2000 AIMETA International Tribology Conference, September 20-22, 2000, L'Aquila, Italy

Experimental Study of the Influence of Changes in Load Direction on the Performance of a Crown Bearing

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ABSTRACT

Laboratory tests have been carried out in order to investigate the influence of small changes in load direction on the performance of a steadily loaded hydrodynamic crown journal bearing. Hydrodynamic pressures, temperature distribution on the bush internal surface, shaft temperature, oil flow rate and bush torque were measured for fixed sets of operating conditions, at three different groove locations (ϕ) with respect to load line of load (ϕ= 0°, -30°, +30°). For each groove location quantitative information is provided concerning the influence of applied load and shaft rotational speed on the performance characteristics. Changing the location of the groove around the load line did affect flow rate and bush torque, but had only a slight or moderate effect on bush maximum temperature.

NOMENCLATURE

- a : Length of supply groove (m)
- b : Bearing length (m)
- C_d : Diametrical bearing clearance at 20°C (m)
- d : Bearing nominal diameter (m)
- D : Bush outer diameter (m)
- N : Shaft speed (rpm)
- P_s : Oil supply pressure (kPa)
- T_a : Laboratory ambient temperature (°C)
- T_s : Oil supply temperature (°C)
- W : Load (kN)
- w : Width of supply groove (m)
- Ψ : Attitude angle (degrees)
- α : Angle measured from the middle of the groove (degrees)

ϕ : Groove location in relation to the line of load (degrees)
μ_{35} : Oil viscosity at 35 °C (Pa.s)
μ_{70} : Oil viscosity at 70 °C (Pa.s)

INTRODUCTION

Nowadays, hydrodynamic bearings are extensively used to support high speed rotating shafts with high specific loads. Under these operating conditions, they can experience a significant variation in oil film temperature due to viscous dissipation. Furthermore, the supplying conditions (supply pressure, supply temperature and groove dimensions and location) dictate the flow rate, affecting the oil temperature inside the bearing. As a consequence, oil viscosity and bearing characteristics such as power loss, minimum film thickness and load carrying capacity will be affected.

In hydrodynamic journal bearings working on a steady state regime, oil can be supplied to the bearing trough one axial groove on the line load, allowing for shaft rotation in both directions. Very often in practical applications, slight changes in load direction may eventually occur. This situation will affect clearance at the inlet, affecting in this way the oil flow supplied to the bearing and, therefore, the bearing performance.

Experimental investigation on journal bearing performance has been carried out over many decades [1]. Recent experimental works [2, 3] have pointed out the need for experimental investigation on the effect of oil supply conditions on the thermohydrodynamic bearing performance. Furthermore, thermal effects in hydrodynamic bearings have been the subject of many experimental studies. However, little information is available on combined experimental results of pressure and temperature distributions on crown bearings [2, 4].

This work aims to contribute to a more clear view of the effect of slight changes in load direction on bearing performance and to add results to the stock of thermohydrodynamic data currently available for journal bearings. The influence of load direction on hydrodynamic pressure and temperature distributions over the inner surface of the bush and on some bearing performance characteristics is discussed.

TEST RIG, MEASUREMENTS AND BEARING SPECIFICATIONS

The test rig used had the specific aim of making precise measurements of bearing operating parameters (applied load and shaft speed), and bearing performance parameters (hydrodynamic pressure, bush temperature, shaft temperature, flow rate and bush torque). A schematic representation of the experimental test rig is shown in Fig. 1. Full construction details may be found elsewhere [5]. In the following only the specifications necessary for the interpretation of the experimental results will be given.

The main shaft is driven by a variable speed DC motor (21 kW) by means of a belt multiplying transmission being supported on three preloaded high precision ball bearings, whose mounting arrangement provides proper stiffness as well as high rotational accuracy. Adjusting a controller to the motor shaft speed could be altered.

The loading system is a pneumatic cylinder where high pressure air is applied, being the load intensity calculated by knowing the internal air pressure, the cylinder cross section and the weight of the bush and its appendages.

Oil flow rate to the test bearing was measured using a flowmeter calibrated for the expected range of oil temperature. Oil supply pressure was indicated by a Bourdon type gauge at entry and was controlled by means of a pressure limiting valve. Oil supply temperature was controlled by a water-oil heat exchanger and was measured just before entering the supply groove.

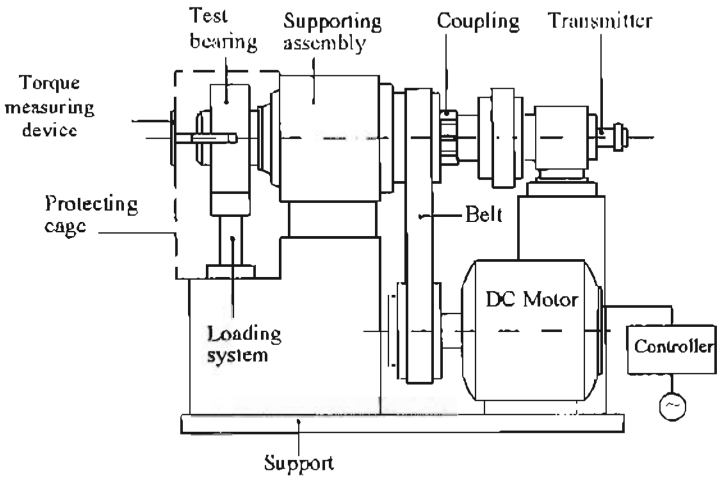
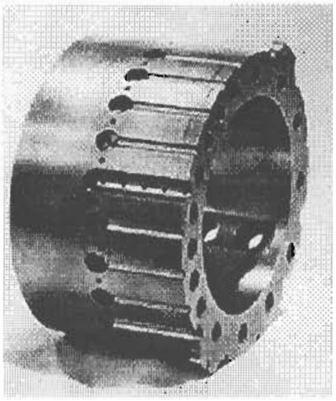


Figure 1. Schematic representation of experimental test rig

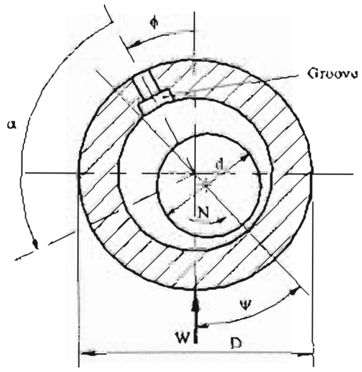
Bush temperatures were measured using type K thermocouples. Table 1 shows the circumferential locations of the thermocouples in the bush inner surface, at the bearing mid-plane. Shaft temperature was measured by three type J thermocouples that were mounted in the shaft and connected to a mercury transmitter. The hydrodynamic pressure distribution was measured in the bearing mid-plane using fifteen holes (Table 1) drilled through the bush. Each pressure hole was connected via a manifold valve to a high precision Bourdon type gauge manometer. The friction torque in the bush was measured using a torque-measuring device mounted on a two-armed assembly connected to the bush.
The test bush (Fig. 2(a)) was made in bronze having one axial groove. The bearing geometry is indicated in Fig. 2(b) and the geometric details are listed in Table 2. Using always the same bush, changes in load direction with respect to the groove location could be attained during the bush mounting by fixing the angle ϕ. Angular locations of the groove corresponding to ϕ equal $-30°$, $0°$ and $+30°$ have been used.
Details concerning measurements, procedure and uncertainty analyses (Table 3) have been described elsewhere [2]. Oil characteristics and test operation conditions are listed in Table 2. Data for each test condition were recorded after achieving thermal equilibrium in the bearing.

Thermocouples location (degrees)							
12.5	45	75	105	135	150	165	180
195	210	225	240	260	285	315	347.5
Pressure holes location (degrees)							
30	60	90	120	142.5	157.5	172.5	187.5
202.5	217.5	232.5	247.5	270	300	330	

Table 1 Circumferential location of thermocouples and holes for pressure measurements in the mid-plane of the bush (Angle origin at the middle of the groove)



(a)



(b)

Figure 2. Photography of bush without instrumentation (a) and bearing geometry (b)

Bearing diameter	d	m	0.10
Bearing length	b	m	0.08
Bush outer diameter	D	m	0.20
Diametrical clearance at 20°C	C_d	10^{-3} m	0.187
Length of supply groove	a	10^{-3} m	70
Width of supply groove	w	10^{-3} m	16
Groove location	ϕ	degrees	-30, 0.0, +30
Speed	N	rpm	2000 ... 4000
Load	W	kN	2 ... 8
Oil supply pressure	P_s	kPa	140
Oil supply temperature	T_s	°C	35
Lubricant type	-	-	ISO VG 32
Oil viscosity at 35 °C	μ_{35}	Pa.s	0.0358
Oil viscosity at 70 °C	μ_{70}	Pa.s	0.0111
Laboratory ambient temperature	T_a	°C	18 ... 24

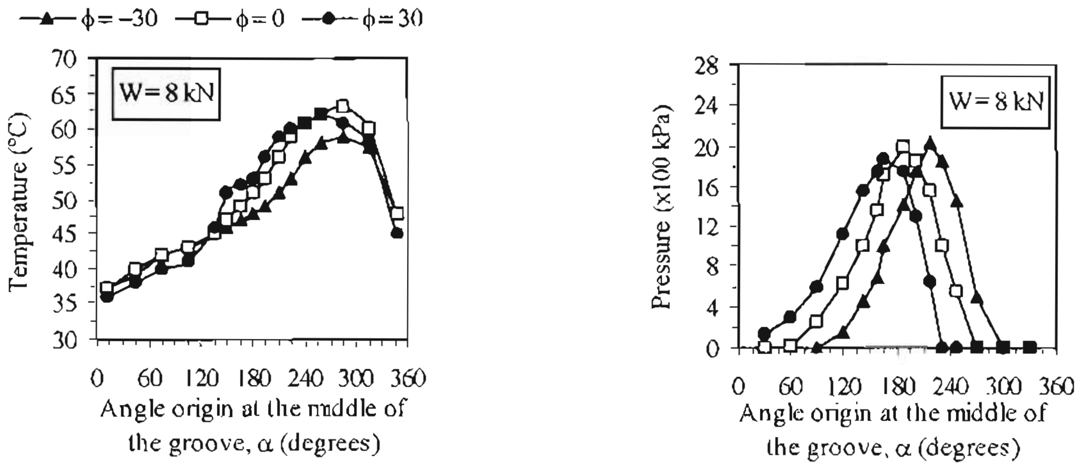
Table 2 Essential bearing dimensions and test conditions

Temperature	*Pressure*	*Flow rate*	*Torque*	*Load*	*Speed*	*Clearance*
± 0.5 °C	± 3%	± 0.05 litres/min	± 20 %	± 0.05 kN	± 10 rpm	± 5 μm

Table 3 Errors associated with the measurements

RESULTS AND DISCUSSION

Typical circumferential temperature profiles of the inner surface of the bush and circumferential hydrodynamic pressure profiles in the bearing mid-plane for different load directions are plotted in Fig. 3. The applied load was 8 kN and the shaft rotational speed 3000 rpm. It is interesting to note that for ϕ= -30° there is a noticeable decrease in the temperature of the inner surface of the bush. As indicated in Fig. 3(b), for ϕ= -30° the measured hydrodynamic pressure only started raising at 90° downstream of the groove. As a consequence oil film reformation occurred at about the same angular location, thus viscous dissipation is undoubtedly lower than for the other groove locations. This can explain the observed decrease in maximum temperature of the bush.



(a) (b)

Figure 3. Circumferential temperature profiles of the inner surface of the bush (a) and circumferential hydrodynamic pressure profiles (b) in the bearing mid-plane, for different load directions, with W= 8 kN and N= 3000 rpm [ϕ values in degrees]

The combined influence of load direction and applied load on the bearing performance is depicted in Fig. 4. Let us look to the influence of load first. Except maximum temperature of the bush, all bearing performance characteristics were significantly affected by load. An increase in applied load will cause an increase in clearance at oil inlet and an increase in maximum pressure inside the bearing. This explains the increase in oil flow rate. The increasing eccentricity associated with increasing load originates an increase of shear rate in the oil film and the augmentation of the extent of the oil cavited zone. The latter seemed to be preponderant, causing a decrease of heat generated at the film-shaft interface that would explain the decrease on shaft temperature. The increase in shear rate as load increases is the cause for increasing bush torque. Nevertheless maximum bush temperature was not significantly affected. As load increase oil supplied to the bearing increases and recirculated flow decreases. For each load, maximum temperature of the bush is the result of a balance between oil shear rate, recirculating flow and fresh oil flow supplied to the bearing.
Load direction did not affect shaft temperature and maximum hydrodynamic pressure. The effect of load direction on the performance characteristics studied was more significant at moderate loads (6 and 8 kN). The change of groove location from ϕ= 0° to ϕ= -30°, causes a decrease in clearance at oil inlet, originating a decrease in flow rate. This alone would generate a smaller cooling effect of the supply oil, but as observed in Fig. 3(b) for this groove location oil circulates inside the bearing downstream of the groove for about 120° before hydrodynamic pressure starts raising. As a consequence the local amount of heat generated by viscous friction is lower than for ϕ= 0°, resulting in lower maximum bush temperature. The decrease in bush torque may be partly explained by the decrease in oil flow rate in the bearing. The change in groove location from ϕ= 0° to ϕ= +30°, resulted in an increase in flow rate. As may be observed in Fig. 3(b), there is no significant change in the hydrodynamic pressure field with changes of groove location. Therefore, the increase of flow rate must be due to the increase in the oil inlet clearance. Regarding bush torque, it was not found a straight explanation for the decrease observed (Fig. 4(d)). A possible reason is the effect of

local velocity gradients associated with the displacement of the pressure field in relation to de position of minimum film thickness.

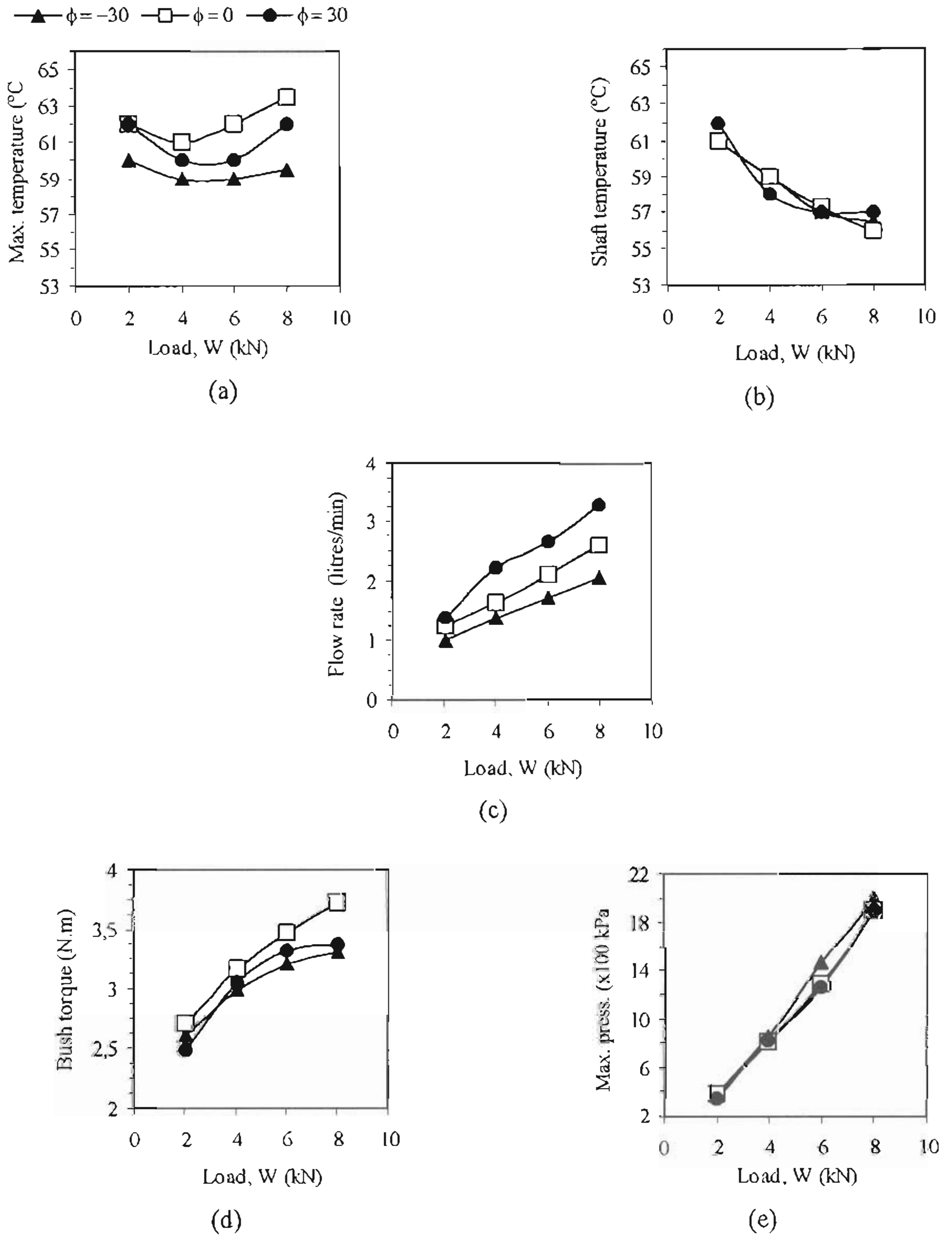


Figure 4. Bearing performance characteristics as a function of load and load direction, with N= 3000 rpm [ϕ values in degrees]

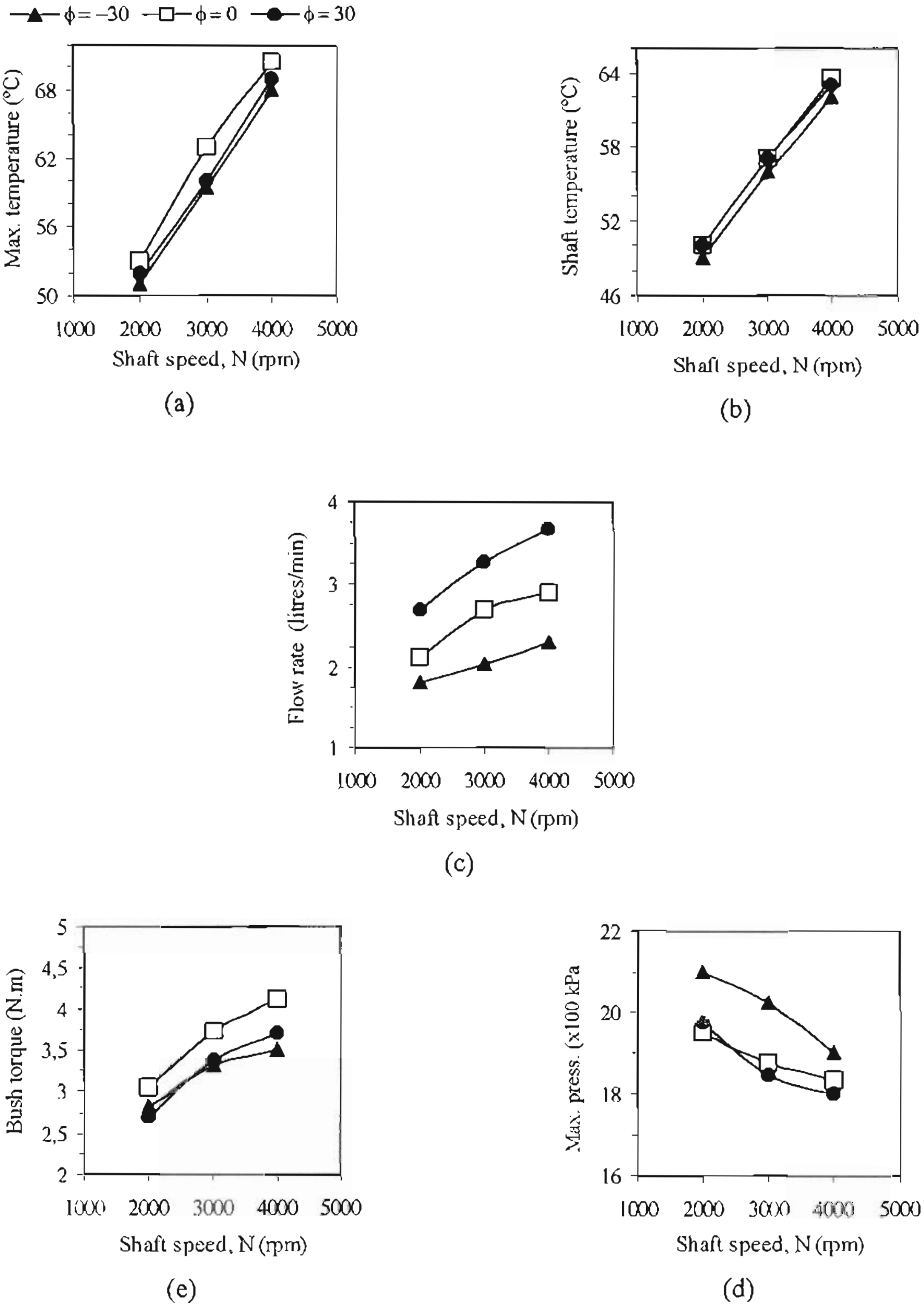


Figure 5. Bearing performance characteristics as a function of load direction and shaft speed for W= 8 kN [ϕ values in degrees]

Fig. 5 shows the combined effect of load direction and shaft speed on the measured bearing characteristics. It is clear that, except for maximum hydrodynamic pressure, increasing shaft speed resulted in a significant increase in the measured characteristics. Maximum bush temperature, shaft temperature and bush torque increased as a consequence of the increase on shear rate in the oil film. The significant increase on flow rate is associated with the increase of oil entrainment by the shaft. Increasing shaft speed originates an increase in load carrying capacity of the bearing. As load is fixed, this will result in a decrease in maximum hydrodynamic pressure and eccentricity (Fig. 5(d)). The influence of load direction on bearing performance for the speed range tested, was the same as observed in Fig. 4 for N= 3000 rpm.

CONCLUSIONS

An experimental study has been carried out using a hydrodynamic crown bearing where changes of groove location (ϕ= -30° and ϕ= +30°) in relation to the crown groove (ϕ= 0°) were tested. The following conclusions can be drawn:

i) For ϕ= -30°, a moderate decrease in maximum bush temperature and a considerable decrease on flow rate and bush torque have been observed.
ii) For ϕ= +30°, a significant increase in flow rate, a moderate decrease in bush torque and a slight decrease in maximum temperature of the bush have been observed.
iii) Maximum hydrodynamic pressure and shaft temperature have been only slightly affected by changes in the groove location.

ACKNOWLEDGEMENTS

The first author wish to thank the *"Fundação para a Ciência e Tecnologia"* (Portugal), for the financial support through PRAXIS XXI Programme/BD/13922/97. Thanks are also extended to *Total Raffinage, S. A.* (France) for the free supply of the lubricating oil used in the experiments.

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