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Effect of Load Direction, Preload, Clearance Ratio, and Oil Flow on the Performance of a 200 mm Journal Pad Bearing©

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Journal pad bearings are widely used in industry, particularly in high-speed applications. Previous experimental work with bearings of this sort has tended to be fragmented and mainly confined to smaller and relatively lightly loaded bearings. The purpose of this paper is to present temperature measurements and derived values of power loss from a new, extensive experimental program involving a 200 mm diameter journal pad bearing, operating at speeds up to 105 m/s and loaded at specific pressures up to 4.14 MPa. The program was designed to isolate some important design parameters and determine their effect on bearing temperatures and energy consumption. Bearing clearance, journal pad preload, the direction of the applied load, and the impact of radically reducing the amount of lubricant supplied to the bearing are all examined. In particular, it is found that load line direction has a substantial impact on maximum pad temperature. It is also shown that large reductions in the volume of oil supplied to the bearing are possible, leading to useful energy savings without compromising reliability. One of the most interesting features of the results presented is the significant effect on pad temperature of an apparent laminar to turbulent transition in the lubricant which occurs at high operating speeds. It is hypothesized that the transition takes place in the oil which flows around the pads rather than in the hydrodynamic films. It is anticipated that the information contained in the paper will be of interest and use to members of the academic community, bearing designers, and bearing operators.

KEY WORDS

Journal Bearings, Tilt Pad Bearings, Hydrodynamic Lubrication

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INTRODUCTION

Tilting pad journal bearings are well established as an important design option particularly appropriate for high-speed applications. The results of experimental work with bearings of this sort have been presented previously by a number of authors (*1*)–(*6*). While all the papers are useful in their own right, taken collectively the published information is somewhat fragmented. None of the available reports provides a comprehensive investigation into the effects that design variables, such as clearance ratio, pad preload, load direction, and oil flow, have on bearing temperatures and energy consumption.

The purpose of this paper is to present the results of a recent wide ranging program of experimental work with a 200 mm diameter, five-shoe tilting pad bearing. The experimental program was designed to complement existing work and fill gaps by providing measurements of journal pad temperature over significant ranges of speed and applied load which extend to duties which might be associated, for example, with a high-speed gearbox. Lubricant supply and drain temperatures have also been monitored and used in conjunction with oil flow to derive figures for the energy absorbed by the bearing. Where possible comparisons are made with published experimental work, and the influence of the design variables mentioned above is evaluated.

EXPERIMENTAL BEARING

The bearing used in the experimental work is shown in Fig. 1 and consists of a steel housing supporting five, center pivoted, tilting journal pads. The pads have whitemetal (babbit) faced working surfaces and center pivots on their reverse sides. The pads are located axially by bronze end plates fitted to the housing. Circumferential movement of the pads is prevented by the stop pins shown in Fig. 1. Dimensional details for the bearing are given in Table 1.

Diametric clearance was obtained by direct measurements at a number of orientations of maximum possible movement of the assembled bearing unit relative to the shaft. The

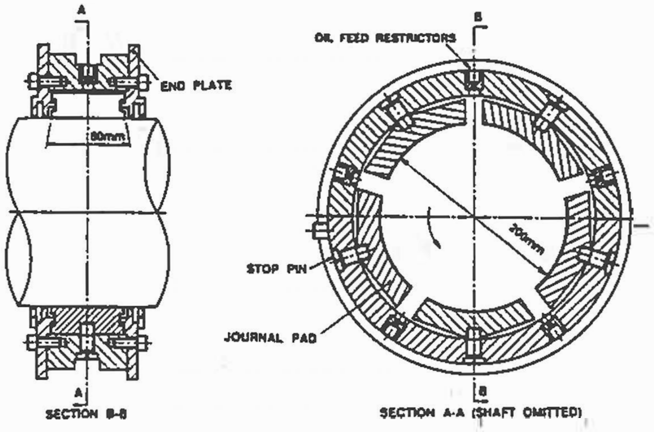

Fig. 1—Tilting pad journal bearing used in experimental work.

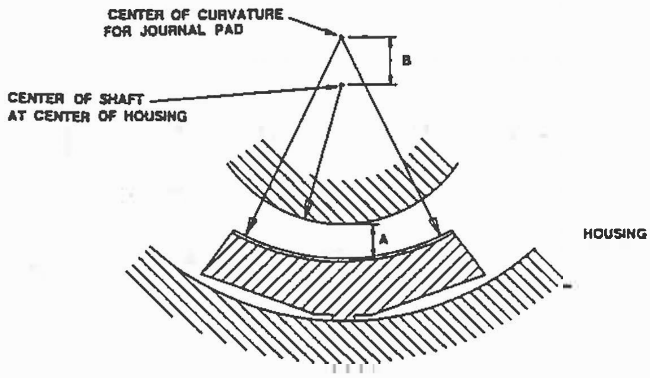

Fig. 2—Definition of preload.

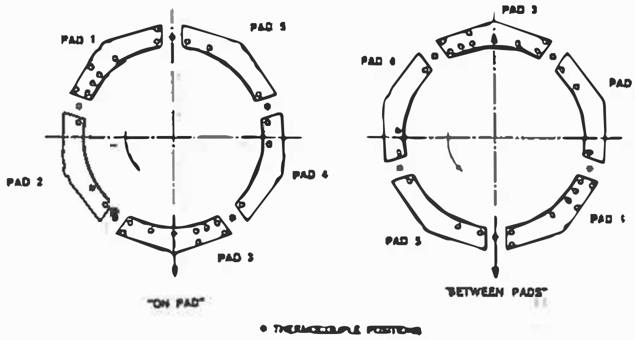

Fig. 3—Direction of applied load.

TABLE 1—DIMENSIONAL DETAILS OF THE EXPERIMENTAL BEARING	
Bearing diameter	200 mm
Number of pads	5
Center pivots	
pad axial length	80 mm
pad subtended angle	60°
nominal pad thickness	
at pivot	27.5 mm
average	24 mm
diametric clearance:	
standard	.23 mm
large	.36 mm
preload ratio:	
standard clearance	.52
large clearance	.22
large clearance, modified	.50

values obtained were verified by comparison with a derived value for clearance gained from inspection measurements of the individual components. It is estimated that the values quoted in Table 1 are accurate to within 0.02 mm. In order to assess the effect of diametric clearance in the bearing, two sets of pads with slightly different radial thicknesses were used to give the standard and large values of clearance referred in Table 1. Standard clearance, 0.23 mm, represents 0.11% bearing diameter and is in keeping with usual industrial practice for the bearings being examined for sliding speeds in the hydrodynamic film up to approximately 60 m/s. The larger clearance, 0.36 mm, is 0.18% of diameter and is typical for bearings with sliding speeds in excess of 60 m/s.

The pads are preloaded, as is normal industrial practice, by being bored to a diameter which is slightly greater than that justified by their radial distance from the axis of the bearing. This has the effect shown by Fig. 2 of ensuring that the working face of each pad can form a converging wedge with the shaft even when unloaded. Preload is defined by the ratio, $B/(A+B)$, indicated in Fig. 2, where 2A is the diametric clearance. Because the two sets of pads used in the experimental work were bored to the same internal diameter but had different thicknesses, they had the different values of preload ratio indicated by Table 1. In order to isolate the effect of preload in the latter part of the experimental program, the bore of the large clearance pads was increased to give a preload ratio equal to that of the standard clearance pads. Care was taken not to alter pad thickness at the pivot and thus change the value of clearance.

The design of the bearing housing was such that it could be mounted as shown in Fig. 1 with one of the pads directly in line with the applied load acting vertically downwards. Alternatively, the bearing could be rotated about the axis of the shaft and repositioned so that the line of the applied load would pass through a gap between two pads. These two configurations are known as the on pad and between pads condition, respectively, and are shown schematically in Fig. 3.

The bearing is lubricated by means of oil supplied under positive pressure from an external source to a distribution annulus in the housing, then through controlling orifices between each pair of neighboring pads into the bearing compartment. The bronze endplates, referred to earlier, are designed to maintain the bearing compartment flooded with oil but not to prevent some leakage along the shaft. A slot provided in each bottom half endplate allows the bulk of oil to drain directly. The proportion of oil which escapes past the endplates is prevented from leaving the casing by means of a shaft flinger and non-contacting, double labyrinth seal arrangement which can be seen in Fig. 4. Since one of the objectives of the experimental work was to in-

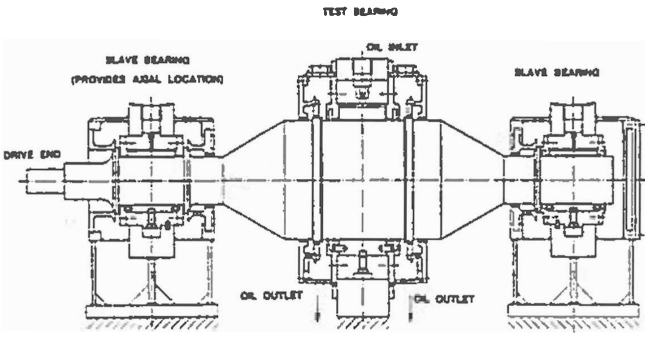

Fig. 4—Experimental arrangement.

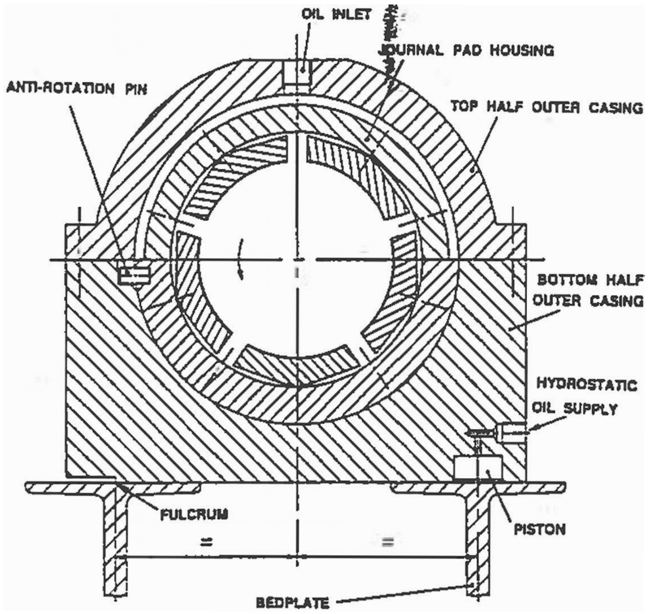

Fig. 5—Cross section of apparatus at axial midpoint of the experimental bearing.

vestigate the effect of changing the amount of lubricant supplied to the bearing, it was necessary to choose a diameter for the five supply orifices which was suitable for the full range of oil flows used in the experiments. It is necessary to maintain a positive lubricant supply pressure under all duties. Operating experience indicates that a supply pressure value of about 0.02 MPa is adequate at the minimum flow through the bearing. Based on this requirement, an orifice diameter of 5.8 mm was chosen and used throughout the work described in this paper.

TEST RIG

The test rig used for the experimental work is shown in Fig. 4. The experimental bearing, mounted in either the on pad or between pads orientation, is carried on a shaft which is supported by two 80 mm diameter slave bearings. Figure 5 is a sectional view of the apparatus taken at the axial midpoint of the experimental bearing. Load is applied using a 50 mm diameter hydraulic piston, shown in Fig. 5, positioned under one side of the casing bottom. The casing foot is relieved on the opposite side to create a fulcrum about which the bearing and its casing can be rotated. Hydrostatic oil pressure is used to force the piston against the bed plate and hence jack the experimental bearing upwards against the shaft, which is constrained by the 80 mm support bearings. The hydrostatic oil pressure, monitored by a gauge accurate to ± 0.17 MPa, provides a means to derive the specific load applied to the bearing.

The apparatus is driven by a thyristor-controlled DC motor capable of providing rotational speeds of up to 10000 rev/min. A standard ISO VG 32 lubricating oil was used throughout the experiments and was supplied from an external system which was adjusted to deliver oil at an inlet temperature of 43°C ± 1.5°C.