7. Starting and running torques

7.1 Preload and starting torque for angular contact ball bearings

Angular contact ball bearings, like tapered roller bearings, are most often used in pairs rather than alone or in other multiple bearing sets. Back-to-back and face-to-face bearing sets can be preloaded to adjust bearing rigidity. Extra light (EL), Light (L), Medium (M), and Heavy (H) are standard preloads. Friction torque for the bearing will increase in direct proportion to the preload.

The starting torque of angular contact ball bearings is mainly the torque caused by angular slippage between the balls and contact surfaces on the inner and outer rings. Starting torque for the bearing M due to such spin is given by,

$$M = M_s \cdot Z \sin\alpha \ (\mathrm{mN \cdot m}),\ \{\mathrm{kgf \cdot mm}\} \quad \cdots\cdots\cdots\cdots\cdots\ (1)$$

where, M_s: Spin friction for contact angle α centered on the shaft,

$$M_s = \frac{3}{8} \mu_s \cdot Q \cdot a\ E(k) \quad (\mathrm{mN \cdot m}),\ \{\mathrm{kgf \cdot mm}\}$$

μ_s: Contact-surface slip friction coefficient

Q: Load on rolling elements (N), {kgf}

a: (1/2) of contact-ellipse major axis (mm)

$E(k)$: With $k = \sqrt{1 - \left(\frac{b}{a}\right)^2}$ as the population parameter, second class complete ellipsoidal integration

b: (1/2) of contact-ellipse minor axis (mm)

Z: Number of balls

α: Contact angle (°)

Actual measurements with 15° angular contact ball bearings correlate well with calculated results using μ_s=0.15 in Equation (1). Fig. 1 shows the calculated friction torque for 70C and 72C series bearings.

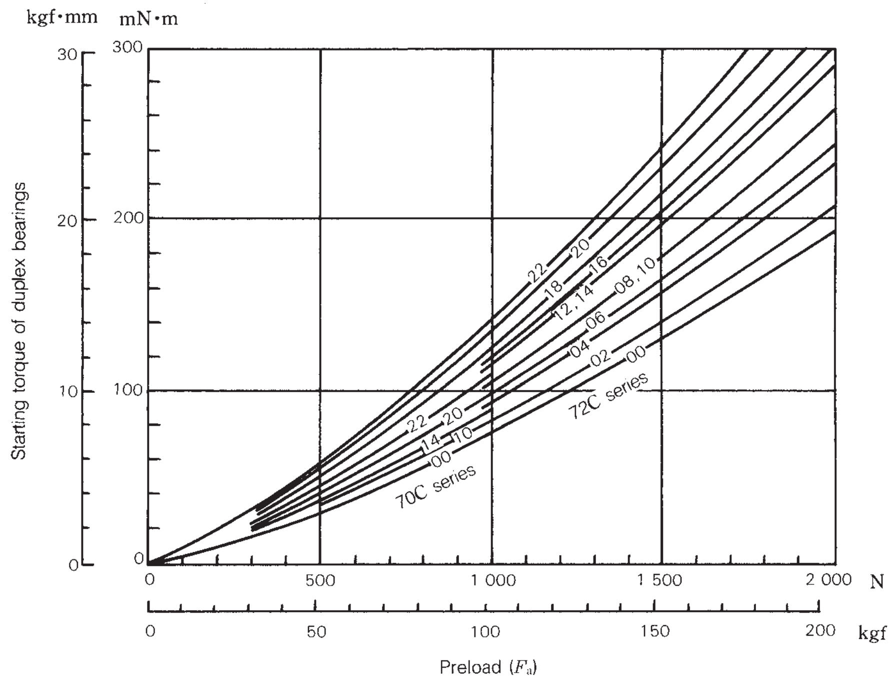

Fig. 1 Preload and starting torque for angular contact ball bearings (α=15°) of DF and DB duplex sets

7.2 Preload and starting torque for tapered roller bearings

The balance of loads on the bearing rollers when a tapered roller bearing is subjected to axial load F_a is expressed by the following three Equations (1), (2), and (3):

$$Q_e=\frac{F_a}{Z\sin\alpha} \quad \cdots\cdots (1)$$

$$Q_i=Q_e\cos 2\beta=\frac{\cos 2\beta}{Z\sin\alpha}F_a \quad \cdots\cdots (2)$$

$$Q_f=Q_e\sin 2\beta=\frac{\sin 2\beta}{Z\sin\alpha}F_a \quad \cdots\cdots (3)$$

where, Q_e: Rolling element load on outer ring (N), {kgf}
Q_i: Rolling element load on inner ring (N), {kgf}
Q_f: Rolling element load on inner-ring large end rib, (N), {kgf} (assume $Q_f \perp Q_i$)
Z: Number of rollers
α: Contact angle...(1/2) of the cup angle (°)
β: (1/2) of tapered roller angle (°)
D_{w1}: Roller large-end diameter (mm) (Fig. 1)
e: Contact point between roller end and rib (Fig. 1)

As represented in Fig. 1, when circumferential load F is applied to the bearing outer ring and the roller turns in the direction of the applied load, the starting torque for contact point C relative to instantaneous center A becomes $e\ \mu_e Q_f$.
Therefore, the balance of frictional torque is,

$$D_{w1}F=e\ \mu_e Q_f \ (\mathrm{mN\cdot m}),\ \{\mathrm{kgf\cdot mm}\} \quad \cdots\cdots (4)$$

where, μ_e: Friction coefficient between inner ring large rib and roller endface

The starting torque M for one bearing is given by,

$$M=F\ Z\ l$$

$$=\frac{e\ \mu_e\, l\sin 2\beta}{D_{w1}\sin\alpha}F_a$$

$$(\mathrm{mN\cdot m}),\ \{\mathrm{kgf\cdot mm}\} \quad \cdots\cdots (5)$$

because, $D_{w1}=2\ \overline{\mathrm{OB}}\sin\beta$, and $l=\overline{\mathrm{OB}}\sin\alpha$.
If we substitute these into Equation (5) we obtain,

$$M=e\ \mu_e\ \cos\beta\ F_a\ (\mathrm{mN\cdot m}),\ \{\mathrm{kgf\cdot mm}\} \quad \cdots\cdots (6)$$

The starting torque M is sought considering only the slip friction between the roller end and the inner-ring large-end rib. However, when the load on a tapered roller bearing reaches or exceeds a certain level (around the preload) the slip friction in the space between the roller end and inner-ring large end rib becomes the decisive factor for bearing starting torque. The torque caused by other factors can be ignored. Values for e and β in Equation (5) are determined by the bearing design. Consequently, assuming a value for μ_e, the starting torque can be calculated.

The values for μ_e and for e have to be thought of as a dispersion, thus, even for bearings with the same number, the individual starting torques can be quite diverse. When using a value for e determined by the bearing design, the average value for the bearing starting torque can be estimated using μ_e=0.20 which is the average value determined from various test results.

Fig. 2 shows the results of calculations for various tapered roller bearing series.

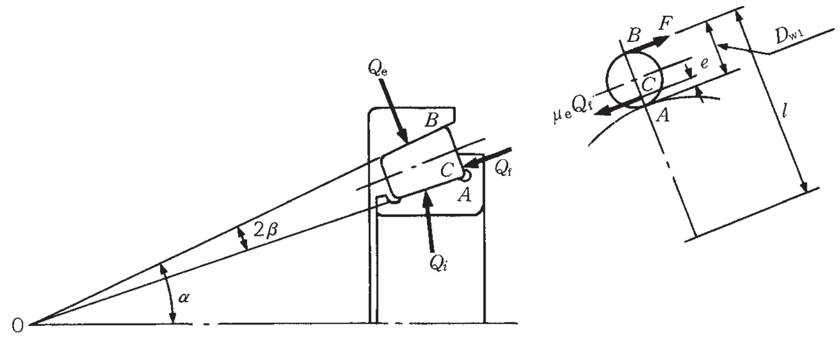

Fig. 1

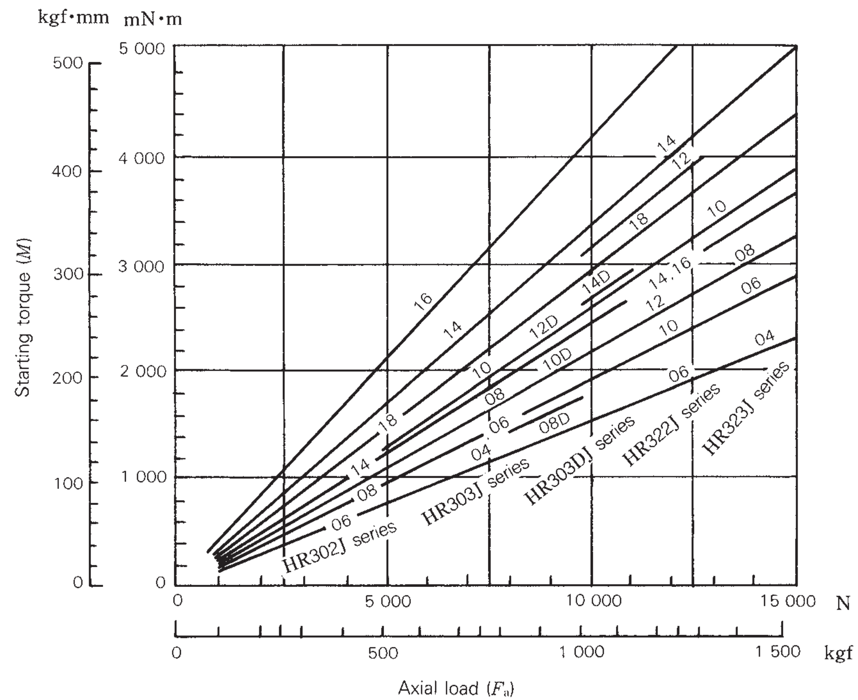

Fig. 2 Axial load and starting torque for tapered roller bearings

7.3 Empirical equation of running torque of high-speed ball bearings

We present here empirical equations for the running torque of high speed ball bearings subject to axial loading and jet lubrication. These equations are based on the results of tests of angular contact ball bearings with bore diameters of 10 to 30 mm, but they can be extrapolated to bigger bearings.

The running torque M can be obtained as the sum of a load term M_l and speed term M_v as follows:

$$M = M_l + M_v \ (\mathrm{mN \cdot m}),\ \{\mathrm{kgf \cdot mm}\} \quad \cdots\cdots (1)$$

The load term M_l is the term for friction, which has no relation with speed or fluid friction, and is expressed by Equation (2) which is based on experiments.

$$\left.\begin{aligned} M_l &= 0.672\times10^{-3} D_{pw}^{\ 0.7} F_a^{\ 1.2} \ (\mathrm{mN \cdot m}) \\ &= 1.06\times10^{-3} D_{pw}^{\ 0.7} F_a^{\ 1.2} \ \{\mathrm{kgf \cdot mm}\} \end{aligned}\right\} \quad \cdots\cdots (2)$$

where, D_{pw}: Pitch diameter of rolling elements (mm)
F_a: Axial load (N), {kgf}

The speed term M_v is that for fluid friction, which depends on angular speed, and is expressed by Equation (3).

$$\begin{aligned} M_v &= 3.47\times10^{-10} D_{pw}^{\ 3} n_i^{\ 1.4} Z_B^{\ a} Q^b \ (\mathrm{mN \cdot m}) \\ &= 3.54\times10^{-11} D_{pw}^{\ 3} n_i^{\ 1.4} Z_B^{\ a} Q^b \ \{\mathrm{kgf \cdot mm}\} \end{aligned} \quad \cdots\cdots (3)$$

where, n_i: Inner ring speed (min^{-1})
Z_B: Absolute viscosity of oil at outer ring temperature (mPa·s), {cp}
Q: Oil flow rate (kg/min)

The exponents a and b, that affect the oil viscosity and flow rate factors, depend only on the angular speed and are given by Equations (4) and (5) as follows:

$$a = 24 n_i^{\ -0.37} \quad \cdots\cdots (4)$$
$$b = 4\times10^{-9} n_i^{\ 1.6} + 0.03 \quad \cdots\cdots (5)$$

An example of the estimation of the running torque of high speed ball bearings is shown in Fig. 1. A comparison of values calculated using these equations and actual measurements is shown in Fig. 2. When the contact angle exceeds 30°, the influence of spin friction becomes big, so the running torque given by the equations will be low.

Calculation Example

Obtain the running torque of high speed angular contact ball bearing 20BNT02 (ϕ20×ϕ47×14) under the following conditions:

$n_i = 70\ 000\ \mathrm{min}^{-1}$
$F_a = 590$ N, {60 kgf}
Lubrication: Jet, oil viscosity: 10 mPa·s {10 cp}
oil flow: 1.5 kg/min

From Equation (2),

$$\begin{aligned} M_l &= 0.672\times10^{-3} D_{pw}^{\ 0.7} F_a^{\ 1.2} \\ &= 0.672\times10^{-3}\times33.5^{0.7}\times590^{1.2} \\ &= 16.6\times10^{-3} \ (\mathrm{mN \cdot m}) \end{aligned}$$
$$M_l = 1.06\times10^{-3}\,33.5^{0.7}\times60^{1.2} = 1.7 \ \{\mathrm{kgf \cdot mm}\}$$

From Equations (4) and (5),

$$a = 24 n_i^{\ -0.37} = 24\times70\ 000^{-0.37} = 0.39$$
$$b = 4\times10^{-9} n_i^{\ 1.6} + 0.03 = 4\times10^{-9}\times70\ 000^{1.6} + 0.03 = 0.26$$

From Equation (3),

$$\begin{aligned} M_v &= 3.47\times10^{-10} D_{pw}^{\ 3} n_i^{\ 1.4} Z_B^{\ a} Q^b \\ &= 3.47\times10^{-10}\times33.5^3\times70\ 000^{1.4}\times10^{0.39}\times1.5^{0.26} \\ &= 216 \ (\mathrm{mN \cdot m}) \end{aligned}$$
$$M_v = 3.54\times10^{-11}\times33.5^3\times70\ 000^{1.4}\times10^{0.39}\times1.5^{0.26} = 22.0 \ \{\mathrm{kgf \cdot mm}\}$$
$$M = M_l + M_v = 16.6 + 216 = 232.6 \ (\mathrm{mN \cdot m})$$
$$M = M_l + M_v = 1.7 + 22 = 23.7 \ \{\mathrm{kgf \cdot mm}\}$$

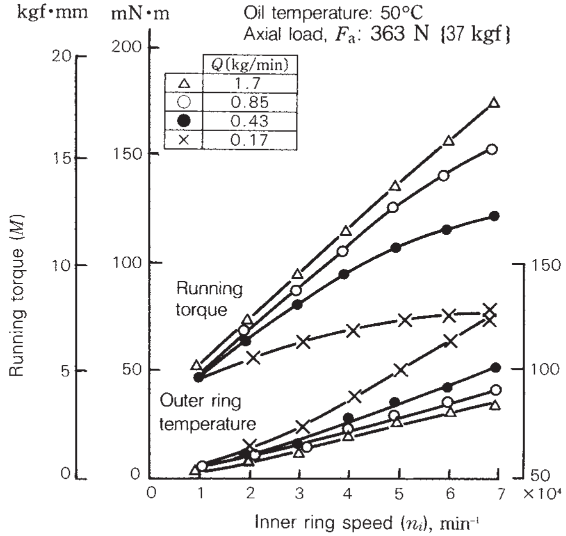

Fig. 1 Typical test example

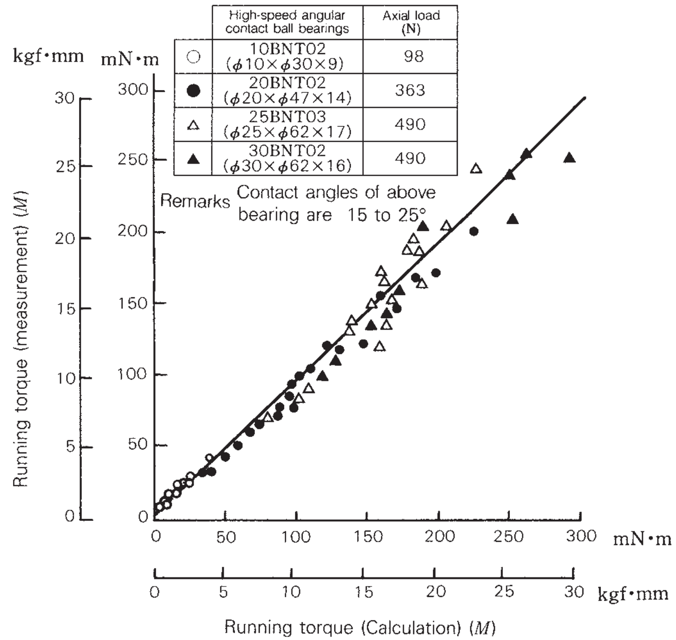

Fig. 2 Comparison of actual measurements and calculated values

7.4 Empirical equations for running torque of tapered roller bearings

When tapered roller bearings operate under axial load, we reanalyzed the torque of tapered roller bearings based on the following two kinds of resistance, which are the major components of friction:

(1) Rolling resistance (friction) of rollers with outer or inner ring raceways — elastic hysteresis and viscous rolling resistance of EHL

(2) Sliding friction between inner ring ribs and roller ends

When an axial load F_a is applied on tapered roller bearings, the loads shown in Fig. 1 are applied on the rollers.

$$Q_e \doteqdot Q_i = \frac{F_a}{Z \sin\alpha} \quad \cdots\cdots (1)$$

$$Q_f = \frac{F_a \sin 2\beta}{Z \sin\alpha} \quad \cdots\cdots (2)$$

where, Q_e: Rolling element load on outer ring
Q_i: Rolling element load on inner ring
Q_f: Rolling element load on inner-ring large end rib
Z: Number of rollers
α: Contact angle…(1/2) of the cup angle
β: (1/2) of tapered roller angle

For simplification, a model using the average diameter D_{we} as shows in Fig. 2 can be used.

Where, M_i, M_e: Rolling resistance (moment)
F_{si}, F_{se}, F_{sf}: Sliding friction
R_i, R_e: Radii at center of inner and outer ring raceways
e: Contact height of roller end face with rib

In Fig. 2, when the balance of sliding friction and moments on the rollers are considered, the following equations are obtained:

$$F_{se} - F_{si} = F_{sf} \quad \cdots\cdots (3)$$

$$M_i + M_e = \frac{D_w}{2} F_{se} + \frac{D_w}{2} F_{si} + \left(\frac{D_w}{2} - e\right) F_{sf} \quad \cdots\cdots (4)$$

When the running torque M applied on the outer (inner) ring is calculated using Equations (3) and (4) and multiplying by Z, which is the number of rollers:

$$M = Z\,(R_e\,F_{se} - M_e)$$

$$= \frac{Z}{D_w}(R_e\,M_i + R_i M_e) + \frac{Z}{D_w} R_e\; e\; F_{sf}$$

$$= M_R + M_S$$

Therefore, the friction on the raceway surface M_R and that on the ribs M_S are separately obtained. Additionally, M_R and M_S are rolling friction and sliding friction respectively.

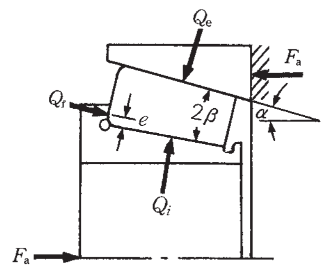

Fig. 1 Loads applied on roller

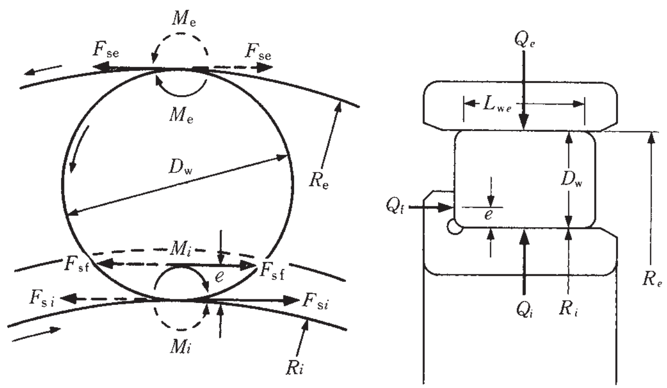

Fig. 2 Model of parts where friction is generated

The running torque M of a tapered roller bearing can be obtained from the rolling friction on the raceway M_R and sliding friction on the ribs M_S.

$$M=M_R+M_S=\frac{Z}{D_w}(R_e M_i+R_i M_e)+\frac{Z}{D_w}R_e\ e\ F_{sf} \quad \cdots\cdots (5)$$

Sliding friction on rib M_S

As a part of M_S, F_{sf} is the tangential load caused by sliding, so we can write $F_{sf}=\mu Q_f$ using the coefficient of dynamic friction μ. Further, by substitution of the axial load F_a, the following equation is obtained:

$$M_S = e\ \mu\ \cos\beta\ F_a \quad \cdots\cdots (6)$$

This is the same as the equation for starting torque, but μ is not constant and it decreases depending on the conditions or running in. For this reason, Equation (6) can be rewritten as follows:

$$M_S=e\ \mu_0\ \cos\beta\ F_a f'\ (\Lambda,\ t,\ \sigma) \quad \cdots\cdots (7)$$

Where μ_0 is approximately 0.2 and $f'\ (\Lambda,\ t,\ \sigma)$ is a function which decreases with running in and oil film formation, but it is set equal to one when starting.

Rolling friction on raceway surface M_R

Most of the rolling friction on the raceway is viscous oil resistance (EHL rolling resistance). M_i and M_e in Equation (5) correspond to it. A theoretical equation exists, but it should be corrected as a result of experiments. We obtained the following equation that includes corrective terms:

$$M_{i,\ e}=\left[f\ (w)\left(\frac{1}{1+0.29L^{0.78}}\right)\frac{4.318}{\alpha_0}(G\cdot U)^{0.658}W^{0.0126}R^2 L_{we}\right]_{i,\ e} \quad \cdots\cdots (8)$$

$$f\ (w)=\left(\frac{kF_a}{E'\ D_w L_{we} Z\sin\alpha}\right)^{0.3} \quad \cdots\cdots (9)$$

Therefore, M_R can be obtained using Equations (8) and (9) together with the following equation:

$$M_R=\frac{Z}{D_w}(R_e M_i+R_i M_e)$$

Running torque of bearings M

From these, the running torque of tapered roller bearings M is given by Equation (10)

$$M=\frac{Z}{D_w}(R_e M_i+R_i M_e)+e\ \mu_0\ \cos\beta\ F_a f'\ (\Lambda,\ t,\ \sigma) \quad \cdots\cdots (10)$$

As shown in Figs. 3 and 4, the values obtained using Equation (10) correlate rather well with actual measurements. Therefore, estimation of running torque with good accuracy is possible. When needed, please consult NSK.

[Explanation of Symbols]

- G, W, U: EHL dimensionless parameters
- L: Coefficient of thermal load
- α_0: Pressure coefficient of lubricating oil viscosity
- R: Equivalent radius
- k: Constant
- E': Equivalent elastic modulus
- α: Contact angle (Half of cup angle)
- R_i, R_e: Inner and outer ring raceway radii (center)
- β: Half angle of roller
- i, e: Indicate inner ring or outer ring respectively
- L_{we}: Effective roller length

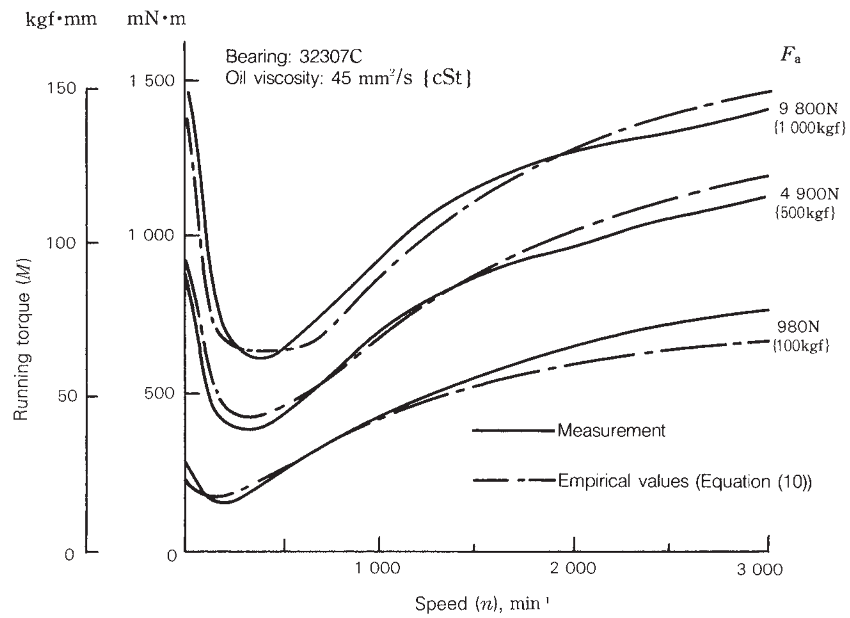

Fig. 3 Comparison of empirical values with actual measurements

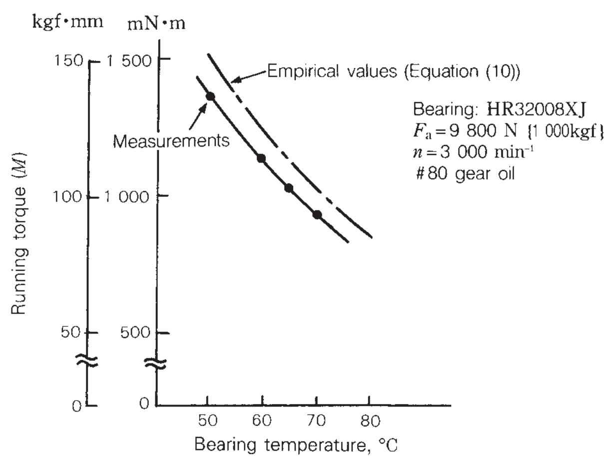

Fig. 4 Viscosity variation and running torque