TECHNICAL TABLES

1. Bearing materials

1.1 Raceway and rolling element materials

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

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

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

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

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

2. External bearing sealing devices

External seals have two main functions: to prevent lubricating oil from leaking out, and, 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. Theref ore friction is negligible,making them suitable forhigh 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 noncontact 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 wi 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 and JIS standards (rolling bearing tolerances). For dimensional accuracy, these standards prescribe the tolerances necessary when installing bearings on shafts or in housings.Running accuracy is defined as the allowable limits for bearing runout during operation.

Table 3.1 Comparison of tolerance classifications of national standards

Standard		Tolerance class				
Japanese industrial standard (JIS)	JIS	class 0,6X	class 6	class 5	class 4	class 2
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 $\triangle_{dmp}$										Single radial plane bore diameter variation V_{dp}									
		class 0		class 6		class 5		class 4 [1]		class 2 [1]		diameter series 9					maxdiameter series 0.1				
												class 0	class 6	class 5 max.	class 4	class 2	class 0	class 6	class 5 max.	class 4	class 2
over	incl.	high	low	high	low	high	low	high	low	high	low										
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} maxdiameter series 2,3,4					Mean single plane bore diameter variation V_{dmp}					Inner ring radial runout K_{ia}					Face runout with bore S_d		
over	incl.	class 0	class 6	class 5 max.	class 4	class 2	class 0	class 6	class 5 max.	class 4	class 2	class 0	class 6	class 5 max.	class 4	class 2	class 5	class 4 max.	class 2
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}[2]			Inner ring width deviation ΔB_s										Inner ring width varietion V_{Bs}				
					normal						modified[3]								
		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.				
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	-250	20	20	5	3.0	1.5
50	80	8	5	2.5	0	-150	0	-150	0	-150	0	-380	0	-250	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 Δds of bore diameter to applied for class 4 and 2 is the same as the tolerance of dimentional difference Δdmp 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 ΔD_{mp}										Single radial plane outside diameter variation V_{Dp}									
												diameter series 9					maxdiameter series 0.1				
		class 0		class 6		class 5		class 4[5]		class 2[5]		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}					Single radial plane outside diameter variation V_{Dp}[6]		Mean single plane outside diameter variation V_{Dmp}				
		maxdiameter series 2.3.4					capped bearings diameter series						
		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}			
over	incl.	class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2	class 5	class 4	class 2	all type	class 0,6	class 5	class 4	class 2
		max.					max.			max.					max.		
6	18	15	8	5	3	1.5	8	4	1.5	8	5	1.5	Identical to $\triangle_{Bs}$ of inner ring of same bearing	Identical to $\triangle_{Bs}$ and V_{bs} of inner ring of same bearing	5	2.5	1.5
18	30	15	9	6	4	2.5	8	4	1.5	8	5	2.5			5	2.5	1.5
30	50	20	10	7	5	2.5	8	4	1.5	8	5	2.5			5	2.5	1.5
50	80	25	13	8	5	4.0	8	4	1.5	10	5	4.0			6	3.0	1.5
80	120	35	18	10	6	5.0	9	5	2.5	11	6	5.0			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 dimensional 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 Cfass 2.
❻ To be applied in case snap rings are not installed on the bearings.
❼ To be applied for Deep Groove Ball Bearings and 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 bearings is that it provides uniform load support over the entire ring circumference without any loss of load carrying capacity. However, with a tight fit, ease of installation and disassembly is lost; And when using a non-separable bearing as the floating-side bearing, axial displacement is not possible. For this reason, a tight fit cannot be recommended in all cases.

4.2 The necessity of a proper fit

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

- Raceway cracking, early peeling and displacement of raceway
- Raceway 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 dependent upon thorough analysis of bearing operating conditions, including consideration of:

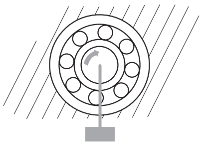
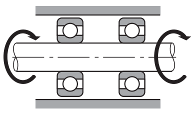
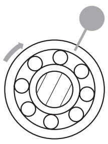
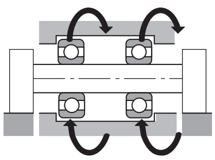
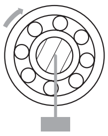
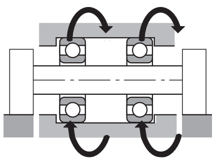
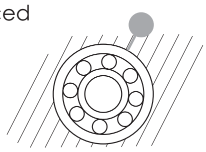
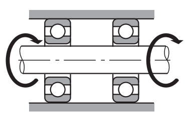
- Shaft and housing material, wall thickness, finished surface accuracy, etc.
- Machinery operating conditions (nature and magnitude of load, rotational speed, temperature, etc.)

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 Deep Groove Ball Bearings, 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
Unbalanced 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
Unbalanced 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 a bearing has before being installed on a shaft or in a housing. The internal clearance valves for NIKO ball bearing classes are shown in tables 5.1 to 5.5

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

Nominal bore diameter d (mm)		C2		CN		C3		C4		C5	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
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 for 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 for 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 bearings for electric motor

(Unit : μm)

Nominal bore diameter d (mm)		Radial internal clearance CM Deep groove 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 a thin oil (or grease) film on the contact surfaces. However, for rolling bearings, lubrication has the following advantages:

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

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

Almost all rolling bearings use either grease or oil lubrication methods, but in some special applicatic solid lubricant such as molybdenum disulfide or graphite may be used.

6.2 Grease lubrication
Grease type lubricants are relatively easy to handle require only the simplest sealing devicesfor these reasons, grease is the most widely used lubricant rolling bearings.

6.2.1 Types and characteristics of grease
Lubricating grease are composed of either a mineral base or a synthetic oil base. To this base a thicks 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 Table 6.2. As performance characteristics of even same type of grease will vary widely from brand, it is best to check the manufacturers' data when selecting a grease.

Table 6.1 Grease varieties and characteristics

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

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

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

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

7.2 Basic rating life and basic dynamic load rating

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

This " life " disparity can be accounted for by the difference in the fatigue of the bearing material itself. This disparity is considered statistically when calculating bearing life, and the basic 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 to identical 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 **NIKO** standard bearing materials, using standard manufacturing techniques. Please consult **NIKO** engineering for basic load ratings of bearings constructed of special materialsor using special manufacturing techniques.

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

$$L_{10} = \left(\frac{C}{P}\right)^{P} \text{..............(7.1)}$$

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), and is calculated as shown in formula (7.2).

$$L_{10h} = 500 f_h^{\,p} \text{..............(7.2)}$$

$$f_h = f_n \frac{C}{P} \text{...............(7.3)}$$

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

where,

L_{10} : Basic rating life, h

f_h : Life factor

f_n : Speed factor

n : Rotational speed, r/min

Ball bearings

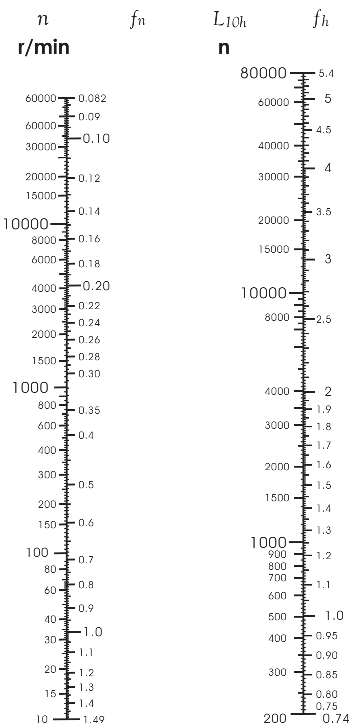


Fig.7.1 Bearing life rating scale

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

$$L_{10h} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^{P} \text{..............(7.5)}$$

The relationship between rotational speed n and speed factor fn as well as the relation between the basic rating life L10h and the life factor fn is shown in Fig. 7.1. When several bearings are incorporated in machines or equipment as complete units, all the bearings in the unit are considered as a whole when computing bearing life (see formula 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.

$$L = \frac{1}{\left(\frac{1}{L_1^{e}} + \frac{1}{L_2^{e}} + \dots \frac{1}{L_n^{e}}\right)^{1/e}} \text{..............(7.6)}$$

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 (7.7).
$L_m = (\Sigma\, \Phi_j / L_j)^{-1}$...............................(7.7)

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 factor; 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.

$L_{na} = a1 \cdot a2 \cdot a3 \cdot (C/P)^P$...........(7.8)

where,
L_{na} : Adjusted life rating in millions of revolutions (10^6)(adjusted for reliability, material and operating conditions)
$a1$: Reliability adjustment factor
$a2$: Material adjustment factor
$a3$: 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 7.1

Table 7.1 Reliability adjustment factor values a_1

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

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 catalog are based on NIKO's standard material and process, therefore, the adjustment factor a_2 =1. When special materials or processes are used the adjustment factor can be larger than 1.

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

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 life.

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

However, when lubricating conditions are particularly unfavorable and the oil film formation on the contact surfaces of the raceway and rolling elements is insufficient, the value of a_3 becomes less than one. This Insufficient oil film formation can be caused, for example, by the lubricating oil viscosity being too low for the operating temperature (below13 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 NIKO 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.7.2.

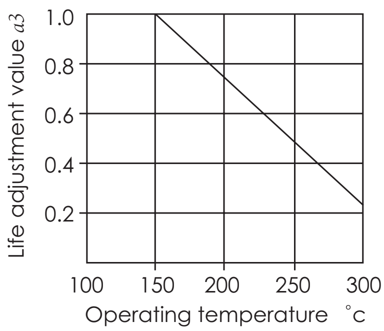


Fig. 7.2 Life adjustment value for operating temperature

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 is impaired.

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

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

For ball bearings (except Self-aligning Ball Bearings)	4,200 Mpa
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, 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 (3.9) and Table 7.3 the safety factor *So* can be determined considering the maximum static equivalent load.

$So = Co/Po$ .... (3.9)

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

8. Bearing handling

Bearing storage

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

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 longercontinue to operate in a stable manner. Therefore, the maximum speed at which it is possible for the bearing to continuously operate without the generation 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 NIKO 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 (LLU 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 NIKO Engineering.

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

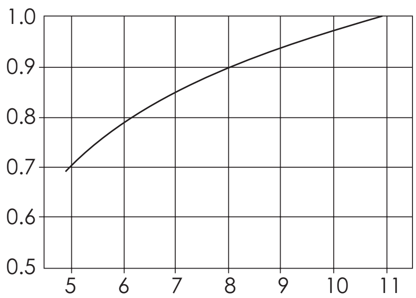


Fig.9.2 Value of adjustment factor Fc depends on combined load

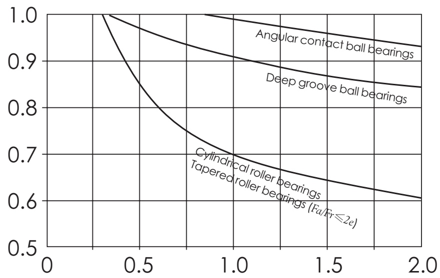


Table 9.1 Adjustment factor, f_B, for allowable number of revolutions

Type of bearing	Adjustment factor f_B
Deep groove ball bearings	3.0
Angular contact ball bearings	2.0