12. General miscellaneous information

12.1 JIS concerning rolling bearings

Rolling bearings are critical mechanical elements which are used in a wide variety of machines. They are standardized internationally by ISO (International Organization for Standardization). Standards concerning rolling bearings can also be found in DIN (Germany), ANSI (USA), and BS (England). In Japan, the conventional JIS standards related to rolling bearings are arranged systematically and revised in accordance to the JIS standards enacted in 1965. Since then, these have been individually revised in reference to the ISO standards or compliance with the actual state of production and sales.

Most of the standard bearings manufactured in Japan are based on the JIS standards. BAS (Japan Bearing Association Standards), on the other hand, acts as a supplement to JIS. Table 1 lists JIS standards related to bearings.

Table 1 JIS related to rolling bearing

No.	Standard classification	Standard No.	Title of Standard
1	General code	B 1511	Rolling bearings — General code
2	Common standards of bearings	B 0005	Technical drawings — Rolling bearings — Part 1: General simplified representation — Part 2: Detailed simplified representation
3		B 0104	Rolling bearings — Vocabulary
4		B 0124	Rolling bearings — Symbols for quantities
5		B 1512	Rolling bearings — Boundary dimensions
6		B 1513	Rolling bearings — Designation
7		B 1514	Rolling bearings — Tolerances of bearings — Part 1: Radial bearings — Part 2: Thrust bearings — Part 3: Chamfer dimensions-Maximum values
8		B 1515	Rolling bearings — Tolerances — Part 1: Terms and definitions — Part 2: Measuring and gauging principles and methods
9		B 1516	Making on rolling bearings and packagings
10		B 1517	Packaging of rolling bearings
11		B 1518	Dynamic load ratings and rating life for rolling bearings
12		B 1519	Static load ratings for rolling bearings
13		B 1520	Rolling bearings — Radial internal clearance
14		B 1548	Rolling bearings — Measuring methods of A-weighted sound pressure levels
15		B 1566	Mounting dimensions and fits for rolling bearings
16		G 4805	High carbon chromium bearing steels
17	Individual standards of bearings	B 1521	Rolling bearings — Deep groove ball bearings
18		B 1522	Rolling bearings — Angular contact ball bearings
19		B 1523	Rolling bearings — Self-aligning ball bearings
20		B 1532	Rolling bearings — Thrust ball bearings with flat back faces
21		B 1533	Rolling bearings — Cylindrical roller bearings
22		B 1534	Rolling bearings — Tapered roller bearings
23		B 1535	Rolling bearings — Self-aligning roller bearings
24		B 1536	Rolling bearings-Boundary dimensions and tolerances of needle roller bearings — Part 1: Dimension series 48, 49 and 69 — Part 2: Drawn cup without inner ring — Part 3: Radial needle roller and cage assemblies — Part 4: Thrust needle roller and cage assemblies, thrust washers — Part 5: Track rollers
25		B 1539	Rolling bearings — Self-aligning thrust roller bearings
26		B 1557	Rolling bearings — Insert bearing units
27		B 1558	Rolling bearings — Insert bearings
28	Standards of bearing parts	B 1501	Steel balls for ball bearings
29		B 1506	Rolling bearings — Rollers
30		B 1509	Rolling bearings — Radial bearings with locating snap ring — Dimensions and tolerances
31	Standards of bearing accessorie	B 1551	Rolling bearing accessories — Plummer block housings
32		B 1552	Rolling bearings — Adapter assemblies, Adapter sleeves and Withdrawal sleeves
33		B 1554	Rolling bearings — Locknuts and locking devices
34		B 1559	Rolling bearings — Cast and pressed housings for insert bearings
35	Reference standard	K 2220	Lubricating grease

12.2 Amount of permanent deformation at point where inner and outer rings contact rolling element

When two materials are in contact, a point within the contact zone develops local permanent deformation if it is exposed to a load exceeding the elastic limit of the material. The rolling and raceway surfaces of a bearing, which appear to be perfect to a human eye, are found to be imperfect when observed by microscope even though the surfaces are extremely hard and finished to an extreme accuracy. Therefore, the true contact area is surprisingly small when compared with the apparent contact area, because the surface is actually jagged and rough with asperities or sharp points. These local points develop permanent deformation when exposed to a relatively small load. Such microscopic permanent deformations seldom affect the function of the bearing. Usually, the only major change is that light is reflected differently from the raceway surface (running marks, etc.).

As the load grows further, the amount of permanent deformation increases corresponding to the degree identifiable on the macroscopic scale in the final stage. Fig. 1 shows the manner of this change. While the load is small, the elastic displacement during point contact in a ball bearing is proportional to the p-th power of the load Q (p=2/3 for ball bearings and p=0.9 for roller bearings) in compliance with the Hertz theory. The amount of permanent deformation grows as the load increases, resulting in substantial deviation of the elastic displacement from the theoretical value.

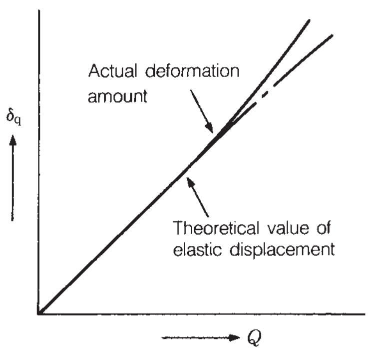


Fig. 1

For normal bearings, about 1/3 of the gross amount of permanent deformation δ_q occurs in rolling element and about 2/3 in the bearing ring.

12.2.1 Ball bearings

The amount of permanent deformation δ_q can also be expressed in relation to the load Q. Equation (1) shows the relationship between δ_q and Q for ball bearings:

$$\delta_q = 1.30\times10^{-7}\frac{Q^2}{D_w}(\rho_{I1}+\rho_{II1})(\rho_{I2}+\rho_{II2}) \quad \text{(N)}$$

$$= 1.25\times10^{-5}\frac{Q^2}{D_w}(\rho_{I1}+\rho_{II1})(\rho_{I2}+\rho_{II2}) \quad \{\text{kgf}\}$$

$$\text{(mm)} \quad \cdots\cdots (1)$$

where, δ_q: Gross amount of permanent deformation between the rolling element and bearing ring (mm)

Q: Load of rolling element (N), {kgf}

D_w: Diameter of rolling element (mm)

ρ_{I1}, ρ_{I2} and ρ_{II1}, ρ_{II2}: Take the reciprocal of the main radius of curvature of the area where materials I and II make contact (Units: 1/mm).

When the equation is rewritten using the relation between δ_q and Q, Equation (2) is obtained:

$$\left.\begin{aligned}\delta_q &= K\cdot Q^2 && \text{(N)}\\ &= 96.2K\cdot Q^2 && \{\text{kgf}\}\end{aligned}\right\}\ \text{(mm)} \quad \cdots\cdots (2)$$

The value of the constant K is as shown for the bearing series and bore number in Table 1. K_i applies to the contact between the inner ring and rolling element while K_e to that between the outer ring and rolling element.

Table 1 Value of the constant K for deep groove ball bearings

Bearing bore No.	Bearing series 60		Bearing series 62		Bearing series 63	
	K_i	K_e	K_i	K_e	K_i	K_e
	$\times10^{-10}$	$\times10^{-10}$	$\times10^{-10}$	$\times10^{-10}$	$\times10^{-10}$	$\times10^{-10}$
00	2.10	4.12	2.01	2.16	0.220	0.808
01	2.03	1.25	0.376	1.13	0.157	0.449
02	1.94	2.21	0.358	1.16	0.145	0.469
03	1.89	2.24	0.236	0.792	0.107	0.353
04	0.279	0.975	0.139	0.481	0.0808	0.226
05	0.270	0.997	0.133	0.494	0.0597	0.218
06	0.180	0.703	0.0747	0.237	0.0379	0.119
07	0.127	0.511	0.0460	0.178	0.0255	0.0968
08	0.417	0.311	0.129	0.0864	0.0206	0.0692
09	0.312	0.234	0.127	0.0875	0.0436	0.0270
10	0.308	0.236	0.104	0.0720	0.0333	0.0207
11	0.187	0.140	0.0728	0.0501	0.0262	0.0162
12	0.185	0.141	0.0547	0.0377	0.0208	0.0218
13	0.183	0.142	0.0469	0.0326	0.0169	0.0105
14	0.119	0.0914	0.0407	0.0283	0.0138	0.00863
15	0.118	0.0920	0.0402	0.0286	0.0117	0.00733
16	0.0814	0.0624	0.0309	0.0218	0.00982	0.00616
17	0.0808	0.0628	0.0243	0.0170	0.00832	0.00523
18	0.0581	0.0446	0.0194	0.0136	0.00710	0.00447
19	0.0576	0.0449	0.0158	0.0110	0.00611	0.00386
20	0.0574	0.0450	0.0130	0.00900	0.00465	0.00292
22	0.0296	0.0225	0.00928	0.00639	0.00326	0.00203
24	0.0293	0.0227	0.00783	0.00544	0.00320	0.00205
26	0.0229	0.0178	0.00666	0.00467	0.00255	0.00164
28	0.0227	0.0179	0.00656	0.00472	0.00209	0.00134
30	0.0181	0.0143	0.00647	0.00477	0.00205	0.00136

As an example, the δ_q and Q relation may be illustrated as shown in Fig. 2 for the 62 series of deep groove ball bearings.

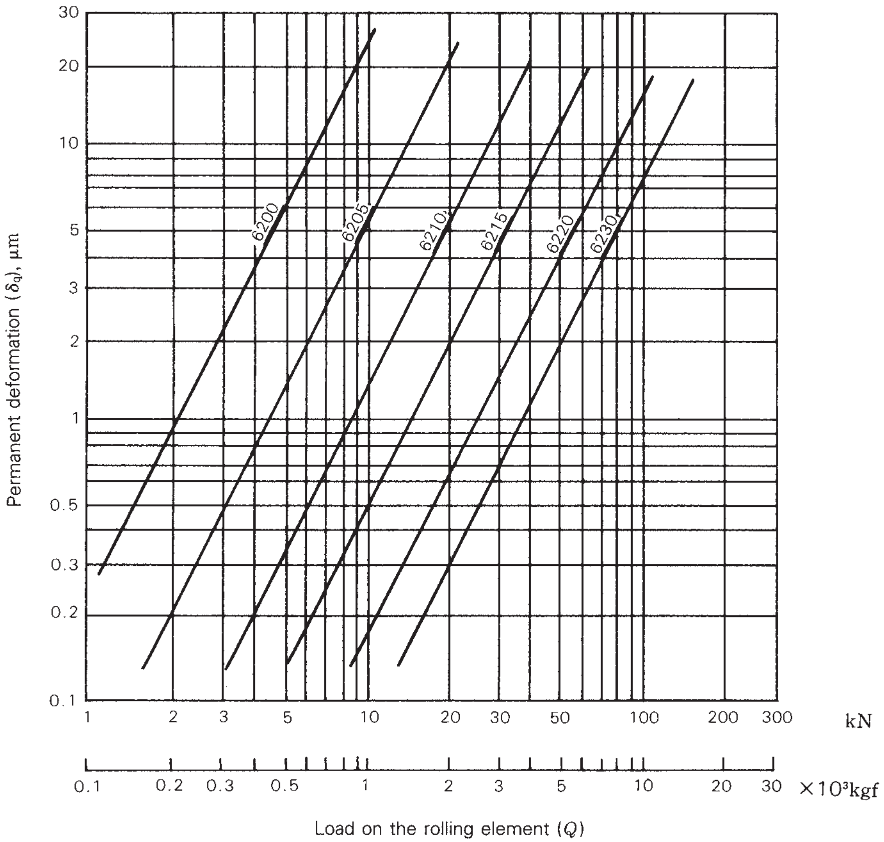


Fig. 2 Load and permanent deformation of rolling element

12.2.2 Roller bearings

In the case of roller bearings, the permanent deformation δ_q and load Q between the rolling element and bearing ring may be related as shown in Equation (3).

$$\left.\begin{aligned}\delta_q &= 2.12\times10^{-11}\cdot\frac{1}{\sqrt{D_w}}\cdot\left(\frac{Q}{L_{we}}\right)^3\cdot(\rho_I+\rho_{II})^{3/2} \quad \cdots\cdots\cdots\cdots\cdots \text{(N)}\\ &= 2.00\times10^{-8}\cdot\frac{1}{\sqrt{D_w}}\cdot\left(\frac{Q}{L_{we}}\right)^3\cdot(\rho_I+\rho_{II})^{3/2} \quad \cdots\cdots\cdots\cdots\cdots \{\text{kgf}\}\end{aligned}\right\}$$

$$\text{(mm)} \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \text{(3)}$$

where, L_{we}: Effective length of roller (mm)
ρ_I, ρ_{II}: Reciprocal of the main radius of curvature at the point where materials I and II contact (1/mm)

Other symbols for quantities are the same as in Equation (1) of 12.2.1. When the equation is rewritten using the relation between δ_q and Q, then the next Equation (4) is obtained:

$$\left.\begin{aligned}\delta_q &= K\cdot Q^3 \quad \text{(N)}\\ &= 943K\cdot Q^2 \quad \{\text{kgf}\}\end{aligned}\right\} \text{(mm)} \cdots\cdots\cdots\cdots \text{(4)}$$

The value of the constant K is as shown for the bearing number in Table 2. K_i applies to the contact between the inner ring and rolling element while K_e to that between the outer ring and rolling element.

As an example, the δ_q and Q relation may be illustrated as shown in Fig. 3 for the NU2 series of cylindrical roller bearings.

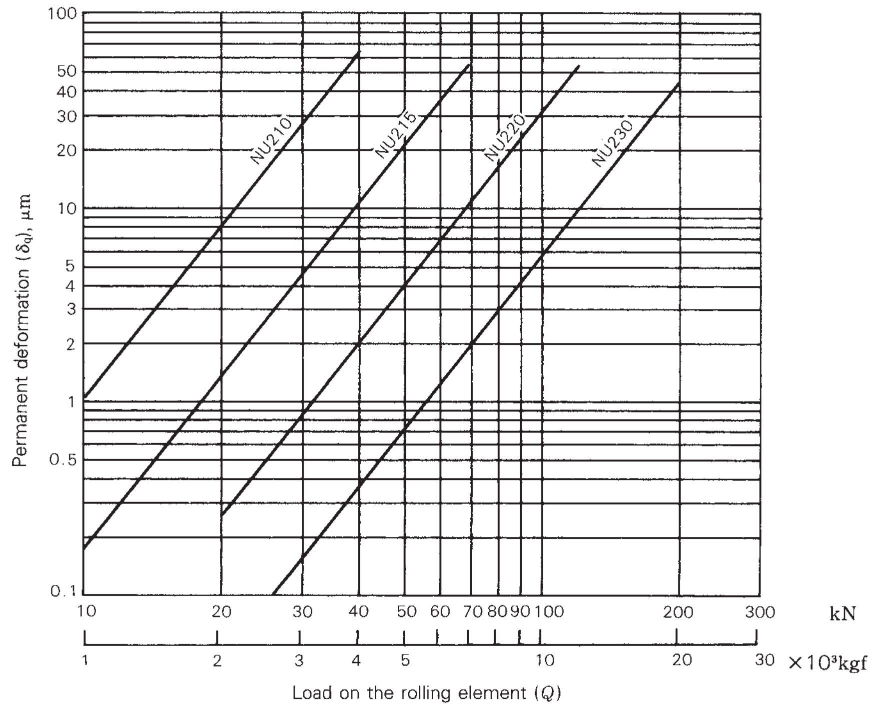


Fig. 3 Load and permanent deformation of rolling element

Table 2 Value of the constant K for cylindrical roller bearings

Bearing series NU2			Bearing series NU3		
Brg No.	K_i	K_e	Brg No.	K_i	K_e
	$\times 10^{-16}$	$\times 10^{-16}$		$\times 10^{-16}$	$\times 10^{-16}$
NU205W	113	67.5	NU305W	20.4	10.9
NU206W	50.7	30.9	NU306W	11.3	6.32
NU207W	19.1	11.4	NU307W	6.83	3.81
NU208W	10.8	6.53	NU308W	4.24	2.43
NU209W	10.6	6.64	NU309W	1.92	1.07
NU210W	10.4	6.74	NU310W	1.51	0.856
NU211W	6.23	4.06	NU311W	0.786	0.435
NU212W	3.93	2.57	NU312W	0.575	0.323
NU213W	2.58	1.69	NU313W	0.460	0.262
NU214W	2.54	1.70	NU314W	0.347	0.200
NU215W	1.74	1.15	NU315W	0.211	0.120
NU216W	1.38	0.915	NU316W	0.207	0.121
NU217W	0.976	0.648	NU317W	0.132	0.0761
NU218W	0.530	0.343	NU318W	0.112	0.0650
NU219W	0.426	0.277	NU319W	0.0903	0.0529
NU220W	0.324	0.210	NU320W	0.0611	0.0357
NU221W	0.249	0.162	NU321W	0.0428	0.0247
NU222W	0.156	0.0995	NU322W	0.0325	0.0187
NU224W	0.123	0.0800	NU324W	0.0176	0.00992
NU226W	0.121	0.0810	NU326W	0.0132	0.00750
NU228W	0.0836	0.0559	NU328W	0.0100	0.00576
NU230W	0.0565	0.0378	NU330W	0.00832	0.00484

Bearing series NU4		
Brg No.	K_i	K_e
	$\times 10^{-16}$	$\times 10^{-16}$
NU405W	4.69	2.28
NU406W	2.09	1.01
NU407W	1.61	0.821
NU408W	0.835	0.418
NU409W	0.607	0.312
NU410W	0.373	0.191
NU411W	0.363	0.194
NU412W	0.220	0.116
NU413W	0.173	0.0926
NU414W	0.0954	0.0509
NU415W	0.0651	0.0342
NU416W	0.0455	0.0237
NU417M	0.0349	0.0178
NU418M	0.0251	0.0130
NU419M	0.0245	0.0132
NU420M	0.0182	0.00972
NU421M	0.0137	0.00729
NU422M	0.0104	0.00559
NU424M	0.00611	0.00323
NU426M	0.00353	0.00185
NU428M	0.00303	0.00161
NU430M	0.00296	0.00163

12.3 Rotation and revolution speed of rolling element

When the rolling element rotates without slip between bearing rings, the distance which the rolling element rolls on the inner ring raceway is equal to that on the outer ring raceway. This fact allows establishment of a relationship among rolling speed n_i and n_e of the inner and outer rings and the number of rotation n_a of rolling elements.

The revolution speed of the rolling element can be determined as the arithmetic mean of the circumferential speed on the inner ring raceway and that on the outer ring raceway (generally with either the inner or outer ring being stationary). The rotation and revolution of the rolling element can be related as expressed by **Equations** (1) through (4).

No. of rotation

$$n_a=\left(\frac{D_{pw}}{D_w}-\frac{D_w\cos^2\alpha}{D_{pw}}\right)\frac{n_e-n_i}{2}\quad(\mathrm{min}^{-1})\quad\cdots\cdots(1)$$

Rotational circumferential speed

$$v_a=\frac{\pi D_w}{60\times10^3}\left(\frac{D_{pw}}{D_w}-\frac{D_w\cos^2\alpha}{D_{pw}}\right)\frac{n_e-n_i}{2}\quad(\mathrm{m/s})\quad\cdots\cdots(2)$$

No. of revolutions (No. of cage rotation)

$$n_c=\left(1-\frac{D_w\cos\alpha}{D_{pw}}\right)\frac{n_i}{2}+\left(1+\frac{D_w\cos\alpha}{D_{pw}}\right)\frac{n_e}{2}\quad(\mathrm{min}^{-1})\quad\cdots\cdots(3)$$

Revolutional circumferential speed
(cage speed at rolling element pitch diameter)

$$v_c=\frac{\pi D_{pw}}{60\times10^3}\left[\left(1-\frac{D_w\cos\alpha}{D_{pw}}\right)\frac{n_i}{2}+\left(1+\frac{D_w\cos\alpha}{D_{pw}}\right)\frac{n_e}{2}\right]\quad(\mathrm{m/s})\quad\cdots\cdots(4)$$

where, D_{pw}: Pitch diameter of rolling elements (mm)
D_w: Diameter of rolling element (mm)
α: Contact angle (°)
n_e: Outer ring speed (min^{-1})
n_i: Inner ring speed (min^{-1})

The rotation and revolution of the rolling element is shown in **Table 1** for inner ring rotating (n_e=0) and outer ring rotating (n_i=0) respectively at 0°≦ α <90° and at α=90°.

As an example, **Table 2** shows the rotation speed n_a and revolution speed n_c of the rolling element during rotating of the inner ring of ball bearings 6210 and 6310.

Table 2 n_a and n_c for ball bearings 6210 and 6310

Ball bearing	γ	n_a	n_c
6210	0.181	$-2.67n_i$	$0.41n_i$
6310	0.232	$-2.04n_i$	$0.38n_i$

Remarks $\gamma=\frac{D_w\cos\alpha}{D_{pw}}$

Table 1 Rolling element's rotation speed n_a, rotational circumferential speed v_a, revolution speed n_c, and revolutional circumferential speed v_c

Contact angle	Rotation/revolution speed	Inner ring rolling (n_e=0)	Outer ring rolling (n_i=0)
0°≦α<90°	n_a (min^{-1})	$-\left(\frac{1}{\gamma}-\gamma\right)\frac{n_i}{2}\cdot\cos\alpha$	$\left(\frac{1}{\gamma}-\gamma\right)\frac{n_e}{2}\cdot\cos\alpha$
	v_a (m/s)	$\frac{\pi D_w}{60\times10^3}n_a$	
	n_c (min^{-1})	$(1-\gamma)\frac{n_i}{2}$	$(1+\gamma)\frac{n_e}{2}$
	v_c (m/s)	$\frac{\pi D_{pw}}{60\times10^3}n_c$	
α=90°	n_a (min^{-1})	$-\frac{1}{\gamma}\cdot\frac{n_i}{2}$	$\frac{1}{\gamma}\cdot\frac{n_e}{2}$
	v_a (m/s)	$\frac{\pi D_w}{60\times10^3}n_a$	
	n_c (min^{-1})	$\frac{n_i}{2}$	$\frac{n_e}{2}$
	v_c (m/s)	$\frac{\pi D_{pw}}{60\times10^3}n_c$	

Reference 1. ±: The "+" symbol indicates clockwise rotation while the "−" symbol indicates counterclockwise rotation.
2. $\gamma=\frac{D_w\cos\alpha}{D_{pw}}$ (0°≦α<90°), $\gamma=\frac{D_w}{D_{pw}}$ (α=90°)

12.4 Bearing speed and cage slip speed

One of the features of a rolling bearing is that its friction is smaller than that of a slide bearing. This may be attributed to the fact that rolling friction is smaller than slip friction. However, even a rolling bearing inevitably develops some slip friction.

Slip friction occurs mainly between the cage and rolling element, on the guide surface of the cage, between the rolling element and raceway surface (slip caused by the elastic displacement), and between the collar and roller end surface in the roller bearing.

The most critical factor for a high speed bearing is the slip friction between the cage and rolling element and that on the guide surface of the cage. The allowable speed of a bearing may finally be governed by this slip friction. The PV value may be used as a parameter to indicate the speed limit in the slide bearing and can also be applied to the slip portion of the rolling bearing. "P" is the contact pressure between the rolling element and cage or that between the guide surface of the cage. "P" is not much affected by the load on the bearing in the normal operation state. "V" is a slip speed.

Accordingly, the speed limit of a rolling bearing can be expressed nearly completely by the slip speed, that is, the bearing size and speed.

Conventionally, the $D_{pw} \times n$ value ($d_m n$ value) has often been used as a guideline to indicate the allowable speed of a bearing. But this is nothing but the slip speed inside the bearing. With the outer ring stationary and the inner ring rotating, the relative slip speed V_e on the guide surface of the outer ring guiding cage is expressed by Equation (1):

$$V_e = \frac{\pi}{120 \times 10^3} (1-\gamma)\, d_{e1} n_i$$

$$= K_e n_i \ \text{(m/s)} \quad \cdots\cdots (1)$$

where, d_{e1}: Diameter of the guide surface (mm)

γ: Parameter to indicate the inside design of the bearing

$$\gamma = \frac{D_w \cos\alpha}{D_{pw}}$$

D_w: Diameter of rolling element (mm)
α: Bearing contact angle (°)
D_{pw} (or d_m): Pitch diameter of rolling elements (mm)
n_i: Inner ring rotating speed (min^{-1})

$$K_e = \frac{\pi\, d_{e1}}{120 \times 10^3} (1-\gamma)$$

Table 2 shows the value of the constant K_e for deep groove ball bearings, 62 and 63 series, and cylindrical roller bearings, NU2 and NU3 series. Assuming V_i for the slip speed of the inner ring guiding cage and V_a for the maximum slip speed of the rolling element for the cage, the relation may be approximated as follows:

$V_i \fallingdotseq (1.15 \text{ to } 1.18)\ V_e$ (diameter series 2)
$\fallingdotseq (1.20 \text{ to } 1.22)\ V_e$ (diameter series 3)
$V_a \fallingdotseq (1.05 \text{ to } 1.07)\ V_e$ (diameter series 2)
$\fallingdotseq (1.07 \text{ to } 1.09)\ V_e$ (diameter series 3)

Example of calculation with deep groove ball bearing

Table 1 shows $D_{pw} \times n$ ($d_m n$) and the slip speed for 6210 and 6310 when n_i=4 500 min^{-1}.

Table 1

Ball bearing	$D_{pw} \times n$ ($\times 10^4$)	V_e(m/s) outer ring guide	V_a(m/s)	V_i(m/s) inner ring guide
6210	31.5	7.5	8.0	8.7
6310	36.9	8.5	9.1	10.3

Remarks
Assuming h_e for the groove depth in Equation (1);

$$d_{e1} = D_{pw} + D_w - 2h_e = D_{pw}\left(1 + \frac{D_w - 2h_e}{D_{pw}}\right)$$

$$V_e = \frac{\pi}{120 \times 10^3} (1-\gamma) \left(1 + \frac{D_w - 2h_e}{D_{pw}}\right) D_{pw} \cdot n$$

$$= K_e' \cdot D_{pw} \cdot n$$

The constant K_e' is determined for each bearing and is approximately within the range shown below:
K_e'=(0.23 ~ 0.245)$\times 10^{-4}$

Table 2 Constant K_e for 62 and 63 series ball bearings and NU2 and NU3 series roller bearings

Bearing bore No.	Bearing series			
	62	63	NU2	NU3
	$\times 10^{-5}$	$\times 10^{-5}$	$\times 10^{-5}$	$\times 10^{-5}$
00	48	49	—	—
01	50	52	—	—
02	59	66	—	—
03	67	74	—	—
04	77	81	79	84
05	92	103	92	102
06	110	121	110	123
07	125	133	126	136
08	142	149	144	155
09	155	171	157	171
10	168	189	172	189
11	184	201	189	206
12	206	218	208	224
13	221	235	226	259
14	233	252	239	261
15	249	270	251	278
16	264	287	270	298
17	281	305	288	314
18	298	323	304	333
19	316	340	323	352
20	334	366	341	376
21	350	379	361	392
22	368	406	378	416
24	400	441	408	449
26	430	475	441	486
28	470	511	478	523
30	510	551	515	559
32	550	585	551	599
34	585	615	588	635
36	607	655	615	670
38	642	695	651	707
40	682	725	689	747

12.5 Centrifugal force of rolling elements

Under normal operating conditions, the centrifugal force on a rolling element is negligible when compared with the load on the bearing and thus not taken into account during calculation of the effective life of the bearing. However, if the bearing is running at high speed, then even if the load is small, the effect of the centrifugal force on the rolling element cannot be ignored. The deep groove ball bearing and cylindrical roller bearing suffer a decrease in the effective life because of the centrifugal force on the rolling element. In the case of an angular contact ball bearing, the contact angle of the inner ring increases and that of the outer ring decreases from the initial value, resulting in relative variation in the fatigue probability.

Apart from details of the effect on the life, the centrifugal force F_c of the rolling element during rotating of the inner ring is expressed by **Equations** (1) and (2) respectively for a ball bearing and roller bearing.

Ball bearing

$$F_c = K_B n_i^2 \quad \cdots\cdots (1)$$

$$K_B = 5.580\times10^{-12} D_w^3 D_{pw} (1-\gamma)^2 \quad \cdots\cdots (N)$$

$$= 0.569\times10^{-12} D_w^3 D_{pw} (1-\gamma)^2 \quad \cdots\cdots \{kgf\}$$

Roller bearing

$$F_c = K_R n_i^2 \quad \cdots\cdots (2)$$

$$K_R = 8.385\times10^{-12} D_w^2 L_w D_{pw} (1-\gamma)^2 \quad \cdots\cdots (N)$$

$$= 0.855\times10^{-12} D_w^2 L_w D_{pw} (1-\gamma)^2 \quad \cdots\cdots \{kgf\}$$

where, D_w: Diameter of roller element (mm)
D_{pw}: Pitch diameter of rolling elements (mm)
γ: Parameter to indicate the internal design of the bearing

$$\gamma = \frac{D_w \cos\alpha}{D_{pw}}$$

α: Contact angle of bearing (°)
L_w: Length of roller (mm)
n_i: Inner ring rotating speed (min^{-1})

Table 1 shows the K values (K_B and K_R) for both series of NU2 & NU3 roller bearings and the 62 & 63 ball bearings.

Table 1 Constant K for 62 and 63 series ball bearings and for NU2 and NU3 series roller bearings

Bearing bore No.	Bearing series 62		Bearing series 63		Bearing series NU2		Bearing series NU3	
	K		K		K		K	
	$\times10^{-8}$	$\times10^{-8}$	$\times10^{-8}$	$\times10^{-8}$	$\times10^{-8}$	$\times10^{-8}$	$\times10^{-8}$	$\times10^{-8}$
00	0.78	{0.08}	2.16	{0.22}	—		—	
01	1.37	{0.14}	3.14	{0.32}	—		—	
02	1.77	{0.18}	4.41	{0.45}	—		—	
03	2.94	{0.30}	6.67	{0.68}	—		—	
04	5.49	{0.56}	9.41	{0.96}	5.00	{0.51}	9.51	{0.97}
05	6.86	{0.70}	15.7	{1.6}	6.08	{0.62}	16.7	{1.7}
06	13.7	{1.4}	29.4	{3.0}	11.8	{1.2}	28.4	{2.9}
07	25.5	{2.6}	47.1	{4.8}	22.6	{2.3}	41.2	{4.2}
08	36.3	{3.7}	73.5	{7.5}	35.3	{3.6}	63.7	{6.5}
09	41.2	{4.2}	129	{13.2}	39.2	{4.0}	109	{11.1}
10	53.9	{5.5}	186	{19.0}	43.1	{4.4}	149	{15.2}
11	84.3	{8.6}	251	{25.6}	63.7	{6.5}	234	{23.9}
12	128	{13.1}	341	{34.8}	91.2	{9.3}	305	{31.1}
13	161	{16.4}	455	{46.4}	127	{12.9}	391	{39.9}
14	195	{19.9}	595	{60.7}	135	{13.8}	494	{50.4}
15	213	{21.7}	765	{78.0}	176	{17.9}	693	{70.7}
16	290	{29.6}	969	{98.8}	233	{23.8}	758	{77.3}
17	391	{39.9}	1 216	{124}	302	{30.8}	1 020	{104}
18	518	{52.8}	1 491	{152}	448	{45.7}	1 236	{126}
19	672	{68.5}	1 824	{186}	559	{57.0}	1 471	{150}
20	862	{87.9}	2 560	{261}	689	{70.3}	1 961	{200}
21	1 079	{110}	3 011	{307}	844	{86.1}	2 501	{255}
22	1 344	{137}	4 080	{416}	1 167	{119}	3 207	{327}
24	1 736	{177}	4 570	{466}	1 422	{145}	4 884	{498}
26	2 177	{222}	6 160	{628}	1 569	{160}	6 257	{638}
28	2 442	{249}	8 140	{830}	2 157	{220}	7 904	{806}
30	2 707	{276}	9 003	{918}	2 903	{296}	9 807	{1 000}
32	2 962	{302}	11 572	{1 180}	3 825	{390}	10 787	{1 100}
34	4 168	{425}	16 966	{1 730}	4 952	{505}	13 925	{1 420}

Remarks The value given in braces { } is the calculated result for constant K in units of kgf.

12.6 Temperature rise and dimensional change

Rolling bearings are extremely precise mechanical elements. Any change in dimensional accuracy due to temperature cannot be ignored. Accordingly, it is specified as a rule that measurement of a bearing must be made at 20°C and that the dimensions to be set forth in the standards must be expressed by values at 20°C.

Dimensional change due to temperature change not only affects the dimensional accuracy, but also causes change in the internal clearance of a bearing during operation. Dimensional change may cause interference between the inner ring and shaft or between the outer ring and housing bore. It is also possible to achieve shrink fitting with large interference by utilizing dimensional change induced by temperature difference. The dimensional change Δl due to temperature rise can be expressed as in Equation (1) below:

$$\Delta l = \Delta T \ \alpha \ l \quad \text{(mm)} \quad \cdots\cdots (1)$$

where, Δl: Dimentional change (mm)
ΔT: temperature rise (°C)
α: Coefficient of linear expansion for bearing steel
$\alpha=12.5\times10^{-6}$ (1/°C)
l: Original dimension (mm)

Equation (1) may be illustrated as shown in Fig. 1. In the following cases, Fig. 1 can be utilized to easily obtain an approximate numerical values for dimensional change:

(1) To correct dimensional measurements according to the ambient air temperature
(2) To find the change in bearing internal clearance due to temperature difference between inner and outer rings during operation
(3) To find the relationship between the interference and heating temperature during shrink fitting
(4) To find the change in the interference when a temperature difference exists on the fit surface

Example
To what temperature should the inner ring be heated if an inner ring of 110 mm in bore is to be shrink fitted to a shaft belonging to the n6 tolerance range class?

The maximum interference between the n6 shaft of 110 in diameter and the inner ring is 0.065. To enable insertion of the inner ring with ease on the shaft, there must be a clearance of 0.03 to 0.04. Accordingly, the amount to expand the inner ring must be 0.095 to 0.105. Intersection of a vertical axis Δl=0.105 and a horizontal axis l=110 is determined on a diagram. ΔT is located in the temperature range between 70°C and 80°C (ΔT≒77°C). Therefore, it is enough to set the inner ring heating temperature to the room temperature +80°C.

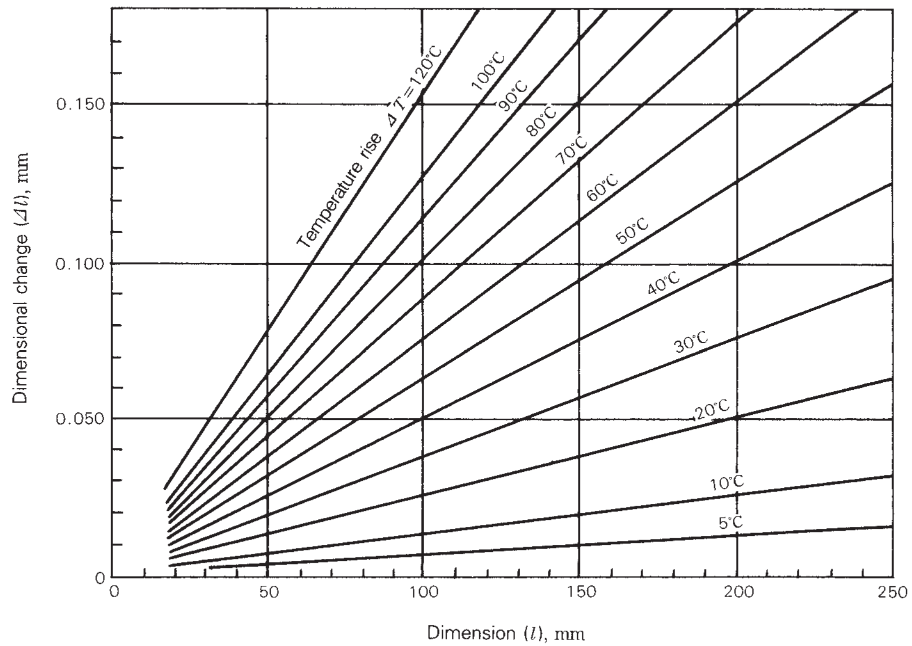


Fig. 1 Temperature rise and dimensional change of bearing steel

12.7 Bearing volume and apparent specific gravity

The bearing bore is expressed by "d" (mm), the bearing outside diameter by "D" (mm), and the width by "B" (mm). The volume "V" of a bearing is expressed as follows:

$$V=\frac{\pi}{4}(D^2-d^2)\,B\times 10^{-3}\ (\text{cm}^3) \quad \cdots\cdots\cdots\cdots \ (1)$$

Table 1 shows the bearing volume for the principal dimension series of radial bearings. In the case of a tapered roller bearing, the volume is a calculated value assuming the assembly width as "B". When the bearing mass is expressed by "W" (kg), $W/V=k$ may be considered as an apparent specific gravity and the value of "k" is nearly constant according to the type of bearings.

Table 2 shows the values of "k" for radial bearings of each dimension series. When the mass of a bearing not included in the standard dimension series is to be determined, the approximate mass value may be known by using the apparent specific gravity "k" if the bearing volume "V" has been determined.

Table 1 Volume of radial bearing

Units: cm^3

Bearing bore No.	Radial bearing (excluding tapered roller bearing)						Tapered roller bearing		
	Dimension series						Dimension series		
	10	30	02	22	03	23	20	02	03
00	3.6	5.4	5.6	8.8	9.7	15.0	—	—	—
01	4.0	6.0	6.9	9.7	11.5	16.3	—	—	—
02	5.4	7.9	8.4	11.0	15.7	20.5	—	—	17.2
03	7.4	10.3	12.3	16.5	21.1	28.6	—	13.6	23.0
04	12.9	17.1	19.9	25.6	27.1	38.0	—	21.7	29.4
05	14.9	20.0	24.5	29.4	43.0	60.6	—	26.5	46.1
06	21.7	31.7	36.9	46.2	63.9	90.8	28.4	39.3	69.8
07	28.8	41.1	52.9	71.5	85.3	126	37.0	56.8	92.4
08	35.6	50.0	67.9	86.7	117	168	45.2	74.5	129
09	45.2	65.0	77.6	93.9	157	225	56.5	84.9	170
10	49.0	70.5	88.0	101	203	301	61.3	95.6	220
11	71.7	104	115	137	259	384	92	125	281
12	76.7	111	147	187	324	480	98	159	350
13	81.6	118	184	249	398	580	104	198	434
14	113	170	202	261	484	705	142	221	525
15	119	179	221	275	580	860	150	241	627
16	159	239	269	342	689	1 020	204	293	750
17	175	270	336	432	810	1 190	230	366	880
18	217	334	412	550	945	1 410	289	446	1 020
19	226	348	500	671	1 095	1 630	301	538	1 200
20	236	362	598	809	1 340	2 080	313	650	1 460
21	298	469	709	985	1 530	2 390	400	767	1 660
22	369	594	833	1 160	1 790	2 860	502	898	1 950
24	396	649	1 000	1 450	2 300	3 590	536	1 090	2 480
26	598	945	1 130	1 810	2 800	4 490	818	1 240	3 080
28	632	1 020	1 415	2 290	3 430	5 640	866	1 540	3 740
30	773	1 240	1 780	2 890	4 080	6 770	1 060	1 940	4 520

Table 2 Bearing type and apparent specific gravity (k)

Bearing type	Principal bearing series	Apparent specific gravity, k
Single row deep groove ball bearing (with pressed cage)	60, 62, 63	5.3
NU type cylindrical roller bearing	NU10, NU2, NU3	6.8
N type cylindrical roller bearing	N10, N2, N3	6.5
Tapered roller bearing	320, 302, 303	5.5
Spherical roller bearing	230, 222, 223	6.4

12.8 Projection amount of cage in tapered roller bearing

The cage of a tapered roller bearing is made of a steel plate press construction and projects perpendicularly from the side of the outer ring as shown in Fig. 1. It is essential to design the bearing mounting to prevent the cage from contacting such parts as the housing and spacer. It is also recommended to employ the dimension larger than specified in JIS B 1566 “Mounting Dimensions and Fit for Rolling Bearing” and S_a and S_b of the bearing catalog in view of securing the grease retaining space in the case of grease lubrication and in view of improving the oil flow in the case of oil lubrication.

However, if the dimension cannot be designed smaller due to a dimensional restriction in the axial direction, then mounting dimensions S_a and S_b should be selected by adding as large as possible space to the maximum projection values δ_1 and δ_2 (Table 1) to the cage from the outer ring side.

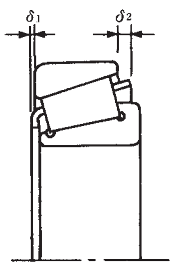


Fig. 1 Projection of a cage

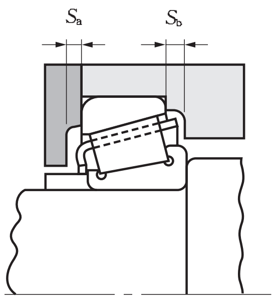


Fig. 2 Dimensions related to bearing mounting

Table 1 Projection of a cage of tapered roller bearing

Units: mm

Bearing bore No.	Bearing series																			
	HR329J		HR320XJ		HR330J		HR331J		HR302J		HR322J		HR332J		HR303J		HR303DJ		HR323J	
	δ_1	δ_2	δ_1	δ_2	δ_1	δ_2	δ_1	δ_2	δ_1	δ_2	δ_1	δ_2	δ_1	δ_2	δ_1	δ_2	δ_1	δ_2	δ_1	δ_2
02	—	—	—	—	—	—	—	—	—	—	—	—	—	—	1.2	3.3	—	—	—	—
03	—	—	—	—	—	—	—	—	0.7	2.0	0.3	3.0	—	—	1.4	3.7	—	—	—	—
04	—	—	1.5	2.9	—	—	—	—	1.0	2.9	0.6	3.5	—	—	0.9	3.7	—	—	1.3	3.2
/22	—	—	1.6	3.1	—	—	—	—	—	—	0.9	3.8	—	—	1.1	2.9	—	—	—	—
05	—	—	1.9	3.5	2.0	3.1	—	—	0.8	2.9	0.9	3.8	2.0	3.3	1.6	3.4	—	—	1.5	4.5
/28	—	—	1.9	3.5	—	—	—	—	1.4	3.4	1.5	3.4	1.8	3.8	1.5	4.8	—	—	—	—
06	—	—	2.0	3.1	2.0	4.0	—	—	1.4	3.4	1.5	3.3	2.1	4.6	2.1	3.9	—	—	1.6	4.0
/32	—	—	2.0	3.9	—	—	—	—	0.7	3.3	1.6	2.8	2.2	4.4	—	—	—	—	—	—
07	1.3	2.7	2.5	3.7	2.2	3.4	—	—	2.0	3.1	1.7	4.3	2.6	4.7	2.9	4.8	2.1	4.8	1.1	4.1
08	1.9	2.9	2.3	4.2	2.2	3.2	—	—	1.1	4.7	1.4	5.1	3.1	5.5	1.8	4.9	2.0	5.0	0.5	4.5
09	—	—	2.9	4.7	—	—	3.3	4.7	1.8	3.9	1.9	5.1	3.7	6.0	2.5	5.1	2.3	5.5	2.0	5.2
10	—	—	3.0	4.6	2.4	4.4	3.3	5.1	1.8	5.5	1.7	6.2	3.3	5.8	2.2	5.9	3.7	6.8	1.5	5.7
11	1.9	3.3	3.1	5.1	2.9	4.8	3.3	6.3	2.7	4.8	2.1	4.5	3.5	6.6	2.6	5.7	3.3	6.0	1.8	6.4
12	2.3	3.5	3.1	5.0	2.9	5.1	—	—	1.2	5.9	3.4	4.2	3.9	7.0	3.1	6.5	3.2	8.0	2.7	6.5
13	—	—	2.5	5.9	3.0	5.1	—	—	3.9	4.8	2.8	4.0	4.9	7.4	3.1	6.2	3.9	10.0	2.3	7.4
14	2.5	4.1	2.9	5.6	3.5	5.5	—	—	3.3	5.3	2.7	5.0	5.5	7.0	3.2	6.5	3.8	8.2	2.1	7.2
15	—	—	3.5	5.5	3.5	5.4	—	—	3.9	5.3	2.8	4.7	5.0	7.9	3.0	7.6	3.7	8.6	1.8	7.7
16	—	—	4.5	7.2	—	—	—	—	3.1	5.5	3.1	4.6	4.7	7.6	2.2	7.8	3.4	9.2	2.2	7.9
17	—	—	4.2	6.5	3.7	6.0	4.8	7.6	3.1	6.3	2.1	5.8	4.6	8.7	3.4	8.5	4.0	10.3	2.8	9.8
18	3.4	5.5	4.4	7.2	—	—	4.8	7.5	3.6	5.1	2.6	5.1	—	—	—	—	3.2	9.6	2.1	8.9
19	3.3	5.2	4.5	7.1	—	—	—	—	3.5	5.9	1.9	5.4	—	—	—	—	3.0	10.3	—	—
20	3.4	5.1	4.5	7.1	—	—	3.8	8.8	3.2	6.9	2.0	5.6	3.8	9.4	—	—	—	—	2.1	10.3

12.9 Natural frequency of individual bearing rings

The natural frequencies of individual bearing rings of a rolling bearing are mainly composed of radial vibration and axial vibration. The natural frequency in the radial direction is a vibration mode as shown in Fig. 1. These illustrated modes are in the radial direction and include modes of various dimensions according to the circumferential shape, such as a primary (elliptical), secondary (triangular), tertiary (square), and other modes.

As shown in Fig. 1, the number of nodes in the primary mode is four, with the number of waves due to deformation being two. The number of waves is three and four respectively in the secondary and tertiary modes. In regards to the radial natural frequency of individual bearing rings, Equation (1) is based on the theory of thin circular arc rod and agrees well with measured values:

$$f_{RiN}=\frac{1}{2\pi}\sqrt{\frac{Eg}{\gamma}\,\frac{I_x}{AR^4}}\;\frac{n\,(n^2-1)}{\sqrt{n^2-1}}\quad \text{(Hz)} \cdots\cdots\cdots (1)$$

where, f_{RiN}: i-th natural frequency of individual bearing rings in the radial directios (Hz)
E: Young's modulus (MPa) {kgf/mm^2}
γ: Specific weight (N/mm^3) {kgf/mm^3}
g: Gravity acceleration (mm/s^2)
n: Number of deformation waves in each mode (i+1)
I_x: Sectional secondary moment at neutral axis of the bearing ring (mm^4)
A: Sectional area of bearing ring (mm^2)
R: Radius of neutral axis of bearing ring (mm)

The value of the sectional secondary moment is needed before using Equation (1). But it is troublesome to determine this value exactly for a bearing ring with a complicated cross-sectional shape. Equation (2) is best used when the radial natural frequency is known approximately for the outer ring of a radial ball bearing. Then, the natural frequency can easily be determined by using the constant determined from the bore, outside diameter, and cross-sectional shape of the bearing.

$$f_{RiN}=9.41\times10^5\,\frac{K\,(D-d)}{\{D-K\,(D-d)\}^2}\times\frac{n\,(n^2-1)}{\sqrt{n^2-1}}\quad \text{(Hz)} \cdots\cdots\cdots (2)$$

where, d: Bearing bore (mm)
D: Bearing outside diameter (mm)
K: Constant determined from the cross-sectional shape
K=0.125 (outer ring with seal grooves)
K=0.150 (outer ring of an open type)

Another principal mode is the one in the axial direction. The vibration direction of this mode is in the axial direction and the modes range from the primary to tertiary as shown in Fig. 2. The figure shows the case as viewed from the side. As in the case of the radial vibration modes, the number of waves of deformation in primary, secondary, and tertiary is two, three, and four respectively. As for the natural frequency of individual bearing rings in the axial direction, there is an approximation Equation (3), which is obtained by synthesizing an equation based upon the theory of circular arc rods and another based on the non-extension theory of cylindrical shells:

$$f_{AiN}=\frac{\frac{\sqrt{3}}{6\pi}\,n\,(n^2-1)\,\rho}{(1-\nu^2)\left(\frac{\rho}{\kappa}\right)^2\frac{n^2\,(n^2+1)\,\rho^2+3}{n^2\rho^2+6\,(1-\nu)}+n^2+\lambda}\times\frac{1}{R}\sqrt{\frac{Eg}{\gamma}}\quad \text{(Hz)} \cdots\cdots\cdots (3)$$

where, $\kappa=H/2R$
$\rho=B/2R$

$$\lambda=\frac{1+\nu}{2-1.26\sigma\,(1-\sigma^4/12)}$$

$$\sigma=\min\left(\frac{\kappa}{\rho},\frac{\rho}{\kappa}\right)$$

where, f_{AiN}: i-th natural frequency of individual bearing rings is the axial direction (Hz)
E: Young's modulus (MPa) {kgf/mm^2}
γ: Specific weight (N/mm^3) {kgf/mm^3}
g: Gravity acceleration (mm/s^2)
n: Number of deformation waves in each mode (i+1)
R: Radius of neutral axis of bearing ring (mm)
H: Thickness of bearing ring (mm)
B: Width of bearing ring (mm)
ν: Poisson's ratio

This equation applies to a rectangular sectional shape and agrees well with actual measurements in the low-dimension mode even in the case of a bearing ring. But this calculation is difficult. Therefore, Equation (4) is best used when the natural frequency in the axial direction is known approximately for the outer ring of the ball bearing. Calculation can then be made using the numerical values obtained from the bearing's bore, outside diameter, width, and outer ring sectional shape.

$$f_{AiN}=\frac{9.41\times10^5\,n\,(n^2-1)\,R_0^2}{B\sqrt{\frac{0.91}{H_0^2}\cdot\frac{n^2\,(n^2+1)\,R_0^2+3}{n^2R_0^2+4.2}+n^2+\frac{1.3}{2-1.26H_0+0.105H_0^5}}}\quad \text{(Hz)} \cdots\cdots\cdots (4)$$

where, $R_0=B/\{D-K\,(D-d)\}$
$H_0=K\,(D-d)/B$

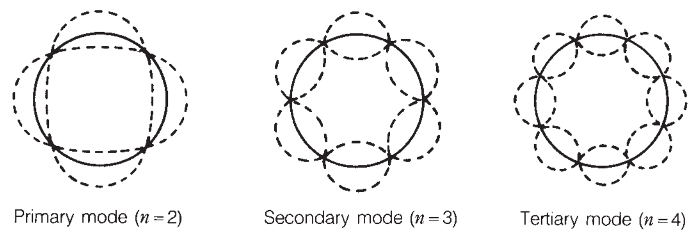


Fig. 1 Primary to tertiary vibration modes in the radial direction

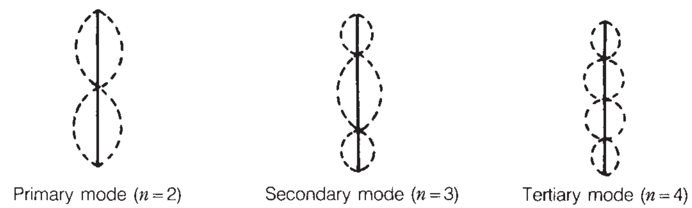


Fig. 2 Primary to tertiary vibration modes in the axial direction

12.10 Vibration and noise of bearings

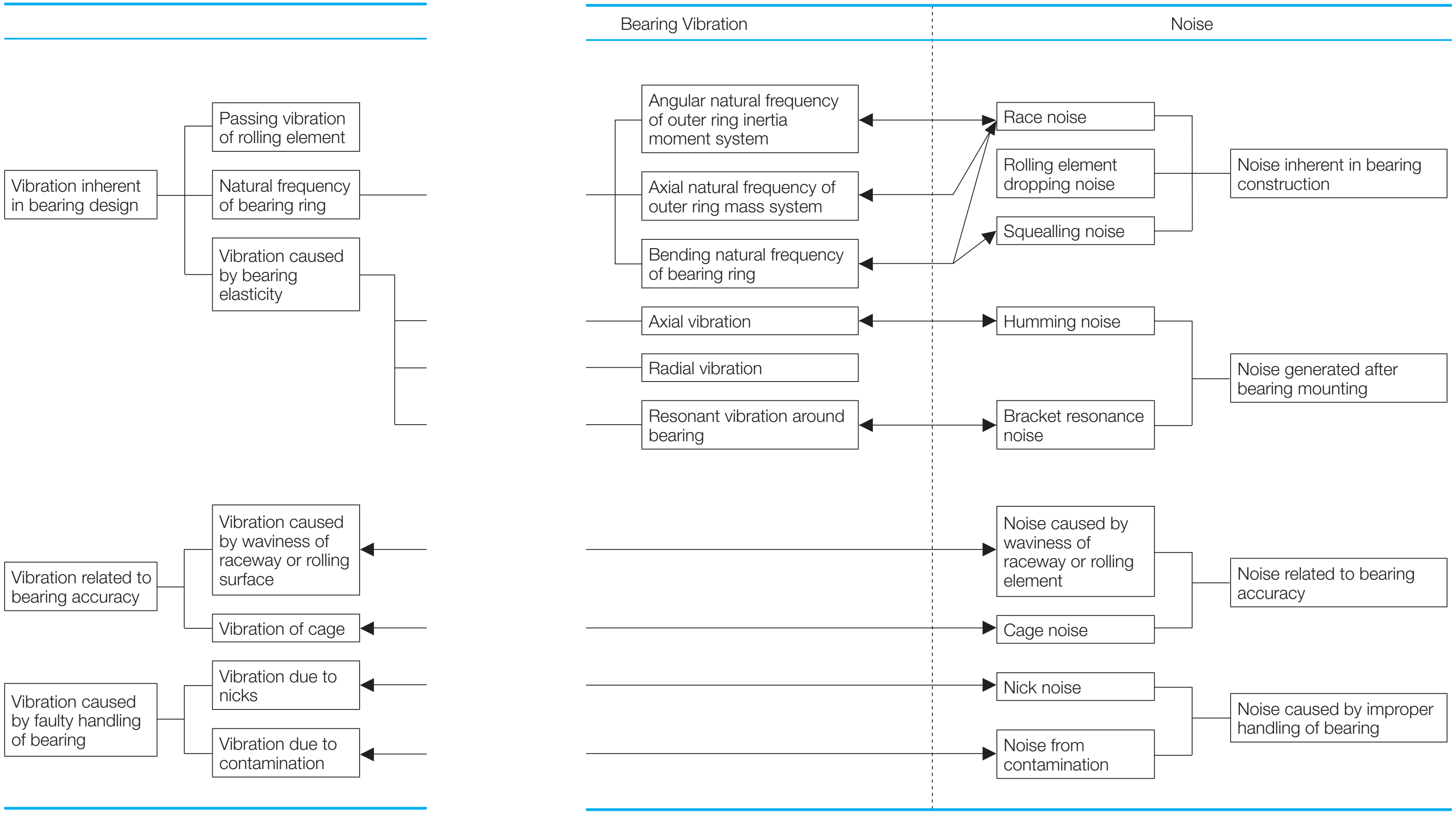
The vibration and noise occurring in a rolling bearing are very diverse. Some examples are shown in Table 1. This table shows the vibration and noise of bearings while classifying them roughly into those inherent in the bearing design which occur regardless of the present superb technology and those caused by other reasons, both are further subdivided into several groups. The boundaries among these groups, however, are not absolute. Although vibration and noise due to the bearing structure may be related to the magnitude of the bearing accuracy, nevertheless, vibration attributed to accuracy may not be eliminated completely by improving the accuracy, because there exists certain effects generated by the parts surrounding the bearing.

Arrows in the table show the relationship between the vibration and noise.

Generally, vibration and noise are in a causal sequence but they may be confused. Under normal bearing running conditions, however, around 1 kHz may be used as a boundary line to separate vibration from noise. Namely, by convention, the frequency range of about 1 kHz or less will be treated as vibration while that above this range will be treated as noise.

Typical vibration and noise, as shown in Table 1, have already been clarified as to their causes and present less practical problems. But the environmental changes as encountered these days during operation of a bearing have come to generate new kinds of vibration and noise. In particular, there are cases of abnormal noise in the low temperature environment, which can often be attributed to friction inside of a bearing. If the vibration and noise (including new kinds of abnormal noise) of a bearing are to be prevented or reduced, it is essential to define and understand the phenomenon by focusing on vibration and noise beforehand. As portable tape recorders with satisfactory performance are commonly available these days, it is recommended to use a tape recorder to record the actual sound of the vibration or noise.

Table 1 Vibration and noise of rolling bearing



12.11 Application of FEM to design of bearing system

Before a rolling bearing is selected in the design stage of a machine, it is often necessary to undertake a study of dynamic and thermal problems (mechanical structure and neighboring bearing parts) in addition to the dimensions, accuracy, and material of the shaft and housing. For example, in the prediction of the actual load distribution and life of a bearing installed in a machine, there are problems with overload or creep caused by differences in thermal deformation due to a combination of factors such as dissimilar materials, or estimation of temperature rise or temperature distribution.

NSK designs optimum bearings by using Finite Element Methods (FEM) to analyze the shaft and bearing system. Let's consider an example where FEM is used to solve a problem related to heat conduction.

Fig. 1 shows an example of calculating the temperature distribution in the steady state of a rolling mill bearing while considering the bearing heat resistance or heat resistance in the fit section when the outside surface of a shaft and housing is cooled with water. In this analysis, the amount of decrease in the internal clearance of a bearing due to temperature rise or the amount of increase needed for fitting between the shaft and inner ring can be found.

Fig. 2 shows a calculation for the change in the temperature distribution as a function of time after the start of operation for the headstock of a lathe. Fig. 3 shows a calculation example for the temperature change in the principal bearing components. In this example, it is predicted that the bearing preload increases immediately after rotation starts and reaches the maximum value in about 10 minutes.

When performing heat analysis of a bearing system by FEM, it is difficult to calculate the heat generation or to set the boundary conditions to the ambient environment. NSK is proceeding to accumulate an FEM analytical database and to improve its analysis technology in order to effectively harness the tremendous power of FEM.

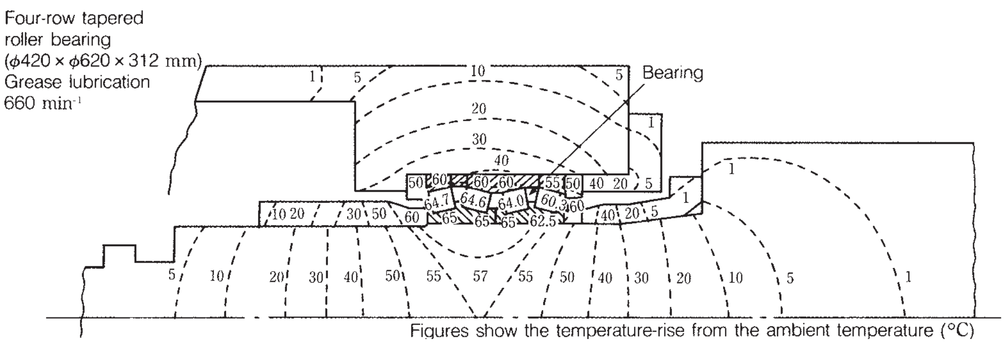


Fig. 1 Calculation example of temperature distribution for the intermediate roll of a rolling mill

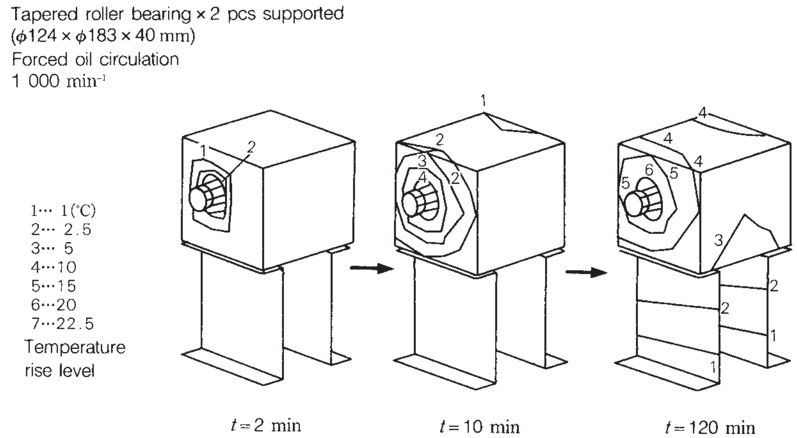


Fig. 2 Calculation example of temperature rise in headstock of lathe

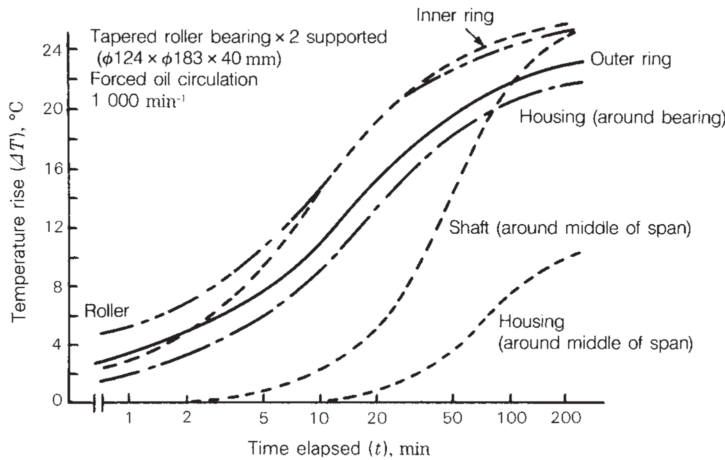


Fig. 3 Calculation example of temperature rise in bearing system

As an example of applying the FEM-based analysis, we introduce here the result of a study on the effect of the shape of a rocker plate supporting the housing of a plate rolling mill both on the life of a tapered roller bearing (ϕ489×ϕ635 in dia. ×321) and on the housing stress. Fig. 4 shows an approximated view of the housing and rocker plate under analysis. The following points are the rerults of analysis made while changing the relief amount l at the top surface of the rocker plate:

(1) The maximum value σ_{max} of the stress (maximum main stress) on a housing occurs at the bottom of the housing.

(2) σ_{max} increases with increase in l. But it is small relative to the fatigue limit of the material.

(3) The load distribution in the rolling element of the bearing varies greatly depending on l. The bearing life reaches a maximum at around l/L=0.7.

(4) In this example, l/L=0.5 to 0.7 is considered to be the most appropriate in view of the stress in the housing and the bearing life. Fig. 5 shows the result of calculation on the housing stress distribution and shape as well as the rolling element load distribution when l/L≒0.55.

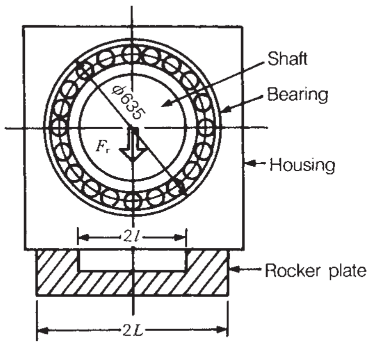


Fig. 4 Rolling mill housing and rocker plate

Fig. 6 shows the result of a calculation on the housing stress and bearing life as a function of change in l.

FEM-based analysis plays a crucial role in the design of bearing systems. Finite Element Methods are applicable in widely-varying fields as shown in Table 1. Apart from these, FEM is used to analyze individual bearing components and contribute to NSK's high-level bearing design capabilities and achievements. Two examples are the analysis of the strength of a rib of a roller bearing and the analysis of the natural mode of a cage.

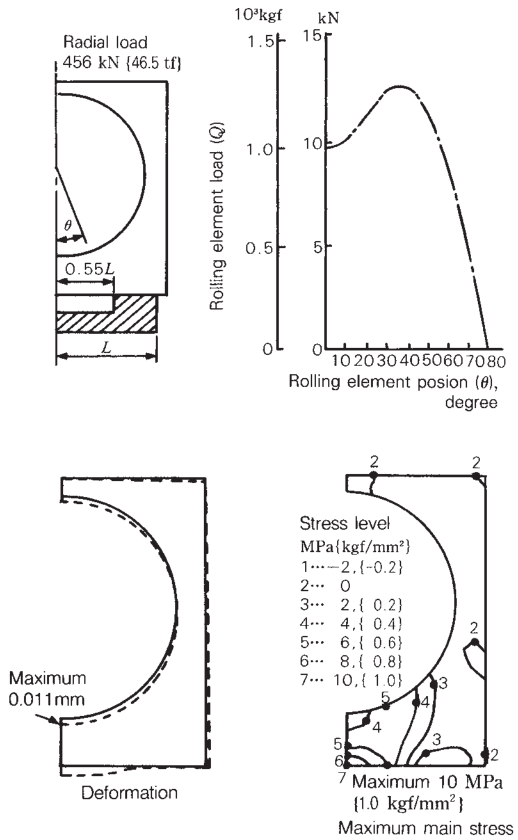


Fig. 5 Calculation example of housing stress and rolling element load distribution of bearing

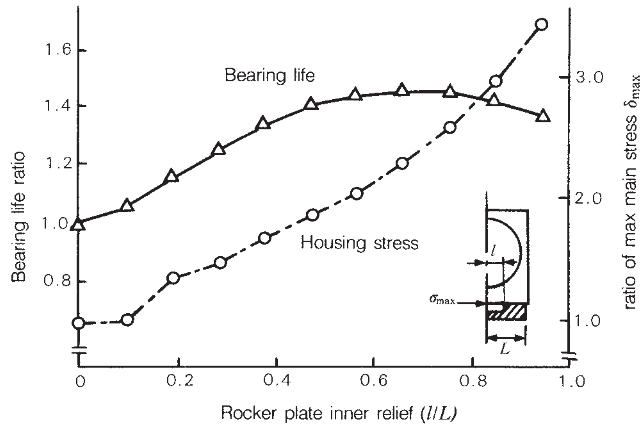


Fig. 6 Calculation of housing stress and bearing life

Table 1 Examples of FEM analysis of bearing systems

Bearing application	Examples	Purpose of analysis
Automobile	•Hub unit •Tension pulley •Differential gear and surrounding •Steering joint	Strength, rigidity, creep, deformation, bearing life
Electric equipment	•Motor bracket •Alternator •Suction Motor Bearing for Cleaner •Pivot Ball Bearing Unit for HDDs	Vibration, rigidity, deformation, bearing life
Steel machinery	•Roll neck bearing peripheral structure (cold rolling, hot rolling, temper mill) •Adjusting screw thrust block •CC roll housing	Strength, rigidity, deformation, temperature distribution, bearing life
Machine tool	•Machining center spindle •Grinding spindle •Lathe spindle •Table drive system peripheral structure	Vibration, rigidity, temperature distribution, bearing life
Others	•Jet engine spindle •Railway rolling stock •Semiconductor-related equipment •Engine block •Slewing bearing's peripheral	Strength, rigidity, thermal deformation, vibration, deformation, bearing life