2. Dynamic load rating, fatigue life, and static load rating

2.1 Dynamic load rating

The basic dynamic load rating of rolling bearings is defined as the constant load applied on bearings with stationary outer rings that the inner rings can endure for a rating life (90% life) of one million revolutions. The basic load rating of radial bearings is defined as a central radial load of constant direction and magnitude, while the basic load rating of thrust bearings is defined as an axial load of constant magnitude in the same direction as the central axis.

This basic dynamic load rating is calculated by an equation shown in Table 1. The equation is based on the theory of G. Lundberg & A. Palmgren, and was adopted in ISO R281 : 1962 in 1962 and in JIS B 1518 : 1965 in Japan in March, 1965. Later on, these standards were established respectively as ISO 281 : 1990 and JIS B 1518 (under revision) after some modification.

The fatigue life of a bearing is calculated as follows:

$$L=\left(\frac{C}{P}\right)^{3} \text{ for a ball bearing} \quad \cdots\cdots (1)$$

$$L=\left(\frac{C}{P}\right)^{10/3} \text{ for a roller bearing} \quad \cdots\cdots (2)$$

where, L: Rating fatigue life (10^6 rev)
P: Dynamic equivalent load (N), {kgf}
C: Basic dynamic load rating (N), {kgf}

The factor f_c used in the calculation of Table 1 has a different value depending on the bearing type, as shown in Tables 2 and 3. The value f_c of a radial ball bearing is the same as specified in JIS B 1518 : 1965, while that of a radial roller bearing was revised to be the maximum possible value. In this way, the factor f_c determined from the processing accuracy and material has been at about the same level for the past 20 years.

During this period, however, bearings have undergone substantial improvement in terms of not only material, but also processing accuracy. As a result, the practical bearing life is extended considerably. It would be easier to use the above equations for calculations with improved bearings because the dynamic load rating already reflects the life extension factor. This concept of ISO 281 : 1990 and JIS B 1518 (under revision) has led to the increase of the basic dynamic load rating by multiplying by the rating factor b_m. The value of the rating factor b_m is as shown in Table 4.

Table 1 Calculation equation of basic dynamic load rating

Classification		Ball bearing	Roller bearing
Radial bearing		$b_m f_c (i \cos\alpha)^{0.7} Z^{2/3} D_w^{1.8}$	$b_m f_c (i L_{we} \cos\alpha)^{7/9} Z^{3/4} D_{we}^{29/27}$
Single-row thrust bearing	$\alpha=90°$	$b_m f_c Z^{2/3} D_w^{1.8}$	$b_m f_c L_{we}^{7/9} Z^{3/4} D_{we}^{29/27}$
	$\alpha \neq 90°$	$b_m f_c (\cos\alpha)^{0.7} \tan\alpha\, Z^{2/3} D_w^{1.8}$	$b_m f_c (L_{we} \cos\alpha)^{7/9} \tan\alpha\, Z^{3/4} D_{we}^{29/27}$
Quantity symbols in equations		b_m: Rating factor depending on normal material and manufacture quality f_c: Coefficient determined from shape, processing accuracy, and material of bearing parts i: Number of rows of rolling elements in one bearing α: Nominal contact angle (°) Z: Number of rolling elements per row D_w: Diameter of ball (mm) D_{we}: Diameter of roller used in calculation (1) (mm) L_{we}: Effective length of roller (mm)	

Note (1) Diameter at the middle of the roller length
Tapered roller: Arithmetic average value of roller large and small end diameters assuming the roller without chamfers
Convex roller (asymmetric): Approximate value of roller diameter at the contact point of the roller and ribless raceway (generally outer ring raceway) without applying load

Remarks When D_w>25.4 mm, $D_w^{1.8}$ becomes 3.647 $D_w^{1.4}$

Table 2 f_c value of radial ball bearings

$\frac{D_w \cos\alpha}{D_{pw}}$ (1)	f_c		
	Single-row deep groove ball bearing, single/double row angular contact ball bearing	Double-row deep groove ball bearing	Self-aligning ball bearing
0.05	46.7 {4.76}	44.2 {4.51}	17.3 {1.76}
0.06	49.1 {5.00}	46.5 {4.74}	18.6 {1.90}
0.07	51.1 {5.21}	48.4 {4.94}	19.9 {2.03}
0.08	52.8 {5.39}	50.0 {5.10}	21.1 {2.15}
0.09	54.3 {5.54}	51.4 {5.24}	22.3 {2.27}
0.10	55.5 {5.66}	52.6 {5.37}	23.4 {2.39}
0.12	57.5 {5.86}	54.5 {5.55}	25.6 {2.61}
0.14	58.8 {6.00}	55.7 {5.68}	27.7 {2.82}
0.16	59.6 {6.08}	56.5 {5.76}	29.7 {3.03}
0.18	59.9 {6.11}	56.8 {5.79}	31.7 {3.23}
0.20	59.9 {6.11}	56.8 {5.79}	33.5 {3.42}
0.22	59.6 {6.08}	56.5 {5.76}	35.2 {3.59}
0.24	59.0 {6.02}	55.9 {5.70}	36.8 {3.75}
0.26	58.2 {5.93}	55.1 {5.62}	38.2 {3.90}
0.28	57.1 {5.83}	54.1 {5.52}	39.4 {4.02}
0.30	56.0 {5.70}	53.0 {5.40}	40.3 {4.11}
0.32	54.6 {5.57}	51.8 {5.28}	40.9 {4.17}
0.34	53.2 {5.42}	50.4 {5.14}	41.2 {4.20}
0.36	51.7 {5.27}	48.9 {4.99}	41.3 {4.21}
0.38	50.0 {5.10}	47.4 {4.84}	41.0 {4.18}

Note (1) D_{pw} is the pitch diameter of balls.
Remarks Figures in { } for kgf unit calculation

Table 3 f_c value of radial roller bearings

$\frac{D_{we} \cos\alpha}{D_{pw}}$ (2)	f_c
0.01	52.1 {5.32}
0.02	60.8 {6.20}
0.03	66.5 {6.78}
0.04	70.7 {7.21}
0.05	74.1 {7.56}
0.06	76.9 {7.84}
0.07	79.2 {8.08}
0.08	81.2 {8.28}
0.09	82.8 {8.45}
0.10	84.2 {8.59}
0.12	86.4 {8.81}
0.14	87.7 {8.95}
0.16	88.5 {9.03}
0.18	88.8 {9.06}
0.20	88.7 {9.05}
0.22	77.2 {9.00}
0.24	87.5 {8.92}
0.26	86.4 {8.81}
0.28	85.2 {8.69}
0.30	83.8 {8.54}

Note (2) D_{pw} is the pitch diameter of rollers.
Remarks 1. The f_c value in the above table applies to a bearing in which the stress distribution in the length direction of roller is nearly uniform.
2. Figures in { } for kgf unit calculation

Table 4 Value of rating factor b_m

Bearing type		b_m
Radial Bearings	Deep groove ball bearing	1.3
	Magneto bearing	1.3
	Angular contact ball bearing	1.3
	Ball bearing for rolling bearing unit	1.3
	Self-aligning ball bearing	1.3
	Spherical roller bearing	1.15
	Filling slot ball bearing	1.1
	Cylindrical roller bearing	1.1
	Tapered roller bearing	1.1
	Solid needle roller bearing	1.1
Thrust Bearings	Thrust ball bearing	1.3
	Thrust spherical roller bearing	1.15
	Thrust tapered roller bearing	1.1
	Thrust cylindrical roller bearing	1
	Thrust needle roller bearing	1

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2.2 Dynamic equivalent load

In some cases, the loads applied on bearings are purely radial or axial loads; however, in most cases, the loads are a combination of both. In addition, such loads usually fluctuate in both magnitude and direction.

In such cases, the loads actually applied on bearings cannot be used for bearing life calculations; therefore, a hypothetical load should be estimated that has a constant magnitude and passes through the center of the bearing, and will give the same bearing life that the bearing would attain under actual conditions of load and rotation. Such a hypothetical load is called the equivalent load.

Assuming the equivalent radial load as P_r, the radial load as F_r, the axial load as F_a, and the contact angle as α, the relationship between the equivalent radial load and bearing load can be approximated as follows:

$$P_r = XF_r + YF_a \quad \cdots\cdots (1)$$

where, X: Radial load factor } See Table 1
Y: Axial load factor }

The axial load factor varies depending on the contact angle. Though the contact angle remains the same regardless of the magnitude of the axial load in the cases of roller bearings, such as single-row deep groove ball bearings and angular contact ball bearings experience an increase in contact angle when the axial load is increased. Such change in the contact angle can be expressed by the ratio of the basic static load rating C_{0r} and axial load F_a. Table 1, therefore, shows the axial load factor at the contact angle corresponding to this ratio. Regarding angular contact ball bearings, the effect of change in the contact angle on the load factor may be ignored under normal conditions even if the contact angle is as large as 25°, 30° or 40°.

For the thrust bearing with the contact angle of α=90° receiving both radial and axial loads simultaneously, the equivalent axial load P_a becomes as follows:

$$P_a = XF_r + YF_a \quad \cdots\cdots (2)$$

Table 1 Value of factors X and Y

Bearing type		$\frac{C_{0r}}{F_a}$	Single-row bearing $F_a/F_r \leqq e$ X	Y	Single-row bearing $F_a/F_r > e$ X	Y	Double-row bearing $F_a/F_r \leqq e$ X	Y	Double-row bearing $F_a/F_r > e$ X	Y	e
Single-row deep groove ball bearings		5	1	0	0.56	1.26	—	—	—	—	0.35
		10				1.49					0.29
		15				1.64					0.27
		20				1.76					0.25
		25				1.85					0.24
		30				1.92					0.23
		50				2.13					0.20
Angular contact ball bearings	15°	5	1	0	0.44	1.10	1	1.23	0.72	1.79	0.51
		10				1.21		1.36		1.97	0.47
		15				1.28		1.43		2.08	0.44
		20				1.32		1.48		2.14	0.42
		25				1.36		1.52		2.21	0.41
		30				1.38		1.55		2.24	0.40
		50				1.44		1.61		2.34	0.39
	25°	—	1	0	0.41	0.87	1	0.92	0.67	1.41	0.68
	30°	—	1	0	0.39	0.76	1	0.78	0.63	1.24	0.80
	40°	—	1	0	0.35	0.57	1	0.55	0.57	0.93	1.14
Self-aligning ball bearings		—	—	—	—	—	1	0.42cotα	0.65	0.65cotα	1.5tanα
Magnet ball bearings		—	1	0	0.5	2.5	—	—	—	—	0.2
Tapered roller bearings Spherical roller bearings		—	1	0	0.4	0.4cotα	1	0.45cotα	0.67	0.67cotα	1.5tanα
Thrust ball bearings	45°	—	—	—	0.66	1	1.18	0.59	0.66	1	1.25
	60°	—	—	—	0.92	1	1.90	0.55	0.92	1	2.17
Thrust roller bearings		—	—	—	tanα	1	1.5tanα	0.67	tanα	1	1.5tanα

Remarks 1. Two similar single-row angular contact ball bearings are used.
(1) DF or DB combination: Apply X and Y of double-row bearing. However, if obtain the axial load ratio of C_{0r}/F_a, C_{0r} should be half of C_{0r} for the bearing set.
(2) DT combination: Apply X and Y of single-row bearing. C_{0r} should be half of C_{0r} for the bearing set.
2. This table differs from JIS and ISO standards in the method of determining the axial load ratio C_{0r}/F_a.

2.3 Dynamic equivalent load of triplex angular contact ball bearings

Three separate single-row bearings may be used side by side as shown in the figure when angular contact ball bearings are to be used to carry a large axial load. There are three patterns of combination, which are expressed by combination symbols of DBD, DFD, and DTD.

As in the case of single-row and double-row bearings, the dynamic equivalent load, which is determined from the radial and axial loads acting on a bearing, is used to calculate the fatigue life for these combined bearings.

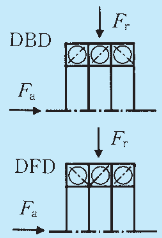
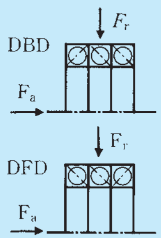
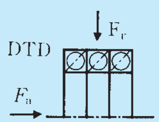
Assuming the dynamic equivalent radial load as P_r, the radial load as F_r, and axial load as F_a, the relationship between the dynamic equivalent radial load and bearing load may be approximated as follows:

$$P_r = XF_r + YF_a \quad \cdots\cdots (1)$$

where, X: Radial load factor } See Table 1
Y: Axial load factor

The axial load factor varies with the contact angle. In an angular contact ball bearing, whose contact angle is small, the contact angle varies substantially when the axial load increases.

A change in the contact angle can be expressed by the ratio between the basic static load rating C_{0r} and axial load F_a. Accordingly, for the angular contact ball bearing with a contact angle of 15°, the axial load factor at a contact angle corresponding to this ratio is shown. If the angular contact ball bearings have contact angles of 25°, 30° and 40°, the effect of change in the contact angle on the axial load factor may be ignored and thus the axial load factor is assumed as constant.

Arrangement	Load direction
3 row matched stack, axial load is supported by 2 rows. (Symbol DBD or DFD)	
3 row matched stack, axial load is supported by 1 row. (Symbol DBD or DFD)	
3 row tandem matched stack (Symbol DTD)	

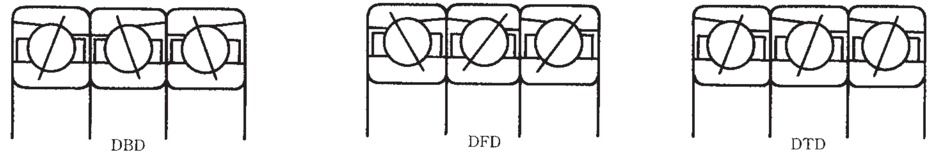


Table 1 Factors X and Y of triplex angular contact ball bearing

Contact angle α	j	$\frac{C_{0r}}{jF_a}$	$\frac{F_a}{F_r} \leqq e$		$\frac{F_a}{F_r} > e$		e	Basic load rating of 3 row ball bearings	
			X	Y	X	Y		C_r	C_{0r}
		5		0.64		1.46	0.51		
		10		0.70		1.61	0.47		
		15		0.74		1.70	0.44		
15°	1.5	20	1	0.76	0.58	1.75	0.42		
		25		0.78		1.81	0.41	2.16 times of single bearing	3 times of single bearing
		30		0.80		1.83	0.40		
		50		0.83		1.91	0.39		
25°	—	—	1	0.48	0.54	1.16	0.68		
30°	—	—	1	0.41	0.52	1.01	0.80		
40°	—	—	1	0.29	0.46	0.76	1.14		
		5		2.28		2.37	0.51		
		10		2.51		2.61	0.47		
		15		2.64		2.76	0.44		
15°	3	20	1	2.73	0.95	2.85	0.42		
		25		2.80		2.93	0.41	2.16 times of single bearing	3 times of single bearing
		30		2.85		2.98	0.40		
		50		2.98		3.11	0.39		
25°	—	—	1	1.70	0.88	1.88	0.68		
30°	—	—	1	1.45	0.84	1.64	0.80		
40°	—	—	1	1.02	0.76	1.23	1.14		
		5				1.10	0.51		
		10				1.21	0.47		
		15				1.28	0.44		
15°	1	20	1	0	0.44	1.32	0.42		
		25				1.36	0.41	2.16 times of single bearing	3 times of single bearing
		30				1.38	0.40		
		50				1.44	0.39		
25°	—	—	1	0	0.41	0.87	0.68		
30°	—	—	1	0	0.39	0.76	0.80		
40°	—	—	1	0	0.35	0.57	1.14		

2.4 Average of fluctuating load and speed

When the load applied on a bearing fluctuates, an average load which will yield the same bearing life as the fluctuating load should be calculated.

(1) When the relation between load and rotating speed can be partitioned into groups of rectangles (Fig. 1),

Load F_1; Speed n_1; Operating time t_1
Load F_2; Speed n_2; Operating time t_2
⋮ ⋮ ⋮
Load F_n; Speed n_n; Operating time t_n

then the average load F_m may be calculated using the following equation:

$$F_m = \sqrt[p]{\frac{F_1{}^p n_1 t_1 + F_2{}^p n_2 t_2 + \cdots + F_n{}^p n_n t_n}{n_1 t_1 + n_2 t_2 + \cdots\cdots + n_n t_n}} \quad \cdots\cdots\cdots\cdots\cdots\cdots (1)$$

where, F_m: Average of fluctuating load (N), {kgf}
p=3 for ball bearings
p=10/3 for roller bearings

The average speed n_m may be calculated as follows:

$$n_m = \frac{n_1 t_1 + n_2 t_2 + \cdots\cdots + n_n t_n}{t_1 + t_2 + \cdots\cdots + t_n} \quad \cdots\cdots\cdots\cdots\cdots\cdots (2)$$

(2) When the load fluctuates almost linearly (Fig. 2), the average load may be calculated as follows:

$$F_m \fallingdotseq \frac{1}{3}\left(F_{min} + 2F_{max}\right) \quad \cdots\cdots\cdots\cdots\cdots\cdots (3)$$

where, F_{min}: Minimum value of fluctuating load (N), {kgf}
F_{max}: Maximum value of fluctuating load (N), {kgf}

(3) When the load fluctuation is similar to a sine wave (Fig. 3), an approximate value for the average load F_m may be calculated from the following equation:

In the case of Fig. 3 (a)
$F_m \fallingdotseq 0.65\ F_{max}$ ························· (4)
In the case of Fig. 3 (b)
$F_m \fallingdotseq 0.75\ F_{max}$ ························· (5)

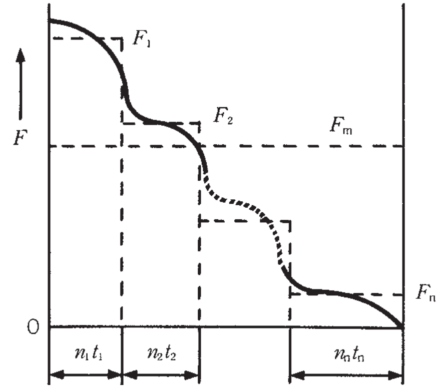


Fig. 1 Incremental load variation

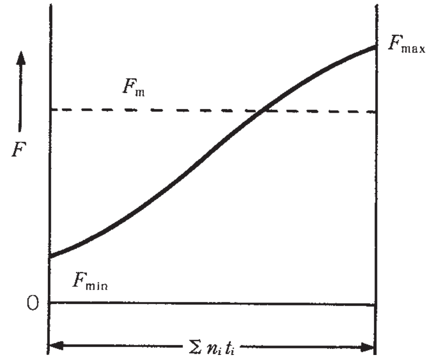


Fig. 2 Simple load fluctuation

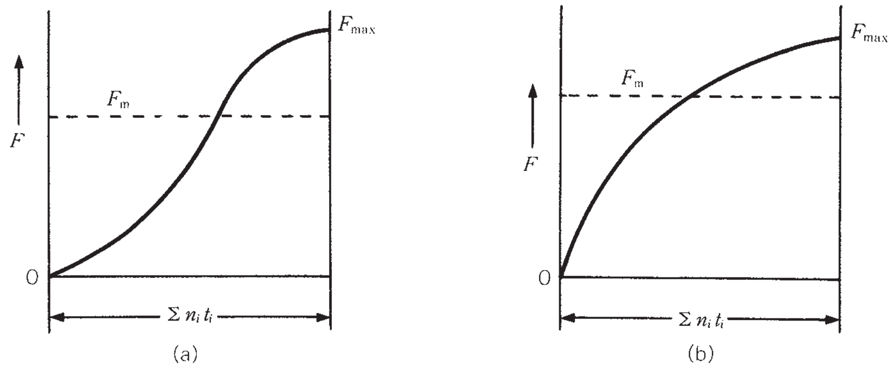


Fig. 3 Sinusoidal load variation

2.5 Combination of rotating and stationary loads

Generally, rotating, static, and indeterminate loads act on a rolling bearing. In certain cases, both the rotating load, which is caused by an unbalanced or a vibration weight, and the stationary load, which is caused by gravity or power transmission, may act simultaneously. The combined mean effective load when the indeterminate load caused by rotating and static loads can be calculated as follows. There are two kinds of combined loads; rotating and stationary which are classified depending on the magnitude of these loads, as shown in Fig. 1.

Namely, the combined load becomes a running load with its magnitude changing as shown in Fig. 1 (a) if the rotating load is larger than the static load. The combined load becomes an oscillating load with a magnitude changing as shown in Fig. 1 (b) if the rotating load is smaller than the stationary load.

In either case, the combined load F is expressed by the following equation:

$$F=\sqrt{F_R^2+F_S^2-2F_RF_S\cos\theta} \quad \cdots\cdots (1)$$

where, F_R: Rotating load (N), {kgf}
F_S: Stationary load (N), {kgf}
θ: Angle defined by rotating and stationary loads

The value F can be approximated by Load Equations (2.1) and (2.2) which vary sinusoidally depending on the magnitude of F_R and F_S, that is, in such a manner that F_R+F_S becomes the maximum load F_{max} and F_R-F_S becomes the minimum load F_{min} for $F_R \gg F_S$ or $F_R \ll F_S$.

$$F_R \gg F_S,\ F=F_R-F_S\cos\theta \quad \cdots\cdots (2.1)$$
$$F_R \ll F_S,\ F=F_S-F_R\cos\theta \quad \cdots\cdots (2.2)$$

The value F can also be approximated by Equations (3.1) and (3.2) when $F_R \fallingdotseq F_S$.

$F_R>F_S$,

$$F=F_R-F_S+2F_S\sin\frac{\theta}{2} \quad \cdots\cdots (3.1)$$

$F_R>F_S$,

$$F=F_S-F_R+2F_R\sin\frac{\theta}{2} \quad \cdots\cdots (3.2)$$

Curves of Equations (1), (2.1), (3.1), and (3.2) are as shown in Fig. 2.

The mean value F_m of the load varying as expressed by Equations (2.1) and (2.2) or (3.1) and (3.2) can be expressed respectively by Equations (4.1) and (4.2) or (5.1) and (5.2).

$$F_m=F_{min}+0.65\,(F_{max}-F_{min})$$
$$F_R \geqq F_S,\ F_m=F_R+0.3F_S \quad \cdots\cdots (4.1)$$
$$F_R \leqq F_S,\ F_m=F_S+0.3F_R \quad \cdots\cdots (4.2)$$

$$F_m=F_{min}+0.75\,(F_{max}-F_{min})$$
$$F_R \geqq F_S,\ F_m=F_R+0.5F_S \quad \cdots\cdots (5.1)$$
$$F_R \leqq F_S,\ F_m=F_S+0.5F_R \quad \cdots\cdots (5.2)$$

Generally, as the value F exists somewhere among Equations (4.1), (4.2), (5.1), and (5.2), the factor 0.3 or 0.5 of the second terms of Equations (4.1) and (4.2) as well as (5.1) and (5.2) is assumed to change linearly along with F_S/F_R or F_R/F_S. Then, these factors may be expressed as follows:

$$0.3+0.2\frac{F_S}{F_R},\ 0 \leqq \frac{F_S}{F_R} \leqq 1$$

or $$0.3+0.2\frac{F_R}{F_S},\ 0 \leqq \frac{F_R}{F_S} \leqq 1$$

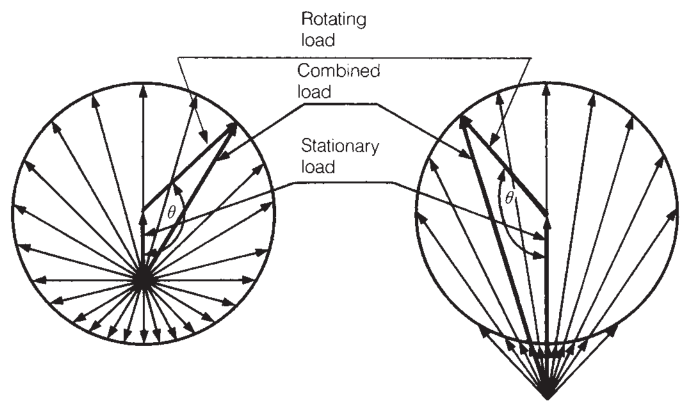
Accordingly, F_m can be expressed by the following equation:

$F_R \geqq F_S$,

$$F_m=F_R+\left(0.3+0.2\frac{F_S}{F_R}\right)F_S$$
$$=F_R+0.3F_S+0.2\frac{F_S^2}{F_R} \quad \cdots\cdots (6.1)$$

$F_R \geqq F_S$,

$$F_m=F_S+\left(0.3+0.2\frac{F_R}{F_S}\right)F_R$$
$$=F_S+0.3F_R+0.2\frac{F_R^2}{F_S} \quad \cdots\cdots (6.2)$$



(a) Rotating load > stationary load (b) Rotating load < stationary load

Fig. 1 Combined load of rotating and stationary loads

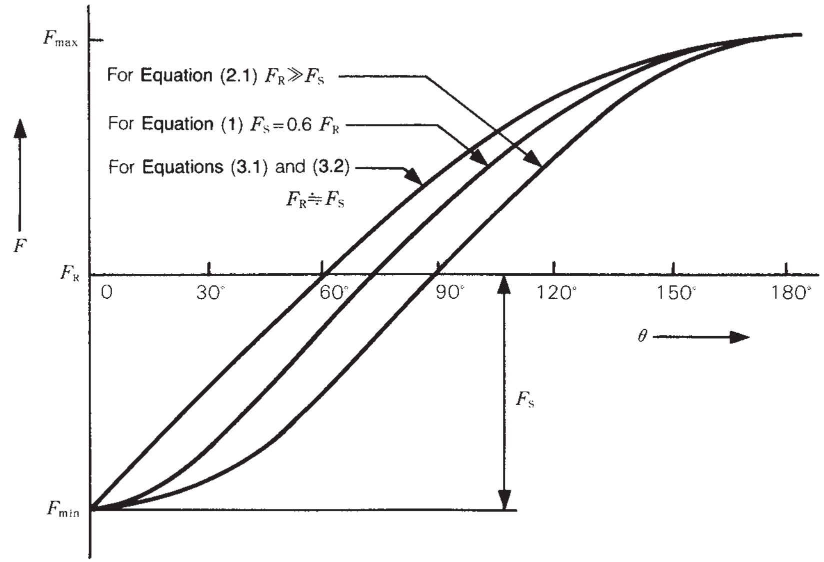


Fig. 2 Chart of combined loads

2.6 Life calculation of multiple bearings as a group

When multiple rolling bearings are used in one machine, the fatigue life of individual bearings can be determined if the load acting on individual bearings is known. Generally, however, the machine becomes inoperative if a bearing in any part fails. It may therefore be necessary in certain cases to know the fatigue life of a group of bearings used in one machine.

The fatigue life of the bearings varies greatly and our fatigue life calculation equation

$L=\left(\frac{C}{P}\right)^{p}$ applies to the 90% life (also called the rating fatigue life, which is either the gross number of revolution or hours to which 90% of multiple similar bearings operated under similar conditions can reach).

In other words, the calculated fatigue life for one bearing has a probability of 90%. Since the endurance probability of a group of multiple bearings for a certain period is a product of the endurance probability of individual bearings for the same period, the rating fatigue life of a group of multiple bearings is not determined solely from the shortest rating fatigue life among the individual bearings. In fact, the group life is much shorter than the life of the bearing with the shortest fatigue life.

Assuming the rating fatigue life of individual bearings as L_1, L_2, L_3 ... and the rating fatigue life of the entire group of bearings as L, the below equation is obtained:

$$\frac{1}{L^e}=\frac{1}{L_1{}^e}+\frac{1}{L_2{}^e}+\frac{1}{L_3{}^e}+\cdots\cdots \quad (1)$$

where, e=1.1 (both for ball and roller bearings)

L of Equation (1) can be determined with ease by using Fig. 1.

Take the value L_1 of Equation (1) on the L_1 scale and the value of L_2 on the L_2 scale, connect them with a straight line, and read the intersection with the L scale. In this way, the value L_A of

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

is determined. Take this value L_A on the L_1 scale and the value L_3 on the L_2 scale, connect them with a straight line, and read an intersection with the L scale.
In this way, the value L of

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

can be determined.

Example

Assume that the calculated fatigue life of bearings of automotive front wheels as follows:

280 000 km for inner bearing
320 000 km for outer bearing

Then, the fatigue life of bearings of the wheel can be determined at 160 000 km from Fig. 1. If the fatigue life of the bearing of the right-hand wheel takes this value, the fatigue life of the left-hand wheel will be the same. As a result, the fatigue life of the front wheels as a group will become 85 000 km.

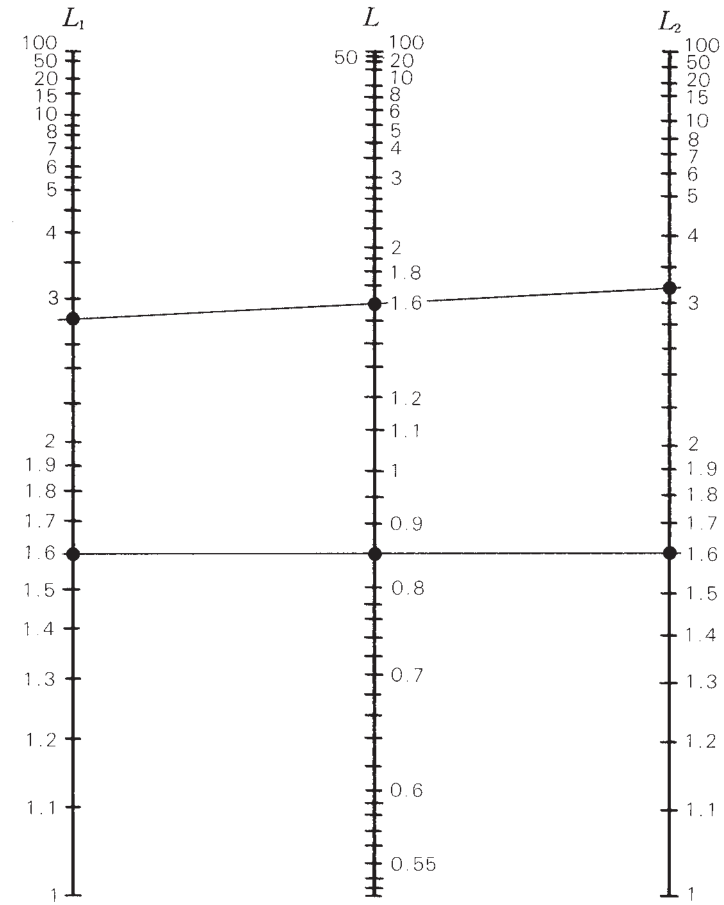


Fig. 1 Chart for life calculation

2.7 Load factor and fatigue life by machine

Loads acting on the bearing, rotating speed, and other conditions must be taken into account when selecting a bearing for a machine. Basic loads acting on a bearing are considered normally to include the weight of a rotating body supported by the bearing, load developed by power transmitted gears and belt, and other loads which can be estimated by calculation.
Actually, in addition to the above loads, there are loads caused by unbalance of a rotating body, load developed due to vibration and shock during operation, etc., which are, however, difficult to determine accurately. In order to assume the dynamic equivalent load P necessary for selection of the bearing, therefore, the above basic load F_c is converted into a practical mean effective load by multiplying it by a certain factor. This factor is called the load factor f_w, which is an empirical value. Table 1 shows the guideline of load factor f_w for each machine and operating conditions. For example, when a part incorporating a bearing is subject to a radial load of F_{rc} and an axial load of F_{ac}, the dynamic equivalent load P can be expressed as follows, with load factors assumed respectively as f_{w1} and f_{w2}:

$$P=Xf_{w1}F_{rc}+Yf_{w2}F_{ac} \quad \cdots\cdots (1)$$

Setting an unnecessarily long fatigue life during selection of a bearing is not economical because it will lead to a larger bearing. Moreover, the fatigue life of a bearing may not be the sole standard in certain cases in view of the strength, rigidity, and mounting dimensions of the shaft. In general, the bearing design life is set as a guideline for each machine and operation conditions to ensure selection of an adequate yet economical bearing.

Such a design life requires an empirical value called the fatigue life factor f_h. Table 2 shows the fatigue life factors which are summarized for each machine and operating conditions. It is therefore necessary to determine the basic load rating C from the fatigue life factor f_h appropriate to the bearing application purpose while using the equation as follows:

$$C \geqq \frac{f_h \cdot P}{f_n} \quad \cdots\cdots (2)$$

where, C: Basic dynamic load rating (N), {kgf}
f_n: Speed factor

The bearing must satisfy the calculated basic dynamic load rating C as shown above.

Table 1 Value of load factor f_w

Running conditions	Typical machine	f_w
Smooth operation free from shock	Electric motors, machine tools, air conditioners	1 to 1.2
Normal operation	Air blowers, compressors, elevators, cranes, paper making machines	1.2 to 1.5
Operation exposed to shock and vibration	Construction equipment, crushers, vibrating screens, rolling mills	1.5 to 3

Table 2 Fatigue life factor f_h for various bearing applications

Operating periods	Fatigue life factor f_h and machine				
	≤3	2 to 4	3 to 5	4 to 7	≥6
Infrequently or only for short periods	•Small motors for home appliances like vacuum cleaners •Hand powered tools	•Agricultural equipment			
Only occasionally but reliability is important		•Motors for home heaters and air conditioners •Construction equipment	•Conveyors •Elevators		
Intermittently for relatively long periods	•Rolling mill roll necks	•Small motors •Deck cranes •General cargo cranes •Pinion stands •Passenger cars	•Factory motors •Machine tools •Transmissions •Vibrating screens •Crushers	•Crane sheaves •Compressors •Specialized transmissions	
Intermittently for more than eight hours daily		•Escalators	•Centrifugal separators •Air conditioning equipment •Blowers •Woodworking machines •Large motors •Axle boxes on railway rolling stock	•Mine hoists •Press fly-wheels •Railway traction motors •Locomotive axle boxes	•Papermaking machines
Continuously and high reliability is important					•Waterworking pumps •Electric power station •Mine draining pumps

2.8 Radial clearance and fatigue life

As shown in the catalog, etc., the fatigue life calculation equation of rolling bearings is Equation (1):

$$L=\left(\frac{C}{P}\right)^{p} \quad \cdots\cdots (1)$$

where, L: Rating fatigue life (10^6rev)
C: Basic dynamic load rating (N), {kgf}
P: Dynamic equivalent load (N), {kgf}
p: Index Ball bearing p=3,

Roller bearing $p=\frac{10}{3}$

The rating fatigue life L for a radial bearing in this case is based on a prerequisite that the load distribution in the bearing corresponds to the state with the load factor ε = 0.5 (Fig. 1). The load factor ε varies depending on the magnitude of load and bearing internal clearance. Their relationship is described in 5.2 (Radial Internal Clearance and Load Factor of Ball Bearing).

The load distribution with ε=0.5 is obtained when the bearing internal clearance is zero. In this sense, the normal fatigue life calculation is intended to obtain the value when the clearance is zero. When the effect of the radial clearance is taken into account, the bearing fatigue life can be calculated as follows. Equations (2) and (3) can be established between the bearing radial clearance Δ_r and a function $f(\varepsilon)$ of load factor ε:

For deep groove ball bearing

$$f(\varepsilon)=\frac{\Delta_r \cdot D_w^{1/3}}{0.00044\left(\frac{F_r}{Z}\right)^{2/3}} \quad \cdots\cdots (N)$$

$$f(\varepsilon)=\frac{\Delta_r \cdot D_w^{1/3}}{0.002\left(\frac{F_r}{Z}\right)^{2/3}} \quad \cdots\cdots \{kgf\}$$

$\cdots\cdots$ (2)

For cylindrical roller bearing

$$f(\varepsilon)=\frac{\Delta_r \cdot L_{we}^{0.8}}{0.000077\left(\frac{F_r}{Z \cdot i}\right)^{0.9}} \quad \cdots\cdots (N)$$

$$f(\varepsilon)=\frac{\Delta_r \cdot L_{we}^{0.8}}{0.0006\left(\frac{F_r}{Z \cdot i}\right)^{0.9}} \quad \cdots\cdots \{kgf\}$$

$\cdots\cdots$ (3)

where, Δ_r: Radial clearance (mm)
F_r: Radial load (N), {kgf}
Z: Number of rolling elements
i: No. of rows of rolling elements
D_w: Ball diameter (mm)
L_{we}: Effective roller length (mm)
L_ε: Life with clearance of Δ_r
L: Life with zero clearance, obtained from Equation (1)

The relationship between load factor ε and $f(\varepsilon)$, and the life ratio L_ε/L, when the radial internal clearance is Δ_r can also be obtained as shown in Table 1.

Fig. 2 shows the relationship between the radial clearance and bearing fatigue life while taking 6208 and NU208 as examples.

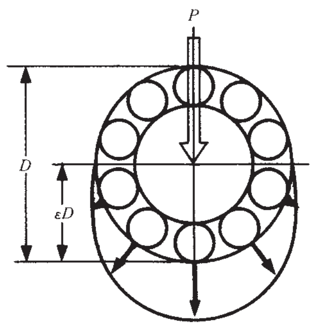


Fig. 1 Load distribution with ε=0.5

Table 1 ε and $f(\varepsilon)$, L_ε/L

ε	Deep groove ball bearing		Cylindrical roller bearing	
	$f(\varepsilon)$	$\frac{L_\varepsilon}{L}$	$f(\varepsilon)$	$\frac{L_\varepsilon}{L}$
0.1	33.713	0.294	51.315	0.220
0.2	10.221	0.546	14.500	0.469
0.3	4.045	0.737	5.539	0.691
0.4	1.408	0.889	1.887	0.870
0.5	0	1.0	0	1.0
0.6	− 0.859	1.069	− 1.133	1.075
0.7	− 1.438	1.098	− 1.897	1.096
0.8	− 1.862	1.094	− 2.455	1.065
0.9	− 2.195	1.041	− 2.929	0.968
1.0	− 2.489	0.948	− 3.453	0.805
1.25	− 3.207	0.605	− 4.934	0.378
1.5	− 3.877	0.371	− 6.387	0.196
1.67	− 4.283	0.276	− 7.335	0.133
1.8	− 4.596	0.221	− 8.082	0.100
2.0	− 5.052	0.159	− 9.187	0.067
2.5	− 6.114	0.078	−11.904	0.029
3	− 7.092	0.043	−14.570	0.015
4	− 8.874	0.017	−19.721	0.005
5	−10.489	0.008	−24.903	0.002
10	−17.148	0.001	−48.395	0.0002

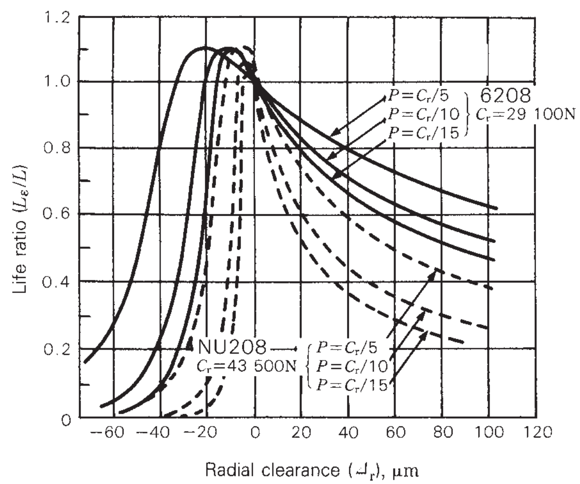


Fig. 2 Radial clearance and bearing life ratio

2.9 Misalignment of inner/outer rings and fatigue life of deep-groove ball bearings

A rolling bearing is manufactured with high accuracy, and it is essential to take utmost care with machining and assembly accuracies of surrounding shafts and housing if this accuracy is to be maintained. In practice, however, the machining accuracy of parts around the bearing is limited, and bearings are subject to misalignment of inner/outer rings caused by the shaft deflection under external load.

The allowable misalignment is generally 0.0006 ~ 0.003 rad (2′ to 10′) but this varies depending on the size of the deep-groove ball bearing, internal clearance during operation, and load.

This section introduces the relationship between the misalignment of inner/outer rings and fatigue life. Four different sizes of bearings are selected as examples from the 62 and 63 series deep-groove ball bearings.

Assume the fatigue life without misalignment as $L_{\theta=0}$ and the fatigue life with misalignment as L_θ. The effect of the misalignment on the fatigue life may be found by calculating $L_\theta/L_{\theta=0}$. The result is shown in Figs. 1 to 4.

As an example of ordinary running conditions, the radial load F_r (N) {kgf} and axial load F_a (N) {kgf} were assumed respectively to be approximately 10% (normal load) and 1% (light preload) of the dynamic load rating C_r (N) {kgf} of a bearing and were used as load conditions for the calculation. Normal radial clearance was used and the shaft fit was set to around j5. Also taken into account was the decrease of the internal clearance due to expansion of the inner ring.

Moreover, assuming that the temperature difference between the inner and outer rings was 5°C during operation, inner/outer ring misalignment, $L_\theta/L_{\theta=0}$ was calculated for the maximum, minimum, and mean effective clearances.

As shown in Figs. 1 to 4, degradation of the fatigue life is limited to 5 to 10% or less when the misalignment ranges from 0.0006 to 0.003 rad (2′ to 10′), thus not presenting much problem.

When the misalignment exceeds a certain limit, however, the fatigue life degrades rapidly as shown in the figure. Attention is therefore necessary in this respect.

When the clearance is small, not much effect is observed as long as the misalignment is small, as shown in the figure. But the life decreases substantially when the misalignment increases. As previously mentioned, it is essential to minimize the mounting error as much as possible when a bearing is to be used.

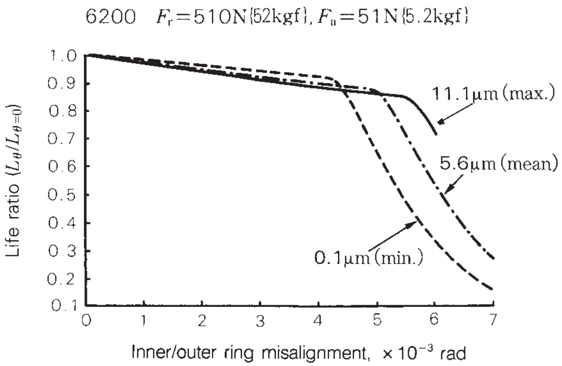


Fig. 1

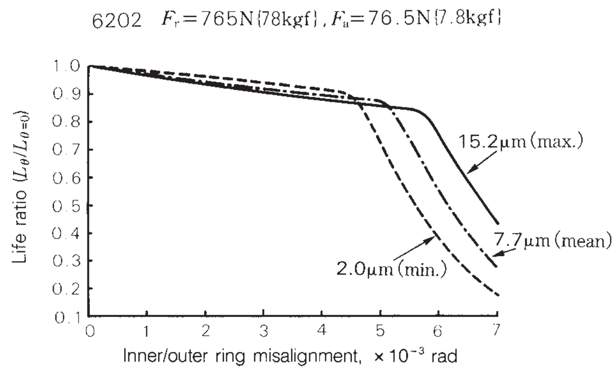


Fig. 2

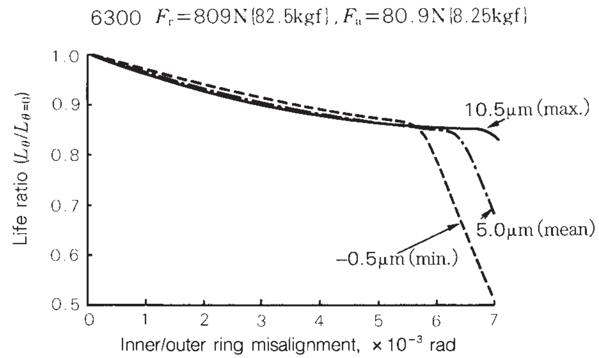


Fig. 3

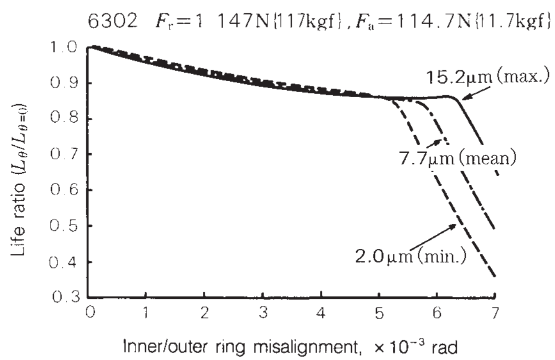


Fig. 4

2.10 Misalignment of inner/outer rings and fatigue life of cylindrical roller bearings

When a shaft supported by rolling bearings is deflected or there is some inaccuracy in a shoulder, there arises misalignment between the inner and outer rings of the bearings, thereby lowering their fatigue life. The degree of life degradation depends on the bearing type and interior design but also varies depending on the radial internal clearance and the magnitude of load during operation.

The relationship between the misalignment of inner/outer rings and fatigue life was determined, as shown in Figs. 1 to 4, while using cylindrical roller bearings NU215 and NU315 of standard design.

In these figures, the horizontal axis shows the misalignment of inner/outer rings (rad) while the vertical axis shows the fatigue life ratio $L_\theta / L_{\theta=0}$. The fatigue life without misalignment is $L_{\theta=0}$ and that with misalignment is L_θ.

Figs. 1 and 2 show the case with constant load (10% of basic dynamic load rating C_r of a bearing) for each case when the internal clearance is a normal, C3 clearance, or C4 clearance. Figs. 3 and 4 show the case with constant clearance (normal clearance) when the load is 5%, 10%, and 20% of the basic dynamic load rating C_r.

Note that the median effective clearance in these examples was determined using m5/H7 fits and a temperature difference of 5°C between the inner and outer rings.

The fatigue life ratio for the clearance and load shows the same trend as in the case of other cylindrical roller bearings. But the life ratio itself differs among bearing series and dimensions, with life degradation rapid in 22 and 23 series bearings (wide type). It is advisable to use a bearing of special design when considerable misalignment is expected during application.

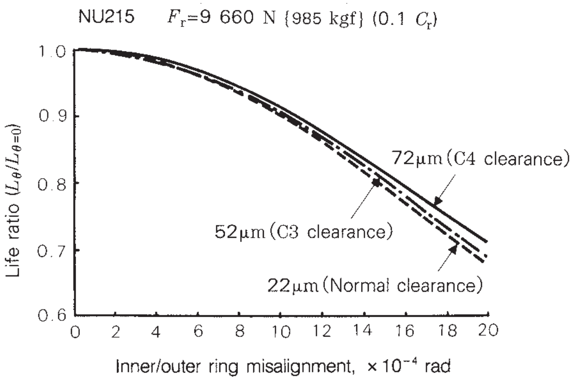


Fig. 1

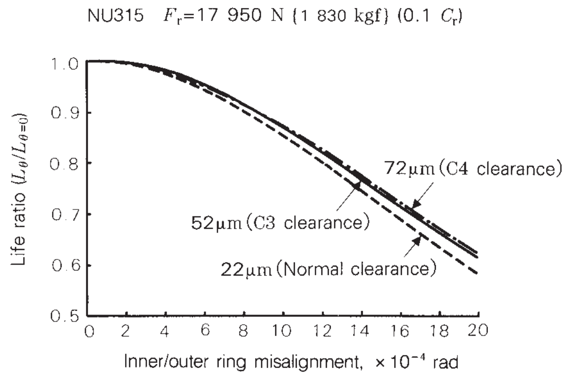


Fig. 2

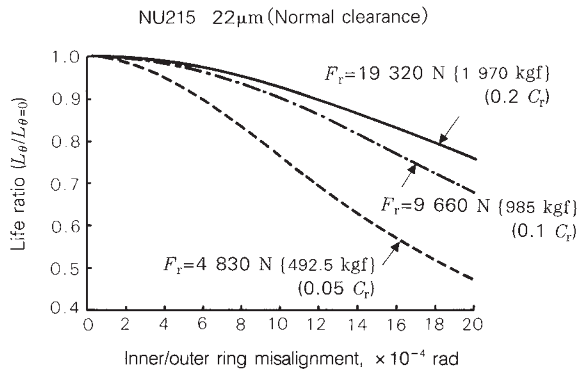


Fig. 3

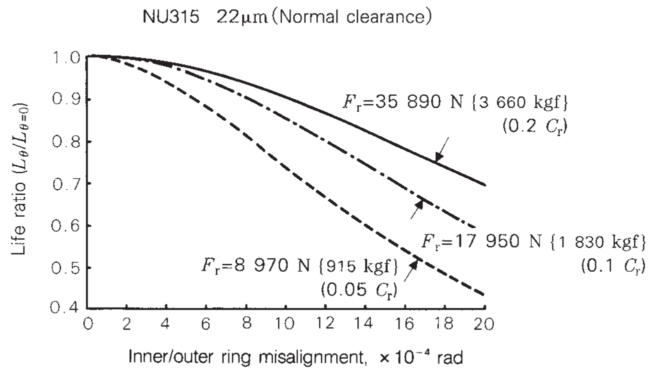


Fig. 4

2.11 Fatigue life and reliability

Where any part failure may result in damage to the entire machine and repair of damage is impossible, as in applications such as aircraft, satellites, or rockets, greatly increased reliability is demanded of each component. This concept is being applied generally to durable consumer goods and may also be utilized to achieve effective preventive maintenance of machines and equipment.

The rating fatigue life of a rolling bearing is the gross number of revolutions or the gross rotating period when the rotating speed is constant for which 90% of a group of similar bearings running individually under similar conditions can rotate without suffering material damage due to rolling fatigue. In other words, fatigue life is normally defined at 90% reliability. There are other ways to describe the life. For example, the average value is employed frequently to describe the life span of human beings. However, if the average value were used for bearings, then too many bearings would fail before the average life value is reached. On the other hand, if a low or minimum value is used as a criterion, then too many bearings would have a life much longer than the set value. In this view, the value 90% was chosen for common practice. The value 95% could have been taken as the statistical reliability, but nevertheless, the slightly looser reliability of 90% was taken for bearings empirically from the practical and economical viewpoint. A 90% reliability however is not acceptable for parts of aircraft or electronic computers or communication systems these days, and a 99% or 99.9% reliability is demanded in some of these cases.

The fatigue life distribution when a group of similar bearings are operated individually under similar conditions is shown in Fig. 1. The Weibull equation can be used to describe the fatigue life distribution within a damage ratio of 10 to 60% (residual probability of 90 to 40%). Below the damage ratio of 10% (residual probability of 90% or more), however, the rolling fatigue life becomes longer than the theoretical curve of the Weibull distribution, as shown in Fig. 2. This is a conclusion drawn from the life test of numerous, widely-varying bearings and an analysis of the data.

When bearing life with a failure ratio of 10% or less (for example, the 95% life or 98% life) is to be considered on the basis of the above concept, the reliability factor a_1 as shown in the table below is used to check the life. Assume here that the 98% life L_2 is to be calculated for a bearing whose rating fatigue life L_{10} was calculated at 10 000 hours. The life can be calculated as L_2=0.33×L_{10}=3 300 hours. In this manner, the reliability of the bearing life can be matched to the degree of reliability required of the equipment and difficulty of overhaul and inspection.

Table 1 Reliability factor

Reliability, %	90	95	96	97	98	99
Life, L_a	L_{10} rating life	L_5	L_4	L_3	L_2	L_1
Reliability factor, a_1	1	0.62	0.53	0.44	0.33	0.21

Apart from rolling fatigue, factors such as lubrication, wear, sound, and accuracy govern the durability of a bearing. These factors must be taken into account, but the endurance limit of these factors varies depending on application and conditions.

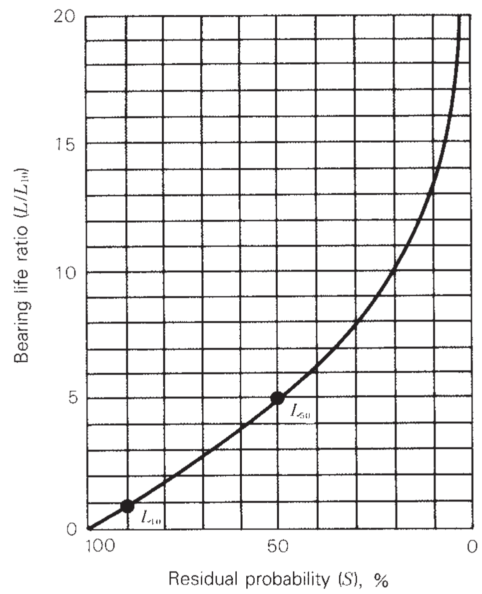


Fig. 1 Bearing life and residual probability

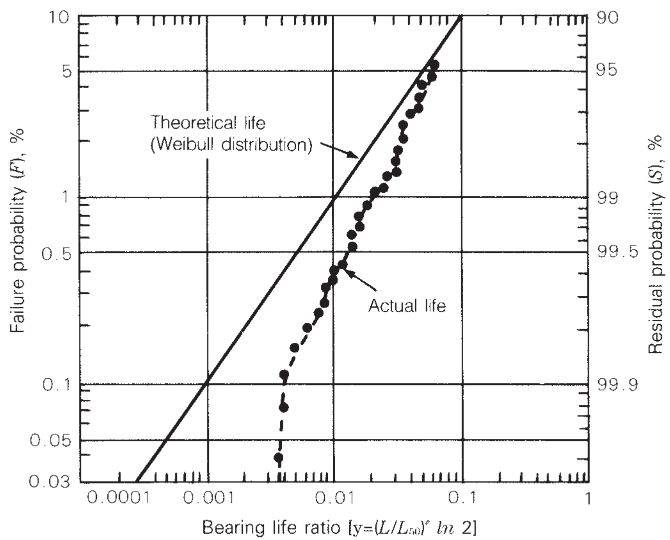


Fig. 2 Life distribution in the low failure ratio range

2.12 Oil film parameters and rolling fatigue life

Based on numerous experiments and experiences, the rolling fatigue life of rolling bearings can be shown to be closely related to the lubrication.

The rolling fatigue life is expressed by the maximum number of rotations, which a bearing can endure, until the raceway or rolling surface of a bearing develops fatigue in the material, resulting in flaking of the surface, under action of cyclic stress by the bearing. Such flaking begins with either microscopic non-uniform portions (such as non-metallic inclusions, cavities) in the material or with microscopic defect in the material's surface (such as extremely small cracks or surface damage or dents caused by contact between extremely small projections in the raceway or rolling surface). The former flaking is called sub-surface originating flaking while the latter is surface-originating flaking.

The oil film parameter (Λ), which is the ratio between the resultant oil film thickness and surface roughness, expresses whether or not the lubrication state of the rolling contact surface is satisfactory. The effect of the oil film grows with increasing Λ. Namely, when Λ is large (around 3 in general), surface-originating flaking due to contact between extremely small projections in the surface is less likely to occur. If the surface is free from defects (flaw, dent, etc.), the life is determined mainly by sub-surface originating flaking. On the other hand, a decrease in Λ tends to develop surface-originating flaking, resulting in degradation of the bearing's life. This state is shown in Fig. 1.

NSK has performed life experiments with about 370 bearings within the range of Λ=0.3 ~ 3 using different lubricants and bearing materials (● and ▲ in Fig. 2). Fig. 2 shows a summary of the principal experiments selected from among those reported up to now. As is evident, the life decreases rapidly at around Λ≒1 when compared with the life values at around Λ=3 ~ 4 where life changes at a slower rate. The life becomes about 1/10 or less at Λ≦0.5. This is a result of severe surface-originating flaking.

Accordingly, it is advisable for extension of the fatigue life of rolling bearings to increase the oil film parameter (ideally to a value above 3) by improving lubrication conditions.

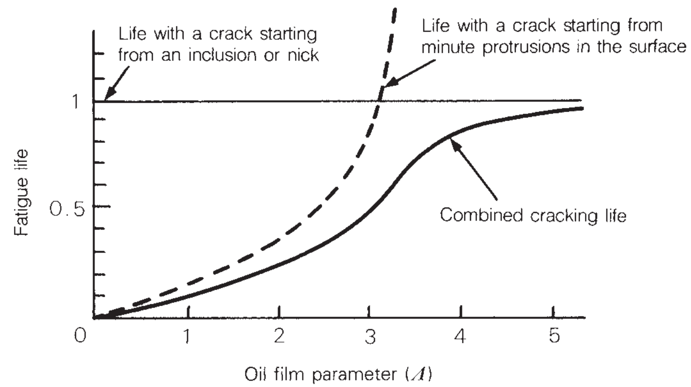


Fig. 1 Expression of life according to Λ (Tallian, et al.)

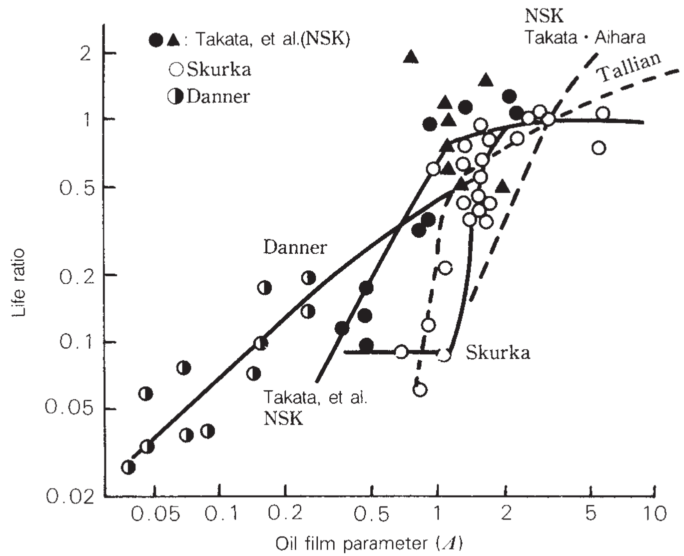


Fig. 2 Typical experiment with Λ and rolling fatigue life
(Expressed with reference to the life at Λ=3)

2.13 EHL oil film parameter calculation diagram

Lubrication of rolling bearings can be expressed by the theory of elastohydrodynamic lubrication (EHL). Introduced below is a method to determine the oil film parameter (oil film − surface roughness ratio), the most critical among the EHL qualities.

2.13.1 Oil film parameter

The raceway surfaces and rolling surfaces of a bearing are extremely smooth, but have fine irregularities when viewed through a microscope. As the EHL oil film thickness is in the same order as the surface roughness, lubricating conditions cannot be discussed without considering this surface roughness. For example, given a particular mean oil film thickness, there are two conditions which may occur depending on the surface roughness. One consists of complete separation of the two surfaces by means of the oil film (Fig. 1 (a)). The other consists of metal contact between surface projections (Fig. 1 (b)). The degradation of lubrication and surface damage is attributed to case (b). The symbol lambda (Λ) represents the ratio between the oil film thickness and roughness. It is widely employed as an oil film parameter in the study and application of EHL.

$$\Lambda = h/\sigma \quad \text{(1)}$$

where h: EHL oil film thickness
σ: Combined roughness ($\sqrt{\sigma_1^2+\sigma_2^2}$)

σ_1, σ_2: Root mean square (rms) roughness of each contacting surface

The oil film parameter may be correlated to the formation of the oil film as shown in Figs. 2 and the degree of lubrication can be divided into three zones as shown in the figure.

2.13.2 Oil film parameter calculation diagram

The Dowson-Higginson minimum oil film thickness equation shown below is used for the diagram:

$$H_{min} = 2.65\frac{G^{0.54}U^{0.7}}{W^{0.13}} \quad \text{(2)}$$

The oil film thickness to be used is that of the inner ring under the maximum rolling element load (at which the thickness becomes minimum).

Equation (2) can be expressed as follows by grouping into terms (R) for speed, (A) for viscosity, (F) for load, and (J) for bearing technical specifications. t is a constant.

$$\Lambda = t \cdot R \cdot A \cdot F \cdot J \quad \text{(3)}$$

R and A may be quantities not dependent on a bearing. When the load P is assumed to be between 98 N {10 kgf} and 98 kN {10 tf}, F changes by 2.54 times as $F \propto P^{-0.13}$. Since the actual load is determined roughly from the bearing size, however, such change may be limited to 20 to 30%. As a result, F is handled as a lump with the term J of bearing specifications [F=F (J)]. Traditional Equation (3) can therefore be grouped as shown below:

$$\Lambda = T \cdot R \cdot A \cdot D \quad \text{(4)}$$

where, T: Factor determined by the bearing Type
R: Factor related to Rotation speed
A: Factor related to viscosity (viscosity grade α: Alpha)
D: Factor related to bearing Dimensions

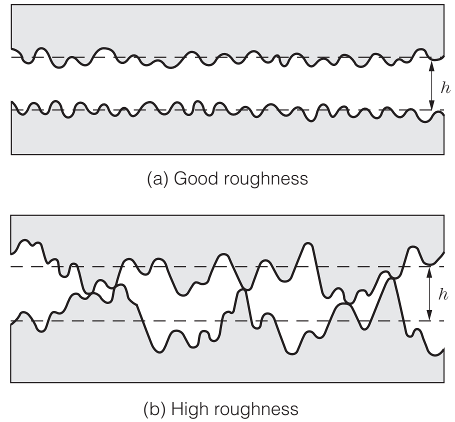


Fig. 1 Oil film and surface roughness

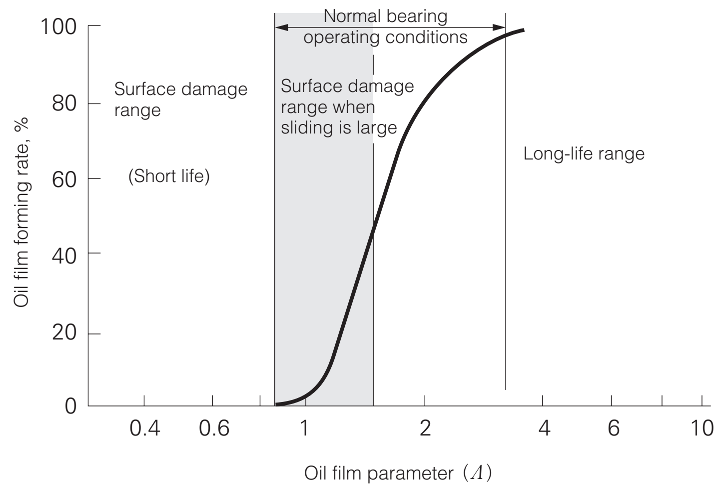


Fig. 2 Effect of oil film on bearing performance

The oil film parameter Λ, which is most vital among quantities related to EHL, is expressed by a simplified equation shown below. The fatigue life of rolling bearings becomes shorter when Λ is smaller.

In the equation $\Lambda = T \cdot R \cdot A \cdot D$ terms include A for oil viscosity η_0 (mPa·s, {cp}), R for the speed n (min^{-1}), and D for bearing bore diameter d (mm). The calculation procedure is described below.

(1) Determine the value T from the bearing type (Table 1).

(2) Determine the R value for n (min^{-1}) from Fig. 3.

(3) Determine A from the absolute viscosity (mPa·s, {cp}) and oil kind in Fig. 4. Generally, the kinematic viscosity ν_0 (mm^2/s, {cSt}) is used and conversion is made as follows:

$$\eta_0 = \rho \cdot \nu_0 \quad \cdots\cdots (5)$$

ρ is the density (g/cm^3) and uses the approximate value as shown below:

Mineral oil ρ=0.85
Silicon oil ρ=1.0
Diester oil ρ=0.9

When it is not known whether the mineral oil is naphthene or paraffin, use the paraffin curve shown in Fig. 4.

(4) Determine the D value from the diameter series and bore diameter d (mm) in Fig. 5.

(5) The product of the above values is used as an oil film parameter.

Table 1 Value T

Bearing type	Value T
Ball bearing	1.5
Cylindrical roller bearing	1.0
Tapered roller bearing	1.1
Spherical roller bearing	0.8

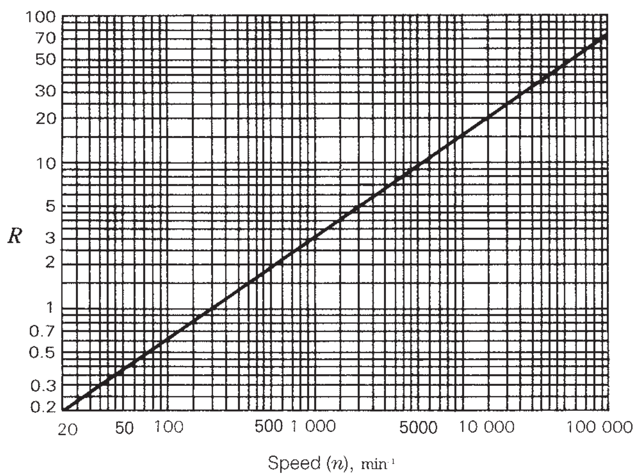


Fig. 3 Speed term, R

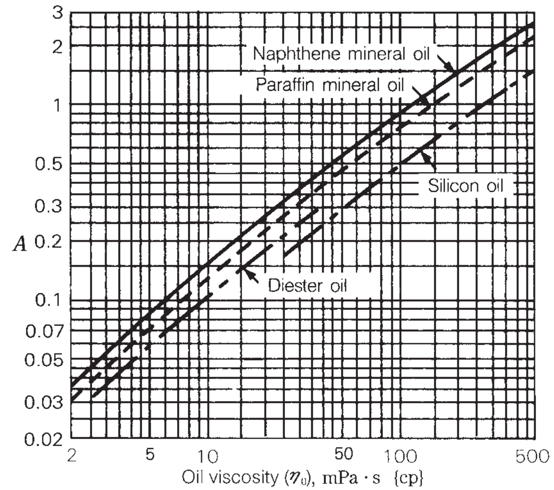


Fig. 4 Term related to lubricant viscosity, A

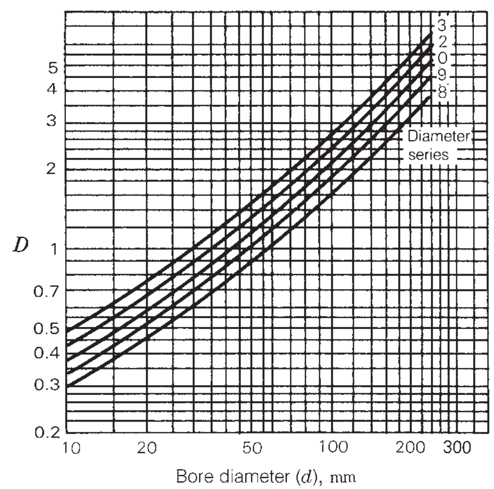


Fig. 5 Term related to bearing specifications, D

Examples of EHL oil film parameter calculation are described below.
(Example 1)
The oil film parameter is determined when a deep groove ball bearing 6312 is operated with paraffin mineral oil (η_0=30 mPa·s, {cp}) at the speed n =1 000 min^{-1}.

(Solution)
d=60 mm and D=130 mm from the bearing catalog.
T=1.5 from Table 1
R=3.0 from Fig. 3
A=0.31 from Fig. 4
D=1.76 from Fig. 5
Accordingly, Λ=2.5

(Example 2)
The oil film parameter is determined when a cylindrical roller bearing NU240 is operated with paraffin mineral oil (η_0=10 mPa·s, {cp}) at the speed n=2 500 min^{-1}.

(Solution)
d=200 mm and D=360 mm from the bearing catalog.
T=1.0 from Table 1
R=5.7 from Fig. 3
A=0.13 from Fig. 4
D=4.8 from Fig. 5
Accordingly, Λ=3.6

2.13.3 Effect of oil shortage and shearing heat generation

The oil film parameter obtained above is the value when the requirements, that is, the contact inlet fully flooded with oil and isothermal inlet are satisfied. However, these conditions may not be satisfied depending on lubrication and operating conditions.

One such condition is called starvation, and the actual oil film paramerer value may become smaller than determined by Equation (4). Starvation might occur if lubrication becomes limited. In this condition, a guideline for adjusting the oil film parameter is 50 to 70% of the value obtained from Equation (4).

Another effect is the localized temperature rise of oil in the contact inlet due to heavy shearing during high-speed operation, resulting in a decrease of the oil viscosity. In this case, the oil film parameter becomes smaller than the isothermal theoretical value. The effect of shearing heat generation was analyzed by Murch and Wilson, who established the decrease factor of the oil film parameter. An approximation using the viscosity and speed (pitch diameter of rolling element set D_{pw} × rotating speed per minute n as parameters) is shown in Fig. 6. By multiplying the oil film parameter determined in the previous section by this decrease factor Hi the oil film parameter considering the shearing heat generation is obtained.
Nameny;

$$\Lambda = Hi \cdot T \cdot R \cdot A \cdot D \quad \cdots\cdots (6)$$

Note that the average of the bore and outside diameters of the bearings may be used as the pitch diameter D_{pw} (d_m) of rolling element set.

Conditions for the calculation (Example 1) include $d_m n$=9.5 × 10^4 and η_0=30 mPa·s, {cp}, and Hi is nearly equivalent to 1 as is evident from Fig. 6. There is therefore almost no effect of shearing heat generation.
Conditions for (Example 2) are $d_m n$=7×10^5 and η_0=10 mPa·s, {cp} while Hi=0.76, which means that the oil film parameter is smaller by about 25%. Accordingly, Λ is actually 2.7, not 3.6.

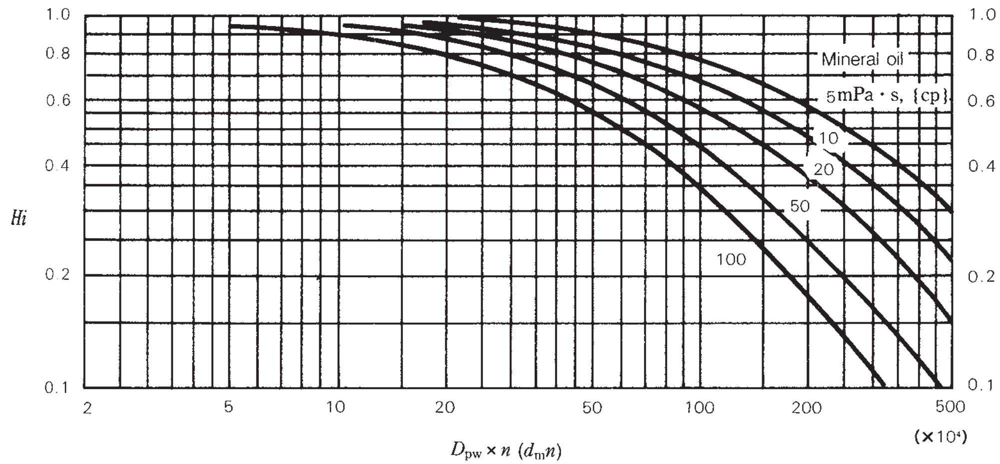


Fig. 6 Oil film thickness decrease factor Hi due to shearing heat generation

2.14 Fatigue analysis

It is necessary for prediction of the fatigue life of rolling bearings and estimation of the residual life to know all fatigue break-down phenomena of bearings. But, it will take some time before we reach a stage enabling prediction and estimation. Rolling fatigue, however, is fatigue proceeding under compressive stress at the contact point and known to develop extremely great material change until breakdown occurs. In many cases, it is possible to estimate the degree of fatigue of bearings by detecting material change. However, this estimation method is not effective in the cases where the defects in the raceway surface cause premature cracking or chemical corrosion occurs on the raceway. In these two cases, flaking grows in advance of the material change.

2.14.1 Measurement of fatigue degree

The progress of fatigue in a bearing can be determined by using an X-ray to measure changes in the residual stress, diffraction half-value width, and retained austenite amount.

These values change as the fatigue progresses as shown in Fig. 1. Residual stress, which grows early and approaches the saturation value, can be used to detect extremely small fatigue. For large fatigue, change of the diffraction half-value width and retained austenite amount may be correlated to the progress of fatigue. These measurements with X-ray are put together into one parameter (fatigue index) to determine the relationship with the endurance test period of a bearing.

Measured values were collected by carrying out endurance test with many ball, tapered roller, and cylindrical roller bearings under various load and lubrication conditions. Simultaneously, measurements were made on bearings used in actual machines.

Fig. 2 summarizes the data. Variance is considerable because data reflects the complexity of the fatigue phenomenon. But, there exists correlation between the fatigue index and the endurance test period or operating hours. If some uncertainty is allowed, the fatigue degree can be handled quantitatively.

Description of "sub-surface fatigue" in Fig. 2 applies to the case when fatigue is governed by internal shearing stress. "Surface fatigue" shows correlation when the surface fatigue occurs earlier and more severely than sub-surface fatigue due to contamination or oil film breakdown of lubricating oil.

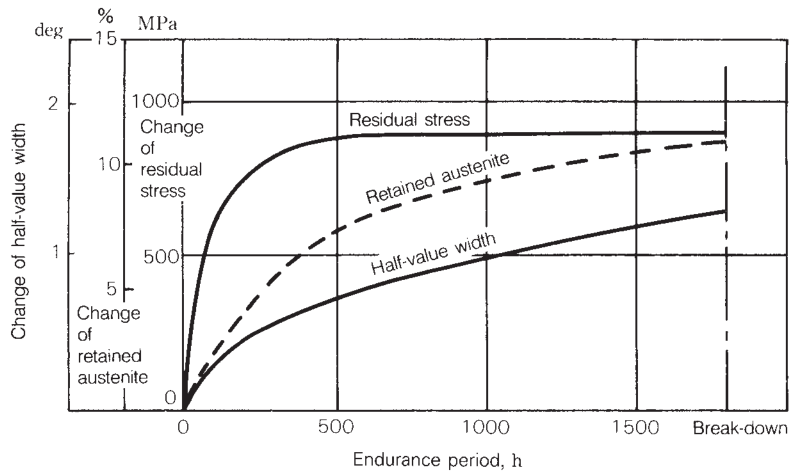


Fig. 1 Change in X-ray measurements

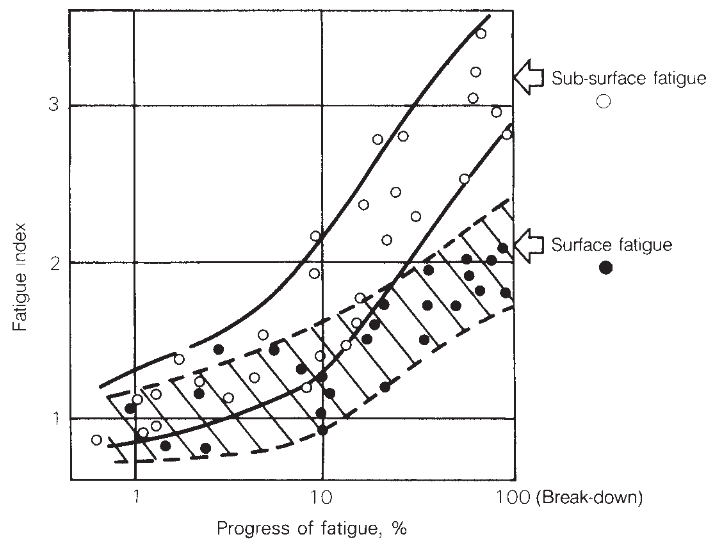


Fig. 2 Fatigue progress and fatigue index

2.14.2 Surface and sub-surface fatigues

Rolling bearings have an extremely smooth finish surface and enjoy relatively satisfactory lubrication conditions. It has been considered that internal shearing stress below the rolling surface governs the failure of a bearing.

Shearing stress caused by rolling contact becomes maximum at a certain depth below the surface, with a crack (which is an origin of break-down) occurring initially under the surface. When the raceway is broken due to such sub-surface fatigue, the fatigue index as measured in the depth direction is known to increase according to the theoretical calculation of shearing stress, as is evident from an example of the ball bearing shown in Fig. 3.

The fatigue pattern shown in Fig. 3 occurs mostly when lubrication conditions are satisfactory and oil film of sufficient thickness is formed in rolling contact points. The basic dynamic load rating described in the bearing catalog is determined using data of bearing failures according to the above internal fatigue pattern.

Fig. 4 shows an example of a cylindrical roller bearing subject to endurance test under lubrication conditions causing unsatisfactory oil film. It is evident that the surface fatigue degree rises much earlier than the calculated life.

In this test, all bearings failed before sub-surface fatigue became apparent. In this way, bearing failure due to surface fatigue is mostly attributed to lubrication conditions such as insufficient oil film due to excessively low oil viscosity or entry of foreign matters or moisture into lubricant.

Needless to say, bearing failure induced by surface fatigue occurs in advance of that by sub-surface fatigue. Bearings in many machines are exposed frequently to danger of initiating such surface fatigue and, in most of the cases, failure by surface fatigue prior to failure due to sub-surface fatigue (which is the original life limit of bearings).

Fatigue analysis of bearings used in actual machines shows not the sub-surface fatigue pattern, but the surface fatigue pattern as shown in the figure in overwhelmingly high percentage.

In this manner, knowing the distribution of the fatigue index in actually used bearings leads to an understanding of effective information not only on residual life of bearings, but also on lubrication and load conditions.

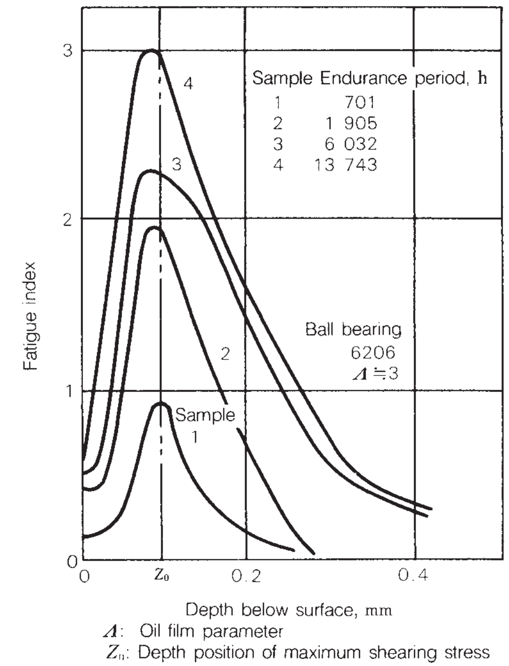


Fig. 3 Progress of sub-surface fatigue

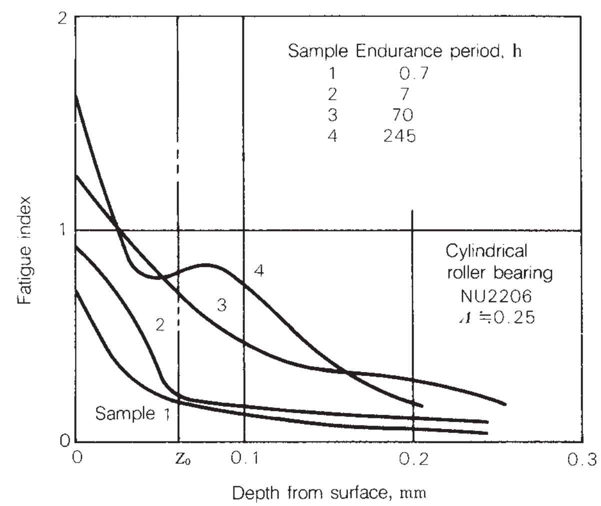


Fig. 4 Progress of surface fatigue

2.14.3 Analysis of practical bearing (1)

Bearings for automotive transmissions must overcome difficult problems of size and weight reduction as well as extension of durable life to meet the ever increasing efforts to conserve energy.

Fig. 5 shows an example of fatigue analysis of bearings used in the transmission of an actual passenger car. Analysis of the transmission bearings of various vehicles reveals the surface fatigue pattern as shown in Fig. 5, but almost no progress of sub-surface fatigue which is used as a criterion for calculating the bearing life.

In other words, these bearings develop less fatigue under external bearing loads but they eventually suffer damage due to fatigue by surface force on the rolling surface though they can be used for a long time.

This may be attributable to indentations caused by inclusions of extremely small foreign matters from the gear oil, which in turn cause excessive fatigue of the surface.

As is evident from Fig. 5, the fatigue index is heaviest in the counter front bearing where the load is most severe, followed by the counter rear bearing where the load is lightest. This surprising fact may be due to the fact that the counter shaft bearings are immersed in gear oil which often has many foreign particles suspend in it. Thus, metal chips in the gear oil eventually contact and abrade the counter shaft bearings.

Fig. 6 shows the result of the durability test and fatigue analysis data with two kinds of bearings used in a transmission of an actual vehicle.

As is known from above the analytical result, a bearing with a special seal (sealed clean bearing), which prevents entry of foreign matter in gear oil while allowing entry of oil only, offers remarkably extended life. In this way, the life is extended by more than ten times when compared with an open type bearing without a seal.

The fatigue pattern shows change in the sub-surface fatigue pattern in the sealed clean bearing, indicating that reduction of surface fatigue contributes greatly to the remarkable life extension.

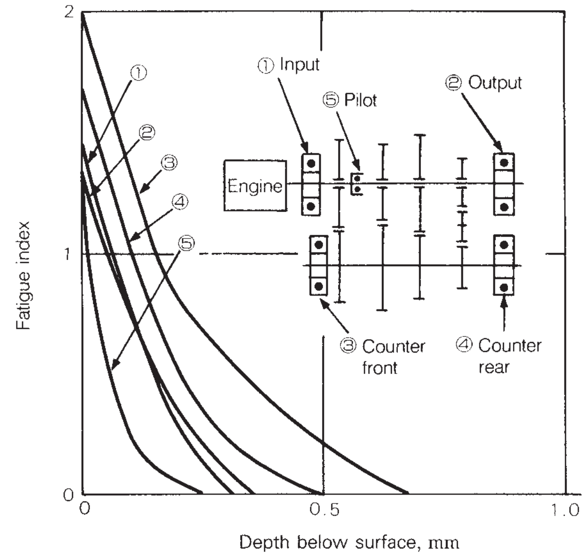


Fig. 5 Fatigue index distribution of transmission bearings (used in actual vehicle)

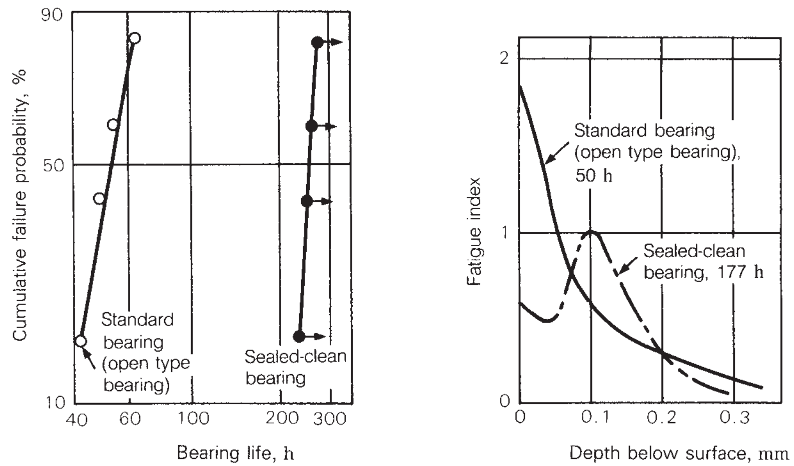


Fig. 6 Comparison between open type bearing and sealed-clean bearing in transmission durability test

2.14.4 Analysis of practical bearing (2)

As shown in the above examples, the cause of fatigue failure can be presumed through measurement of the fatigue index. There are also other applications. Namely, prediction of residual life, prediction of breakdown life by parts (inner and outer rings, rolling elements, etc.), and understanding of surface or sub-surface fatigue are possible. This information can be utilized for improved design. Specifically, this information may be used to reduce the size and weight, to optimize lubrication conditions, expand the applications for sealed clean bearings, and to enhance the load rating. Measurement of the fatigue index begins also to be applied in roller bearings to prevent the edge load of rollers to obtain a more ideal linear contact state. Improvements in the accuracy of fatigue index measurements can lead to better bearing designs.

In the future, prediction of residual life may be used to shorten the durability test period and optimize the interval between replacements.

Fig. 7 shows the measurement of the fatigue index distribution on the raceway surface of a pinion gear shaft incorporating a needle roller bearing. Heavy fatigue is observed on the raceway end nearest to the gear, indicating the necessity of a countermeasure against edge load of the roller.

Fig. 8 shows an example of estimating the durable life on the Weibull chart while interrupting the bearing durability test and predicting the life from the respective measurements of the fatigue index.

Practical examples as above introduced are expected to increase as fatigue analysis technology moves forward.

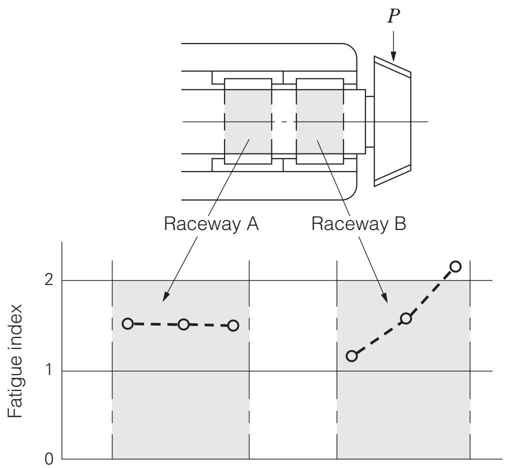


Fig. 7 Distribution of fatigue index in the raceway surface of the pinion shaft

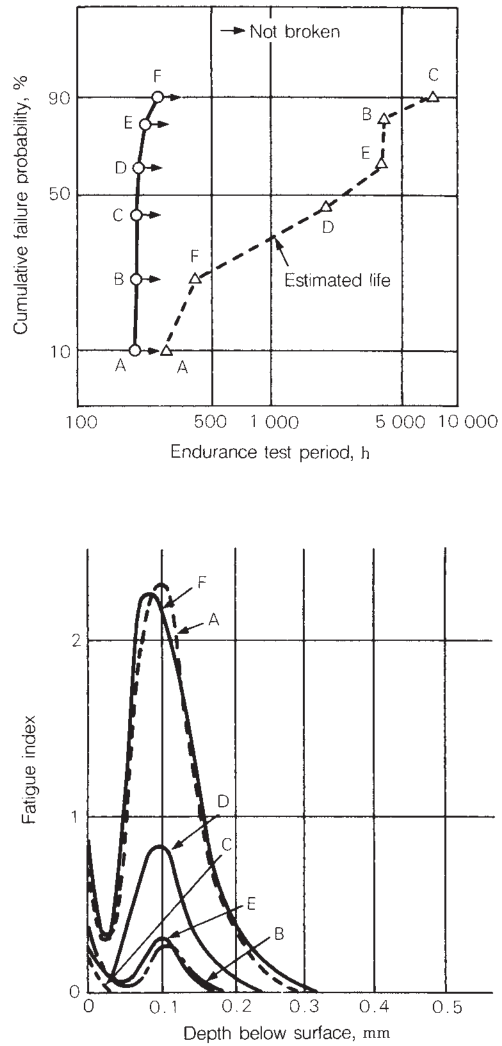


Fig. 8 Estimation of life for bearings whose durability test was interrupted halfway

2.15 Conversion of dynamic load rating with reference to life at 500 min^{-1} and 3 000 hours

The basic dynamic load rating of rolling bearings is the load without fluctuation in the direction and magnitude, at which the rating fatigue life of a group of similar bearings operated with inner rings rotating and outer rings stationary reaches one million revolutions. The standard load rating is 33.3 min^{-1} and 500 hours ($33.3\times500\times60=10^6$). The calculation equation is specified in JIS B 1518. Some other makers, however, are using a calculation equation of dynamic load rating unique and different from ours, and comparison may encounter difficulties due to difference in standards. One of these differences is concerned with the gross number of revolutions.

For example, TIMKEN of the USA determines the dynamic load rating of tapered roller bearing by using the gross number of revolutions for the rotating speed of 500 min^{-1} for 3 000 hours, that is, 500×3 000×60=90 000 000 revolutions, as a standard.
TORRINGTON uses 33.3 min^{-1} and 500 hours, that is, 33.3×500×60=1 000 000 revolutions as a standard as in the case of JIS. Assuming that the dynamic load rating calculation equation is basically similar between both companies, except for the gross number of revolutions standard, the difference in the gross number of revolutions may be converted as follows in terms of dynamic load rating:

$$L_T=\left(\frac{C_T}{P_T}\right)^p\times n_T \quad \cdots\cdots (1)$$

$$L_R=\left(\frac{C_R}{P_R}\right)^p\times n_R \quad \cdots\cdots (2)$$

where, L: Rating fatigue life as expressed by gross number of revolutions
C: Basic dynamic load rating (N), {kgf}
P: Load (N), {kgf}
p: Power
n: Standard of gross number of revolutions
A suffix "T" stands for TIMKEN while "R" for TORRINGTON.

Assuming that the internal specification of a bearing is completely similar between both companies, setting loads $P_T=P_R$ lead to the following equation from Equations (1) and (2):

$$\frac{L_T}{L_R}=\frac{\left(\frac{C_T}{P_T}\right)^p\times n_T}{\left(\frac{C_R}{P_R}\right)^p\times n_R}=1 \quad \cdots\cdots (3)$$

$$C_R{}^p=\frac{n_T}{n_R}C_T{}^p \quad \cdots\cdots (4)$$

Set n_T=90 000 000, n_R=1 000 000, and power $p=\frac{10}{3}$ (applicable to roller bearings) in Equation (4):

$$C_R=\left(\frac{n_T}{n_R}\right)^{\frac{1}{p}}C_T$$

$$=\left(\frac{90\ 000\ 000}{1\ 000\ 000}\right)^{\frac{3}{10}}C_T$$

$$=90^{\frac{3}{10}}C_T \quad \cdots\cdots (5)$$

Namely, the value 3.857 times the dynamic load rating C_T of TIMKEN is equivalent to C_R of TORRINGTON. Actually, however, the internal specification is not necessarily the same because every bearing maker undertakes design and manufacture from its unique viewpoint. In case of a difference in the units (SI unit and pound), simple conversion is enough.

Relationship among Equations (3) to (5) can be established only when the dynamic load rating calculation equation is basically the same as described.
If it is evident that the equation is based on different standards, comparison or conversion using apparent numerical figures should be considered only as reference. Reasonable judgement is possible only by performing recalculation according to a similar calculation method.

2.16 Basic static load ratings and static equivalent loads

(1) Basic static load rating

When subjected to an excessive load or a strong shock load, rolling bearings undergo a local permanent deformation of the rolling elements and raceway surface if the elastic limit is exceeded. The nonelastic deformation increases in both area and depth as the load increases, and when the load exceeds a certain limit, the smooth running of the bearing is impeded.

The basic static load rating is defined as that static load which produces the following calculated contact stress at the center of the contact area between the rolling element subjected to the maximum stress and the raceway surface.

For self-aligning ball bearings
 4 600 MPa {469 kgf/mm^2}
For other ball bearings
 4 200 MPa {428 kgf/mm^2}
For roller bearings
 4 000 MPa {408 kgf/mm^2}

In this most heavily stressed contact area, the sum of the permanent deformation of the rolling element and that of the raceway is nearly 0.0001 times the rolling element's diameter. The basic static load rating C_0 is written C_{0r} for radial bearings and C_{0a} for thrust bearings in the bearing tables.

In addition, following the modification of the criteria for the basic static load rating by ISO, the new C_0 values for NSK's ball bearings became about 0.8 to 1.3 times the past values and those for roller bearings about 1.5 to 1.9 times. Consequently, the values of permissible static load factor f_s have also changed, so please pay attention to this.

In the above description, the static load rating is not the load for failure (crack) of rolling element and bearing ring. Since the load necessary to crush a rolling element is more than seven times the static load rating, this is a sufficient safety factor against failure load when considering general machine equipment.

(2) Static equivalent load

The static equivalent load must be considered for radial bearings exposed to synthetic loads or axial loads only and for thrust bearings exposed to axial loads and slight radial loads.

The static equivalent load is a hypothetical load of a magnitude causing a contact stress (equivalent to the maximum contact stress occurring under actual load conditions) in the contact between the rolling element and raceway under maximum load when the bearing is stationary (including extremely low speed rotation and low-speed oscillation). This equivalent load is the radial load acting through a bearing center for radial bearings and the axial load in a direction aligned to the central axis in the case of thrust bearings.

(a) Static equivalent load of radial bearings
The static equivalent load of radial bearings is taken as the larger value of the two values obtained from the two equations below:

$$P_0=XF_r+Y_0F_a \quad \cdots\cdots (1)$$

$$P_0=F_r \quad \cdots\cdots (2)$$

where, P_0: Static equivalent load (N), {kgf}
 F_r: Radial load (N), {kgf}
 F_a: Axial load (N), {kgf}
 X_0: Static radial load factor
 Y_0: Static axial load factor

(b) Static equivalent load of thrust bearings

$$P_0=X_0F_r+F_a \quad \alpha\neq 90° \quad \cdots\cdots (3)$$

where, P_0: Static equivalent load (N), {kgf}
 α : Nominal contact angle

Note that the accuracy of this equation decreases when $F_a<X_0F_r$.
Values X_0 and Y_0 of Equations (1) and (3) are as shown in Table 2.
Note that $P_0=F_a$ for a thrust bearing with α=90°

(3) Static allowable load factor

The static equivalent load allowed for bearings varies depending on the basic static load rating, requirements of the bearings and bearing operating conditions.

The allowable static load factor f_s for review of the safety factor against the basic static load rating is determined from Equation (4). Generally recommended values of f_s are shown in Table 1.

Due care must be taken during application because the value of f_s for roller bearings (particularly, with the large C_0 value) has been changed along with the change of the static load rating.

$$f_s=\frac{C_0}{P_0} \quad \cdots\cdots (4)$$

where, C_0: Basic static load rating (N), {kgf}
 P_0: Static equivalent load (N), {kgf}

Normally, $f_s \geqq 4$ applies to spherical thrust roller bearings.

Table 1 Value of permissible static load factor, f_s

Running conditions	Lower limit of f_s	
	Ball bearings	Roller bearings
Low-noise applications	2	3
Vibration and shock loads	1.5	2
Standard running conditions	1	1.5

Table 2 Static epuivalent load

Bearing type		Single row		Double row	
		X_0	Y_0	X_0	Y_0
Deep groove ball bearings		0.6	0.5	0.6	0.5
Angular contact ball bearings	$\alpha=15°$	0.5	0.46	1	0.92
	$\alpha=20°$	0.5	0.42	1	0.84
	$\alpha=25°$	0.5	0.38	1	0.76
	$\alpha=30°$	0.5	0.33	1	0.66
	$\alpha=35°$	0.5	0.29	1	0.58
	$\alpha=40°$	0.5	0.26	1	0.52
	$\alpha=45°$	0.5	0.22	1	0.44
Self-aligning ball bearings Tapered roller bearings Spherical roller bearings	$\alpha\neq 0$ $\alpha\neq 0$ $\alpha\neq 0$	0.5	0.22 cot α	1	0.44 cot α
Cylindrical roller bearings	$\alpha=0$	$P_0=F_r$			
Thrust ball bearings Thrust roller bearings	$\alpha=90°$ $\alpha=90°$	$P_{0a}=F_a$			
Thrust ball bearings Thrust roller bearings	$\alpha\neq 90°$ $\alpha\neq 90°$	$P_{0a}=F_a+2.3F_r\tan\alpha$ (where, $F_a>2.3F_r\tan\alpha$)			