1. Bearing materials

1.1 Raceway and rolling element materials

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

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

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

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

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

1.2.3 Steel cages
For temperatures exceeding 120°C, steel cages are required to use.

2. External bearing sealing devices

External seals have two main functions: to prevent lubricating oil/grease which will leaking out, also to prevent dust, water, and other contaminants from entering the bearing. 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 and resultant heat increase, and cost.

Sealing devices for rolling bearings fall into two main classifications: non-contact seals and contact seals.

2.1 Non-contact seals:
Non-contact seals utilize a small clearance between the shaft and the housing cover. Therefore friction is negligible, making them suitable for high speed applications. In order to improve sealing capability, clearance spaces are often filled with lubricant.

2.2 Contact seals:

Contact seals accomplish their sealing action through the contact pressure of a resilient the seal (the lip is often made of synthetic rubber) the sealing surface. Contact seals are generally far superior to non-contact seals in sealing efficieny, although their friction torque and temperature rise coefficients are higher. Furthermore, because the portion of a contact seal rotates while in contact with the shaft, the allowable seal peripheral speed varie depending on seal type.

3. Ball bearing tolerances

3.1 Standard of tolerances

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

Table 3.1 Comparison of tolerance classifications of national standards

Standard		Tolerance class				
International Organization for Standardization (ISO)	ISO	Normal class Class 6X	Class 6	Class 5	Class 4	Class 2
Deutsches Institut fur Normung (ISO)	DIN	P0	P6	P5	P4	P2
American National Standards Institute (ANSI)	ANSI/ABMA	ABEC-1	ABEC-3	ABEC-5	ABEC-7	ABEC-9

3.2 Tolerances for radial bearings

Table 3.2 Inner rings **(Unit : μm)**

Nominal bore diameter d mm		Single plane mean bore diameter deviation $\triangle d_{mp}$										Single radial plane bore diameter variation V_{dp}									
		class 0		class 6		class 5		class 4 ❶		class 2 ❶		diameter series 9					maxdiameter series 0.1				
												class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2
over	incl.	high	low	high	low	high	low	high	low	high	low	max.					max.				
2.5	10	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5
10	18	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5
18	30	0	-10	0	-8	0	-6	0	-5	0	-2.5	13	10	6	5	2.5	10	8	5	4	2.5
30	50	0	-12	0	-10	0	-8	0	-6	0	-2.5	15	13	8	6	2.5	12	10	6	5	2.5
50	80	0	-15	0	-12	0	-9	0	-7	0	-4.0	19	15	9	7	4.0	19	15	7	5	4.0
80	120	0	-20	0	-15	0	-10	0	-8	0	-5.0	25	19	10	8	5.0	25	19	8	6	5.0

Table 3.3 Inner rings

Nominal bore diameter *d* mm		Single radial plane bore diameter variation V_{dp} max diameter series 2,3,4					Mean single plane bore diameter variation V_{dmp}					Inner ring radial runout K_{ia}					Face runout with bore S_d		
		class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2
over	incl.	max.					max.					max.					max.		
2.5	10	6	5	4	3	2.5	6	5	3	2.0	1.5	10	6	4	2.5	1.5	7.0	3.0	1.5
10	18	6	5	4	3	2.5	6	5	3	2.0	1.5	10	7	4	2.5	1.5	7.0	3.0	1.5
18	30	8	6	5	4	2.5	8	6	3	2.5	1.5	13	8	4	3.0	2.5	8.0	4.0	1.5
30	50	9	8	6	5	2.5	9	8	4	3.0	1.5	15	10	5	4.0	2.5	8.0	4.0	1.5
50	80	11	9	7	5	4.0	11	9	5	3.5	2.0	20	10	5	4.0	2.5	8.0	5.0	1.5
80	120	15	11	8	6	5.0	15	11	5	4.0	2.5	25	13	6	5.0	2.5	9.0	5.0	2.5

Table 3.4 Inner rings

Nominal bore diameter *d* mm		Inner ring axial runout (with side) S_{ia} ❷			Inner ring width deviation $\triangle B_s$										Inner ring width varietion V_{Bs}				
					normal						modified ❸								
		class 5	class 4	class 2	class 0,6		class 5,4		class 2		class 0,6		class 5,4		class 0	class 6	class 5	class 4	class 2
over	incl.				high	low	high	low	high	low	high	low	high	low	max.				
2.5	10	7	3	1.5	0	-120	0	-40	0	-40	0	-250	0	-250	15	15	5	2.5	1.5
10	18	7	3	1.5	0	-120	0	- 80	0	- 80	0	-250	0	-250	20	20	5	2.5	1.5
18	30	8	4	2.5	0	-120	0	-120	0	-120	0	-250	0	-250	20	20	5	2.5	1.5
30	50	8	4	2.5	0	-120	0	-120	0	-120	0	-380	0	-380	20	20	5	3.0	1.5
50	80	8	5	2.5	0	-150	0	-150	0	-150	0	-380	0	-380	25	25	6	4.0	1.5
80	120	9	5	2.5	0	-200	0	-200	0	-200	0	-380	0	-380	25	25	7	4.0	2.5

Note: ❶ The dimensional difference $\triangle ds$ of bore diameter to applied for class 4 and 2 is the same as the tolerance of dimentional difference $\triangle dmp$ of average bore diameter. However, the dimensional difference is applied to diameter series 0, 1, 2, 3 and 4 against Class 4, and to all the diameter series against Class 2.

❷ To be applied for deep groove ball bearing and angular contact ball bearings.

❸ To be applied for individual raceway rings manufactured for combined bearing use.

Table 3.5 Outer rings

(Unit : μm)

Nominal Outside diameter *D* mm		Single plane mean outside diameter deviation $\triangle D_{mp}$										Single radial plane outside diameter variation V_{Dp}									
												diameter series 9					max diameter series 0.1				
		class 0		class 6		class 5		class 4 ❺		class 2 ❺		class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2
over	incl.	high	low	high	low	high	low	high	low	high	low	max.					max.				
6	18	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5
18	30	0	-9	0	-8	0	-6	0	-5	0	-4.0	12	10	6	5	4.0	9	8	5	4	4.0
30	50	0	-11	0	-9	0	-7	0	-6	0	-4.0	14	11	7	6	4.0	11	9	5	5	4.0
50	80	0	-13	0	-11	0	-9	0	-7	0	-4.0	16	14	9	7	4.0	13	11	7	5	4.0
80	120	0	-15	0	-13	0	-10	0	-8	0	-5.0	19	16	10	8	5.0	19	16	8	6	5.0
120	150	0	-18	0	-15	0	-11	0	-9	0	-5.0	23	19	11	9	5.0	23	19	8	7	5.0
150	180	0	-25	0	-18	0	-13	0	-10	0	-7.0	31	23	13	10	7.0	31	23	10	8	7.0
180	250	0	-30	0	-20	0	-15	0	-11	0	-8.0	38	25	15	11	8.0	38	25	11	8	8.0

Table 3.6 Outer rings

Nominal Outside diameter D mm		Single radial plane outside diameter variation V_{Dp} maxdiameter series 2.3.4					Single radial plane outside diameter variation V_{Dp} ❻ capped bearings diameter series		Mean single plane outside diameter variation V_{Dmp}				
		class 0	class 6	class 5	class 4	class 2	2,3,4 class 0	0,1,2,3,4 class 6	class 0	class 6	class 5	class 4	class 2
over	incl.	max.					max.		max.				
6	18	6	5	4	3	2.5	10	9	6	5	3	2.0	1.5
18	30	7	6	5	4	4.0	12	10	7	6	3	2.5	2.0
30	50	8	7	5	5	4.0	16	13	8	7	4	3.0	2.0
50	80	10	8	7	5	4.0	20	16	10	8	5	3.5	2.0
80	120	11	10	8	6	5.0	26	20	11	10	5	4.0	2.5
120	150	14	11	8	7	5.0	30	25	14	11	6	5.0	2.5
150	180	19	14	10	8	7.0	38	30	19	14	7	5.0	3.5
180	250	23	15	11	8	8.0	—	—	23	15	8	6.0	4.0

Table 3.7 Outer rings

Nominal Outside diameter D mm		Outer ring radial runout K_{ea}					Outside surface inclination S_D			Outside ring axial runout S_{ea} ❼			Outer ring width deviation $\triangle_{Cs}$	Outer ring width variation V_{Cs}			
		class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2	class 5	class 4	class 2		class 0,6	class 5	class 4	class 2
over	incl.	max.					max.			max.			all type		max.		
6	18	15	8	5	3	1.5	8	4	1.5	8	5	1.5	Identical to	Identical to	5	2.5	1.5
18	30	15	9	6	4	2.5	8	4	1.5	8	5	2.5	$\triangle_{Bs}$ of inner	$\triangle_{Bs}$ and V_{bs}	5	2.5	1.5
30	50	20	10	7	5	2.5	8	4	1.5	8	5	2.5	ring of same	of inner	5	2.5	1.5
50	80	25	13	8	5	4.0	8	4	1.5	10	5	4.0	bearing	ring of same	6	3.0	1.5
80	120	35	18	10	6	5.0	9	5	2.5	11	6	5.0		bearing	8	4.0	2.5
120	150	40	20	11	7	5.0	10	5	2.5	13	7	5.0			8	5.0	2.5
150	180	45	23	13	8	5.0	10	5	2.5	14	8	5.0			8	5.0	2.5
180	250	50	25	15	10	7.0	11	7	4.0	15	10	7.0			10	7.0	4.0

Note: ❺ The dimension al difference $\triangle_{Ds}$ of outer diameter to be applied for classes 4 and 2 is the same as the tolerance of dimensional difference $\triangle_{Dmp}$ of average outer diameter. However, the dimensional difference is applied to diameter series 0,1.2.3 and 4 against Class 4, and also to all the diameter series against Class 2.

❻ To be applied in case snap rings are not installed on the bearings.

❼ To be applied for Ball Bearings and Single-row Angular Contact Ball Bearings.

4. Bearing fits

4.1 Interference

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

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

4.2 The necessity of a proper fit

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

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

4.3 Fit selection

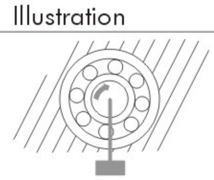
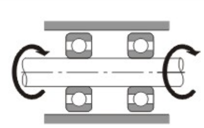
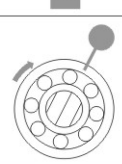
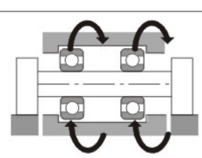
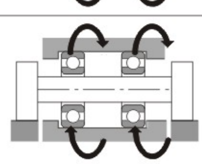
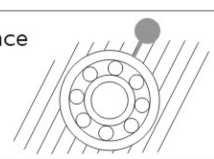
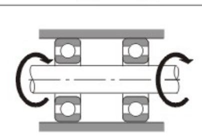
Selection of a proper fit is depend upon thorough analysis of bearing's operating conditions, including consideration of:

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

4.3.1 "Tight fit," "transition fit," or "loose fit"

For raceways under rotating loads, a tight fit is necessary. (Refer to Table 4.1) "Raceways under rotating loads " refers to raceways receiving loads rotating relative to their radial direction. For raceways under static loads, on the other hand, a loose fit is sufficient. (Example) Rotating inner ring load the direction of the radial load on the inner ring is rotating relatively. For non-separable bearings, such as ball bearing, it is generally recommended that either the inner ring or outer ring be given a loose fit.

Table 4.1 Radial load and bearing

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

5. Ball bearing internal clearance

Ball bearing internal clearance (initial clearance) is the amount of internal clearance of a bearing before install on a shaft or in a housing. The classification of internal clearance values for **SLB** ball bearings are shown in tables 5.1 to 5.5

Table 5.1 Radial internal clearance of Ball Bearings **(Unit : µm)**

Nominal bore diameter d (mm)		C2		Normal		C3		C4		C5	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
2.5	6	0	7	2	13	8	23	-	-	-	-
6	10	0	7	2	13	8	23	14	29	20	37
10	18	0	9	3	18	11	25	18	33	25	45
18	24	0	10	5	20	13	28	20	36	28	48
24	30	1	11	5	20	13	28	23	41	30	53
30	40	1	11	6	20	15	33	28	46	40	64
40	50	1	11	6	23	18	36	30	51	45	73
50	65	1	15	8	28	23	43	38	61	55	90
65	80	1	15	10	30	25	51	46	71	65	105
80	100	1	18	12	36	30	58	53	84	75	120
100	120	2	20	15	41	36	66	61	97	90	140

Table 5.2 Radial internal clearance of Self-aligning Ball Bearings (for bearing with cylindrical bore) **(Unit : µm)**

Nominal bore diameter d (mm)		Bearing with cylindrical bore							
		C2		Normal		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
6	10	2	9	6	17	12	25	19	33
10	14	2	10	6	19	13	26	21	35
14	18	3	12	8	21	15	28	23	37
18	24	4	14	10	23	17	30	25	39
24	30	5	16	11	24	19	35	29	46
30	40	6	18	13	29	23	40	34	53
40	50	6	19	14	31	25	44	37	57
50	65	7	21	16	36	30	50	45	69
65	80	8	24	18	40	35	60	54	83

Table 5.3 Radial internal clearance of Self-aligning Ball Bearings (for bearing with tapered bore) **(Unit : μm)**

Nominal bore diameter d (mm)		Bearing with tapered bore							
		C2		Normal		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
6	10	—	—	—	—	—	—	—	—
10	14	—	—	—	—	—	—	—	—
14	18	—	—	—	—	—	—	—	—
18	24	7	17	13	26	20	33	28	42
24	30	9	20	15	28	23	39	33	50
30	40	12	24	19	35	29	46	40	59
40	50	14	27	22	39	33	52	45	65
50	65	18	32	27	47	41	61	56	80
65	80	23	39	35	57	50	75	69	98

Table 5.4 Radial internal clearance of Double Row Angular Contact Ball Bearings **(Unit : μm)**

Nominal bore diameter d (mm)		C2		Normal		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
-	10	6	12	8	15	15	22	22	30
10	18	6	12	8	15	15	24	30	40
18	30	6	12	10	20	20	32	40	55
30	50	8	14	14	25	25	40	55	75

Table 5.5 Radial internal clearance of Ball Bearings (60, 62 and 63 series) for electric motors **(Unit : μm)**

Nominal bore diameter d (mm)		Radial internal clearance CM Ball Bearings	
over	incl.	min.	max.
10(incl.)	18	4	11
18	24	5	12
24	30	5	12
30	40	9	17
40	50	9	17
50	65	12	22
65	80	12	22

6. Lubrication

6.1 Lubrication of rolling bearings

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

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

In order to achieve the above effects, the most effective lubrication method for the operating conditions must be selected. Also a good quality, reliable lubricant must be selected. In addition, an effectively designed sealing system that prevents the intrusion of damaging elements (dust, dirt and water, etc.) into the bearing interior, removes other impurities from the lubricant, and prevents lubricant from leaking to the outside, to be also a requirement.

6.2 Grease lubrication

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

6.2.1 Types and characteristics of greases

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

Standard greases and their characteristics are quite important. As performance characteristics of even same type of grease will vary widely choose, it is better to check the catalogue data when selecting a grease.

The standard grease selection of **SLB** Ball Bearings are:
(1) Alvania S2 : -25°C ~ +120°C
(2) Multemp SRL : -40°C ~ +150°C

Please also refer to page T-28 Greases for more reference.

7. Load rating and life

7.1 Bearing life

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

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

7.2 Basic rating life and basic dynamic load rating

A group of identical bearings when subjected to identical load and operating conditions will exhibit a wide diversify in their durability.

This " life " disparity can be accounted for by the difference in the fatigue of the bearing material itself. This disparity is considered statistically when calculating bearing life, and the basic rating life is defined as follows:

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

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

The relationship between the basic rating life, the basic dynamic load rating and the bearing load is given in formula (Fig. 7.1), also refer to table 7.1.

(Fig. 7.1) .... $L10 = \left(\frac{C}{P} \right)^{P}$

where,

$P = 3$ .... For ball bearings

L_{10} : Basic rating life 106 revolutions

C : Basic dynamic rating load, n

(Cr: radial bearings)

P : Equivalent dynamic load, n

(Pr: radial bearings)

The basic rating life can also be expressed in terms of hours of operation (revolution), it is calculated as shown in formula (7.2).

(Fig. 7.2) .... $L10h = 500 fh^{p}$

(Fig. 7.3) .... $fh \frac{C}{P} = fn$

(Fig. 7.4) .... $fn = \left(\frac{33.3}{n} \right)^{1/p}$

where,

$L10$: Basic rating life, h

fh : Life factor

fn : Speed factor

n : Rotational speed, r/min

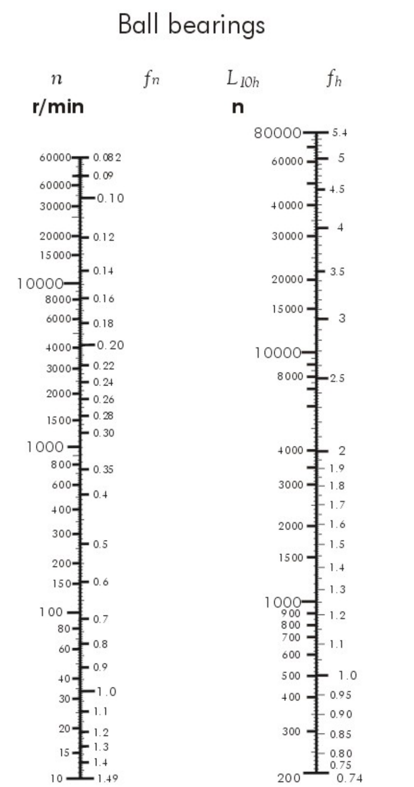


Table 7.1 Bearing life rating scale

(Fig. 7.2) can also be expressed as shown in formula (Fig. 7.5).

(Fig. 7.5) ········· $L_{10h} = \frac{10^6}{60n} (\frac{C}{P})^P$

The relationship between rotational speed n and speed factor fn as well as the relation between the basic rating life L_{10h} and the life factor fn is shown in Fig. 7.1. When several bearings are incorporated in machines or equipments as complete units, all the bearings in the unit are considered as a whole when computing bearing life (see Fig. 7.6). The total bearing life of the unit is a life rating based on the viable lifetime of the unit before even one of the bearings fails due to rolling contact fatigue.

(Fig. 7.6) ········· $L = \frac{1}{(\frac{1}{L_1^e} + \frac{1}{L_2^e} + ... \frac{1}{L_n^e})^{1/e}}$

where,

e = 10/9.......................For ball bearings

L = Total basic rating life or entire unit, h

$L_1, L_2...L_n$: Basic rating life or individual bearings, 1, 2,...n, h

When the load conditions vary at regular intervals, the life can be given by formula (Fig. 7.7).

(Fig. 7.7) ········· $L_m = (\Sigma\ \phi_j / L_j)^{-1}$

where,

ϕj : Frequency of individual load conditions

Lj : Life under individual conditions

7.3 Machine applications and requisite life

When selecting a bearing, it is essential that the requisite life of the bearing be established in relation to the operating conditions.The requisite life of the bearing is usually determined by the type of machine in which the bearing will be used, and duration of service and operational reliability requirements. When determining bearing size, the fatigue life of the bearing is an important actor; however, besides bearing life, the strength and rigidity of the shaft and housing must also be taken into consideration.

7.4 Adjusted life rating factor

The basic bearing life rating (90 % reliability factor) can be calculated through the formulas mentioned earlier in Section 7.2. However, in some applications a bearing fife factor of over 90% reliability may be required. To meet these requir ements, bearing life can be lengthened by the use of specially improved bearing materials or special construction techniques. Moreover, according to elastohydrodynamic lubrication theory, it is clear that the bearing operating conditions (lubrication, temperature, speed, etc.) all exert an effect on bearing life. All these adjustment factors are taken into consideration when calculating bearing life, the adjusted bearing life can be determined.

(Fig. 7.8) ········· $L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot (C/P)^P$

where,

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

a_1 : Reliability adjustment factor

a_2 : Material adjustment factor

a_3 : Operating condition adjustment factor

7.4.1 Life adjustment factor for reliability a_1

The values for the reliability adjustment factor ai (for a reliability factor higher than 90%) can be found in table 7.2

Table 7.2 Reliability adjustment factor values a_1

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

7.4.2 Life adjustment factor for material a_2

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

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

SLB bearings can generally be rotated up to 120˚C ~175˚C, which depends on series. If bearings are operated at a higher temperature, the bearing must be specially heat treated (stabilized) so that inadmissible dimensional change does not occur due to changes in the micro-structure. This special heat treatment might cause the reduction of bearing life because of a hardness change.

7.4.3 Life adjustment factor a_3 for operating conditions

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

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

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

As the operating temperature of the bearing increases, the hardness of the bearing material decreases. Thus, the bearing life correspondingly decreases. The opera-ting temperature adjustment values are shown in Fig.7.9.

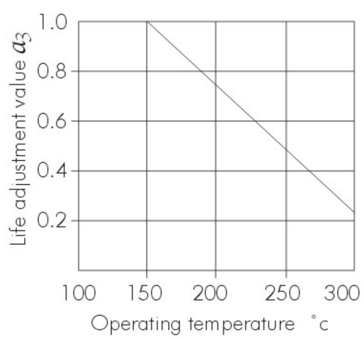


Fig. 7.9 Life adjustment value for operating temperature

7.5 Basic static load rating

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

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

The basic rating of the static load refers to a fixed static load limit at which the specified amount of permanent deformation occurs. It applies to pure radial loads for radial bearings and to pure axial loads for thrust bearings. The maximum applied load values for contact stress occurring at the rolling element and raceway contact points are given on top of page T-19.

For Ball Bearings 4,200 Mpa
(except Self-aligning Ball Bearings)
For Self-aligning Ball Bearings 4,600 Mpa

7.6 Allowable static equivalent load

Generally the static equivalent load which can be permitted is limited by the basic static load rating. However, it will be depending on requirements regarding friction and smooth operation, these limits may be greater or lesser than the basic static rating load in the following formula (Fig. 7.10) and Table 7.3. The safety factor So can be determined considering the maximum static equivalent load.

$So = Co/Po$ .... (Fig. 7.10)

Where,
So : Safety factor
Co : Basic static rating load, N (radial bearings: Cor)
$Po\ max$: Maximum static equivalent load, N (radial: $Por\ max$)

Table 7.3 Minimum safety factor values So

Operating conditions	Ball Bearings
High rotational accuracy demand	2
Normal rotating accuracy demand (Universal application)	1
Slight rotational accuracy deterioration permitted (Low speed, heavy loading, etc.)	0.5

7.7 Selecting bearing size using the life equations - **SLB** rating life

The nominal or basic rating life can deviate significantly from the actual service life in a given application. Service life in a particular application depends on a variety of influencing factors including lubrication, the degree of contamination, misalignment, proper installation and environmental conditions.

Therefore it contains a modified life equation to supplement the basic rating life. This life calculation makes use of a modification factor to account for the lubrication and contamination condition of the bearing and the fatigue limit of the material.

It also makes provisions for bearing manufacturers to recommend a suitable method for calculating the life modification factor to be applied to a bearing, based on operating conditions. The life modification factor applies the concept fatigue load limit P_u analogous to that used when calculating other machine components. The values of the fatigue load limit are provided in the product tables. Furthermore, the life modification factor makes use of the lubricant conditions (viscosity ratio κ) and a factor η_c for the contamination level to reflect the application's operating conditions.

The below equation for rating life is in accordance:

$$L_{nm} = a_1\, a_{\mathbf{SLB}}\, L_{10} = a_1\, a_{\mathbf{SLB}}\, (C/P)^p$$

If the speed is constant, the life can be expressed in operating hours, using the equation

$$L_{nmh} = a_1\, a_{\mathbf{SLB}}\, 10^6/(60n)\, L_{10}$$

where

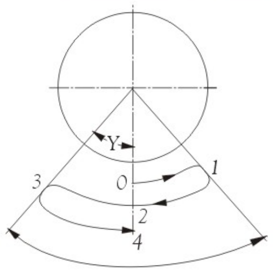
L_{nm} = **SLB** rating life (at 100-n[1]) % reliability) [millions of revolutions]
L_{nmh} = **SLB** rating life (at 100-n[1]) % reliability) [operating hours]
L_{10} = basic rating life (at 90% reliability) [millions of revolutions]
a_1 = life adjustment factor for reliability (table 7.4)
$a_{\mathbf{SLB}}$ = **SLB** life modification factor (Fig. 7.11 and 7.12)
C = basic dynamic load rating [kN]
P = equivalent dynamic bearing load [kN]
n = rotational speed [r/min]
p = exponent of the life equation
3 for ball bearings
10/3 for roller bearings

1) The factor n represents the failure probability, i.e. the difference between the requisite reliability and 100% In some cases it is preferable to express bearing life in units other than millions of revolutions or hours. For example, bearing life for axle bearings used in road and rail vehicles is commonly expressed in terms of kilometres travelled. To facilitate the calculation of bearing life into different units, table 7.5 provides the conversion factors commonly used.

Table 7.4 Life adjustment factor a_1 or SLB rating life equation

Reliability (%)	Failure probability (%) n	**SLB** rating life L_{nm}	Life adjustment factor a_1
90	10	L_{10m}	1.00
95	5	L_{5m}	0.62
96	4	L_{4m}	0.53
97	3	L_{3m}	0.44
98	2	L_{2m}	0.33
99	1	L_{1m}	0.21

Table 7.5 Units conversion factors for bearing life

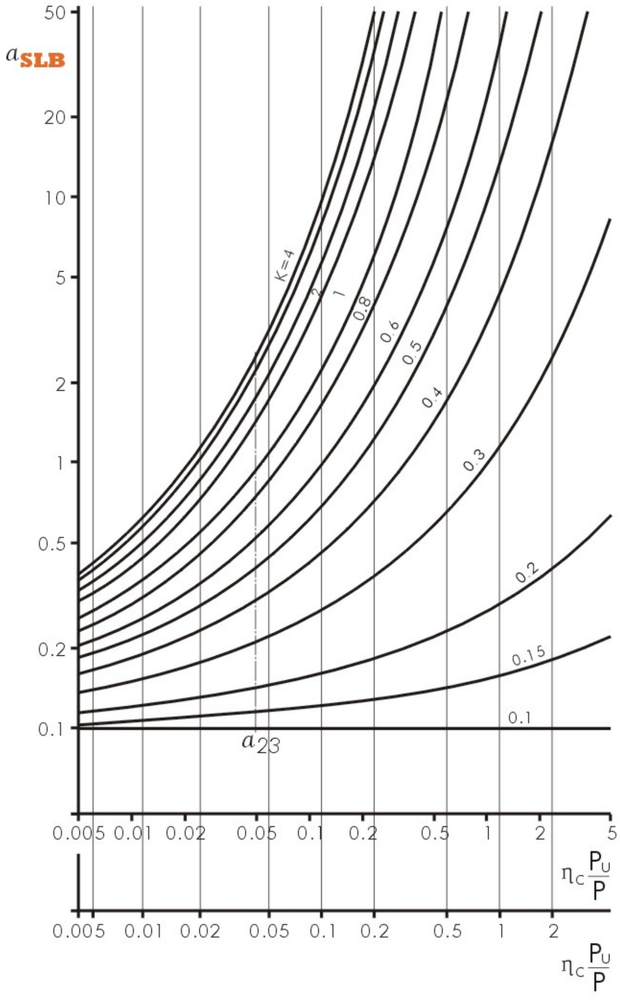


The complete oscillation = 4 γ
i.e. from point 0 to point 4

Basic units	Conversion factor		Million kilometres travelled	Millions of oscillation cycles[1)]
	Millions of revolutions	Operating hours		
1 million revolutions	1	$10^6/(60\,n)$	$\pi\,D/10^3$	$180/(2\,\gamma)$
1 operating hour	$60\,n/10^6$	1	$60\,n\,\pi\,D/10^9$	$180x60\,n/(2\,\gamma\,10^6)$
1 million kilometres	$10^3/(\pi\,D)$	$10^9/(60\,n\,\pi\,D)$	1	$180x10^3/(2\,\gamma\,\pi\,D)$
1 million oscillation cycles[1)]	$2\,\gamma/180$	$2\,\gamma\,10^6/(180x60\,n)$	$2\,\gamma\,\pi\,D/(180x10^3)$	1

D = vehicle wheel diameter, m
n = rotational speed, r/min
γ = oscillation amplitude (angle of max. deviation from centre position), degrees
[1)] Not valid for small amplitudes ($\gamma < 10°$)

Fig. 7.11 Factor a_{SLB} for radial ball bearings

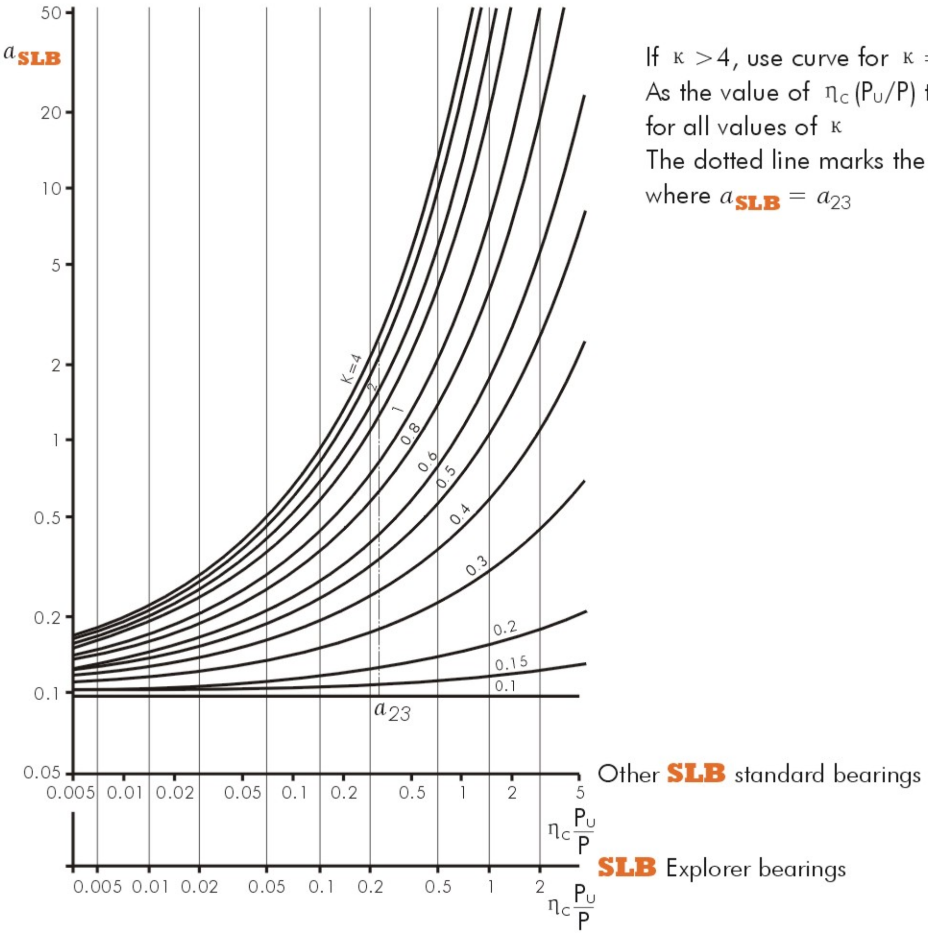


If κ >4, use curve for κ = 4
As the value of η_c (P_u/P) tends to zero, a_{SLB} tends to 0,1 for all values of κ
The dotted line marks the position of the old a_{23} (κ) scale where $a_{SLB} = a_{23}$

Other **SLB** standard bearings

SLB Explorer bearings

Fig. 7.12 Factor a_{SLB} for radial roller bearings



If $\kappa > 4$, use curve for $\kappa = 4$
As the value of $\eta_c (P_u/P)$ tends to zero, a_{SLB} tends to 0,1 for all values of κ
The dotted line marks the position of the old a_{23} (κ) scale where $a_{SLB} = a_{23}$

8. Bearing handling

Bearing storage
Most of the **SLB** bearings are coated with a rust preventative before being packed and shipped, and they should be stored at room temperature below 23 ° C, with a relative humidity of less than 60%.

9. Allowable speed

As bearing speed increases, the temperature of the bearing also increases due to friction heat generated in the bearing interior. If the temperature continues to rise and exceeds certain limits, the efficiency of the lubricant start to fail down drastically, and the bearing can no longer continue to operate in a stable manner. Therefore, the maximum speed at which it is possible for the bearing to continuously operate without the generati on of excessive heat beyond specified limits, is called the allowable speed (r/min). The allowable speed of a bearing depends on the type of bearing, bearing dimensions, type of cage, load, lubricating conditions, and cooling conditions.

The allowable speeds listed in the bearing tables for grease and oil lubrication are for standard **SLB** bearings under normal operating conditions, correctly installed, using the suitable lubricants with adequate supply and proper maintenance. Moreover, these values are based on normal load conditions ($P \leqslant 0.09C$, $Fa/Fr \leqslant 0.3$). For ball bearings with contact seals (2RSR type), the allowable speed is determined by the peripheral lip speed of the seal.

For bearings to be used under heavier than normal load conditions, the allowable speed values listed in the bearing tables must be multiplied by an adjustment factor. The adjustment factors f_L and f_C are given in Figs. 9.1 and 9.2.

Also, when radial bearings are mounted on vertical shafts, lubricant retentions and cage guidance are not favorable compared to horizontal shaft mounting.

Therefore, the allowable speed should be reduced to approximately 80% of the listed speed.

It is possible to operate precision bearings with high speed specification cages at speeds higher than those listed in the bearing tables , if special precautions are taken. These precautions should include the use of forced oil circulation methods such as oil jet or oil mist lubrication.

Under such high speed operating conditions , when special care is taken , the standard allowable speeds given in the bearing tables can be adjusted upward. The maximum speed adjustment values, /B, by which the bearing table speeds can be multiplied, are shown in Table 9.1. However, for any application requiring speeds in excess of the standard allowable speed, please consult **SLB** Engineering.

Fig.9.1 Value of adjustment factor FL depends on bearing load

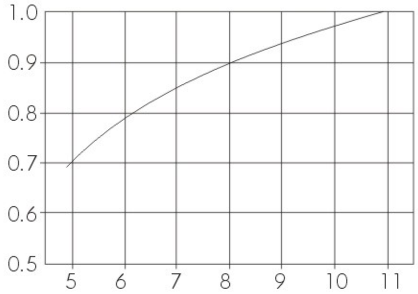


Fig.9.2 Value of adjustment factor Fc depends on combined load

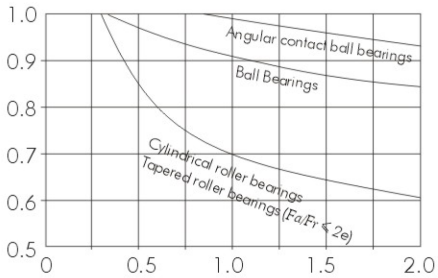


Table 9.1 Adjustment factor, f_B, for allowable number of revolutions

Types of bearing	Adjustment factor f_B
Ball bearings	3.0
Angular contact ball bearings	2.0

10. Vibration and noise value

SLB also supplies bearings for air conditioners, domestic ceiling fans and electric power tools etc. As a rule, the vibration or noise level of these bearings should be carefully controlled and checked. To be a part of our quality control system, **SLB** well equipped with two types of testing instrument S0910-1 (For standard application) and BVT1-1A (For precision or quiet applications). Relatively, here it gives out both vibration and noise standard of these bearings for your reference (Please refer to Table 10.1).

The vibration and noise of SLB bearings are classified in three classes as Z1, Z2 and Z3, it is measured by the instruments of SO910-1. Details please find in the following table:

Table 10.1 Specifications of vibration and noise.

Inner bores (mm)	Vibration acceleration (SO910-1)								
	60 Series			62 Series			63 Series		
	Z1⩽	Z2⩽	Z3⩽	Z1⩽	Z2⩽	Z3⩽	Z1⩽	Z2⩽	Z3⩽
4	34	32	28	35	32	30	36	33	31
5	36	34	30	37	34	32	37	35	33
6	36	34	30	37	34	32	37	35	33
7	38	35	31	38	36	34	-	-	-
8	38	35	31	38	36	34	-	-	-
9	40	36	32	40	37	35	-	-	-
10	42	38	33	42	39	35	44	40	37
12	43	39	34	43	39	35	45	40	37
15	44	40	35	44	41	36	46	42	38
17	44	40	35	45	41	36	47	42	38
20	45	41	36	46	42	38	48	43	39
25	46	42	38	47	43	40	49	44	41
30	47	43	39	48	44	41	50	45	42
35	49	45	41	50	46	43	51	47	44
40	51	46	42	52	47	44	54	49	45
45	53	48	45	54	49	46	56	51	47
50	54	50	47	55	51	48	57	53	49
55	56	52	49	57	53	50	59	54	51
60	58	54	51	59	54	51	61	56	53

For special requirement, it is measured by BVT-1 A and SLB bearings are also classified in three classes as V1, V2 and V3. Details please find in the following table:

Table 10.2 Specifications of vibration and noise.

Inner bores (mm)	Vibration acceleration BVT-1 (0.001/s)								
	V1≤			V2≤			V3≤		
	Low	Medium	High	Low	Medium	High	Low	Medium	High
4	60	35	32	48	26	22	31	16	15
5	74	48	40	58	36	30	35	21	18
6	74	48	40	58	36	30	35	21	18
7	92	66	54	72	48	40	44	28	24
8	92	66	54	72	48	40	44	28	24
9	92	66	54	72	48	40	44	28	24
10	120	80	70	90	60	50	55	35	30
12	120	80	70	90	60	50	55	35	30
15	150	100	85	110	78	60	65	46	35
17	150	100	85	110	78	60	65	46	35
20	180	125	100	130	100	75	80	60	45
25	180	125	100	130	100	75	80	60	45
30	200	150	130	150	120	100	90	75	60
35	200	150	130	150	120	100	90	75	60
40	240	180	160	180	150	130	110	90	80
45	240	180	160	180	150	130	110	90	80
50	280	200	200	210	160	160	125	100	100
55	280	220	200	210	180	180	125	110	110
60	320	220	240	240	180	200	145	110	130