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Experimental study on the effects of oil groove location on temperature and pressure profiles in journal bearing lubrication ☆

Mohamad Ali Ahmad [a,*], Salmiah Kasolang [a], R.S. Dwyer-Joyce [b]

[a] Faculty of Mechanical Engineering, Universiti Teknologi MARA, Malaysia
[b] The Leonardo Centre for Tribology, Department of Mechanical Engineering, University of Sheffield, Sir Frederick Mappin Building, Mappin Street, Sheffield, UK

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ABSTRACT

In the present study, an experimental work was conducted to determine the effect of oil groove location on the temperature and pressure in hydrodynamic journal bearings. A journal with a diameter of 100 mm and a length-to-diameter ratio of ½ was used. The oil supply pressure was set at 0.20–0.25 MPa. The groove was positioned at seven different locations, namely -45°, -30°, -15°, 0°, $+150$, $+30^{\circ}$ and $+45^{\circ}$. Measurements of temperature and pressure were obtained for speeds of 300, 500 and 800 rpm at different radial loads. Changes in oil groove location were shown to affect the temperature and pressure to some extent.

1. Introduction

A single groove for a plain journal bearing is common in industrial applications. The groove is used to distribute oil over the length of the journal and to improve the temperature field. Oil enters the groove through an oil supply hole and flows either by gravity or under pressure. The oil supply conditions (pressure, temperature, groove dimensions and location) influence the flow rate. Theoretically, the conditions will affect the oil temperature inside the bearing as well. When the temperature changes, the viscosity is altered, subsequently affecting the film thickness.

In a study regarding the effect of groove location and supply pressure on the THD performance of a steadily loaded journal bearing, Costa et al. found that locating the groove at 30° led to a reduction in maximum temperature [1]. In this study, the length-to-diameter ratio was 0.8, and the lubricant type was ISO VG 32. In another study, Majumdar and Saha [2] observed that the maximum temperature occurred near the position of minimum film thickness. The authors also concluded that thermal effects on journal bearing performance cannot be neglected and that the assumption of an isothermal lubricant is thus inadequate for evaluating bearing performance. This effect becomes critical in the case of high speeds and loads.

The pressure profile in a journal bearing can be predicted using the Reynolds equation [3,4]. Many experimental and theoretical studies have been conducted to predict the pressure profiles of journal bearings. Wang and Khonsari [5,6] used an analytical solution and static performance to study the effect of oil inlet pressure and the position of an axially grooved oil supply hole. Previous studies by the authors regarding pressure profiles [7] were described, and the experimental values obtained were compared to theoretical profiles resulting from the charts of Raimondi and Boyd [8]. It was also observed that changes in the oil inlet pressure tend to affect the maximum pressure [9].

In the present study, extensive experimental work has been conducted to determine the effect of oil groove location on temperature and pressure profiles in hydrodynamic lubrication around a journal bearing.

2. Background

2.1. Oil groove supply

In hydrodynamic analysis, the oil supply is assumed to be available to flow into the bearing at least as fast as it leaks out. In this study, oil was fed into the system by an oil supply hole and groove. Ideally, the groove should be as long as the bearing, but this would cause all the lubricant to leak from the sides of the groove [10]. In this experimental

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* Correspondence to: Faculty of Mechanical Engineering, Universiti Teknologi MARA, 40450 Shah Alam, Selangor, Malaysia.
Tel.: +60 12 3319244; fax: +60 3 55435160.
E-mail address: alie_76_02@yahoo.com (M.A. Ahmad).

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study, a short angle groove type was used. The lubricant oil supplied to the bearing was pressurised. A pressurised lubricant supply can reduce lubricant heating and viscosity loss; it also prevents shaft-to-bush contact during starting and stopping and modifies vibration stability. Costa et al. [1] reported that an increased oil supply pressure can reduce the operating temperature and increase the maximum circumferential hydrodynamic pressure. This result is consistent with the findings reported in [11], which concluded that the oil supply pressure and the geometry of the feed control determine the cooling effects.

2.2. Temperature in a journal bearing

Temperature monitoring is a well-established technique for detecting overheating and preventing hydrodynamic bearing damage [12]. As reported by Moreno et al. [13], a significant number of numerical studies have focused on the steady-state temperature in journal bearings. In their study, the primary difficulty was associated with the exponential dependency of the viscosity on temperature. Another previous study on temperature distributions in journal bearings demonstrated that the load capacity is generally less than that predicted by classical isothermal theory [14].

An effective temperature is commonly used to calculate the effective viscosity in operating journal bearings. The effective temperature, T_{eff}, can be calculated as

$$T_{\text{eff}} = T_{\text{in}} + \Delta T/2 \tag{1}$$

where T_{in} is the input temperature and ΔT is the temperature rise.

In this study, the validity of using an effective temperature was investigated for various oil groove positions.

2.3. Pressure in a journal bearing

The pressure in the bearing can be plotted by solving the Reynolds equation [3,4]. This differential equation governs the pressure distribution in fluid film lubrication using an incompressible fluid, as shown in Fig. 1.

From this differential equation, parameters such as the geometry of the surface, the relative sliding velocity, the properties of the fluids and the magnitude of the normal load can be determined. In this study, the ratio of the bearing length L over the bearing diameter D (L/D) is equal to 0.5. From this value, the Sommerfeld number was calculated as [15]

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} \tag{2}$$

where μ is the viscosity (Pa s), N is the speed (rps), r is the journal radius (m), c is the radial clearance (m) and P is the radial load per unit of projected bearing area (N/m^2).

Bearing number, S calculated from Eq. (2), was used to obtain the predicted values of eccentricity ratio, friction coefficient, maximum film pressure, position of maximum film pressure and position of minimum film thickness from the charts of Raimondi and Boyd. These predicted values are used for validation purposes with the following assumptions:

- The flow is isothermal.
- The surfaces are smooth.
- The fluid is Newtonian and the flow is laminar.

3. Apparatus

The journal bearing test rig used in this study to characterise the temperature and pressure profiles is shown in Fig. 2. A journal with a 100-mm diameter and a length-to-diameter ratio of 1/2 was used. The bearing piece was modified to fix 12 thermocouples and 12 pressure transducers around the journal bearing circumference at 30° intervals, as shown in Fig. 3. The journal was then mounted horizontally into the bearing. A pneumatic bellows was used to apply the required load. The maximum speed of the journal test rig was 1000 rpm, and the speed used for testing was 300, 500 and 800 rpm.

A single groove, 40 mm long, 10 mm wide and 5 mm deep, was used in this study. During the tests, the journal bearing was run at different loads (10 and 20 kN). The oil supply groove was positioned at −45°, −30°, −15°, 0°, 15°, 30° and 45°. Details regarding the test bearing dimensions, lubricant properties and operating parameters are given in Table 1. The pressure transducers measured the fluid pressure developed through holes bored to within 0.5 mm from the bearing surface [12,16]. These holes are generally called as pressure taps. Three additional thermocouples were installed to measure the room temperature as well as the lubricant inlet and outlet temperatures. The oil inlet supply pressure was regulated using a power pack lubrication system and was maintained between 0.2 to 0.25 MPa throughout the experiments. These inlet pressures were monitored using a PSAN-L1CPV digital pressure sensor.

4. Results

Experimental results for the temperature and pressure profiles are plotted in Figs. 4–15. Fig. 4(a) shows temperature profiles for speeds of 300, 500 and 800 rpm at loads of 10 and 20 kN at different groove locations.

The groove position has been defined to be positive to the right side of the vertical line passing through the centre of the journal and negative to the left as shown in Fig. 3. The groove positions were set at −45°, −15°, 15° and 45° for all cases. Fig. 4(b) shows the corresponding temperature profiles for groove positions of −30°, 0° and 30°. The Sommerfeld number was calculated using Eq. 2. From this bearing characteristic number, the minimum film thickness position was obtained from the Raimondi–Boyd chart and is indicated in each figure.

The effective temperature was calculated using Eq. (1) based on the inlet and outlet temperatures recorded when the oil supply

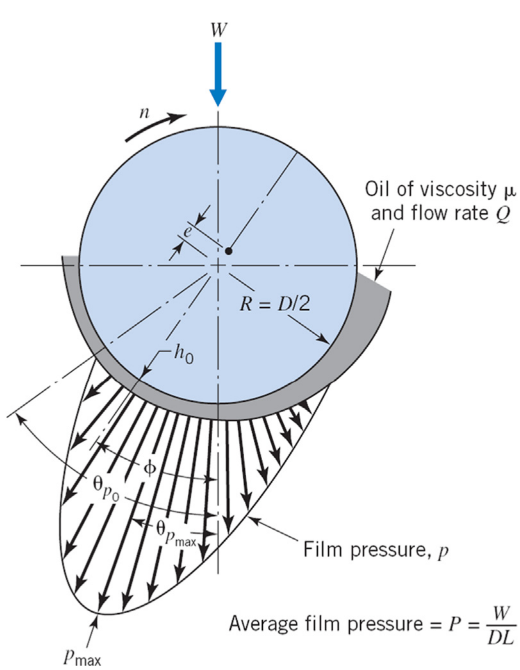

Fig. 1. Pressure distribution schematic, adapted from [15].