4 Rating Life of Bearings

When ball bearing units are installed and operated on a piece of machinery eventually a failure will occur. The period of operation until the unit cannot be used due to failure is called the bearing life.

Bearing failure is caused by two main reasons. The first is fatigue of bearing material, and the second is lubricant degradation. The life is figured on whichever fails first.

Proper bearing lubrication will eliminate grease degradation and allow full bearing life to be achieved. If the bearing units are run without replenishment of the grease the bearing life will have to be factored by either the grease life or the bearing life. During installation, care must be taken not to damage the bearing. Proper bearing maintenance and lubrication will ensure long bearing life.

4.1 Basic Rating Life and Basic Load Rating

4.1.1 Basic rating life

When a bearing is rotated under load the raceways and the rolling elements are continuously exposed to load. Damage, such as scaling (flaking or peeling), eventually appears on the material, and the total rotating frequency until the damage appears is called the "fatigue limit of the bearing". Fatigue limit of the bearing can vary greatly even if the bearings have the same structure, dimensions, materials, machining methods, and are operated under the same conditions.

To account for this variation, a group of the same bearings operating under the same conditions are tested, and the total rotating frequency of 90% of the bearings operating with no damage due to rotating fatigue (90% reliability) is called the basic load rating.

4.1.2 Basic load rating

Dynamic ratings are determined by placing a pure radial load on a radial bearing or by placing a central axial load on a thrust bearing. The dynamic rating is the load that the bearing will withstand for one million cycles before failure of the bearing.

These ratings are referred to as the **basic dynamic radial load rating** (C_r) or the **basic dynamic axial load rating** (C_a). These values are indicated in the catalog as the basic dynamic radial load rating (C_r), and the value is shown in the dimensional table.

4.2 Calculation of Rating Life

The relationship between the basic rating life, the basic dynamic load rating, and the dynamic equivalent load of the ball bearing is indicated in **Formula (4.1)**. If the ball bearing unit is being used at a fixed rotating speed, the life is indicated as time. This is shown in **Formula (4.2)**.

(Total rotating frequency) $\quad L_{10} = \left(\dfrac{C_r}{P_r}\right)^3$ .... (4.1)

(Time) $\quad L_{10h} = \dfrac{10^6}{60n}\left(\dfrac{C_r}{P_r}\right)^3$ .... (4.2)

Whereas,

L_{10}: Basic rating life, 10^6 rotations

L_{10h}: Basic rating life, hr

C_r: Basic dynamic load rating, N

P_r: Dynamic equivalent load, N
(see **"5 Bearing load"**)

n: Rotating speed, min^{-1}

Calculation of the basic rating life using the life factor (f_h) and the speed factor (f_n) in **Formula (4.2)** are shown below.

$L_{10h} = 500\ f_h{}^3$ .... (4.3)

Life factor $\quad f_h = f_n \cdot \dfrac{C_r}{P_r}$ .... (4.4)

Speed factor $\quad f_n = \left(\dfrac{10^6}{500 \times 60n}\right)^{1/3}$

$\qquad\qquad = (0.03n)^{-1/3}$ .... (4.5)

Values of f_n, f_h and L_{10h} can be found using the nomogram of **Fig. 4.1**.
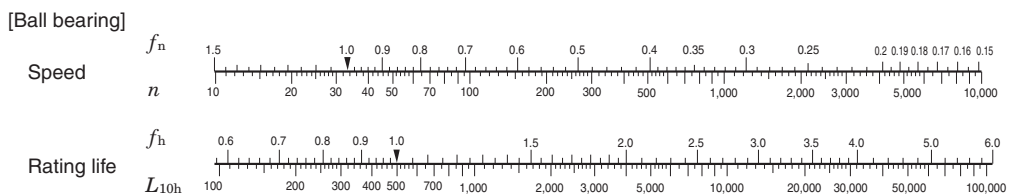
[Ball bearing]

Fig. 4.1   Relation between basic rating life (L_{10h}) and rotating speed (n), speed factor (f_n), and life factor (f_h)

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4 Rating Life of Bearings

4.2.1 Correction of basic load rating due to temperature

If a ball bearing unit is used at a relatively high tempera-ture the physical composition of the bearing material is changed leading to decreased hardness. This decreased hardness leads to the basic dynamic load rating being reduced. Once the structure of the bearing material has been changed, it will remain this way for the life of the unit, even when it returns to room temperature.

When using a ball bearing unit at 150 ºC or more, the basic load rating must be corrected by multiplying the basic dynamic load rating shown in the dimensional table by the temperature factor shown in **Table 4.1**.

Table 4.1   Temperature factor

Bearing temperature, ºC	125	150	175	200	250
Temperature factor	1	1	0.95	0.9	0.75

4.2.2 Corrected rating life

The basic L_{10} rating life shown in **Formula (4.1)** is the fatigue life of a bearing with 90% reliability however; there are circumstances where bearings need greater reliability. The bearing life may be extended by using special materi-als.

Other conditions, including lubrication, may influence the bearing life.

The corrected rating life is found by taking the basic rating life and taking the factors in **Formula (4.6)** into consideration.

$$L_{na} = a_1 a_2 a_3 L_{10} \quad\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \text{ (4.6)}$$

Whereas,

L_{na}: Corrected rating life, 10^6 rotations

Bearing characteristics and operating condi-tions are taken into consideration with reli-ability $100-n\%$ (breakage probability)

L_{10}: Basic load rating, 10^6 rotations

Life with 90% reliability

a_1: Reliability factor .... see (1)

a_2: Bearing characterization factor .... see (2)

a_3: Operating condition factor .... see (3)

(1) Reliability factor a_1

Table 4.2 shows the values used when a corrected bearing life that has less than a 10% breakage probabil-ity is necessary.

Table 4.2   Reliability factor a_1

Reliability, %	L_{na}	a_1
90	L_{10a}	1
95	L_{5a}	0.62
96	L_{4a}	0.53
97	L_{3a}	0.44
98	L_{2a}	0.33
99	L_{1a}	0.21

(2) Bearing characterization factor a_2

The material make-up of a bearing can have an affect on its basic rating life. Factors that can influence the bearing include bearing material (type of steel), produc-tion procedures, and bearing design. Bearing character-ization is shown as factor a_2.

FYH ball bearing inserts use high quality vacuum degassed bearing steel as standard material, and this material allows for a longer rating life. For FYH ball bearing units, the bearing characterization factor a_2 is 1 ($a_2 = 1$). When bearings with special materials are used for a longer fatigue limit this characterization factor can be shown as a_2 being greater than 1 ($a_2 > 1$).

(3) Operating condition factor a_3

Operating conditions may directly influence the life of the bearing (especially proper or improper lubrication). The basic rating life should be corrected using the oper-ating condition factor a_3. If lubrication is being maintained the factor $a_3 = 1$. If excellent re-lubrication practices are being maintained the factor $a_3 > 1$ should be applied.

If any of the following operating conditions are appli-cable the condition should be applied as $a_3 < 1$.
(1) Kinematic viscosity of lubricant during operation is low:
   Ball bearing: 13 mm^2/s or less,
   Roller bearing: 20 mm^2/s or less
(2) Rotating speed is low:
   $d_m n$: 10,000 or less
Note: d_m (Pitch dia. of ball set in mm) $\times n$ (Rotating speed)
(3) Foreign matters are mixed in lubricant

Even if the bearing characterization factor is improved i.e., $a_2 > 1$, the life of the bearing must still be down-rated if the combination, $a_2 \times a_3 > 1$.

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4.2.3 Required lifetime of bearings

At some point, the economical nature of a ball bearing begins to decline. The operating conditions, type of bearing used, and type of machine the bearing is used on all influence the operational life of the bearing.

The required lifetime of the ball bearing is shown in **Table 4.3**.

Table 4.3 Required life time of ball bearing units (reference)

Operating conditions	Machines used	Required life time, hrs
Operated in short periods or intermittently	Home electric appliances, electric tools, agricultural machinery, hoist, etc.	4,000 – 8,000
Operated for several minutes or hours at a time, but less than 8 hours per day	Factory motor, ordinary gearing, etc.	12,000 – 20,000
Constantly operated for 8 hours or longer per day or operated continuously for long periods	General machinery, blowers, etc.	20,000 – 30,000
Operated continuously for 24 hours, no fault is allowed	Power plants, mine drainage facility, etc.	100,000 –200,000

4.3 Grease Life

The grease life for ball bearing units is influenced by: the level of the load, rotating speed of the bearing, and the operating temperature.

The grease life for ball bearing units being used under appropriate operating conditions can be found by the formula shown below.

$$\log L = 6.10 - 4.40 \times 10^{-6}\, d_{\mathrm{m}}n - 2.50 \left(\frac{P_{\mathrm{r}}}{C_{\mathrm{r}}} - 0.05 \right)$$
$$- (0.021 - 1.80 \times 10^{-8}\, d_{\mathrm{m}}n)\, T \ \cdots\cdots\cdots\cdots (4.7)$$

Whereas,

L: Grease life, hr

d_{m}: Pitch dia. of ball set, mm

$$d_{\mathrm{m}} = \frac{(D + d)}{2}$$

$\left\{ \begin{array}{l} D\text{: Nominal bearing outer dia.,} \\ d\text{: Nominal bearing bore dia.} \end{array} \right\}$

n: Rotating speed of bearing, min^{-1}

P_{r}: Dynamic equivalent radial load, N (see **"5 Bearing load"**)

C_{r}: Basic dynamic radial load rating of bearing, N

T: Operating temperature of bearing, ºC

Applicable conditions for the **Formula (4.7)** are shown below.

1) Operating temperature of bearing: T ˚C

To be applied if the following condition is satisfied: $T \le 100$

$\left\{ \begin{array}{l} \text{If } T \text{ is smaller than 50 } (T < 50), \\ \text{following condition should be applied: } T = 50. \end{array} \right\}$

If T is larger than 100 ($T > 100$), contact FYH.

2) Rotating speed of bearing: $d_{\mathrm{m}}n$

To be applied if the following condition is satisfied: $d_{\mathrm{m}}n \le 30 \times 10^4$

$\left\{ \begin{array}{l} \text{If } d_{\mathrm{m}}n \text{ is smaller than } 12.5 \times 10^4 \ (d_{\mathrm{m}}n < 12.5 \times 10^4), \\ \text{following condition should be applied:} \\ d_{\mathrm{m}}n = 12.5 \times 10^4 \end{array} \right\}$

If $d_{\mathrm{m}}n$ is larger than 30×10^4 ($d_{\mathrm{m}}n > 30 \times 10^4$), contact FYH.

3) Load condition of bearing: $\dfrac{P_{\mathrm{r}}}{C_{\mathrm{r}}}$

To be applied if the following condition is satisfied: $\dfrac{P_{\mathrm{r}}}{C_{\mathrm{r}}} \le 0.2$

$\left\{ \begin{array}{l} \text{If } \frac{P_{\mathrm{r}}}{C_{\mathrm{r}}} \text{ is smaller than } 0.05 \ (\frac{P_{\mathrm{r}}}{C_{\mathrm{r}}} < 0.05), \\ \text{following condition should be applied: } \frac{P_{\mathrm{r}}}{C_{\mathrm{r}}} = 0.05 \end{array} \right\}$

If $\dfrac{P_{\mathrm{r}}}{C_{\mathrm{r}}}$ is larger than 0.2 ($\frac{P_{\mathrm{r}}}{C_{\mathrm{r}}} > 0.2$), contact FYH.
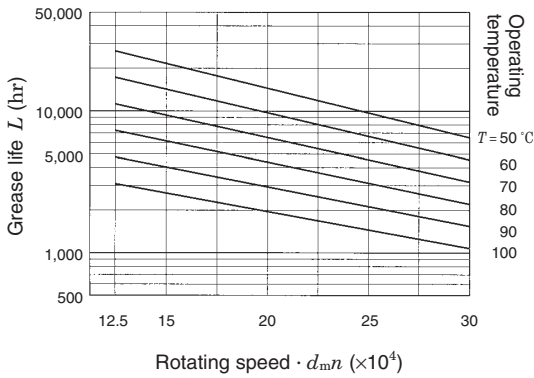
Reference figure of grease life obtained by the **Formula (4.7)** is shown in **Fig. 4.2**.

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4 Rating Life of Bearings

(1) Bearing load $\cdot \dfrac{P_r}{C_r} = 0.05$
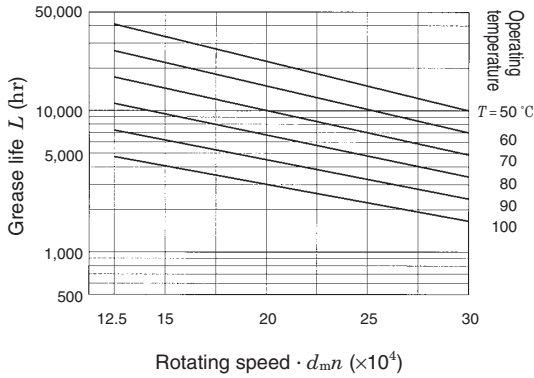
(2) Bearing load $\cdot \dfrac{P_r}{C_r} = 0.125$

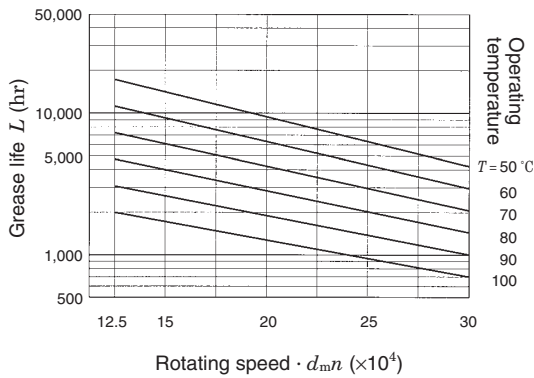(3) Bearing load $\cdot \dfrac{P_r}{C_r} = 0.2$

Fig. 4.2 Relation of grease life to bearing load, rotating speed, and operating temperature (reference)

5 Bearing Load

Loads that are applied to bearings come from a variety of sources.

In addition to the primary load, other resultant loads include the weight of complementary objects including shafting, gears, pulleys, torsion from chain and belts, and so on. Shock or dynamic load can also be derived from these sources.

In many cases, these loads cannot be determined by a simple or single calculation; and since these loads often fluctuate in intensity, it is difficult to determine the exact magnitude of them prior to actual lab or field measurements on the machinery in question.

However, in order to approximate the loads involved prior to putting a machine into operation, the technique shown below should be used. This technique uses multiplication factors that have been determined empirically from sample measurements taken on actual machines in operation.

5.1 Loads Applied to Bearings

5.1.1 Load factor

Even if the static radial load and the axial load can be accurately calculated, the actual loads are generally greater than the calculated figures. This is due to the presence of vibration and shock load during actual machine operation.

To find the loads actually applied to a bearing, multiply the values determined for the static load by the following load factors.

$$F = f_w \cdot F_c \quad \text{...} \quad (5.1)$$

Whereas,

F: Load actually applied to bearing, N

F_c: Theoretically calculated load, N

f_w: Load factor (see **Table 5.1**)

Table 5.1 Load factor f_w

Operating conditions	Applications	f_w
Virtually no vibration or impact	Electric machines and instruments	1 −1.2
Ordinary operation (light impact)	Agricultural machines and blower	1.2–2
Great vibration and impact	Construction machines and grinders	2 −3

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5.1.2 Loads from belts or chain drives

The load calculated for the bearing is equal to the tensile load of the belt. However, this load must be multiplied by the load factor (f_w), which accounts for vibration and impact of the machine and a belt factor (f_b), which accounts for the vibration and impact generated through the belt.

When calculating loads for a chain drive, use the same factor (f_b) as used for belt drives.

$$F_b = \frac{2M}{D_p} \cdot f_w \cdot f_b$$

$$= \frac{19.1 \times 10^6\, W}{D_p \cdot n} \cdot f_w \cdot f_b \quad \text{.................................. (5.2)}$$

Whereas,

F_b: Load actually applied to pulley shaft or sprocket shaft, N

M: Torque applied to pulley or sprocket, mN · m

W: Transmitted power, kW

D_p: Pitch circle dia. of pulley or sprocket, mm

n: Rotating speed, min^{-1}

f_w: Load factor (see **Table 5.1**)

f_b: Belt factor (see **Table 5.2**)

Table 5.2   Belt factor f_b

Belt type	f_b
Toothed belt	1.3–2
V belt	2 –2.5
Flat belt (with tension pulley)	2.5–3
Flat belt	4 –5
Chain	1.2–1.5

5.1.3   Load of gear transmissions

Gear transmissions have a load in the tangential direction (K_t), a load in the radial direction (K_r), and an axial load (K_a). Different types of gears are calculated differently.

The following is a sample of a calculation for an ordinary spur gear arrangement. A flat spur gear will not support an axial load.

(1) Load applied to gear in tangential direction (tangential line force)

$$K_t = \frac{2\,M}{D_p} = \frac{19.1 \times 10^6\, W}{D_p n} \quad \text{.............................. (5.3)}$$

(2) Load applied to gear in radius direction (separating force)

$$K_r = K_t \tan \alpha \quad \text{... (5.4)}$$

(3) Synthetic load applied to gear

$$K_g = \sqrt{K_t{}^2 + K_r{}^2} = K_t \sec\alpha \quad \text{................................. (5.5)}$$

Whereas,

K_t: Load applied to gear in tangential direction (tangential line force), N

K_r: Load applied to gear in radius direction (separating force), N

K_g: Synthetic load applied to gear, N

M: Torque applied to gear, mN · m

D_p: Pitch circle dia. of gear, mm

W: Transmission power, kW

n: Rotating speed, min^{-1}

α: Pressure angle of gear, °

Note that the actual gear load must be found by multiplying the theoretical load by the load factor (f_w) obtained by taking into consideration the vibration and impact loads generated while the machine is in operation. The gear factor (f_g) is determined by taking into consideration the accuracy of machining and the finish of the gears.

$$F_g = f_w \cdot f_g \cdot K_g \quad \text{... (5.6)}$$

Whereas,

F_g: Load actually applied to gear, N

K_g: Theoretical synthetic load applied to gear, N

f_w: Load factor (see **Table 5.1**)

f_g: Gear factor (see **Table 5.3**)

Table 5.3   Gear factor f_g

Gear type	f_g
Precision gear (both pitch error and tooth profile error should be 0.02 mm or less)	1 –1.1
Ordinary gear (both pitch error and tooth profile error should be 0.1 mm or less)	1.1–1.3

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