한국윤활학회지(1995), 제 11 권 제 5 호.
Journal of KSTLE(1995) Vol. 11, No. 5, pp.122~127

Heat Generation of Angular Contact Ball Bearings

Rhim, Yoon Chul and Na, Hee Hyeong*

Department of Mechanical Engineering Yonsei University
**Graduate School, Yonsei University*

Abstract—The heat generations of angular contact ball bearings are studied experimentally with numerical simulations. The temperature variations of inner and outer races and the temperature increase distributions are measured by using thermocouples for the spindle rotational speeds, preloads, viscosity of oils, and lubrication methods. The measured values from experiments are used to estimate the heat generation rates. The heat generation is focused mainly on the dominant sources which are rubbings due to spin and gyro-moments of bearing balls, applied load and viscous friction. Oil-jet and oil-air lubrication methods are adopted using oils with different viscosities.

Key words : Heat generation, Angular contact ball bearing, Oil-air lubrication

1. Introduction

Angular-contact ball bearings, cylindrical or tapered roller bearings are used for the spindle of the machine tools and quite recently magnetic bearings are introduced for some grinding machines [1]. Angular-contact ball bearings are mainly used for the spindle which requires high rotational speed, precision and stiffness. To maintain high precision and stiffness at high rotational speed, it is required about the information of the heat generation at the bearing because the temperature increase affects the stiffness and deformation of the spindle. Also the preload is an important factor to keep the angular contact bearing stiff. Preload is a function of the contact angle and applied force, which is altered during the operation due to the heat generation.

The balls in an angular-contact bearing are subjected to spin and gyro moments due to the rotation with its rotating axis tilted. Due to these moments, balls are sliding on the raceways that causes heat generation. Also the heat is generated by the applied load and the viscous dissipation of the oil.

The heat generated raises the temperature of the bearing, housing and spindle so that the thermal expansion occurs and finally affects the precision. Therefore, the estimation for the heat generation rate is one of the basis of designing a high speed precision machine tool.

Walters conducted dynamic analysis of the ball bearing elements numerically [2], Palmgren proposed empirical relations between heat generation rate and moments applied to the bearing [3], Harris analyzed the motions of the balls in an angular-contact bearings and calculated the heat generation rates [4-6].

In this study, the temperature increases and distributions for the angular-contact ball bearing system are analyzed experimentally and then the heat generation rates are computed based on the experimental data.

2. Experiments and numerical simulations

2-1. Experimental set up

Fig. 1 shows the schematic diagram of the experimental setup. A 2.1Kw, 3-phase induction motor is used to rotate the simulated spindle and the rotational speed is controlled by a frequency inverter for the range up to 9,000 rpm. An angular contact ball bearing (SKF 71914 CDGA/P4A) is assembled in the middle of the spindle as a test bearing and the radial and thrust bearings are used to support the spindle system. The preload for the test bearing is applied using a pushing mechanism with screw and is measured from the average value of four 120Ω strain gages attached around the specially designed aluminum spacer.

A nozzle for the oil-air lubrication system is installed on the bearing housing block. Three kinds of

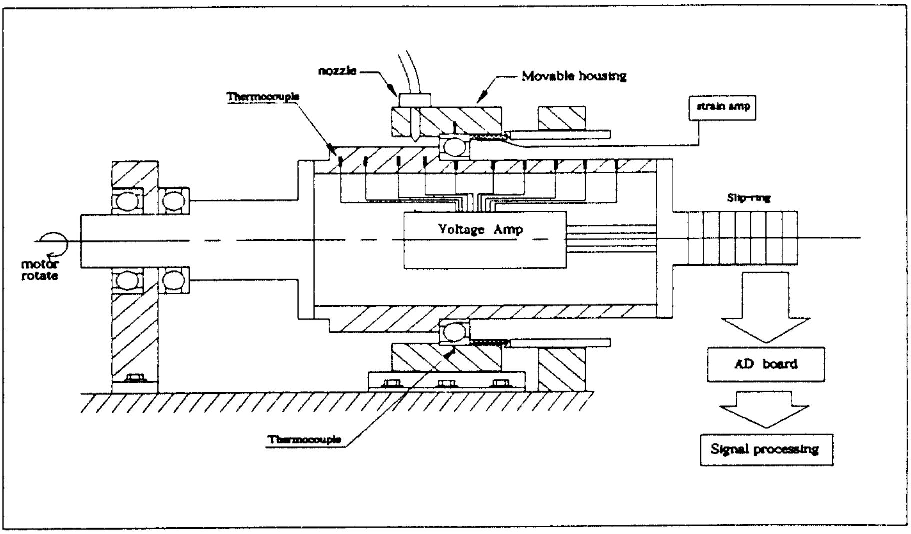


Fig. 1. Test system layout.

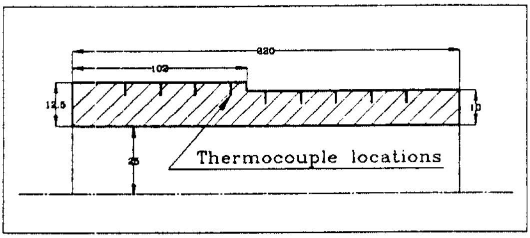


Fig. 2. Cross sectional view of the spindle.

oils having different viscosities are used with the compressed air. The pressure of the compressed air is set to 0.25 MPa by a pressure regulator. The oil is injected into the air tube through a solenoid valve. The period and the duration of the opening of the solenoid valve are controlled by a separately designed driving circuit. Another nozzle for the oil-jet lubrication system is prepared to investigate the effects of the lubrication methods.

T-type thermocouples are installed to measure the temperature distributions along the spindle since it is not easy to measure the heat flux from the bearing directly. The temperature of the inner and outer races are also measured via thermocouples. Signals from the thermocouples are transmitted to the A/D board via slip ring. However, signals from the thermocouples are so weak that they can hardly be detected out of the noise when slip ring is used, especially for the high rotational speed region. A 12-channel voltage amplifier is installed inside the spindle to amplify the signals from the thermocouples so that quite reliable outputs can be obtained for high rotational speed region.

Fig. 2 shows the cross sectional view of the simulated spindle. There are 9 locations where the thermocouples are installed.

2-2. Experiments and numerical simulations

The experiments are conducted at every 1,000 rpm starting from 3,000 rpm to 9,000 rpm. The temperature distributions are measured at every two-minute intervals. The measurement is done for one second and the each value of every thermocouples is taken and stored at the computer. The temperature rise is so slow that it takes long time to get the steady state temperature distribution for specific condition.

Viscosities and densities of oils used are given in Table 1.

One of the main interests of this study is to investigate the heat generation rate at the bearing. The heat is generated at the interfaces between the balls and races and then is conducted into the ball, inner and outer races and also is convected into the air which passes by. It is assumed that one quarter of the total heat generated is conducted into the inner and outer races, respectively and the rest is conducted into the balls and then convected out. Once

Table 1. Properties of oils used in experiment

	Viscosity (cSt)		Density
	40°C	100°C	(kg/m^3)
Oil #1	3.995	1.013	0.8849
Oil #2	8.307	2.228	0.8788
Oil #3	20.28	4.176	0.8557

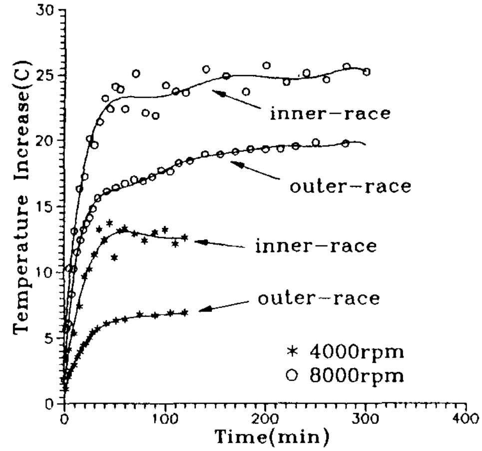
the heat conducted into the race is known, the total heat generation can be computed. However, it is quite complicate to measure the heat generation rate directly, especially for the rotating system. One way to solve this problem is to compute the temperature distributions along the spindle numerically and then compare the results with the experiment to find the proper boundary conditions, heat flux from the inner race to the spindle, inversely. The finite difference method with line SOR is adopted to solve the heat conduction equation for the spindle [7], i.e.,

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = 0$$

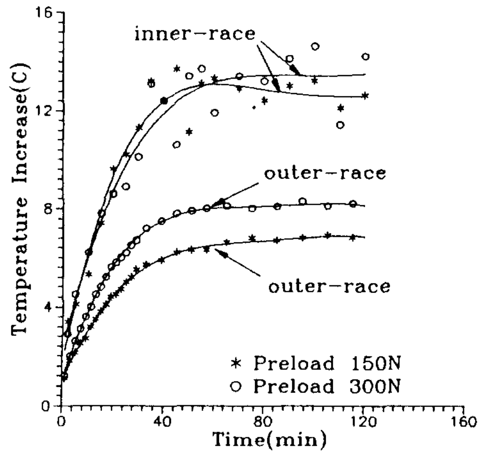
with the proper boundary conditions except the heat flux. Since the air flows axially outside of the spindle, convective heat transfer coefficients are required to establish the boundary conditions. Churchill and Bernstein's equation for the Nusselt number expression is employed to get proper heat convection coefficients [8].

3. Results and Discussion

Fig. 3 shows the temperature variations of inner and outer races with respect to time. Solid line is the curve fit of the experimental data represented by symbols. Data from inner race look a little bit scattered because of the noise from the slip ring. It shows that more than an hour is required to get a steady temperature for a certain condition. The temperature increase of the inner race is higher than that of the outer race and the differences in temperature increase depend on the operating parameters such as rotational speed, preload, viscosity of the oil, and lubrication methods(all is not shown here). The temperature increase gets larger as the preload and rotational speed increase, latter gives more dominant effect on temperature rise. The temperature of the inner race is always higher than that of the outer race because the outer race is easier to remove heat to the environment than inner race.



(a) Preload is set to 150N



(b) Rotational speed is 4,000rpm

Fig. 3. Temperature variation of races with respect to time (Oil #2 is used with oil-air lubrication).

Fig. 4 shows the temperature increases of inner and outer races at steady state when three types of oils with different viscosities are used at 9000 rpm. It shows that there must be an optimal viscosity which generates the least heat. More viscous oil generates heat by viscous dissipation and less viscous oil is easy to break the oil film so that direct contact between balls and races generates heat by spin moment.

Fig. 5 shows the temperature variations of races

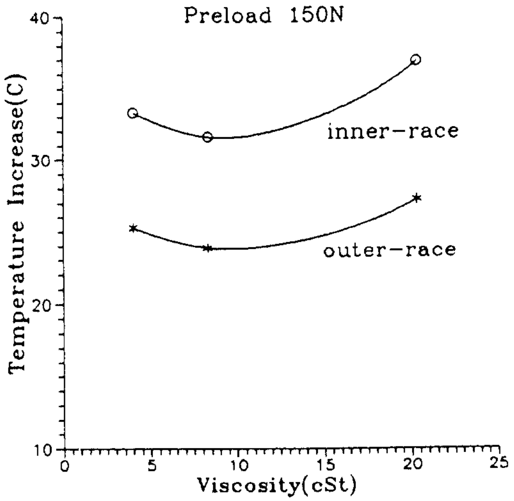


Fig. 4. Temperature variations of races with respect to the oil viscosity at 9,000 rpm when oil-air lubrication is applied.

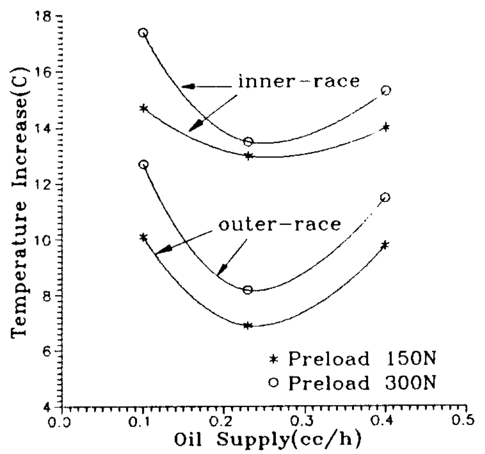


Fig. 5. Temperature increases of races for oil supply rate variations at 4,000 rpm (Oil #2 is used with oil-air lubrication).

for different oil supply rates. It also shows that there is an optimal oil supply rate to minimize the heat generation. Less oil supply results in lack of oil quantity to make an oil film between balls and races. On the other hand, too much oil covers the bearing elements' surfaces so that the heat is not easily transferred to the environments. From the results of Fig. 4 and 5, the optimal viscosity and supply rate of oil are around 10cSt and 0.25 cm^3/hour, respectively for minimizing the heat generation.

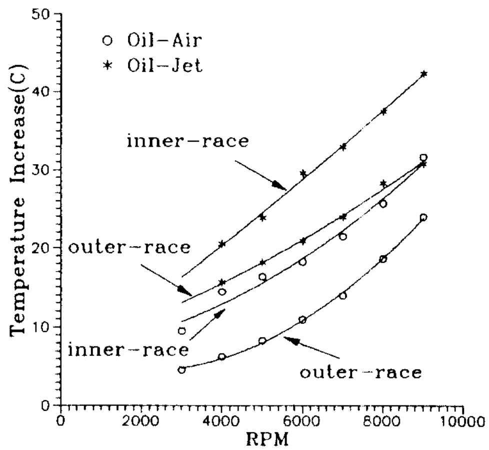


Fig. 6. Temperature increase variations of races with respect to the rotational speed for different lubrication methods.

Fig. 6 shows the temperature increase variations of races with respect to the rotational speed for the oil-jet and the oil-air lubrication methods. The temperature increase varies almost linearly with the rotational speed when the oil-jet lubrication is adopted while it varies quadratically for the oil-air lubrication method. The oil-jet lubrication method supplies sufficient oil to the bearing so that the oil film is surely formed but the redundant oils cover the bearing elements to keep the heat from being convected to the environment. Since the oil-air lubrication method uses the optimal oil supply rate for 4,000 rpm, heat generation becomes larger as the rotational speed increases.

Fig. 7 shows the idea of how to predict the heat generation rate numerically. Once the temperature distribution along the spindle is measured, the heat conduction equation is solved for the spindle with the appropriate boundary conditions and assumed heat flux where the bearing is located. Numerical solutions have the similar temperature distributions to the measured values by adjusting the boundary conditions including heat flux from the bearing. As shown in Fig. 7, the maximum temperature is dependent on the heat flux and temperature variation along the spindle depends on the convective heat transfer coefficients.

Fig. 8 compares the temperature distributions along the spindle for different lubrication methods at 6,000 rpm. The heat generation rate with oil-jet lu-

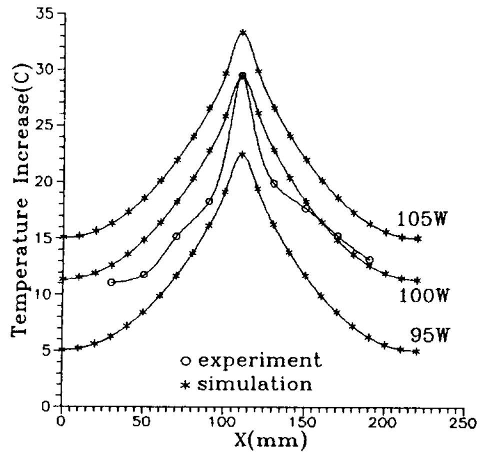


Fig. 7. Comparison of temperature increase distribution from the experiment and numerical solutions with boundary conditions.

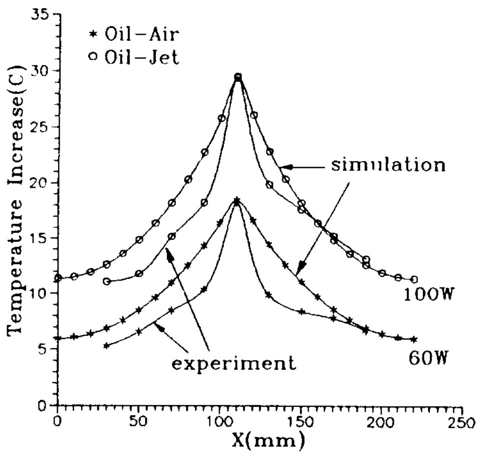


Fig. 8. Comparison of the temperature distribution for different lubrication methods at 6,000 rpm.

brication is about 100 watt while only 60 watt is generated with oil-air lubrication. The temperature distribution is not symmetrical because of the unsymmetry of the test spindle. The results from the computer simulation show relatively large differences near the bearing. The reason is thought to be the unsuitable modeling of the turbulence due to the rotating balls and retainer. It is left for further study.

Fig. 9 shows the trends of estimated heat generation rates with respect to the rotational speed of the spindle. The oil-air lubrication method gen-

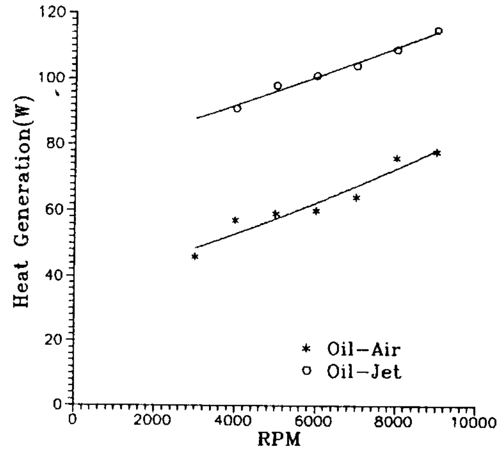


Fig. 9. Heat generation rate variations with respect to the rotational speed for different lubrication methods.

erates 40~30% less heat than oil-jet lubrication method for the tested rotational speed range. Both heat generation rate curves look linear. This means that the gyro-moment effect for the heat generation is not counted yet. The test bearing used in the experiment has small size balls relative to the other bearings whose diameter of the inner race are 70 mm and the preload of 150N is large enough to keep the balls from sliding between the races due to the gyro-moment. Since the heat generation rate shown in Fig. 9 is one quarter of the total heat generated in the bearing, the estimated total heat generation rate for this angular contact ball bearing is around 300 watt for the conditions of rotational speed being 9,000 rpm, 150N of preload with oil-air lubrication method.

4. Conclusions

(1) Temperature increase at the bearing races largely depends on the lubrication method and the rotational speed relative to the preload and the viscosity of the oil used.

(2) For the oil-air lubrication method, the suggested supply rate and viscosity of the oil is 0.2~0.3 cm^3/hour and 10 cSt at 40°C, respectively to minimize the heat generation.

(3) Heat generation rate can be estimated numerically using the temperature data by an inverse method.

References

1. T.Higuchi, *et. al.*, "Application of Periodic Learning Control with Inverse Transfer Function Compensation in Totally Active Magnetic Bearings," Proc. of the 2nd International Symposium on Magnetic Bearings, Tokyo, Japan, pp. 257-264, 1990.
2. C.T. Walters, "The Dynamics of Ball Bearing," Journal of Lubrication Technology, pp. 257-264, Jan. 1971.
3. Arbid Palmgren, Ball and Roller Bearing Engineering, 1959.
4. T.A. Harris, "An Analytical Method to Predict Skidding in Thrust Loaded Angular Contact Ball Bearings," Journal of Lubrication Technology, pp. 17-24, Jan. 1971.
5. T. A. Harris, "Ball Motion in Thrust Loaded Angular Contact Bearings with Coulomb Friction," Journal of Lubrication Technology, pp. 32-38, Jan. 1971.
6. Harris, Rolling Bearing Analysis, John Wiley & Sons, 1991.
7. D.A. Anderson, J.C. Tannehill, Computational Fluid Mechanics and Heat Transfer, McGraw Hill, 1984.
8. Incropera & Dewitt, Introduction to Heat Transfer, John Wiley & Sons, 1985.