • Excessive rotation speed, sudden acceleration and deceleration • Poor lubrication • Temperature rise	• Check the mounting method • Check the temperature, rotation, and load conditions • Reduce the vibration • Select a cage type • Select a lubrication method and lubricant

LUBRICATION SERVICE

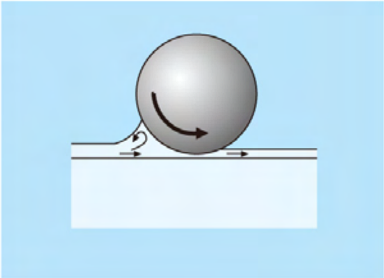
Adequate lubricant is necessary for the successful performance of all bearings. Properly selected lubricants can:

- Reduce running friction by providing a viscous hydrodynamic film of sufficient strength to support the load.
- Separate the balls from the raceways, preventing metal-to-metal contact.
- Minimize retainer wear by reducing sliding friction, thus reducing the development of noise.
- Prevent oxidation and corrosion of the internal elements.
- Act as an additional barrier to contaminants and serve as a heat transfer agent, conducting heat out of the bearing.

Lubrication can be accomplished by using either oil or grease. The most satisfactory bearing performance will be achieved by selecting the method most suitable for a specific application.

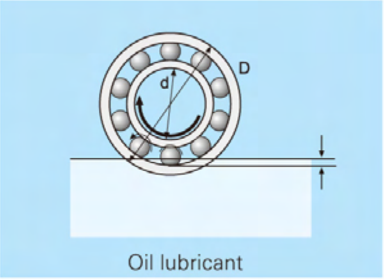
● Grease lubrication

Grease type lubricants are easier to handle than oil based lubricants since the bearings are pre-lubricated at the factory. The prelubricated bearing with grease is often more than adequate for the service life of the application and requires simple sealing devices. However, grease is hard to change since the usual method is to pump grease fitting and let the new grease push out the old grease. Generally, greases with low viscosity base oil are best suited for low temperatures and high speeds; while greases made from high viscosity base oils are suited for loads.



● Oil lubrication

Oil lubrication is considered better suited for high speed and high temperature applications. It is very effective for applications requiring bearing generated heat to dissipate outside. Viscosity of the oil determines the oil's lubricating efficiency. If the viscosity is too low, the oil film will not be sufficiently formed, and if the viscosity is too high, the viscosity resistance will also be high and cause temperature rise. For higher speed, low viscosity oil should be used, and for heavy loads, a higher viscosity oil should be used.



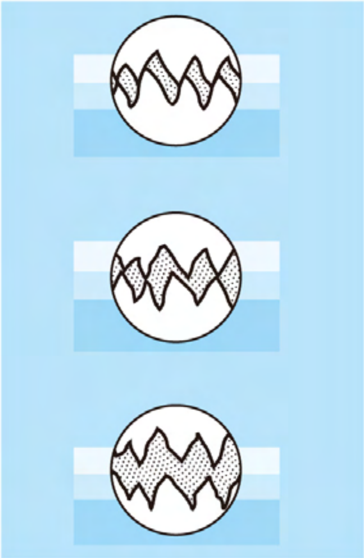
Oil lubricant

Oil lubrication is superior in lubricating efficiency; however, grease lubrication allows a simpler structure around the bearings. The following table compares oil and grease lubrication.

Operating factor	Grease lubrication	Oil lubrication
Housing Structure and Sealing method	Simple	Maybe complex Careful maintenance required
Speed	Limiting speed is 65% to 80% of that of oil lubrication	High limiting Speed
Cooling effect	Poor	Heat transfer is possible using forced oil circulating lubrication.
Fluidity	Poor	Good
Long lasting	Good	Poorer than grease
Full lubricant replacement	Sometimes difficult	Easy
Removal of foreign matter	Removal of particles from grease is impossible	Easy
External contamination due to leakage	Surrounding seldom contaminated by leakage	Often leaks without proper countermeasures. Not suitable if external contamination must be avoided

Lubricating conditions

The following lubricating conditions exist in a rolling bearing.



a) Boundary lubrication

Under boundary lubrication conditions, load is transmitted via continuous surface contact. In combination with poor lubrication this may lead to extreme wear and premature bearing failure. Some heavy-duty grease containing suitable additives for wear protection can also be considered.

b) Mixed lubrication

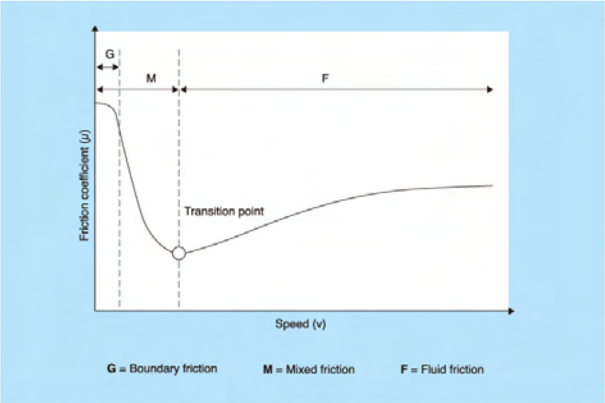
Under mixed-friction lubrication conditions, variable load are transmitted partly between the surface asperities across the roughness peaks in contact with one another, and partly via a lubricant film. Wear protection additives should be used under these conditions to prevent excessive wear.

c) Full-film lubrication

Full-film lubrication is the optimum lubrication condition characterized by complete separation of the surfaces by a load-bearing lubricant film. Depending on the internal friction of the lubricant, extremely low friction coefficients can be attained.

● Friction

Theoretically, the resistance met by the rolling elements when contacting the bearing raceways is assumed to be of a purely rolling nature. In practice, however, partial sliding may occur between the rolling elements and the raceways. Sliding between the cage and the rolling elements can also occur resulting in churning or displacement of the lubricant.



The frictional moments, and hence the friction coefficient μ, are dependent on the load, the lubricating condition and the bearing speed. The stribeck diagram left shows the friction coefficient μ relative to the speed v. In the diagram, we can discern the three different lubricating conditions: boundary lubrication, mixed lubrication and full-film lubrication.

The curve of the friction coefficient μ indicates the increase and decrease of the frictional moments that occur in line with a temperature increase and decrease in the bearing.

● Viscosity ratio

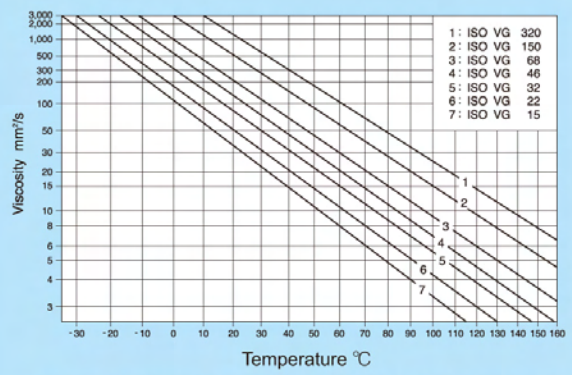
The importance of the oil viscosity for the formation of an oil film is to separate the bearing surfaces and thus for the life of the bearing is dealt with the viscosity ratio K as following:

$$K = \frac{V}{V_1}$$

where,
K = 3 viscosity ratio
V = actual operating viscosity of the lubricant mm²/s
V_1 = rated viscosity depending on the bearing mean diameter and rotational speed, mm²/s

Note: The information applies equally to the base oil viscosity of greases.The base oil of viscosity normally used for rolling bearings lies between 15 and 500 mm²/s at 40 °C. Greases based on oils having higher viscosities than 1 000 mm²/s at 40 °C. Bleed oil so slowly that the bearing will not be adequately lubricated. Therefore, if a calculated viscosity well above 1000 mm²/s at 40 °C. is required because of low speeds, it is better to use a grease with a maximum viscosity of 1000 mm²/s and good oil bleeding properties or to apply oil lubrication.

Oil Lubrication



Relation between lubricating oil viscosity and temperature

For lubricating oils, viscosity is one of the most important properties and determines an oil's lubricating efficiency. If viscosity is too low, formation of the oil film will be insufficient, and damage will occur to the load carrying surfaces of the bearing. If viscosity is too high, viscous resistance will also be great and result intemperature increases and friction loss. In general, for higher speed applications lower viscosity oil should be used; for heavier load applications, higher viscosity oil should be used. In regard to operating temperature and lubrication, the left chart shows an oil viscosity operating temperature comparison for the purpose of selecting a lubrication oil with viscosity characteristics appropriate to an application.

Examples of Selection Lubricating Oils

Operating Temperature	Speed	Light or normal Load	Heavy or Shock Load
-30 to 0°C	Less than limiting speed	ISO VG 15, 22, 32 (refrigerating machine oil)	–
0 to 50°C	Less than 50% of limiting speed 50 to 100% of limiting speed More than limiting speed	ISO VG 32, 46, 68 (bearing oil, turbine oil) ISO VG 15, 22, 32 (bearing oil, turbine oil) ISO VG 10, 15, 22 (bearing oil)	ISO VG 46, 68, 100 (bearing oil, turbine oil) ISO VG 22, 32, 46 (bearing oil, turbine oil) –
50 to 80°C	Less than 50% of limiting speed 50 to 100% of limiting speed More than limiting speed	ISO VG 100, 150, 220 (bearings oil) ISO VG 46, 68, 100 (bearing oil, turbine oil) ISO VG 32, 46, 68 (bearing oil, turbine oil)	ISO VG 150, 220, 320 (bearing oil) ISO VG 68, 100, 150 (bearing oil, turbine oil) –
80 to 110°C	Less than 50% of limiting speed 50 to 100% of limiting speed More than limiting speed	ISO VG 320, 460 (bearing oil) ISO VG 150, 220 (bearing oil) ISO VG 68, 100 (bearing oil, turbine oil)	ISO VG 460, 680 (bearing oil, gear oil) ISO VG 220, 320 (bearing oil)

Note: If the operating temperature is near the high end of the temperature range listed in the left column, select high viscosity oil.

Oil lubrication methods

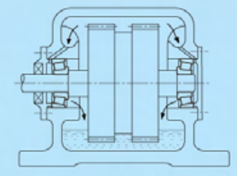
Several methods are available and are described below. The best method to use will depend on operating conditions.

● Oil bath lubrication

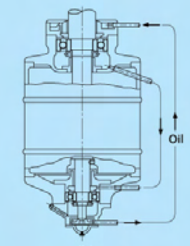
This is a common method used where bearings are operating below the listed oil limiting speed. The housing is designed to provide a sump through which the rolling elements of the bearing will pass. Generally, the oil level should be no higher than the center point of the lowest rolling element. If speed is high, lower oil levels should be used to reduce churning. Gages or controlled elevation drains are used to achieve and maintain the proper oil level.

● Drip feed lubrication

This lubrication method is often used for small bearings operated at relatively high speeds. In the illustration, a visible oiler is used. The oil drip rate is controlled by a screw valve located at the top of the oil cup. In case of drop lubrication, the number of drops should be appropriately adjusted according to the specific conditions; however, several drops per minute should be sufficient under normal conditions.

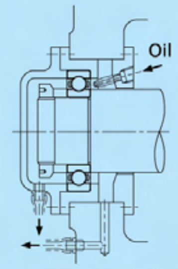


Splash Lubrication

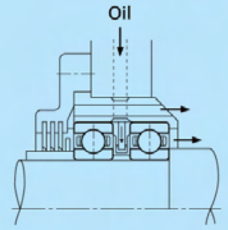


(b)

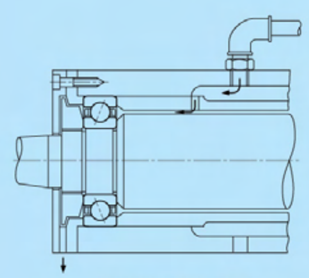
Circulating Lubrication



Jet Lubrication (a)



Jet Lubrication (b)



Oil Mist Lubrication

● Splash lubrication

In this lubricating method, oil is splashed onto the bearings by gears or by a simple rotating disc. This method is commonly used in automobile transmissions, differentials and gear boxes. The illustration shows splash lubrication used on a reduction gear.

● Circulating system

This method is commonly used for high speed operation and for bearings used at high temperatures. A typical circulating oil system consists of an oil reservoir, pump, piping and filter. As shown in the illustration, oil from the supply pipe circulates through the bearings and exits to an external reservoir. After cooling in the reservoir, it returns to the bearing through a pump and filter. In this system, the oil outlet should be larger in diameter than the supply pipe so that an excessive amount of oil will not remain in the housing.

This system has the advantages of:

An adequate supply of oil for both cooling and lubrication.

Metered control of the quantity of oil delivered to each bearing.

Removal of contaminants and moisture from the bearing by flushing action.

Suitability for multiple bearing installations.

Large reservoir, which reduces deterioration.

● Jet Lubrication

Jet lubrication is often used for ultra-high speed bearings, such as the bearings in machine tool spindles. In this method, lubricating oil is sprayed under pressure from one or more nozzles directly into the rolling elements of the bearing. The lubricant quantity will be adjusted by the oil pressure and the nozzle bore diameter. It is important to ensure that the supplied oil will not accumulate at the bearing section. The illustration shows an example of typical jet lubrication.

● Oil-mist lubrication.

Oil-mist lubrication is called also oil fog lubrication, uses air to atomize the oil and carry it into the bearing. This system permits close control of the amount of lubricant reaching the bearings. The air is filtered and supplied under sufficient pressure to assure adequate lubrication of the bearings. This method is used in high-speed, continuous operation applications. To ensure "wetting" of the bearings and to prevent possible damage to the rolling elements and races, it is imperative that the oil mist system be turned on for several minutes before the equipment is started. The importance of "wetting" the bearing before starting cannot be overstated and has particular significance for equipment that has been idled for extended periods of time.

● Oil replacement

The need for oil replacement depends on operating conditions. If a bearing is used at a temperature of 50°C or lower in a favorable environment with little dust or dirt, an oil change intervals up to one year should be sufficient. If the bearing is used at an operating temperature exceeding 100°C with an operating temperature exceeding 100°C with an external heat source, the oil should be replaced every two or three months or more frequently, even if the used oil is thermally stable.

Typical oil lubrication guidelines

The properties and characteristics of lubricants for typical open type deep groove ball bearings and tapered roller bearing applications are listed. These general characteristics have resulted from long successful performance in these applications.

Suggested General Purpose R&O lubricating oil properties	
Base stock	Solvent refined,High viscosity-index petroleum oil
Additives	Corrosion and oxidation inhibitors
Viscosity index	80 min
Pour point	-10 ° C max.
Viscosity grades	ISO through 32 to 220

Suggested industrial EP gear oil properties	
Base stock	Solvent refined, high viscosity index petroleum oil
Additives	Corrosion and oxidation inhibitors. Extreme pressure (EP) additive*
Viscosity index	80 min
Pour point	-10 ° C max.
Viscosity grades	ISO 100, 150, 220, 320, 460

Industrial extreme pressure (EP) gear oil

Extreme pressure gear oils are used to lubricate HCH bearings in all types of heavily loaded industrial equipment. They should be capable of withstanding heavy loads including abnormal shock loads common in heavy-duty equipment.

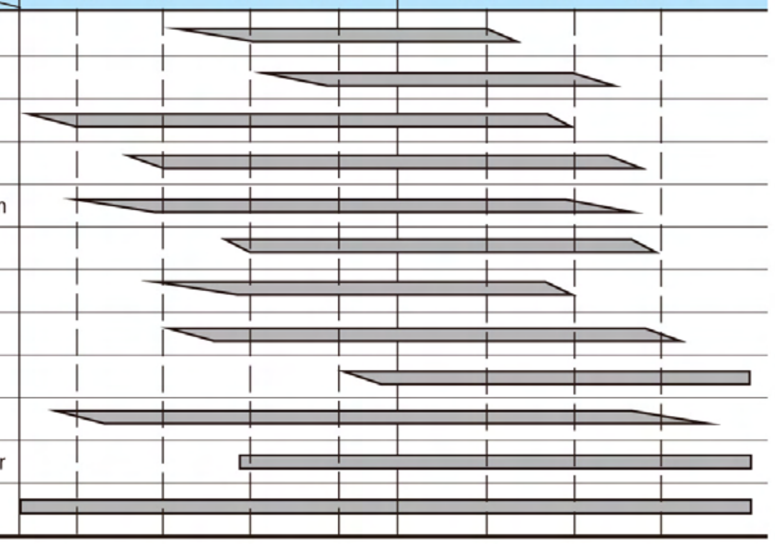
Industrial EP gear oils should be composed of a highly refined petroleum oil-based stock plus appropriate inhibitors and additives. They should not contain materials that are corrosive or abrasive to bearings. The inhibitors should provide long-term protection from oxidation and protect the bearing from corrosion in the presence of moisture. The oils should resist foaming in service and have good water separation properties. An EP additive protects against scoring under boundary-lubrication conditions. The viscosity grades suggested represent a wide range. High temperature and/or slow-speed applications generally require the higher viscosity grades. Low temperatures and/or high speeds require the use of lower viscosity grades.

Oil varieties and characteristics

Lubricating oil used for bearing is usually a highly refined petroleum or synthetic oil which has a high film strength and superior oxidation and corrosion resistance. Petroleum oils have physical and chemical properties that can help in the selection of the correct oil for any bearing application. Synthetic oils cover a broad range of categories, and include polyalphaolefins, silicones, polyglycols, and various esters. In general, synthetic oils are less prone to oxidation and can operate at extreme hot or cold temperatures.

Physical properties such as pressure-viscosity coefficients tend to vary between oil types and caution should be used when making oil selections. The polyalphaolefins (PAO) have hydrocarbon chemistry, which parallel petroleum oil both in their chemical structures and pressure-viscosity coefficients. The silicone, ester and polyglycol oils have an oxygen based chemistry that is structurally quite different from petroleum oils and PAO oils. This difference has a profound effect on reductions in bearing fatigue life and increases in bearing wear could result from this reduction of lubricant film thickness.

Synthetic oil	Item	Usage temperature range (Low-temperature property) -60 -40 -20 0 ℃ (High-temperature property) 100 150 200 ℃
Mineral oil	Low-viscosity mineral oil	
	High-viscosity mineral oil	
Ester	Diester	
	Polyol ester	
Synthetic hydrocarbon	Low and medium viscosity hydrocarbon	
	High-viscosity hydrocarbon	
Polyglycol		
Phenyl ether	Diphenyl ether	
	Poly phenyl ether	
Silicone		
PFAE	High-viscosity perfluoro alkyl polyether	
	High-VI perfluoro alkyal polyether	



Synthetic oil	Item	Lubricity	Temperature viscosity	Heat-resisting property	Oxidation stability	Rubber compatibility	Remarks
Mineral oil	Low-viscosity mineral oil	○	△	×	×	△	Not recommended for use at high temperatures.
	High-viscosity mineral oil	○	○	◎	△	○	Not recommended for use at low temperatures.
Ester	Diester	◎	◎	△	△	×	Excellent low-temperature performance; May swell rubber.
	Polyol ester	◎	◎	○	○	×	Performs well over a wide temperature range; May swell rubber.
Synthetic hydrocarbon	Low and medium viscosity hydrocarbon	○	◎	○	○	◎	Performs well over a wide temperature range with little adverse effect on rubber and plastics because of its molecular structure without a polar group.
	High-viscosity hydrocarbon	○	◎	◎	○	○	However, not recommended for natural rubber and EPDM parts.
Polyglycol		△		○	△	△	Little adverse effect on rubber including natural rubber and EPDM; Shows little electric resistance.
Phenyl ether	Diphenyl ether	○	◎	◎	◎	○	Excellent heat and oxidation stabilities and better anti-radiation property.
	Poly phenyl ether	○	×	◉	◉	○	Excellent heat and oxidation stabilities and better anti-radiation property; Low viscosity index, resulting in somewhat poor low-temperature fluidity.
Silicone		×	◉	◉	◉	◉	Excellent in heat and oxidation stabilities, while somewhat poor in lubricating boundary surface of steel vs. steel.
PFAE	High-viscosity perfluoro alkyl polyether	○	◎	◉	◉	◉	Has the best heat and oxidation stabilities among all oils available; Excellent in chemical resistance,insoluble in organic solvent, and of low vapor pressure.
	High-VI perfluoro alkyal polyether	○	◉	◉	◉	◉	

◉ : Excellent ◎ : Very good ○ : Good △ : Fair × : Poor

Grease lubrication

Lubricating grease consists of a mineral or synthetic oil combined with a thickener. The thickeners are usually metallic soaps. However, other thickeners, e.g. polyurea can be used for superior performance in certain areas, i.e. high temperature applications. Additives can also be included to enhance certain properties of the grease. The consistency of the grease depends largely on the type and concentration of the thickener used and on the operating temperature of the application. When selecting a grease, the consistency, operating temperature range, viscosity of the base oil, rust inhibiting properties and the load carrying ability are the most important factors to be considered. Detailed information of these properties is as follows.

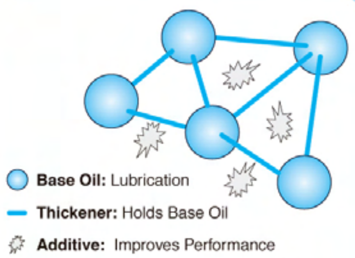
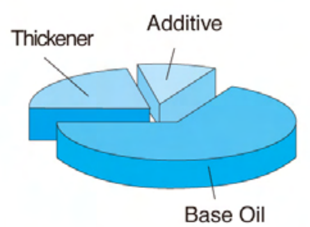


Image of Grease Structure



Composition of Grease

● 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 low-viscosity base oil is suitable for low temperatures or low loads, while grease with high-viscosity base oil is suitable for high temperature and, or high loads. Grease using high-viscosity base oil has superior high-temperature and high-load characteristics.

● Thickening agents

Thickening agents are compounded with base oils to maintain the semi-solid state of the grease. Thickening agents consist of two types of bases, metallic soaps and non-soaps. Metallic soap thickeners include: lithium, sodium, calcium, etc. Non-soap base thickeners are divided into two groups: inorganic (silica gel, bentonite, etc.) and organic (polyurea, fluorocarbon, etc.). The various special characteristics of a grease, such as limiting temperature range, mechanical stability, water resistance, etc., depending largely on the type of thickening agent used. For example, sodium-based grease is generally poor in water resistance properties, while greases with bentone, polyurea and other non-metallic soaps as the thickening agent are generally superior in high temperature properties.

● Additives

Refer to an agent that provides extreme pressure and rust resistance. Grease often contains a variety of additives such as anti-oxidants, rust preventatives, and extreme pressure enhancers to give it special properties. Various additives are added to grease to improve various properties and efficiency. For example, there are anti-oxidants, high-pressure additives (EP additives), rust preventives and anti-corrosives. For bearings subject to heavy loads and shock loads, grease containing high-pressure additives should be used. For comparatively high operating temperatures or in applications where the grease cannot be replenished for long periods, grease with an oxidation stabilizer is best to use.

● Oil seperation

All grease has a tendency to "bleed". This means the seperation of the base oil from the thickener. The amount of oil seperation varies with the type of grease, its oil viscosity and thickener characteristics. If there is an extended standstill period during operation, oil seperatioin can occur if the wrong grease was selected.

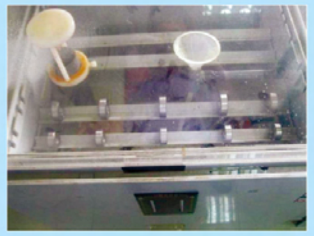
Salt Spray Test Report

Test Date:	October 27, 2012	Test No.:	1210003
Test Duration:	from 3:00pm on 27th to 3:00pm on 28th	Total:	24 hours
Spray time:	24 hours		

1. Sodium chloride quality	Analytically pure (pecial for Salt spray test)
2. Distilled water quality	Pure drinking water
3. Spray device:	1.6 ml /80cm/h
3.1 spray volume	1.5
3.2 Collect the concentration or proportion of solution under room temp.	1.0339(5% sodium chloride)
3.3 PH	6.9
4.Sample test:	cover the bearing surface with FUCHS anti-rust oil No.230L (coating compactness and rust-preventing characteristic of FUCHS No.230L anti-rust oil)
4.1 Type	High carbon chromium bearing steel
4.2 Shape	6204 ZZ&6205 ZZ
4.3 scale	
4.4 number	10pcs
5. The compressed air pressure	1.00Kgf/cm
6. Laboratory relative humidity	86%
7. Laboratory temperature	35°C
8. Pressure barrel temperature	47°C
9. Salt water barrel temperature	35°C
10. Others(room temperature)	22°C
Judgment: Qualified	1. Make judgment in accordance with the standard drawing : no peeling of, no blistering, no rust or any other phenomenon
According to GB/72423.17-93	2. In accordance with the other methods : Qualified
Operator: Rong Guoxiang	Audit:Chen Yinjun

Attached images


The appearance of the salt spray chamber


Testing process


Bearing appearance after testing


Bearing appearance after testing

Grease brands and their nature

It should be noted that even for the same type of grease composition, different brands of grease may have different properties. There are hundreds of types of grease can be found in HCH for a specific application. Following are common types in HCH for your choice. Further information please consult HCH engineers.

● Frequently used grease brands and properties:

Manufacturer	No.	Product	Base Oil	Thickener	Viscosity		Working temperature range	0.1 mm Worked Penetration	Dropping point	Remark
					40℃	100℃				
kyodo Yushi	1	Multemp SRL	Polyol ester+Diester	Lithium hydroxy Stearate	26	5.1	-50℃~+150℃	250	190℃	Low noise, Long life, High temperature
	2	Multemp SB-M	Synthetic hydrocarbon	Diurea	47.6	8.9	-40℃~+200℃	220	260℃	High temperature, High speed operation, Low noise property
	3	Raremax SUPER N	Mineral oil +Synthetic	Diurea	95.9	10.5	-40℃~+180℃	260	255℃	Heat resistance, Oxidation stability
	4	Multemp PS2	Diester oil + mineral oil	Lithium	15.3	4.7	-50℃~+130℃	275	195℃	Heat resistance, Oxidation stability
	5	Mutemp SRH	Polyol ester	Lithium	83	—	-40℃~+150℃	250	200℃	Low noise, Long Life, Wide temperature change
	6	Multemp SRM	Polyol ester	Lithium	55	—	-50℃~+150℃	250	195℃	Low noise, Long Life, Corrosion resistance, Low start torque
	7	Multemp ET-K	Synthetic ether+Polyol ester	Aromatic diurea	95.1	12.3	-40℃~+200℃	300	230℃	High temperature, Long life, Anti-radiation
Kluber	8	BQH72-102	PAO, Ester oil	Polyurea	100	11	-40℃~+180℃	265~285	> 250℃	Low noise, Long life, High temperature
	9	BEP72-82	PAO, Ester oil	Polyurea	70	9.4	-40℃~+180℃	250~280	> 250℃	High temperature, Available for auto generator bearings
	10	ASONIC GLY32	PAO, Ester oil	Lithium	25	5	-50℃~+140℃	265~295	> 190℃	Low noise, Long life, High temperature
	11	ASONIC Q74-73	PAO, Ester oil	Polyurea	67.5	10	-40℃~+160℃	220~250	> 190℃	Low noise, Long life, High temperature
	12	Isoflex LDS 18 special A	Ester oil, mineral oil	Lithium	15	3.5	-50℃~+120℃	265~295	> 190℃	Available for pump bearings, motor and its accessory
	13	ISOFLEX TOPAS NB 52	Synthetic ether+Polyol ester	Barium complex soap	30	5.5	-50℃~+120℃	265~295	> 190℃	Available for pump bearings, motor and its accessory
	14	PETAMO GHY133	Mineral oil, PAO	Polyurea	150	18	-30℃~+160℃	265~295	> 250℃	High temperature, Available for auto generator bearings
	15	BE 41-542	Mineral oil	Lithium	540	32	-20℃~+140℃	265~295	> 230℃	High load-carrying capacity,excellent wear protection
	16	BE 31-502	Mineral oil	Polyurea	500	31	-10℃~+140℃	245~275	> 190℃	High resistance to mechanodynamical loads,good adhesion
Shell	17	AV2	Mineral oil	Lithium	130	12.2	-25℃~+120℃	275	185℃	Wide applications available
	18	RLQ2	Mineral oil	Lithium	75	8.3	-20℃~+120℃	266	195℃	Wide applications available
	19	RL2	Mineral oil	Lithium	75	8	-20℃~+120℃	275	180℃	Low noise, High temperature, Corrosion resistance
	20	RL3	Mineral oil	Lithium	75	8	-30℃~+120℃	235	180℃	Not good for transmission
Exxon Mobil	21	Polyrex EM	Mineral	Diurea	115	12.2	-29℃~+177℃	305	288℃	Low temperature
	22	Unirex N3	Mineral oil	Lithium	115	—	-40℃~+190℃	235	230℃	Long Life, Corrosion resistance, Water resistance
	23	BEACON325	Diester	Lithium	12	4	-54℃~+120℃	285	183℃	High speed, Low noise, Corrosion resistance
	24	Unirex N2	Mineral oil	Lithium	115	—	-40℃~+190℃	280	230℃	Long life, Corrosion resistance, Water resistance
	25	Mobil grease 28	Diester	Micro adhesive	29.3	—	-60℃~+177℃	280	310℃	Very stable in high temperature
Cosmo	26	EMQ2(SBR)	Mineral oil	Lithium	110.8	11.98	-30℃~+130℃	265	195℃	Water resistance, Prominent oxidation stability
	27	PNG	Mineral oil	Lithium	73.5	6.88	-30℃~+130℃	265	197℃	Low noise, Long life
Chevron	28	SRI 2	ISOSYN	Polyurea	100	11	-30℃~+150℃	280	243℃	Water resistance, Corrosion resistance, High temperature
Lubcon	29	N2	PAO, Ester oil	Polyurea	150	22	-40℃~+180℃	265~295	> 250℃	High temperature, Low noise
Dupont	30	Krytox240	Fluorinated	PTFE	200	25	-34℃~+288℃	285	—	High temperature
Jinzhi	31	Hangu 2	Mineral oil	Lithium	150	12	-20℃~+120℃	265~295	198℃	Wide applications available

Grease characteristics

There are several types of thickeners, each with its own special characteristics and advantages for specific applications. The most common kinds of thickeners are lithum soap and polyeurea. Please check the following chart for the detailed characteristics of geases that are composed of varities of different thickeners and base oils.

● Grease varieties and characteristics

Thickener	Base oil	Service temperature range [℃]	Drop point	Suitability for rolling bearings
Lithium soap	Mineral oil	-35 to 130	< 200	◉
	PAO	-50 to 150	< 200	◎
	Ester oil	-65 to 150	< 200	◉
	Silicone oil	-60 to 170	< 200	◎
Aluminium complex soap	Mineral oil	-30 to 160	> 230	◉
Barium complex soap	Mineral oil	-30 to 140	> 220	◉
	Synthetic hydrocarbon	-50 to 150	> 220	◉
Sodium complex soap	Mineral oil	-30 to 160	> 220	◉
	Silicone oil	-50 to 200	> 220	◉
Calcium complex soap	Mineral oil	-30 to 130	> 220	◉
	Ester oil	-40 to 120	> 220	◎
Lithium complex soap	Mineral oil	-30 to 140	> 230	◉
	PG	-30 to 150	> 230	◎
	Ester oil	-40 to 180	> 230	◉
Bentonite	Mineral oil	-20 to 160	–	◎
Polyurea	Mineral oil	-20 to 160	> 250	◎
	PAO	-40 to 160	> 230	◎
				◉
	Ester oil	-40 to 180	> 230	
Synthetic (PE, PTFE, FEP)	Silicone oil	-50 to 200	> 230	◎
	Alkoxy fluorine oil	-40 to 250	not measurable	◎


Thickener	Water resistance	Corrosion protection	Pressure resistance	Machenical stability	Application notes
Lithium soap	◉	◎	◎	◉	General purpose grease
	◎	◎	◎	◎	Low-temperature grease
	○	○	○	◉	High & low temperature grease
					High speed & wear characteristics
	◎	△	△	◎	High & low temperature grease
Aluminium complex soap	◉	◉	◎	◉	High-temperature grease
Barium complex soap					EP grease
					High-speed & long-term grease
Sodium complex soap	○	◉	◎	◉	EP grease & high-speed grease
	◎	○	△	◉	High-speed & low temperature grease
					Low-temperature grease
Calcium complex soap	◎	◉	◎	◉	High-temperature grease
	◎	◎	◎	◎	Long-term grease
Lithium complex soap	○	◎	◎	◉	General purpose grease
	○	◎	◎	◎	EPDM-compatible
	◎	◎	◎	◉	High-temperature & long–term grease
Bentonite	◎	△	△	◎	High-temperature grease
Polyurea	○	△	◉	◎	High-temperature grease
	◉	◎	○	◎	Long-term lubrication
	◉	◎	○	◉	High-temperature & lifetime grease
Synthetic (PE, PTFE, FEP)	◉	○	△	◎	High-temperature grease
	◉	◎	◎	◎	High-temperature grease

◉ : Excellent ◎ : Very good ○ : Good △ : Fair × : Poor

● Compatibility with elastomers and plastics

Besides miscibility with other greases, the compatibility of a lubricant with elastomers and plastics should be tested prior to its use. Such tests normally consist of measuring shore hardness, tensile strength and elongation at tear, etc..

a) Elastomers

	Mineral oil	Synth hydro-carbon	Ester oil	Poly-glycol	Silicone oil	PFPE	Polyphenyl ether
NBR	+	+*	+/-	+/-	+	+	+
HNBR/NEM	+	+	+/-	+/-	+	+	+
FPM/FKM	+	+	+	+	+	+	+
EPDM	-	-	-	+	+	+	-
ACM	+	+	+/-	+	+	+	+
AU	+	+/-	+/-	+/-	+	+	+/-

* slight shrinkage in most cases ** with white oil *** without additives

b) Plastics

	Mineral oil	Synth hydro-carbon	Ester oil	Poly-glycol	Silicone oil	PFPE	Polyphenyl ether
POM	+	+	+	+	+/-	+	+
PA	+	+	+	+	+	+	+
PE	+/-	+/-	+/-	+	+	+	+/-
PC	+**	+	-	+	+	+	-
ABS	+	+***	-	+/-	+	+	-
PTFE	+	+	+	+	+	+	+

\+ resistant +/- partially resistant - not resistant

Calculation of the bearing free space & grease volume

Bearing free space can be determined by means of the formula below. The formula uses volume units [cm^3, dm^3] instead of weight units [g, kg] in order to avoid calculation errors (up to 100%) due to the different densities of greases.

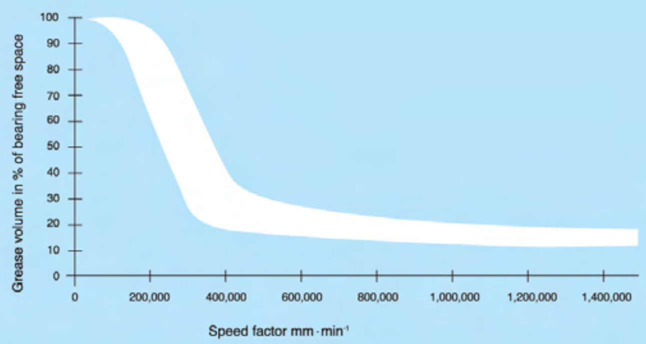
$$V=[\pi/4 \cdot B \cdot (D^2 - d^2) \cdot 10^{-9} - G/7800]\ m^3$$

d = bearing bore diameter [mm]
D = bearing outer diameter [mm]
B = bearing width [mm]
G = bearing weight [kg]

Notes: Because of different bearing types, cages and designs, the a.m. formula does only give a rough estimation.

Once the bearing free space has been determined, the required grease quantity is calculated as a percentage of the available free space. The correct quantity is important to ensure proper lubrication of all contact surfaces.Over lubrication can be just as detrimental as under lubrication: for example, too much grease in high-speed bearings may lead to overheating or the development of higher starting and running torque. Generally, grease quantity in a bearing should occupy about 25% to 35% of the free internal space. In some extremely high speed applications, the lubricant can be as low as 10% to 15%.

A rule that should be generally observed is: low operating temperatures = long service life of both the grease and the bearing. The right table provides an overview of the required grease quantities as a percentage of bearing free space for various speed factors in [mm · min1].



● Consistency

The consistency is the degree of the grease plasticity. Grease with low consistency should be selected for application in very low-temperature ranges in order to guarantee the continuous release of oil. At higher temperatures, a higher consistency is recommendable, so that the grease does not soften too much. Greases with a stiffer consistency are preferred and beneficial for applications with outer ring rotation where centrifugal force tends to sling grease out of the bearing, and those vertical axis applications where gravity pulls grease away from its intended position.

Notes: Penetration is a measure which indicates consistency - the solidity of grease. A measurement device has a cone with a specified time. Penetration is the depth to which the cone penetrates (in units of 1/10 mm).

NLGL NO.	ASTM WORKED (penetration)
0	355~385
1	310~340
2	265~295
3	220~250
4	175~205
5	130~160
6	85~115

Grease is numbered differently by the grease manufacturers. Numbers 250 and 300 of cup and fiber grease generally use penetration (at 25℃), while most versatile grease employ NLGI penetration numbers such as 0, 1 and 2.

Running torque

The grease characteristics (viscosity penetration, etc.) and grease quantity can influence the bearing running torque; further to influence the bearing running speed and result in an increase of bearing temperature.
Generally speaking, if the lubricant quantity is increased, the running torque increases, which may result in temperature generation and being speed fluctuation. Also, with the same grease volume, bearing with high viscosity will come out more running torque than those with low viscosity.

Following table lists some frequent problems which are caused by increased running torque.

Types	Description	Causes	Countermeasures
Temperature Generation	Bearing temperature is going to be higher and higher.	Churning from several lubricant couplings. Churning type lubricants cause temperature generation because of the shearing effect.	Reduce the lubricant quantity. Change to a channeling type of lubricant.
Failure to Reach Speed	Sometimes motors cannot reach the designed nominal speed.	Excessive grease quantity.	Selection of a soft churning type of lubricant.
Excessive Power Consumption	The motors or engines are higher consuming the power than that was estimated.	The reasons for this are the same as "Failure to Reach Speed" above. Lubricants with high worked penetration can also add to the problem.	Reduce the grease quantity. Change lubricant type.
Excessive Reach Running Current	The high running current of the motors during operation.	Grease quantity. Worked penetration of lubricant softness level of a churning type of lubricant.	Selection of a soft churning type of lubricant. Reduce the grease quantity.
Speed Fluctuations	When bearing is within high rotating condition, speed fluctuations come.	This is caused by grease "slumping" into the pathway of the ball.	Reduce grease quantity. Use a channeling type grease. Use a much softer type of churning grease.

● The speed factor n · dm for lubricating greases

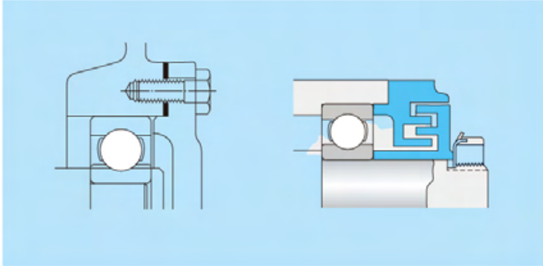
The attainable speed factor of lubricating grease depends largely on its base oil type, viscosity, thickener type, and of course the bearing type used. Under high-speed bearing operating conditions it is important to achieve a constant oil supply.

Grease types	Base oil viscosity at approx, 40℃ [mm^2/s]	Speed factor $n \cdot d_m$
Mineral / lithium / MoS2	1000 to 1500	50,000
Mineral / lithium complex	400 to 500	200,000
Mineral / lithium / MoS2	150 to 200	400,000
Ester / polyurea	70 to 100	700,000
Ester / lithium complex	15 to 30	1,600,000
Ester / polyurea	15 to 30	2,000,000

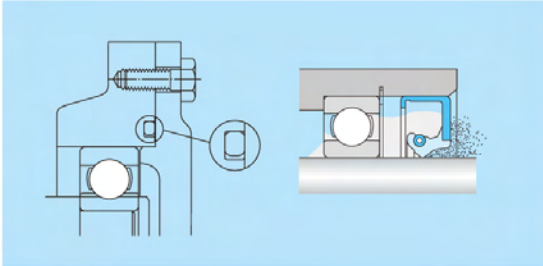
External sealing arrangement

External seals also have two main functions: to prevent lubricating oil from leaking out, and to prevent dust, water and other contaminants from entering the bearing. Similar to internal bearing seals, the external bearing sealing devices also divided into two types, non-contact type and contact type.

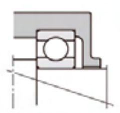
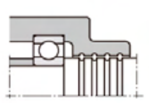
Non-contact seals: Non-contact radial shaft seals function by virtue of the sealing effect of a narrow, relatively long gap which can be arranged axially, radially or in combination. Non-contact seals, which range from simple gap-type seals to multistage labyrinths are practically without friction and do not wear, making them suitable for high speed applications. In order to improve sealing capability, clearance spaces are often filled with lubricant.



Contact seals: Seals in contact are used to seal passages between machine components. These dynamic seals accomplish their sealing action through the contact pressure of a resilient part of the seal (the lip is often made of synthetic rubber) and the sealing surface. These contact seals are generally far superior to non-contact seals in sealing efficiency, although their friction torque and temperature rise coefficients are higher.

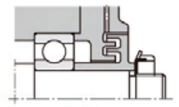
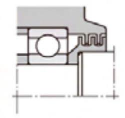
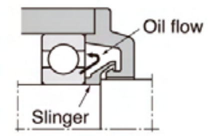
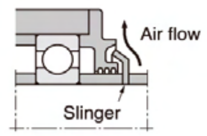
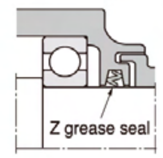
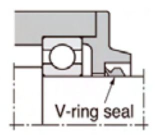
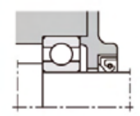
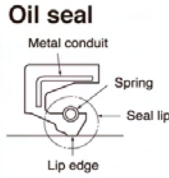
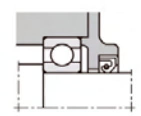


The following chart lists the special characteristics of seals and other points to be considered when choosing an appropriate seal.

Type	Seal construction	Name	Seal characteristics	Selection considerations
Non-contact seals		**Clearance seal**	This is an extremely simple seal design with a small radial clearance.	**Cautionary points regarding selection** ● In order to improve sealing efficiency, clearances between the shaft and housing should be minimized. However, care should be taken to confirm shaft/bearing rigidity and other factors to avoid direct shaft-housing contact during operation.
		**Oil groove seal** (oil grooves on housing side)	Several concentric oil grooves are provided on the housing inner diameter to greatly improve the sealing effect.	**Oil groove clearance (reference)**
		**Oil groove seal** (oil grooves on shaft and housing side)	Oil grooves are provided on both the shaft outer diameter and housing inner diameter for a seal with even greater sealing efficiency.	● Oil groove width, depth (reference) width : 2 5 mm depth : 4 5 mm ● Three or more oil grooves should be provided. ●Sealing efficiency can be further improved by filling the oil groove portion with grease of which the viscosity grade is 150 to 200.

Oil groove clearance (reference)

Shaft diameter mm	Clearance mm
Up to 50	0.2 ~ 0.4
50 or above	0.5 ~ 1.0

Type	Seal construction	Name	Seal characteristics	Selection considerations
Non-contact seals		**Axial labyrinth seal**	This seal has a labyrinth passageway on the axial side of the housing.	Labyrinth clearance (reference) ●Grease is generally used as the lubricant for labyrinth seals, ●Sealing efficiency can be further improved by filling the labyrinth passageway with grease of which the viscosity grade is 150 to 200. ● Labyrinth seals are suitable for high speed applications.
		**Radial labyrinth seal**	A labyrinth passageway is affixed to the radial side of the housing. For use with split housings. This offers better sealing efficiency than axial labyrinth seals.	
		**Internal slinger**	By providing a slinger inside the housing, centrifugal force guides the lubricant flow back on the bearing and helps prevent it from dirtying the work environment.	**Cautionary points regarding selection** ● By installation on the revolving shaft, these seal types make use of centrifugal force to aid lubrication, seal in lubricant and prevent the entrance of contaminants. ● Installation of a slinger inside the outside of the housing further enhances the sealing in of lubricants.
		**External slinger**	By mounting a slinger on the outside of the housing, centrifugal force helps to prevent dust and other solid contaminants from entering.	●Installation of a slinger slinger on the outside of the housing will provide greater protection against dust and other bearing contaminants.
Contact seals		**Z grease seal**	In cross section resembling the letter "Z," this seal's empty spaces are filled with grease. The seal is commonly used with a plummer block (bearing housing).	**Shaft surface roughness (reference)**
		**V-ring seal**	This design enhances sealing efficiency with a lip that seals from the axial direction. With the aid of centrifugal force, this seal also offers effective protection against dust, water, and other contaminants entering the bearing. Grease can be used on both sides of the seal.	**Shaft material (reference)**
	 For dust proof	**Oil seal** 	Oil seals are widely used, and their shapes and dimensions are standardized under JIS B 2402. In this design, a ring-shaped spring is installed in the lip section. As a result, optimal contact pressure is exerted between the lip edge and shaft surface, and sealing efficiency is good.	**Cautionary points regarding selection** When the oil seal and the bearing are in very close proximity, internal bearing clearances are sometimes too small to accommodate the additional heat generated by friction between the seal and shaft. In addition to considering the heat generated by contact seals at various peripheral speeds, internal bearing clearances must also be selected with caution.
	 For preventing lubricant leakage		Depending upon the direction in which the lip faces (in toward the bearing or away from the bearing) contact oil seals are very effective at preventing lubricant leakage from the housing or contaminants from infiltrating the bearing.	

Labyrinth clearance (reference)

Shaft diameter mm	Clearance mm	
	Radial direction	Axial direction
Up to 50	0.2 ~ 0.4	1.0 ~ 2.0
50 200	0.5 ~ 1.0	3.0 ~ 5.0

Shaft surface roughness (reference)

Peripheral speed m/s	Surface roughness	
	Ra	Rmax
Up to 50	0.8a	3.2s
5 10	0.4a	1.6s
10	0.2a	0.8s

Shaft material (reference)

Material	Machine structural carbon steel Low carbon alloy steel Stainless steel
Surface hardness	HRC 40 or higher necessary HRC 55 or higher advisable
Processing mothod	Final grinding without repeat (moving), or buffed after hard chrome plating

BEARING LOAD CALCULATION

To compute bearing loads, the forces which act on the shaft being supported by the bearing must be determined. These forces include the inherent dead weight of the rotating body (the weight of the shafts and components themselves), loads generated by the working forces of the machine, and loads arising from transmitted power.

Bearing load distribution

For shafting, the static tension is considered to be supported by the bearing, and any loads acting on the shafts are distributed to the bearings. The applied bearing loads can be found by using following formulas:

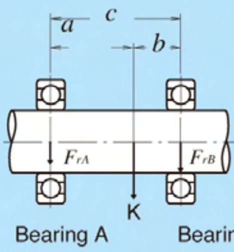
$$F_{rA} = \frac{b}{c} K \qquad F_{rB} = \frac{a}{c} K$$

where

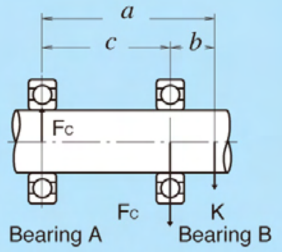
F_{rA} : Radial load applied on bearingA(N), {kgf}

F_{rB} : Radial load applied on bearingB (N), {kgf}

K : Shaft load (N), {kgf}



Radial Load Distribution (1)



Radial Load Distribution (2)

When two or more loads are applied simultaneously, The radial loads on bearings A and B can be calculated using the following equations:

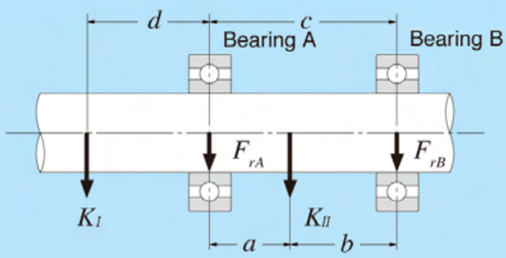
$$F_{rA} = \frac{d+c}{c} K_I + \frac{b}{c} K_{II} \qquad F_{rB} = -\frac{d}{c} K_I + \frac{a}{a+b} K_{II}$$

Where,

F_{rA} : Radial load on bearing A, N

F_{rB} : Radial load on bearing B, N

K_I , K_{II} : Radial load on shaft N



Chain / belt shaft load

The tangential loads on sprockets or pulleys when power (load) is transmitted by means of chains or belts can be calculated by formula:

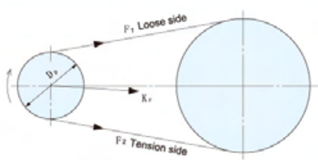
$$K_t = \frac{19.1 \text{x} 10^6 \cdot H}{Dp \cdot n} \text{ N} = \frac{1.95 \text{ x} 10^6 \cdot H}{Dp \cdot n} \text{ kgf}$$

where,

K_t : Sprocket/pulley tangential load, N

H : Transmitted force, kW

D_p: Sprocket/pulley pitch diameter, mm



$$K_r = f_b \cdot K_t$$

where,

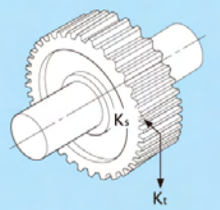
K_r : Sprocket or pulley radial load, N

f_b : Chain or belt factor

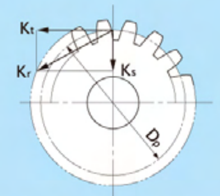
For belt drives, an initial tension is applied to give sufficient constant operating tension on the belt and pulley. Taking this tension into account, the radial loads acting on the pulley are expressed by formula. For chain drives, the same formula can also be used if vibrations and shock loads are taken into consideration.

Chain or belt type	f_b
Chain (single)	1.2~1.5
V-belt	1.5~2.0
Timing belt	1.1~1.3
Flat belt (w / tension pulley)	2.5~3.0
Flat belt	3.0~4.0

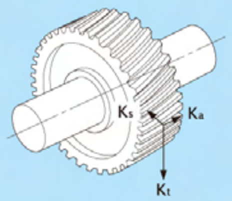
Load generated under gear transmission



Spur gear loads



Radial resultant forces



Helical gear loads

The loads operating on gears can be divided into three main types according to the direction in which the load is applied; i.e. tangential (Kt), radial (Kr), and axial (Ka). Loads acting on planetary shaft gears are depicted in the pictures.The load magnitude can be found by using or formulas:

$$K_t = \frac{19.1 \times 10^6 \cdot H}{D_p \cdot n} \text{ N} = \left\{\frac{1.95 \times 10^6 \cdot H}{D_p \cdot n}\right\} \text{ kgf}$$

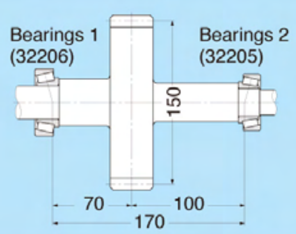
$$K_s = K_t \cdot \tan\alpha \text{ (Spur gear)} = K_t \cdot \frac{\tan\alpha}{\cos\beta} \text{ (Helical gear)}$$

$$K_r = \sqrt{K_t^2 + K_s^2} \qquad K_a = K_t \cdot \tan\beta \text{ (Helical gear)}$$

where,

K_t : Tangential gear load (tangential force), N
K_s : Radial gear load (separating force), N
K_r : Right angle shaft load (resultant force of tangential force and separating force), N
K_a : Parallel load on shaft, N
H : Transmission force , kW
n : Rotational speed, r/min
D_p : Gear pitch circle diameter, mm
α : Gear pressure angle
β : Gear helix angle

Spur gear diagram



Example: What are the rated lives of the two tapered roller bearings supporting the shaft shown in left chart? Bearing1 is an 32206 with a Cr = 54.5 kN, and bearing 2 is an 32205 with a Cr = 42.0 kN. The spur gear shaft has a pitch circle diameter Dp of 150 mm, and a pressure angle of 20°. The gear transmitted force HP = 150 kW at 2,000 r/min (speed factor n).

The gear load from formulas is:

$$K_t = \frac{19.1 \times 10^6 \cdot H}{D_p \cdot n} = \frac{19{,}100 \times 150}{150 \times 2{,}000} = 9.55\text{kN } \{974\text{kgf}\}$$

$$K_s = K_t \cdot \tan\alpha = 9.55 \times \tan 20^\circ = 3.48\text{kN } \{355\text{kgf}\}$$

$$K_r = \sqrt{K_t^2 + K_s^2} = \sqrt{9.55^2 + 3.48^2} = 10.16\text{kN } \{1040\text{kgf}\}$$

The radial loads for bearings are:

$$F_{r1} = \frac{100}{170} K_r = \frac{100}{170} \times 10.16 = 5.98\text{kN } \{610\text{kgf}\}$$

$$F_{r2} = \frac{70}{170} K_r = \frac{70}{170} \times 10.16 = 4.18\text{kN } \{426\text{kgf}\}$$

$$\frac{0.5F_{r1}}{Y_1} = 1.87 > \frac{0.5F_{r2}}{Y_2} = 1.31$$

The equivalent radial load is:

$$P_{r1} = F_{r1} = 5.98\text{kN } \{610\text{kgf}\}$$

$$P_{r2} = XF_{r2} + Y_2 \frac{0.5F_{r1}}{Y_1} = 0.4 \times 4.18 + 1.67 \times 1.87 = 0.466\text{kN } \{475\text{kgf}\}$$

From formula the life factor, fh, for each bearing is:

$$f_{h1} = f_n \frac{C_{r1}}{P_{r1}} = 0.293 \times 54.5/5.98 = 2.67$$

$$f_{h2} = f_n \frac{C_{r2}}{P_{r2}} = 0.293 \times 42.0/4.66 = 2.64$$

Therefore: Tapered roller bearings life

$$L_{h1} = 13{,}200 \times a_2 = 13{,}200 \times 1.4 = 18{,}480 \text{ ore}$$

$$L_{h2} = 12{,}700 \times a_2 = 12{,}700 \times 1.4 = 17{,}780 \text{ ore}$$

The combined bearing life, Lh, from formula is:

$$L_{h1} = \frac{1}{\left[\frac{1}{L_{h1}^{e}} + \frac{1}{L_{h2}^{e}}\right]^{1/e}} = \frac{1}{\left[\frac{1}{18{,}480^{9/8}} + \frac{1}{17{,}780^{9/8}}\right]^{8/9}} = 9{,}780 \text{ hour}$$

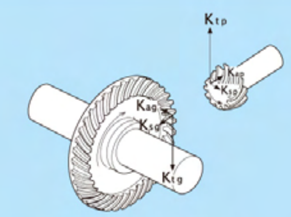
● Gear factor f_g

Because the actual gear load also contains vibrations and shock loads as well, the theoretical load obtained by the above formula should also be adjusted by the gear factor f_g as shown in the table below:

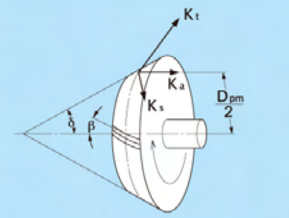
Gear type	f_g
Precision ground gears (Pitch and tooth profile errors of less than 0.02mm)	1.05～1.1
Ordinary machined gears (Pitch and tooth profile errors of less than 0.1mm)	1.1～1.3

● Load on bevel gears

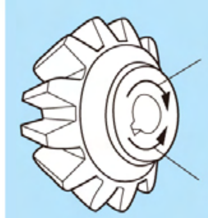
For spiral bevel gears, the direction of the load varies depending on the direction of the helix angle, the direction of rotation, and which side is the driving side or the driven side. The directions for the separating force (Ks) and axial load (Ka) shown in the left picture are positive directions. The direction of rotation and the helix angle direction are defined as viewed from the large end of the gear. The gear rotation direction in the picture is assumed to be clockwise (right).



Loads on bevel gears



Bevel gear diagram



Clockwise rotation

Counterclockwise rotation

The calculation methods for these gear loads are shown in table as following:

Pinion		Rotation direction	Clockwise	Counter clockwise	Clockwise	Counter clockwise
		Helix direction	Right	Left	Left	Right
Tangential load K_t			$K_t = \frac{19.1 \times 10^6 \cdot H}{D_{pm} \cdot n}$ $\left\{\frac{1.95 \times 10^6 \cdot H}{D_{pm} \cdot n}\right\}$			
Straight[1] bevel gears	Separating force K_s	Driving side	$K_s = K_t \left[\tan\alpha \frac{\cos\delta}{\cos\beta}\right]$			
	Axial load K_a	Driven side	$K_a = K_t \left[\tan\alpha \frac{\sin\delta}{\cos\beta}\right]$			
Spiral[1,2] bevel gears	Separating force K_s	Driving side	$K_s = K_t \left[\tan\alpha \frac{\cos\delta}{\cos\beta} + \tan\beta \sin\delta\right]$		$K_s = K_t \left[\tan\alpha \frac{\cos\delta}{\cos\beta} - \tan\beta \sin\delta\right]$	
		Driven side	$K_s = K_t \left[\tan\alpha \frac{\cos\delta}{\cos\beta} - \tan\beta \sin\delta\right]$		$K_s = K_t \left[\tan\alpha \frac{\cos\delta}{\cos\beta} + \tan\beta \sin\delta\right]$	
	Axial load K_a	Driving side	$K_a = K_t \left[\tan\alpha \frac{\sin\delta\delta}{\cos\beta\beta} - \tan\beta \cos\delta\right]$		$K_a = K_t \left[\tan\alpha \frac{\sin\delta}{\cos\beta} + \tan\beta \cos\delta\right]$	
		Driven side	$K_a = K_t \left[\tan\alpha \frac{\sin\delta}{\cos\beta} + \tan\beta \cos\delta\right]$		$K_a = K_t \left[\tan\alpha \frac{\sin\delta}{\cos\beta} - \tan\beta \cos\delta\right]$	

where,

D_{pm} : Mean pitch circle diameter, mm
δ : Pitch cone angle

Herein, to calculate gear loads for straight bevel gears, the helix angle β = 0.

Mean load

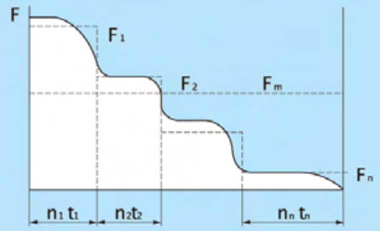
The load on bearings used in machines under normal circumstances will, in many cases, fluctuate according to a fixed time period or planned operation schedule. The load on bearings operating under such conditions can be CONVERTED TO A MEAN LOAD (Fm), this is a load which gives bearings the same life they would have under constant operating conditions.

(1) Fluctuating stepped load

The mean bearing load, F_m, for stepped loads is calculated from following formula. F_1 , F_2 ...... F_n are the loads acting on the bearing; $n_1, n_2 ....n_n$ and $t_1, t_2 .... t_n$ are the bearing speeds and operating times respectively.

$$F_m = \left[\frac{\sum (F_i^{\,p}\, n_i t_i)}{\sum (n_i t_i)}\right]^{1/p}$$

Where,
$p = 3$ For ball bearings
$p = 10/3$ For roller bearings



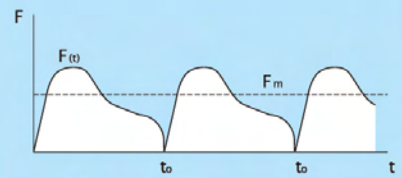
Stepped load

(2) Consecutive series load

Where it is possible to express the function $F_{(t)}$ in terms of load cycle to and time t, the mean load is found by using formula as following:

$$F_m = \left[\frac{1}{t_o}\int_o^{t_o} F(t)^p\, d_t\right]^{1/p}$$

Where,
$p = 3$ For ball bearings
$p = 10/3$ For roller bearings

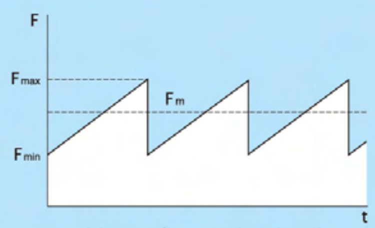


Time function series load

(3) Linear fluctuating load

The mean load, F_m, can be approximated by formula:

$$F_m = \frac{F_{min} + 2F_{max}}{3}$$

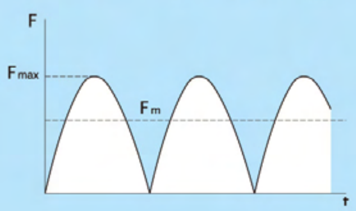


Linear fluctuating load

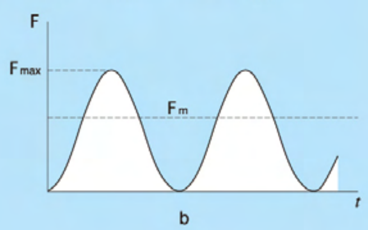
(4) Sinusoidal fluctuating load

The mean load, F_m, can be approximated by following formulas:

case (a) $F_m = 0.75\, F_{max}$



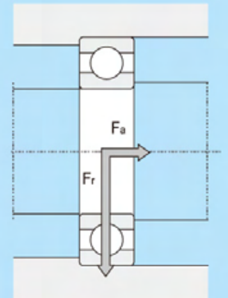
case (b) $F_m = 0.65\, F_{max}$



Sinusoidal variable load

Bearing size selection

Example1:



Deep groove ball bearing: 62 series
Required service life: more than 10000h

Radial load	$F_r = 2\ 000$N
Axial load	$F_a = 300$N
Rotational speed	$n = 1\ 600\ \text{min}^{-1}$

1.The dynamic equivalent load (P_r) is hypothetically calculated

The resultant value, F_a/F_r=300/2000=0.15, is smaller than any other values of e in the bearing specification table

Hence, HCH can consider that $P_r = F_r = 2\ 000$N

2.The required basic dynamic load rating (C_r) is calculated according to equation

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

$$= 2\ 000 \times \left(10\ 000 \times \frac{60 \times 1600}{10^6}\right)^{1/3}$$

$$= 19730 \text{ N}$$

3. Among those covered by the bearing specification table, the bearing of the 62 series with C_r exceeding 19730N is 6206, with bore diameter for 30mm.

4.The dynamic equivalent load obtained at step 1 is confirmed by obtaining value e for 6206.

Where C_{0r} of 6206 is l2.8 kN,and f_0 is 13.8

$$f_0 \times F_a/C_{0r} = 13.8 \times 300/12\ 800 = 0.323$$

Then, value e can be calculated using proportional interpolation

$$e = 0.19 + (0.22\text{-}0.19) \times \frac{(0.323\text{-}0.172)}{(0.345\text{-}0.172)}$$
$$= 0.216$$

As a result,it can be confirmed that

F_a/F_r=0.15<e

Hence, $P_r = F_r$

Example2: Deep groove ball bearing: 63 series
Required service life: more than 10000 h

Radial load	$F_r = 4\ 000$N
Axial load	$F_a = 2\ 400$N
Rotational speed	$n = 1000\ \text{min}^{-1}$

1. The hypothetic dynamic equivalent load (P_r) is calculated: Since $F_a/F_r = 2\ 400/4\ 000$=0.6 is much larger than the value specified in the bearing specification table, it suggests that the axial load affects the dynamic equivalent load. Hence, assuming that $X = 0.56$, Y=1.6 (approximate mean value of Y).

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

2. The required basic dynamic load rating(G_r)is:

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

$$= 6\ 080 \times \left(10\ 000 \times \frac{60 \times 1000}{10^6}\right)^{1/3}$$

$$= 51280 \text{ N}$$

3.From the bearing specification table, a 6310 with a bore diameter of 50 mm is selected as a 63 series bearing with C_r exceeding 51 280 N

4. The dynamic equivalent load and basic rating life are confirmed,by calculating the value e for a 6310 Values obtained using the proportional interpolation are:

where

$f_0 \times F_a/C_{0r} = 13.2 \times 2\ 400/35\ 340 = 0.89$

$e = 0.27$, Y= 1.6

$F_a/F_r = 0.6 > e$

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

$$P_r = XF_r + XF_a$$
$$= 0.56 \times 4\ 000 + 1.61 \times 2400 = 6\ 080\text{N}$$

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

$$= \frac{10^6}{60 \times 1\ 000} \times \left(\frac{57.54\times 10^3}{6\ 080}\right)^3 = 14\ 127 \text{ h}$$

5. The basic rating life of the 6309, using the same steps,is:

$L_{10h} \doteq 9\ 000$ h which does not satisfy the service life requirement.