Experimental Study on Oil Pressures of Hydrodynamic Lubrication in Thrust Bearing Considering Circumferential V-Grooved Pad

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Abstract—This paper studies experimentally the oil pressures on the pad of thrust bearings under hydrodynamic lubrication. The smooth pad and the pad with circumferential grooves surface were considered. The configuration of a groove is V shape. The axial loads were applied to thrust bearings by the pneumatic system that controlled by the electronic board. The pressure transducer was used to measure the oil pressure distributions at the contact area. The oil pressures with varying both axial loads and runner velocities were presented in this paper. The hydrodynamic lubrication model for the smooth surface was used to predict the suitable condition for the testing bearing. The experimental results showed that the oil pressure distribution is affected by the axial load and angular velocity. The oil pressure, especially at the lower angular velocity and higher load, reduces when using the pad with circumferential v-grooves.

Index Terms—Hydrodynamic, thrust bearing, oil pressure, experiment, circumferential grooves

I. INTRODUCTION

Thrust bearings are widely used in rotating machinery for supporting the axial loads. The oil lubrication under the sliding surface, usually forming pressure distributions between a thrust and a pad surfaces, is significant to prevent material wear. Therefore, it is necessary to study the characteristics of oil pressure in the thrust bearing. The oil lubricant with high pressure leads to high friction at the bearing surface. For some machine component, reducing oil pressure is an advantage in preventing material wear. Usually, the experimental method is an alternative to analyze the oil pressure in the contact region. The previous research works considering the experimental study on the fluid film lubrication in a thrust bearing can be summarized as follows. Heinrichson et al. [1] measured the pressure and oil film thickness profiles for the tilting-pad thrust bearing. They proposed the effect of an oil injection pocket on the pressure profile and oil film thickness with considering the smooth pad surface area 100 cm^2. Almqvist et al. [2] investigated the running temperature, power loss, and oil pressure distributions for the pivoted pad thrust bearing. Those parameters were measured and presented as the function of load and rotational speed. Glavatskikh [3] reported measurements of oil film temperature and frictional torque which included in a thrust bearing test rig. Hanka et al. [4] considered the water contamination in the tilting-pad thrust bearing lubrication. The ultrasonic method was applied to measure the oil film in a full-size PTFE-faced thrust pad [5]. Annan et al. [6] proposed the static and dynamic behaviors of tilting-pad thrust bearings. They used the stepper motor to drives a ball screw for applying the axial loads.

This paper applies the pneumatic system controlled by the electronic board for obtaining the axial loads. The pressure transducers were adapted to measure oil pressures. Three positions on a pad were applied to investigate oil pressure behavior.

II. EXPERIMENT APPARATUS

Fig.1 shows the experiment apparatus for studying the oil pressure in the thrust pad under varying axial loads and runner velocities. The experimental apparatus consists of the rotating shaft, the belt, the pneumatic cylinder, the runner plate, the test bearing, the chamber containing oil lubricant, and the pressure gage for controlling an axial load. The load is determined by measurement of the air pressure in the pneumatic system. The rotating shaft and the runner are derived by the belt connecting with the AC motor. By using the inverter, we can control the rotational speed. In order to study the effect of axial load on the oil pressure in the contact region between the runner surface and the pad surface, the pneumatic system is used to generate the axial load. The load is determined by measurement of the air pressure in the pneumatic system. The test bearing is mounted on a steel base that supported by the pneumatic cylinder. The air compressor generates the air pressure in a pneumatic cylinder. When the air pressure reaches the set pressure, the test bearing with a steel base moves up to compress the runner plate.
In order to measure the oil pressure lubricated between a runner surface and a pad bearing surface, the pressure sensors are mounted under the pad surface. Fig. 2 shows the installation of a pressure sensor. The pad bearing surface is drilled for providing the plastic tube into the pad. The small hole diameter is 4 mm. Therefore, the pressure sensor, connecting with a plastic tube, can measure the oil pressure. The electronic board gets the oil pressure data from the sensor. Then the computer records these data respectively. Fig. 3 shows the positions of measuring the oil pressures for both the smooth and grooved pads. The positions at A, B, and C are at the curve of the middle pad bearing with the radial $r_m$ (27.5 mm) as seen in Fig. 3(b). While the positions of A, B, and C correspond to the pad angles 15°, 30° and 45° degrees respectively. The type of oil lubricant is SAE 90. The effect of temperature is neglect for this study due to the testing spends a short time for each experiment.

Fig. 4 shows the example of oil pressures at the measured positions. For the time range from 0 to 5.0 second, the thrust bearing was operating without applying axial load. After that, the axial load performs on the bearing from 5.0 to 25 second for studying the oil pressures. Therefore, the average oil pressure in this period was representative to describe the behavior of fluid film lubrication into a contact region between the runner and pad surfaces. The test bearing would stop when the time was more than 25 second.
III. HYDRODYNAMIC LUBRICATION MODEL FOR SMOOTH PADS

In order to figure out the suitable test bearing conditions, the oil pressures measured at positions as mentioned above compared to the numerical results for the smooth surface. The theory of fluid film lubrication in a thrust bearing has briefly described in this section. The hydrodynamic lubrication model (HL-model) in dimensionless form, used to predict the oil pressure profile between a runner and pad surfaces, is presented in (1).

\[
\frac{1}{R} \frac{\partial}{\partial \theta} \left( \frac{R^3}{R^2} \frac{\partial P}{\partial \theta} \right) + \frac{\partial}{\partial R} \left( \frac{R^3}{R^2} \frac{\partial (PR)}{\partial R} \right) = \left( \frac{6 \mu \sigma \theta_0}{h_0^2 P_0 R} \right) \frac{\partial h}{\partial \theta} \quad (1)
\]

The dimensionless parameters are expressed as

\[
P = p / p_0, \quad h = h / h_0, \quad R = \frac{r - r_i}{r_o - r_i} = \frac{\theta}{\theta_0}, \quad \theta_0 = \pi / 3
\]

where \( p \) is the oil pressure, \( h \) is the oil film thickness, \( h_0 \) is the minimum oil film thickness, \( \omega \) is the angular velocity of runner, \( r \) is the pad radius, \( r_i \) is the inner pad radius, \( r_o \) is the outer pad radius, \( \theta \) is the pad angle, and \( \mu \) is the oil viscosity respectively. The runner surface separates from the pad bearing surface because the oil pressure and the axial load is in an equilibrium condition. The balance force equation can be expressed as

\[
\frac{w}{\mu \sigma h_0^2 P_0} = \int_{\theta_0}^{\theta_0} \int_{r_i}^{r_o} PRdRd\theta
\]

The derivatives in (1) can be approximated by the finite difference method. The oil pressure profile in dimensionless form is obtained by the Newton-raphson iterative scheme. The numerical results are about to be convergence criteria when the oil pressure and load are adopted as follows:

\[
\begin{align*}
\varepsilon_1 &= \sum_{j \neq j'} \sum_{i \neq i'} P_{ij} - P_{ij'} \left| \sum_{i} \sum_{j} P_{ij} \leq 1 \times 10^{-3} \right. , \\
\varepsilon_2 &= \sum_{j \neq j'} \sum_{i \neq i'} P_{ij} \frac{R_{ij} \Delta R \Delta \theta w}{h_0^2 P_0} - \frac{w}{\mu \sigma h_0^2 P_0} \leq 1 \times 10^{-3}
\end{align*}
\]

The example of numerical calculations presenting the oil pressure profiles under varying the axial load was shown in Figs.5(a) to (c). The number of nodes in a radial axis considering one pad is 50 (\( m=50 \)). The number of nodes in an angular axis is 256 (\( n=256 \)). As found in numerical calculations, the oil pressure profile is not symmetry, and the maximum oil pressure appears in the outlet of the contact region. The axial load affects significantly the oil pressure that is the oil pressure increase under increased axial load. Because the pressure sensor is not available to use for high pressures, it is necessary to find out the suitable condition for the testing bearing. Therefore, comparing the experimental results and the numerical results for in case of smooth pads is necessary to decide for the testing condition such as the axial load, and the angular velocity.

The test bearing for this paper is aluminum. The inner pad radius (\( r_i \)) is 0.015 m, the outer pad radius (\( r_o \)) is 0.04 m, and the pad angle (\( \theta \)) is 60° degrees, the number of pads is 5 respectively. The oil lubricant is SAE 90, and the oil viscosity obtained from the kinematic viscometer is 0.08002 Pa.s. The case of grooved pads, the width

IV. RESULTS AND DISCUSSIONS

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grooved \((w_g)\) is 4.0 mm, and the groove depth \((d_g)\) is 3.0 mm.

From comparing the numerical results to the experimental result as found in Fig. 6, the suitable condition for testing bearing can be as follows: the axial load is from 260 N to 433 N, and the runner angular velocity is 157 rad/s to 283 rad/s. For numerical results, the oil pressure increases gradually from the pad angles of 0 to 35\(^\circ\) degrees after that the oil pressure increases immediately from the pad angles of 35\(^\circ\) to 55\(^\circ\) degrees. The peak pressure appears at the pad angle of 55\(^\circ\) degrees. Therefore, the positions measured the oil pressure in this test condition are not the position explained the peak pressure. However, the positions chosen are sufficient to study the tendency of oil pressure in the thrust bearing both the smooth pad and the grooved pad. Figs. 7(a), (b), and (c) show the average oil pressure at positions A, B, and C under considering axial load changed while the angular velocity is 157 rad/s. The results show the comparisons between the smooth pad and the v-grooved pad. Both the smooth and v-grooved pads as shown in Figs. 7(a) to (c), while the axial loads increase, the oil pressures at all positions also increase. Considering the grooved surface with V shape, the oil pressures, measured all positions, are lower than the case of the smooth surface.

Because the velocity of a runner surface is low for this study, the oil film thickness also reduces, and the oil lubricant leaks from the contact region to beside pad under the axial load. Therefore, the pad with the circumferential v-grooves provide the gaps where the oil

Figure 6. Comparison of oil pressures at the central pad \((r_m)\) between the experiments and the calculations at \(\omega=157 \text{ rad/s}\)

Figure 7  Average oil pressure under varying axial loads at \(\omega=157 \text{ rad/s}:\) (a) position A \((\theta=15^\circ)\), (b) position B \((\theta=30^\circ)\), and (c) position C \((\theta=45^\circ)\)

Figure 8  Average oil pressure under varying axial loads at \(\omega=283 \text{ rad/s}:\) (a) position A \((\theta=15^\circ)\), (b) position B \((\theta=30^\circ)\), and (c) position C \((\theta=45^\circ)\)
The oil pressures on the pad bearing under considering the circumferential groove with V shape and the smooth pads were experimentally studied. The main conclusions can be summarized as follows:

1) Using the pad with circumferential v-grooves is effective an alternative for reducing the oil pressure especially in the case of low runner velocity.
2) Both of the axial load and angular velocity impacts the oil pressure significantly. The oil pressure increases when the axial load increases and the angular velocity decreases.

CONFLICT OF INTEREST
The authors declare no conflict of interest.

AUTHOR CONTRIBUTIONS
Puttha Jeenkour and Kittipong Boonlong conducted the research, analyzed the data and wrote the paper. All authors had approved the final version.

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Figure 9 Oil pressure profiles under varying runner velocities for smooth and groove pads (load = 833N)

V. CONCLUSIONS

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