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Heat Generation Model of Angular Contact Ball Bearing with Oil-Air Lubrication

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Abstract : Angular contact ball bearings are mainly used in the spindle, which requires high speed and stiffness. The heat generation is studied by experiments and simulations using a pair of angular contact ball bearings. The temperature variation of inner and outer races and the temperature increment distribution are measured by using thermocouples for the rotational speed, preload, viscosity of lubricant. The measured values from experiments are used to estimate the heat conduction rate. The method of oil-air lubrication is used for the experiment. The amount of conduction heat transfer to the test spindle and the convection heat transfer coefficients along the spindle are computed by using inverse method with temperature increment distribution. Total heat generation rate is estimated with the heat partition rate which is calculated from temperatures of inner and outer races. In addition, the empirical factor of oil-air lubrication method for Palmgren's heat generation model is suggested. The empirical friction coefficients, which are obtained from the experiments, depend on the preload condition, and can give us more accurate estimation of the heat generation in ball bearings.

Key words : heat generation, angular contact ball bearing, oil-air lubrication, inverse method, Palmgren's heat generation model

Introduction

Angular contact ball bearings are mainly used in the machine tool spindle, which requires high speed and stiffness in these days. Heat generation in bearing should be minimized to keep the accuracy of the spindle from the thermal distortion and deformation. The heat generation of bearing should be studied experimentally and numerically in order to find out the optimal lubrication condition for the minimization of the heat generation. Many design parameters for the various types of bearing and the method of lubrication have been proposed, and many engineers have been used them in their design.

Palmgren [1] proposed a model and an empirical equation for the heat generation in ball bearing. His empirical equation is quite good to predict the heat generation so it is widely used in engineering field. But the design parameters for the oil-air lubrication method have not been established yet. So, it is one of the main objects of this study to set up a set of those parameters which could be adopted to the, we call it, modified Palmgren's model.

Walters [2] analyzed dynamics of ball bearing elements numerically, Palmgren [1] proposed empirical equations of moments applied to the bearing, and Harris [3-5] analyzed the rolling motion as well as the sliding motion with friction force under loading. He also calculated the heat generation rate of the angular-contact ball bearing. For high speed region, the heat generation due to the gyro-moment should be considered, but it has not been applied to the Palmgren's heat generation model. Harris had computed the gyro-moment and had proposed the criteria for the gyro-moment to be considered or not. However, he did not count it in the calculation of heat generation. Carmichael and Davis [6] measured the preload variation and friction torque variation using angular contact ball bearing with respect to the rotational speed, the oil supply rate and the lubricant. But their result is unsuitable in high-speed region because the rate of lubricant supply was too high.

In this research, the heat generation in angular contact ball bearing is studied experimentally and numerically using a pair of angular contact ball bearings. The temperature variation of inner and outer races, and the temperature increment distributions are measured by using thermocouples for the rotational speed, preload and viscosity of lubricant. The method of oil-air lubrication is adopted and the heat conduction rate is estimated by experiments and numerical simulations using the inverse method. The amount of heat conducted to the test spindle and convection heat transfer coefficients along the spindle are computed by using the inverse method with temperature increment distribution. Total heat generation rate is estimated with heat partition rate, which is calculated from the temperature of the inner and outer races. In addition, the empirical factor of oil-air lubrication method for the Palmgren's heat generation model is also suggested. The empirical friction coefficients are used for the accurate estimation of the heat generation. The coefficients are obtained from experiments according to the preload conditions.

Heat generation model

Balls in an angular contact bearing are subject to spin and gyro

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moments due to the rotation with its tilted rotating axis. Due to these moments, balls are sliding over the raceways which result in heat generation. Also the applied load and the viscous dissipation of the oil, both of which were suggested by Palmgren, generate the heat.

Spinning moment

When the axial load F_a is applied to the ball with the contact angle α, the radial force Q is computed as the following;

$$Q = F_a/\sin\alpha. \tag{1}$$

Under the condition of pure spin in the normal direction of contact ellipse, the friction moment is changed as well;

$$Ms = \tfrac{3}{8}\,\mu_s Q a E(k) \tag{2}$$

where μ_s is the friction coefficient, a is the semi-major axis of the projected elliptical area of contact, and $E(k)$ is the complete elliptic integral of the second kind which is a function of the elliptical eccentricity parameter k.

Gyroscopic moment

Figure 1 shows coordinates and angular velocity components of the ball for angular-contact ball bearing. The ball rotates as well as revolves on bearing axis since gyroscopic moment is applied to the ball. As we can see in Fig. 1, the infinitesimal element of mass d_m in the ball has the instantaneous position in the rotating system, and the equation of motion for d_m is subject to the Newton's second law. In this study it is assumed that the pivotal motion of the ball due to the gyroscopic moment occurs only when the tangential force on ball due to the gyroscopic moment becomes greater than the normal load times friction coefficient.

The gyroscopic moment M_g is obtained for ball bearings as the following;

$$M_g = J\omega_m\omega_b \sin\beta. \tag{3}$$

In this equation, J is the mass moment of inertia, β is the rotation angle as shown in Fig. 1, ω_m and ω_b are angular velocity of cage and ball on its axis, respectively. The gyroscopic sliding occurs when the tangential force caused by gyroscopic moment is larger than the friction force, then the ball starts to slide and the heat is generated.

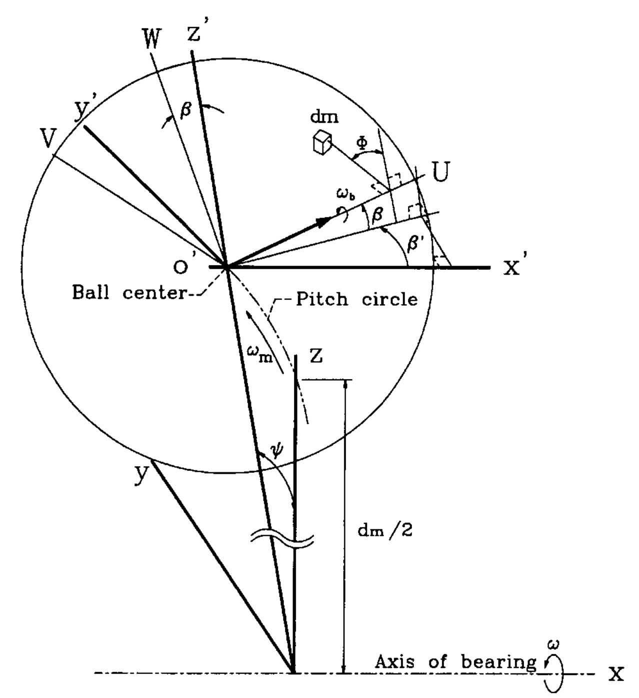

Fig. 1. Instantaneous position of ball mass element dm.

Torque due to applied load

Palmgren [2] determined empirically that the torque due to applied load can be computed asthe following;

$$M_l = f_l F_\beta (r_i + r_b) \tag{4}$$

in which f_l is a parameter depending on bearing design and relative bearing load determined as the following;

$$fl = z(F_s/C_s)^y. \tag{5}$$

For the contact angle of 15°, z and y are 0.001 and 0.33, respectively. C_s is the basic static load which is usually given in manufacturer's catalogs along with the data for the calculation of F_s. F_β shown in Eqn. (4) depends on the magnitude and the direction of the applied load. F_β may be expressed in two ways as follows and larger one is usually selected:

$$F_\beta = 0.9F_a\cot\alpha - 0.1F_r \quad \text{or} \quad F_\beta = F_r. \tag{6}$$

Viscous friction torque

Palmgren [2] empirically determined that the viscous friction torque can be expressed as the following:

$$\begin{aligned} Mv &= 10^2 f_c(\nu n)^{2/3}(r_i+r_b)^3 \quad \text{for } \nu n > 2{,}000 \\ &= 1.6\times 10^4 f_c(r_i+r_b)^3 \quad \text{for } \nu n \le 2{,}000 \end{aligned} \tag{7}$$

where ν is the viscosity, n is the rotational speed (rpm), and fc is the factor depending on the type of bearing and the method of lubrication. In this study, we set $f_c = 6.6$ for oil jet, $f_c = 3.3$ for oil bath, $f_c = 1.7$ for oil mist, and $f_c = 2.0$ for grease lubrication.

Heat generation

The rate of heat generation can be obtained by the following equation:

$$H = \omega M = (2\pi/60)nM, \tag{8}$$

The total heat generation rate of angular-contact ball bearing is calculated as the following:

$$H_{total} = H_{spin} + H_{gyro} + H_{load} + H_{viscous}. \tag{9}$$

Experiments and estimation of heat generation

Experimental setup

Figure 2 shows the schematic of the experimental setup to measure the temperature distribution along the hollow spindle which is supported by two angular-contact ball bearings. Rotational speed, controlled by an inverter, varies from 2,500 to 22,000 rpm. The T-type thermocouple is used at seven

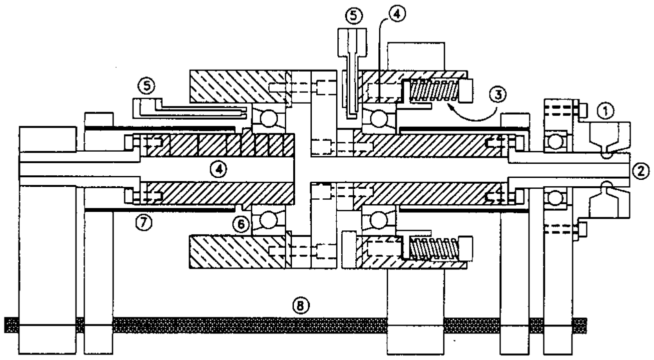

Fig. 2. Test system layout.
① air-nozzle for drive shaft, ② test shaft(air bucket), ③ spring(preload),④thermocouple(T type),⑤oil-air lubrication supply nozzle, ⑥ angular contact ball bearing(7005C), ⑦ steel tube, ⑧ alignment guide

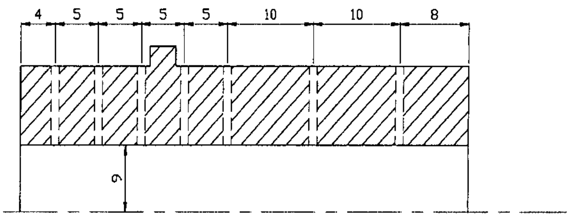

Fig. 3. Cross sectional view of the spindle for the thermocouple setup position.

locations. Since the voltage generated by the thermocouple is too weak to use slip ring, voltage amplifying circuit is embedded inside the hollow spindle.

Experimental and computational procedures

The temperature increase is so slow that it requires a long time to obtain the steady state temperature distribution for the specific condition. Viscosity and density of oils used in experiment are given in Table 1.

One of the main interests in this study is to examine the heat generation rate of the bearing. The heat, generated from the interfaces between the balls and races, is conducted into the ball, inner and outer races, and also is convected into the air passing by. Once the heat conducted into the one of races is known, the total heat generation can be computed. However, it is quite complicated to measure the heat generation rate directly, especially for the rotating system. One way to solve this kind of problem is to compute the heat conducted into the spindle using the inverse method. The inverse method gives the amount of generated heat to satisfy the given temperature distribution along the spindle when we solve the heat conduction equation. The governing equation for the heat conduction in cylindrical coordinates is:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = S \qquad (10)$$

The solution procedure is following: First, generated heat is assumed to be transferred such that 25% of total generated heat at the bearing transferred to inner and outer races, respectively. Temperature distributions of inner and outer races are calculated by solving the Eqn.(10) using assumed convection heat transfer coefficient along the spindle. These temperature distributions of inner and outer races are compared with those of measured values from experiment. The generated heat and convection heat transfer coefficient can be calculated through two different sets of experimental results. Rashid and Seireg's relations are adopted for the calculation of heat partition rates to the inner, outer races and to the air [8].

Table 1. Viscosity and Density

	Viscosity (cSt)		Density (kg/m^3)
	40°C	100°C	
Oil #1	8.307	2.228	878.8
Oil #2	12.02	3.176	865.8
Oil #3	16.05	3.878	858.6

Results and Discussion

Figure 4 shows the variation of bearing friction coefficient when three different viscosity of lubricants are used in oil-air lubrication system. As the preload increases, the increment of friction coefficient becomes larger because the elastic deformation is increased as the load subject to the rolling element increases. Also, the friction coefficient increases as the viscosity of oil and the rotational speed of spindle increase due to the viscous drag in oil film. Pressure in oil film at the contact area increases with the rotational speed of the spindle so that it generates larger elastic deformation. The bearing friction coefficients are ranged 0.008~0.011, 0.01~0.013, and 0.012~

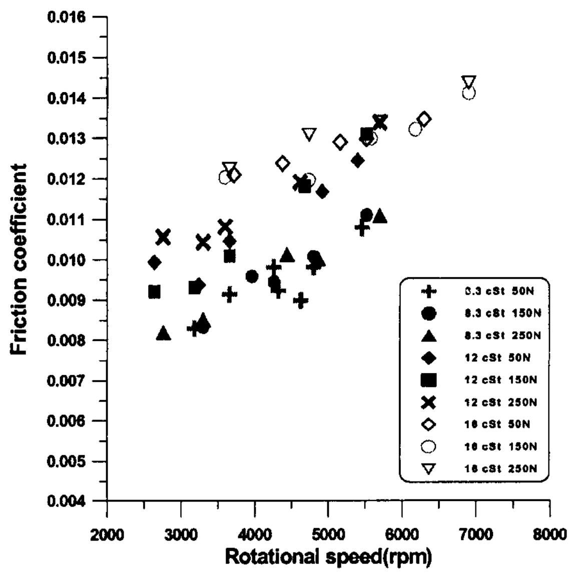

Fig. 4. Friction coefficient of test bearing in oil-air lubrication. (Viscosity 8.3 cSt at 40°C)

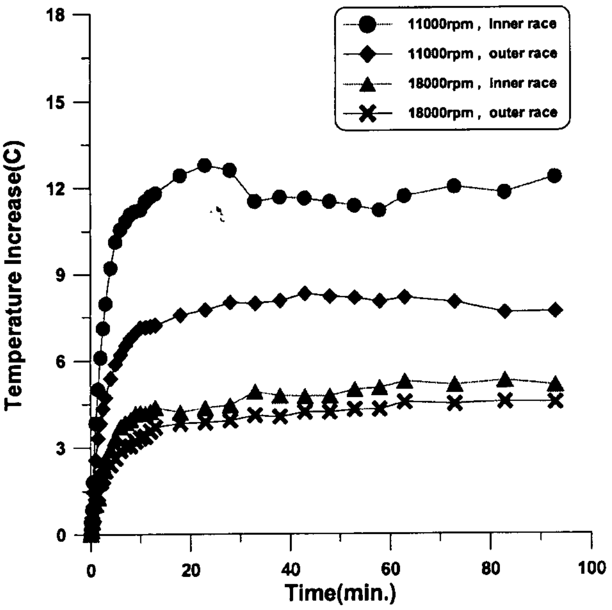

Fig. 5. Temperature variation of races w.r.t. time. (Preload 150 N, Viscosity 8.3 cSt at 40°C)

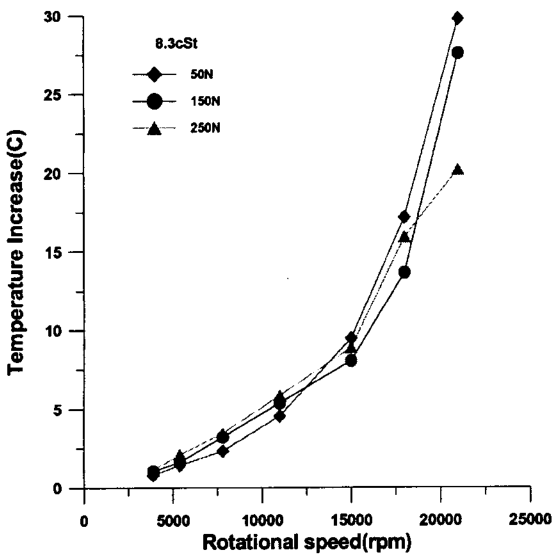

Fig. 6. Temperature variations of inner race w.r.t. spindle rpm. (Viscosity 8.3 cSt at 40°C)

0.0145 when the viscosity of oil used in oil-air system are 8.3cSt, 12cSt, and 16cSt, respectively.

Figure 5 represents the temperature variations of inner and outer races with respect to time. The temperature increase of inner race is higher than that of outer race because outer race easily removes heat to outward than inner race does. The temperature increase becomes larger with the increase of preload and rotational speed. Rotational speed gives more dominant effect on temperature increase. Suggested oil supply rate for oil-air system is about 0.25 cc/h for the minimum heat generation.

Figure 6 shows temperature variation with respect to the rotational speed for three different preloads when oil of 8.3 cSt

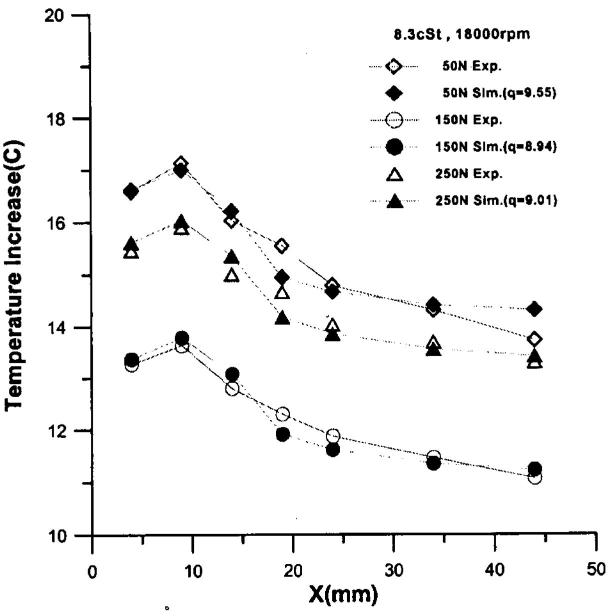

Fig. 7. Temperature distribution of spindle for experiment and simulation in oil-air lubrication. (8.3 cSt at 40°C, 18,000 rpm)

is used as lubricant. When 50 N is applied as a preload, temperature increases abruptly around 11,000 rpm due to the gyro-moment of balls in bearing. However, for the cases of preload of 150 N and 250 N, temperature increases abruptly around 15,000 rpm where the gyro-moment becomes in effective. The onset point where the gyro-moment is effective increases as the preload increases because the preload suppresses balls from slipping between races due to gyro-moment.

Figure 7 represents the idea how to predict the convection heat transfer coefficient numerically. Heat conducted to the spindle and convection coefficient around the spindle are calculated from the temperature distribution along the spindle by using the inverse method. Appropriate guess of heat generation and convection coefficient gives correct temperature distribution which matches with the result from experiment.

Figure 8 shows heat generation rates at the bearing based on the heat conducted to inner race using Rashid and Seireg's heat partition relations for 8.3 cSt oil. The heat generation rates for the case of preload of 50 N is lower than those of other preload conditions below 15,000 rpm but they become higher when the spindle rotates above 15,000 rpm. The reason is that larger preload suppresses gyro-moment effect more.

Figure 9 reveals heat conduction rates transferred to inner race according to different preloads and rotational speeds by Rashid and Seireg's heat partition relations. In particular, the rates increased due to preload change are larger than those due to rotational speed change. Because oil film for lubrication becomes thinner and contact area becomes wider between balls and races.

When 8.3 cSt oil is used, Fig. 10 represents the total heat generation estimated numerically based on experimental data, the Palmgren's heat generation model, and proposed model of

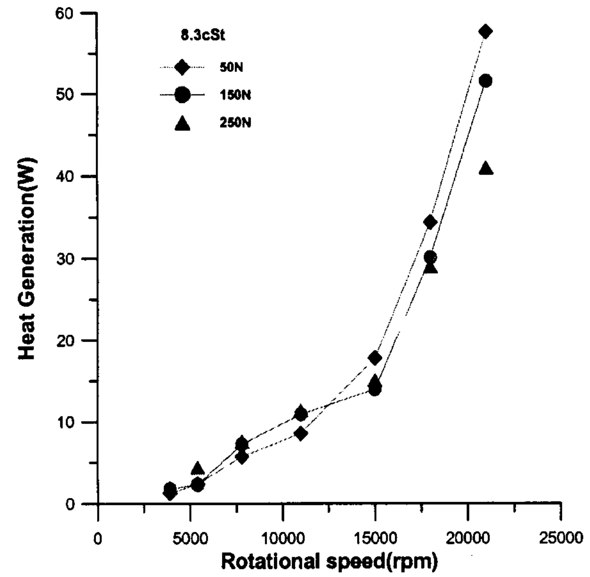

Fig. 8. Variation of heat generation rate at different preload in oil-air lubrication. (8.3 cSt at 40°C)

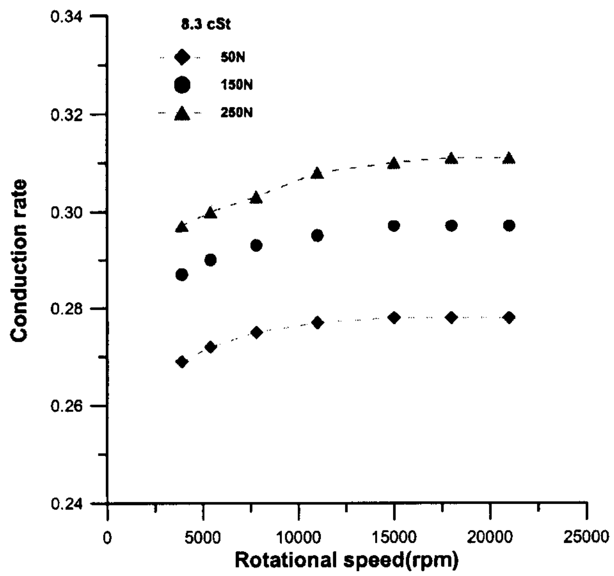

Fig. 9. Variation of heat conduction rate of inner race w.r.t. spindle rpm. (8.3 cSt at 40°C)

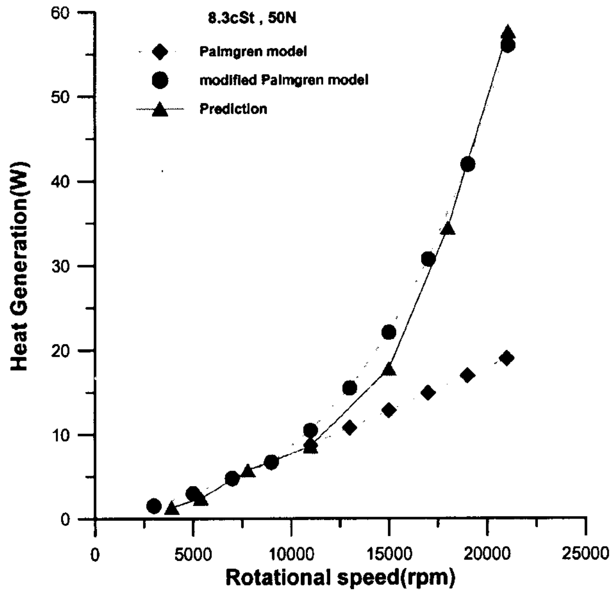

Fig. 10. Comparison of heat generation rate of Palmgren's model and proposed model w.r.t. spindle rpm. (Preload 50 N, Viscosity 8.3 cSt at 40°C)

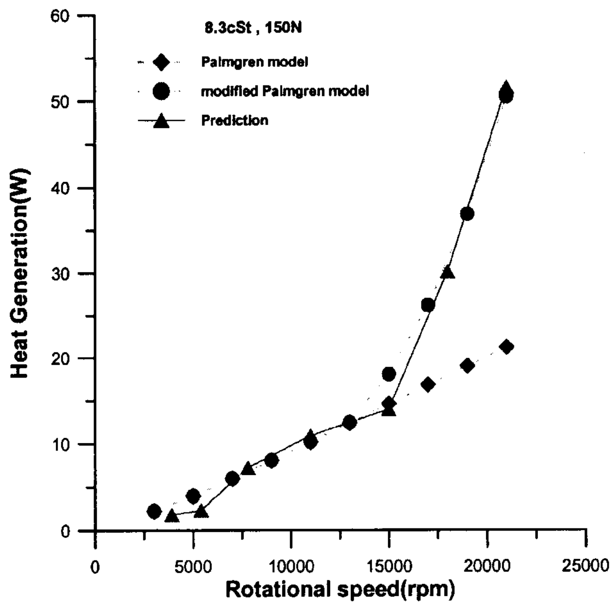

Fig. 11. Comparison of heat generation rate variation of Palmgren's model and proposed model w.r.t. spindle rpm. (Preload 150 N, Viscosity 8.3 cSt at 40°C)

this study, we call it as `modified Palmgren's heat generation model' which considers gyro- and spin moment.

In order to find out the oil-air lubrication factor f_c, heat generation estimated numerically is compared with that from modified Palmgren's heat generation model by varying f_c. The result obtained for f_c is 0.4. In Fig. 11, the result of f_c is 0.33 for the preload of 150 N and 8.3 cSt oil. In Fig. 12, f_c is 0.34 for the preload of 150 N and 12 cSt oil while f_c is 0.11 for the preload of 150 N and 16 cSt oil as shown in Fig.13.

Conclusion

I. Friction coefficients are measured from the friction test to estimate the heat generation rate. Friction coefficients with range of 0.008~0.015 are used to calculate heat generation in Palmgren's model.

II. Total heat generation rates estimated numerically based on experimental data and those from `modified Palmgren's heat generation model' are very similar. Therefore, effects of gyro- and spin moments should be considered in estimation of heat generation in high-speed region.

III. Heat conducted to the spindle and convection coefficient along the spindle are calculated from temperature distribution

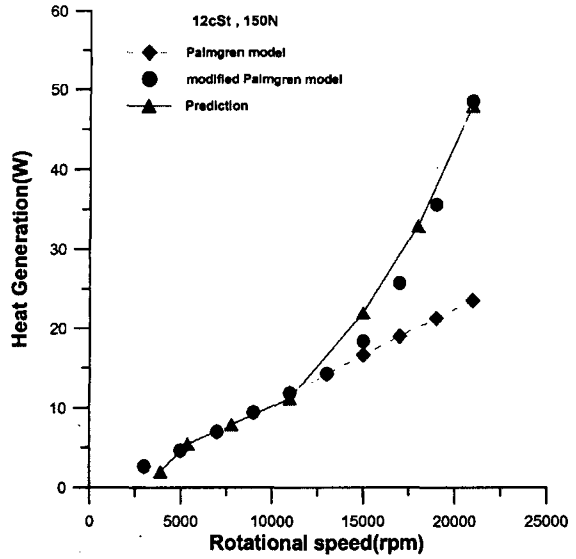

Fig. 12. Comparison of heat generation rate variation of Palmgren's model and proposed model w.r.t. spindle rpm. (Preload 150 N, Viscosity 12 cSt at 40°C)

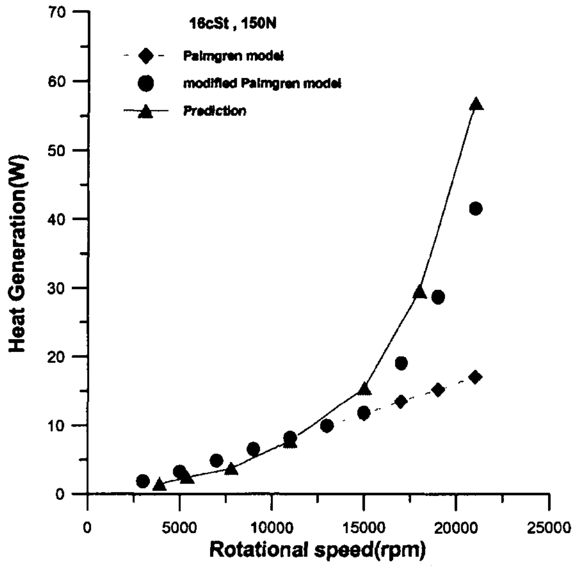

Fig. 13. Comparison of heat generation rate variation of Palmgren's model and proposed model w.r.t. spindle rpm. (Preload 150 N, Viscosity 16 cSt at 40°C)

by using inverse method. Heat partition rates are calculated from measured temperatures of inner and outer races. Heat generation rate can be estimated from these heat partition rates.
IV. In order to find out the oil-air lubrication factor f_c, heat generation estimated numerically is compared with that from modified Palmgren's heat generation model by varying f_c. The f_c varies from 0.1 to 0.4 for preload of 150 N with oils of 8.3 cSt, 12 cSt, and 16 cSt in oil-air lubrication method.

Nomenclature

a : semi-major axis of the projected elliptical area of contact
E(k) : complete elliptic integral of the second kind
f_c : factor depending on type of bearing and method of lubrication
f_l : factor depending on bearing design and relative bearing load
F_a : axial load applied to the ball (N)
F_β : force depends on magnitude and direction of applied load (N)
h : convection heat transfer coefficient (W/m^2 · K)
k : elliptical eccentricity parameter
J : mass moment of inertia (kg · mm^2)
M_s : spinning moment (N · mm)
M_g : gyroscopic moment (N · mm)
M_v : viscous friction torque (N · mm)
M_l : friction torque due to applied load (N · mm)
n : rotational speed (rpm)
r_i : radius of inner ring track (mm)
r_b : radius of rolling element (mm)
T : temperature (°C)
α : contact angle of bearing
μ_s : friction coefficient
v : viscosity of lubricant (cSt)
ω_m : angular velocity of cage (rad/sec)
ω_b : angular velocity of ball on its axis (rad/sec)

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