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Experimental evidence of a two-axial groove hydrodynamic journal bearing under severe operation conditions

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ABSTRACT

In this study, an experimental investigation of the influence of the applied static load and rotational speed on the behaviour of a 160-mm diameter cylindrical journal bearing with two axial grooves was conducted. The rotational speed ranged from 66 rpm to 1440 rpm, and the applied static load varied from 0 kN to 350 kN in the vertical direction. The profiles of the pressure and the oil-film thickness during the shaft rotation have been measured by one proximity probe and one pressure probe installed in the rotating shaft. Measurements of the shaft centre position, dynamic coefficients, hydrodynamic pressure, temperature distributions on the bearing, film thickness, and bearing deformation under several operating conditions are presented and discussed.

1. Introduction

The installation of oil-film plain journal bearings in industrial machines remains a good choice because of their simplicity and low cost compared to rolling element bearings or oil-film TPJBs-Tilting-pad Journal Bearings. Typical applications include reciprocating machines with small diameter shafts, operating at high speeds, where the high dynamic behaviours offered by other bearing types are not required. Critical applications are represented by machines with medium/large diameter shafts, operating at very low Sommerfeld numbers and characterized by both low tangential speeds (less than 1 m/s) and high loads (specific pressure higher than 10 MPa). High viscosity oils are usually employed in these cases (ISO VG grade higher than 100).

In this condition, the so-called mixed or partial lubrication occurs, and the use of simple models, such as hydrodynamic (HD) or thermo-hydrodynamic (THD) during the design phase could lead to the oversizing of the bearing, mainly because of the overestimation of the maximum value of the oil-film pressure. Thermo-elasto-hydrodynamic models (TEHD), improve the prediction of the pressure distribution by considering the bearing deformation under large loads.

The thermal behaviours of a cylindrical journal bearing have been widely studied for several decades. THD and TEHD models as well as experimental activities have been performed to fully understand the behaviour of oil-lubricated [1–10], gas-lubricated [11–13], water-lubricated [14–18], or hybrid [19–21] journal bearings.

Brito et al. [22] studied the behaviour of a journal bearing as a function of the rotational speed and applied static load. The test bearing had a diameter of 100 mm with two axial grooves placed at $\pm 90°$ with reference to the direction of applied static load. The maximum operating conditions were 10 kN for the load and 4000 rpm for the rotational speed. They stated that the journal temperature, the maximum bush temperature and the flow-rate increased linearly with the increase in the rotational speed. In particular, the rising of the static load leaded to the increase of the temperature in the loaded part and the decrease of the temperature in the unloaded part of the bearing, but it did not appear to significantly affect the temperature of the rotating shaft and the oil outlet.

Wang and Khonsari [23,24] used an analytical solution to study the impact of oil inlet pressure and the position of an axially grooved oil supply hole. The Reynolds-Floberg-Jakobsson boundary conditions were assumed to recognize the starting position of the cavitation, the reformation of oil film at the end of cavitation, and the effect of oil inlet pressure and inlet position. It was discovered that the oil inlet pressure had a pronounced impact on the oil film configuration and pressure distribution. With increasing oil inlet pressure, the cavitation region shrinks and the peak hydrodynamic pressure increases.

Gardner [25] determined experimentally the conditions under which the transition from full hydrodynamic film to boundary lubrication occurred in journal bearings. He performed the tests with a two grooves sleeve type and a TPJB for 13″ diameter shaft. He concluded that the lower limit of hydrodynamic lubrication corresponds to the

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Nomenclature

c_{xx}, c_{xy}, c_{yx}, c_{yy}	dynamic damping coefficients (N s/m)
c	radial clearance (m)
D	diameter of bearing (m)
h	oil-film thickness (m)
k_{xx}, k_{xy}, k_{yx}, k_{yy}	dynamic stiffness coefficients (N/m)
L	bearing length (m)
ΔF_x^{oil}, ΔF_y^{oil}	oil-film forces (N)
ΔX, ΔY	relative displacements between shaft and housing support (m)
X, Y	coordinates of the shaft centre (m)
Ω	rotational speed of the shaft (rad/s)
ω	force frequency (rad/s)

Sommerfeld number between 0.005 and 0.006. By experimental tests, he detected that the definite movement of the Babbitt metal (wiping) will occur when the applied static load is above 750 psi and only light polishing of the Babbitt developed in the matter of the applied load is up to 300 psi.

The effect of oil groove location was studied in the experimental work conducted by Ahmad et al. [26] for a 100 mm-diameter journal bearing. The groove was positioned at seven different locations, namely −45°, −30°, −15°, 0°, +150, +30° and +45°. It was discovered that the changes of the oil groove location affected the temperature and pressure to some extent. A THD was considered in [27,28] for the effect of the oil groove location.

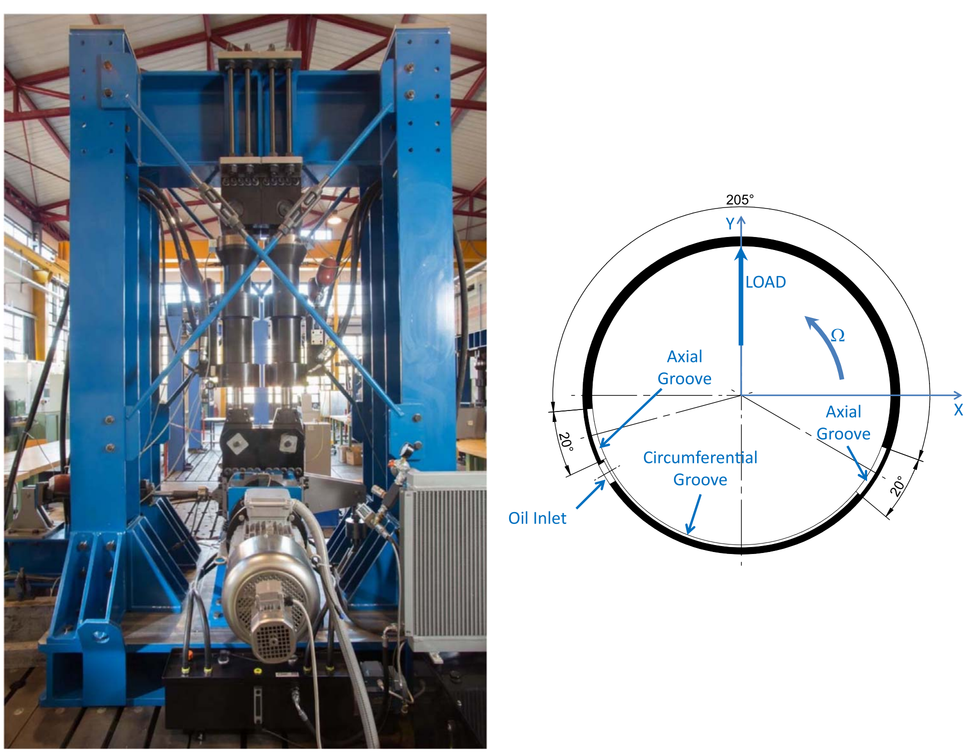
In [29] and [30], the mixed lubrication of a small bearing running at low rotational speed from 2 to 500 rpm with different oil temperatures and applied loads were investigated. In the mixed lubrication regime, the higher lubricant temperature leads to a large friction coefficient. The Sommerfeld number is less than 0.002 in this condition.

Allmaier et al. [31,32] extended a TEHD model from an EHD model to study the differences arising from local temperatures in the journal bearings. With suitable thermal boundary conditions, it should be noted that the TEHD model is able to forecast the occurring temperatures at some different points of the test-rig with very high accuracy. In [33] and [34], the friction and mixed lubrication regime in journal bearings were investigated numerically and experimentally for a large range of operating conditions, lubricants (SAE10, SAE20, SAE30 and SAE40), and rotational speeds from hydrodynamic lubrication regime to mixed lubrication regime with metal-metal contact.

In the present study, extensive experimental work has been conducted to determine the static and the dynamic behaviour of a plain journal bearing working at different operating conditions, namely static load and rotational speed. For confidential reasons, all data have been omitted as well as some results have been normalized with reference to the corresponding maximum value.

2. Test rig and bearing description

The test-rig for characterization of the plain journal bearing is illustrated in Fig. 1a. A 15.0 kW electric motor drives the rigid shaft by through a flexible coupling. The motor speed is adjusted by a frequency controller and can reach a maximum value of 1465 rpm. The shaft speed can also be reduced by installing the additional gearbox with a gear ratio of 1:9. The vertical static load is applied on the top of the bearing case through two hydraulic actuators placed in the vertical direction and working in parallel with the maximum force of 400 kN. This load corresponds to the very high specific pressure of approximately 17 MPa compared to the common values of approximately 2–3 MPa adopted in the industrial field. The vertical static load acting on the housing support is in the downward direction corresponding to a load applied on the shaft in the upward direction. The test rig is also equipped with a hydraulic actuator (maximum force equal to 20 kN) placed in the orthogonal direction (horizontal) respecting to the two



(a) (b)

Fig. 1. Test-rig from the DE side view (a) and sketch of bearing under test from NDE view (b).