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To cite this article: Haiyin Cao *et al* 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **491** 012011

IOP Conf. Series: Materials Science and Engineering **491** (2019) 012011 doi:10.1088/1757-899X/491/1/012011

An Experimental Investigation of the Thermohydrodynamic Performance of a Steady Loaded Hole-Entry Hybrid Journal Bearing

Haiyin Cao[1,2], Ruolin Wu[1,2,*], Huiping Chen[1,2] and Chao Lu[1,2]

[1]State Key Lab of Digital Manufacturing Equipment & Technology, Huazhong University of Science & Technology, 430074, Wuhan, China
[2]School of Mechanical Science and Engineering, Huazhong University of Science & Technology, 430074, Wuhan, China

[*]Corresponding author e-mail: wrl515342@163.com

Abstract. The purpose of this paper is to show the results of parametric experiments about the thermohydrodynamic performance of the steady-state loaded hole-entry hybrid bearing. Because there are many available theoretical in that respect have been studied, it become an urgent need to do related experimental researches. In the present study, the internal pressure and temperature distribution, flow rate and shaft center position under different supply pressures were measured. At the same time, a series of tests under different load and speed conditions were conducted. The experimental results show that bearing performances are significantly affected by the temperature rise of oil film, meanwhile, speed is the most important factor affecting the bearing temperature distribution.

1. Introduction

The hole-entry hybrid journal bearings have the merits of precise positioning, no wear, large direct film stiffness and damping, controllability of critical speeds and harmful vibration attenuation, so they are widely used in high-speed turbo-machinery, machine tool spindles, precision grinding spindles, liquid rocket pumps etc. [1]. To utilize both hydrostatic and hydrodynamic actions in a more efficient way, the oil recess of the hydrostatic bearing has been modified to be shallower, even changed to step oil recess or directly cancelled, meanwhile the land area has been enlarged to fit higher speed. Compared with the traditional recessed bearing, the hole-entry hybrid journal bearing is easier to manufacture, but also retains the advantages of hydrostatic bearing. Geometry of the hole-entry hybrid journal bearing studied in this paper is shown in Fig. 1.

Due to the rapid development of technology, bearings often need to work under severer conditions. When the oil film bearing operate under high speed, heat is generated within the fluid-film for oil-shear, and temperature rise of oil and bearing shell occurs. This, in turn, results in a significant reduction in the viscosity of the lubricating oil, and a lower minimum oil film thickness. As a result, the flow field of the lubricant becomes distorted and the bearing performance is affected. In order to accurately and reliably survey the bearing performance, it becomes imperative to measure the temperature distribution of the bearing systems. In the following paragraphs some recent studies into the hole-entry journal bearing systems, both theoretical and experimental, are reviewed.

The theoretical research on the hole-entry hybrid bearings has acquired remarkable achievements in

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recent decades. Notable among these is the study by Rowe et al. [2] who demonstrated that the hole-entry bearings were particularly effective for better load support and low energy consumption at zero and high-speed operations when compared with other bearing configurations. A comprehensive review about the development of hydrostatic/hybrid journal bearing systems was presented by Rowe et al. [3]. Sharma et al. [1] developed a theoretical model to study the performance of a hole-entry hybrid journal bearing system by considering variation of viscosity due to temperature rise of the lubricant in the analysis, indicated that the variation of viscosity due to temperature rise of the lubricant film had a quite appreciable influence on the static and dynamic performance of a hole-entry hybrid journal bearing system. Garg et al. [4] combined the non-Newtonian behavior of lubricant with temperature rise to study the dynamic performance of a hole-entry hybrid journal bearing system. Very recently, Khatri et al. [5] studied the influence of textured surface on the performance of non-recessed hybrid journal bearing operated with non-Newtonian lubricant.

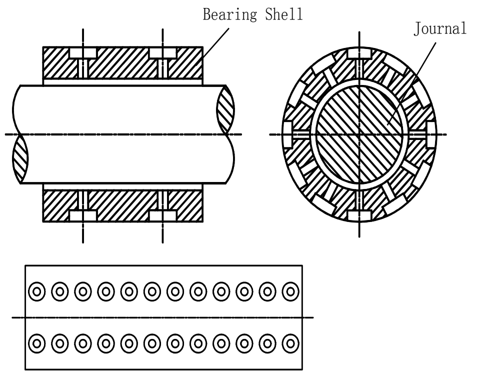

Figure 1. Bearing geometry.

As we know, few experimental researches into the thermohydrodynamic performance of hole-entry hybrid bearings have been performed. Xu et al. [6] designed a bearing test rig to study static and dynamic performance of oil-film journal bearings. The hole-entry bearing studied had 16 uniform distribution holes in double row, with a pair of pressure and displacement sensors orthogonal arranged under the shaft surface to measure the oil film pressure and thickness distribution. But Xu did not consider the effect of oil film temperature rise on the bearing performance. For other types of bearings, a lot of experimental researches appeared in the last few decades. For example, Dowson et al. [7] studied the pressure distribution and the temperature distribution on both bearing shell and shaft of a steady-loaded journal bearing under thermal equilibrium conditions, and concluded that the adiabatic assumption did not agree well with the experimental results and the shaft could be considered isothermal. Ferron et al. [8] studied the theoretical and experimental thermohydrodynamic problem of a finite length journal bearing, concluded that the thermal expansion and deformation of the shaft and the bush must be accounted in the theoretical and experimental processes. Ahmad et al. [9] conducted an experimental study of the effects of groove position on the temperature and pressure distribution of a journal bearing, revealed that when the groove position was in the divergent region, the oil film pressure was lowest. For hydrostatic/hybrid journal bearings, Sawicki et al. [10] experimentally investigated the full linear anisotropic rotordynamic model of a four-pocket, oil-fed, orifice-compensated hydrostatic bearing.

The available theoretical studies clearly indicate that the temperature rise of the lubricant fluid-film affects the performance of fluid-film hydrodynamic journal bearings quite appreciably. So it becomes an urgent need to do related experimental researches. In the present study, extensive experimental work has been conducted to determine the effects of supply pressure, load and speed on the thermohydrodynamic performance of a steady-state loaded hole-entry hybrid bearing.

2. Test Apparatus, Measurements and Bearing Specifications

An original experimental equipment is used in this paper to measure the bearing pressure and temperature profiles, as well as flow rate and steady-state shaft center position. The specific structure is illustrated in Fig. 2, and the photography of the whole test rig is shown in Fig. 3.

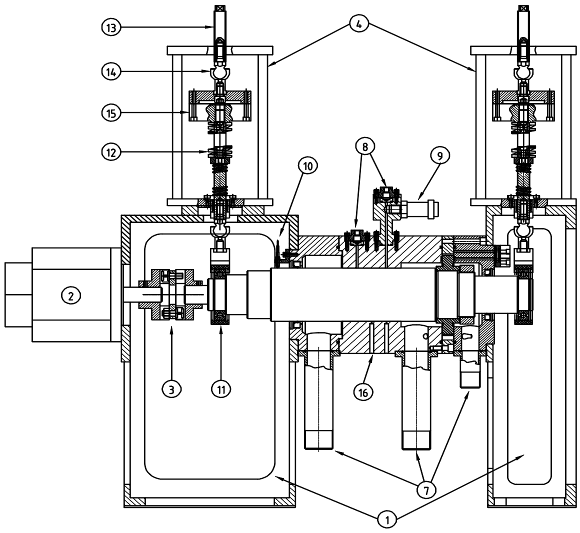

Figure 2. Diagram of the hole-entry hybrid bearing test rig. (1) Support structure, (2) Servo motor, (3) Cross rolling bearing sliding coupling, (4) Screw loading mechanism, (5) Tested bearing, (6) Spindle, (7) Oil return pipe, (8) Orifice restrictor, (9) Pressure transducer, (10) Displacement transducer, (11) Load rolling bearing, (12) Load spring, (13) Adjustment screw, (14) Boll joint universal coupling, (15) Weighing transducer, (16) Temperature transducer.

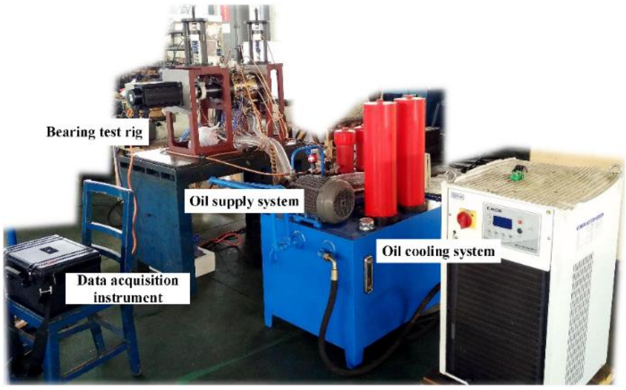

Figure 3. Photography of the hole-entry hybrid bearing test rig.

An orifice compensated hole-entry hybrid bearing with double row supply holes, each row 12 holes symmetrically, are tested. The bearing geometric and operating parameters are presented in Table 1. The bearing bush is fixed to both ends of the support structure (1), while the shaft is driven by servo motor through a self-aligning cross rolling bearing sliding coupling (3) to eliminate the rotor misalignment caused by the fluctuation of the spindle during the experiment. This structure is obviously different with that of Xu [6], and more close to the actual application of the bearing. As tested, the rotation errors of the spindle are within 1.5μm under different operating conditions, proving the validity of the test rig.

The data collected during the experiment include the flow rate of oil supply, the external load, the pressure of the oil inlet hole after restriction, the temperature distribution of the inner surface of the bearing and the steady shaft center position, successively corresponding to sensors such as flow rate transducer, weighing transducer, oil film pressure transducer, temperature transducer and displacement transducer.

For the loading device, in order to ensure the horizontal alignment of the spindle, the equivalent load at both ends of the spindle is imposed synchronously through hard compression springs. With adjustment screw (13) at the top of the support structure (1), the load can be changed through the compression of load spring (12). The load value can be displayed by the weighing transducer (15). Since the spindle is in a state of high speed rotation, the loading devices are in contact with the spindle through rolling bearings (11). In addition, considering that small oscillation of the spindle will occur in the test process, boll joint universal couplings are adopted in the load transfer structure to avoid restrictions of the shaft center position changes. The measurement range of the weighing transducer is up to 2T.

Table 1. Bearing operating and geometric parameters for the hole-entry hybrid bearing.

Parameter	Value
Journal diameter (D)	100 mm
Bearing length (L)	100 mm
External bearing diameter (D_2)	220mm
Radius clearance (c)	0.05 mm
Bearing land width (a_b)	25 mm
Bush material	Brass
Shaft material	Carbon steel
Supply pressure (P_s)	0~6MPa
Lubricant dynamic viscosity (μ_0)	1.806e-3 Pa·s at 40 ℃, 0.817e-3 Pa·s at 100 ℃
Number of restrictors per row (n)	12
Number of rows of holes in each bearing	2
Speed (N)	0~2200 rpm
Load (W)	0~500 Kg
Type of restrictors	orifice
Diameter of orifice (d_o)	0.5 mm

The measurement of oil flow is through a digital display type turbine flow transducer, which is installed at the outlet of the oil pump. According to the results of numerical calculation, the flow meter range is up to 20 L/min. For the oil film pressure must be measured by the contact-type sensor, the pressure sensor is installed in single row of oil supply holes after the orifice restrictors, so as to avoid the damage of the bearing inner surface and collect enough pressure points at the same time. Then the oil inlet pressure is measured to reflect the distribution of oil film pressure. The temperature distribution is nonuniform as that of the oil film, so it is necessary to collect enough data points to completely reflect the temperature distribution of the oil film. In this paper, contact type platinum resistance thermocouples are used, with temperature range of -30~150 ℃ and combined error of ±0.15 ℃. The distribution of the temperature sensor in expanded view is shown in Fig. 4. Eddy current displacement transducers are adopted here to measure the shaft center position changes, with the range of 0~0.5mm, and accuracy of 0.04um at 100Hz. In this rig, the two displacement transducers are installed orthogonally on the support structure, near the drive end of the shaft. All measurements were carried out in equilibrium state. Essential repeated experiments were carried out to verify the repeatability of the test results.

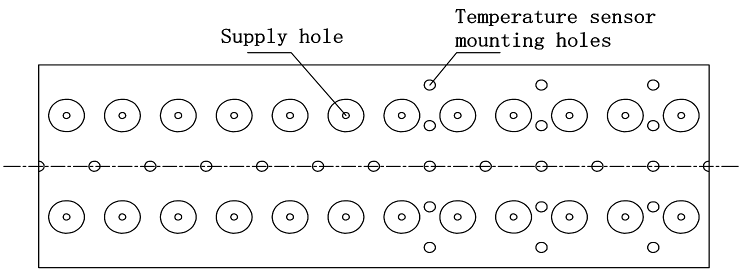

Figure 4. Location of thermocouple and pressure holes.

3. Results and Discussion

The experiment produced a large amount of data, only part of them are presented and analyzed here. Generally, the oil temperature was set at 30 ℃, with several contrast test at 25 and 35 ℃. In order to verify the influence of temperature rise on the bearing performance, transient experiment and steady thermal experiment were carried out in this paper. The difference between them lay in the control of

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temperature rise during the experiment. For the transient experiment, experimental data was collected before thermal equilibrium, and temperature rise was controlled within 1 ℃. While for the steady thermal experiment, the bearing working time must be long enough to allow the bearing to reach thermal equilibrium. The digital acquisition instrument was used to monitor temperature rise of the bearing. During both experiments, the temperature of supply oil could be set and adjusted automatically through the oil cooling system, and supply oil temperature range was controlled within ±0.5 ℃.

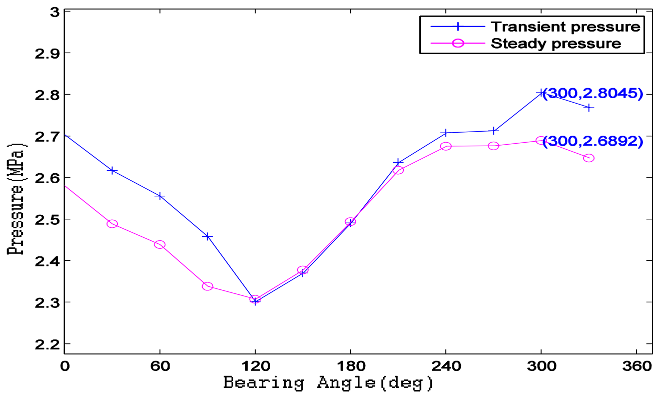

Figure 5. Pressure distribution of transient experiment and steady thermal experiment.

For operating condition of steady load 3KN, supply pressure 4MPa, speed 1000, 1500 and 2000 rpm, the pressure and temperature distribution of transient experiment and steady thermal experiment are shown in Fig. 5. It can be seen that oil film pressure of steady thermal experiment is lower than that of transient experiment. The drop of maximum oil film pressure can reach 4.11%. Data of flow rate and shaft center position were also collected at the same time, flow rate increased from 11.8 L/min to 12.8 L/min, an increase of 8.5%; eccentricity increased from 0.370 to 0.623, an increase of 68.4%. It is verified that the temperature rise of the oil film during the steady operation of the bearing has great influence on the bearing performance. Thus the steady thermal experiment must be carried out to better understand the actual working condition of the bearing.

Under the condition of load 1, 3, 5KN, supply pressure 2, 4, 5MPa, rotating speed 1000, 1500, and 2000 rpm, the pressure measurement results of steady thermal experiment are shown in Fig. 6. Relatively speaking, the effect of rotation speed on the oil film pressure distribution is not obvious. Under high rotational speed (2000 rpm), the oil film pressure is lower than that of low speed, which is caused by the temperature sharp increase at high speed. With the increase of oil supply pressure, the overall trend of the oil film pressure distribution is constant, only increasing in amplitude. The effect of external load on the oil film pressure distribution is obvious, oil film pressure distribution becomes more uneven with the increase of load, the difference between peak and valley values increases.

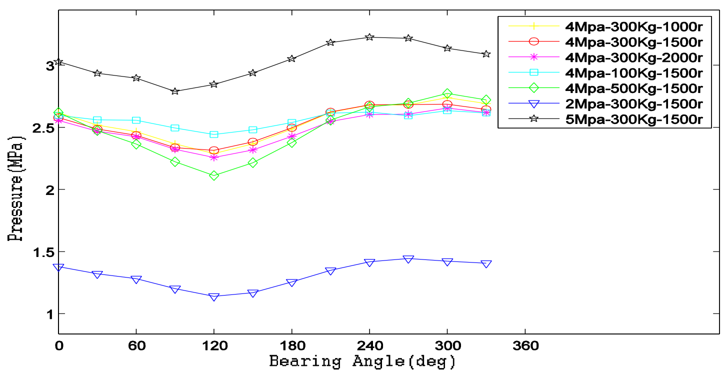

Figure 6. Pressure distribution of steady thermal experiment versus load, supply pressure and speed.

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When the steady load is 3 KN, the oil supply pressure is 4MPa, and the speed is 1500 rpm, the temperature and pressure measurement results of the steady thermal experiment are illustrated in Fig. 7 and Fig. 8. The radial temperature distribution compared to the pressure distribution is more complex, overall speaking the loading region has higher temperature; compared to maximum pressure, the maximum temperature moves forward the direction of rotation with 45°. The axial temperature distribution, as illustrated in Fig. 8, is relatively uniform as a whole, showing the trend of high on both sides and low in the middle, which can be explained by the bearing side leakage flow. The basic trend of pressure and temperature distribution is in accordance with the theoretical results of reference [1].

The temperature measurement results of steady thermal experiment are shown in Fig. 9. The steady-state load is 1, 3, 5 KN, the supply pressure is 2, 4, 5 MPa, and the speed is 1000, 1500 and 2000 rpm, with supply oil temperature 30 ℃. The temperature rise of the oil film can reach 6.8 ℃ under the test conditions. It is clear that the effect of speed on temperature rise is the most obvious. At low speed, due to the drop in loading capacity, the temperature rise of the loading region is more pronounced. The shear effect of the oil film increases at high speed, resulting in significant temperature rise in the whole bearing. When the oil pressure is low, the cooling effect of the lubrication oil is relatively weak, which leads to the heat accumulation in the loading region and the temperature rise is remarkable. With the increase of the oil pressure, the temperature rises of the oil film increases overall, but not obviously. With the increase of load, the temperature distribution of oil film is on the rise as a whole, and the oil film temperature rise in the loading region is apparently increased under the condition of heavy load (5 KN).

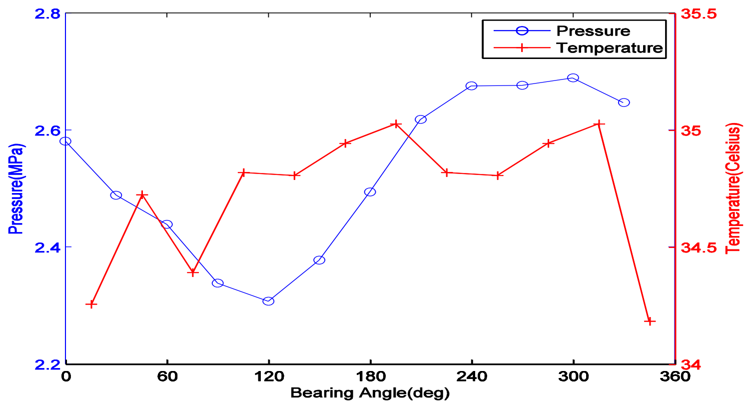

Figure 7. Radial pressure and temperature distribution of steady thermal experiment at load 3 KN, supply pressure 4 MPa and speed 1500 rpm.

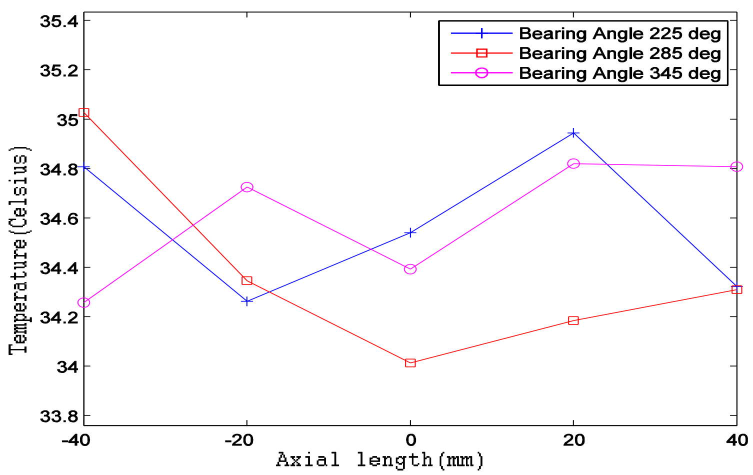

Figure 8. Axial pressure and temperature distribution of steady thermal experiment at load 3 KN, supply pressure 4 MPa and speed 1500 rpm.

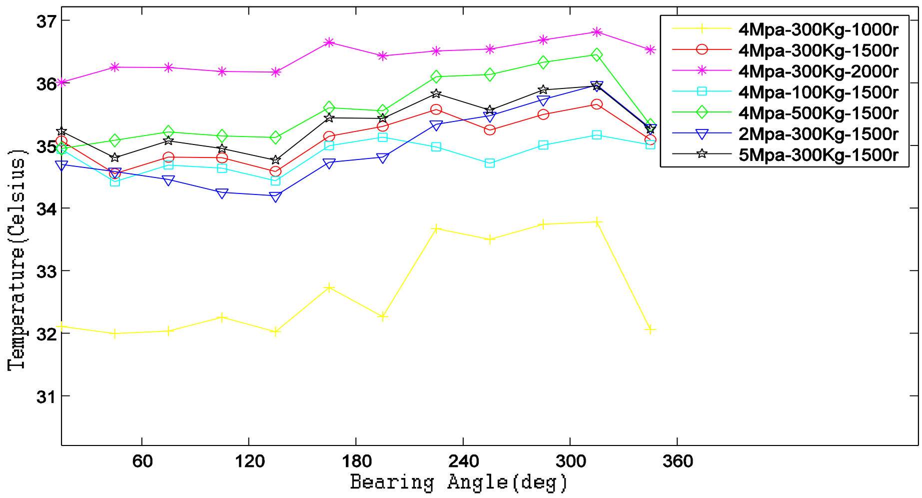

Figure 9. Temperature distribution of steady thermal experiment versus load, supply pressure and speed.

The load capacity of the bearing can be intuitively expressed through the change of the eccentricity ratio, which is illustrated in Fig. 10. The eccentricity ratio increases with the increase of load, and decreases with the increase of speed and oil supply pressure.

It needs to be pointed out that under the experimental conditions, the bearing bush is thick enough and directly contact with the outside space. The effect of heat dissipation is obvious, and the temperature rise is not high. So the influence of the thermal deformation of the bush and shaft is not considered here.

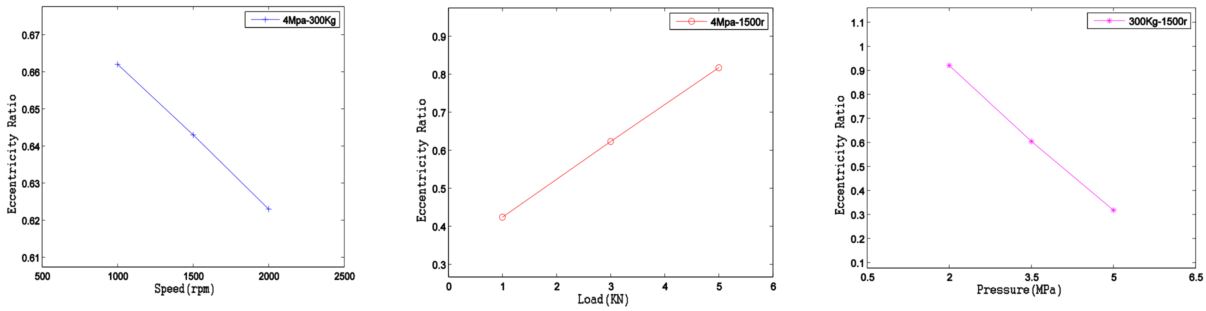

Figure 10. Eccentricity ratio of steady thermal experiment versus load, supply pressure and speed.

4. Conclusions

Experimental measurements of parametric experiments about the thermohydrodynamic performance of a steady-state loaded hole-entry hybrid bearing were performed. Experimental results indicating the effects of temperature rise on bearing performance for different speeds and loads had been presented. For the specific experimental operating conditions employed in this study, the following conclusions can be drawn:

(1) The influence of the oil film temperature rise on the bearing performance cannot be ignored. Bearing internal pressure of steady thermal experiment is lower than that of transient experiment. The drop of maximum oil film pressure can reach 4.11%.

(2) The shaft speed is the main factor affecting the temperature rise of the oil film. With the increase of oil supply pressure, the temperature rises of the oil film changed little, and the lubricating oil acted as the coolant to a certain extent. The oil film temperature rise in the loading region is obviously increased under the condition of heavy load.

(3) The eccentricity ratio, which illustrates load capacity, increases with the increase of load, and decreases with the increase of speed and oil supply pressure.

Acknowledgements
This research is supported by the National Natural Science Foundation of China (NSFC) No. 51875223, Science Challenge Project (TZ2018006-0102-03) and China Postdoctoral Science Foundation (2017M622427).

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