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Performance Experiments with a 200 mm, Offset Pivot Journal Pad Bearing©

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Journal bearings are an important design option, particularly appropriate for high-speed rotating machinery applications. This paper presents results from an extensive program of experimental work with a 200 mm, five-shoe, tilting-pad bearing of this type. The tilting pads were fitted with offset pivots; this is in contrast to most previous work which has concentrated on bearings with pads which have centrally mounted pivots. The new results obtained for the offset pivot pad bearing are compared with recently published, equivalent results for the same bearing fitted with center pivot pads. A substantial temperature reduction in bearing temperature is revealed for the offset pivot case.

KEY WORDS

Hydrodynamic Bearings, Journal Bearings

INTRODUCTION

Tilting-pad, hydrodynamic journal bearings are widely used in industry for a range of machinery and are particularly appropriate for high-speed applications. In many cases, center-pivot pads are used and prove very satisfactory. Pressure to design and operate machinery capable of operating at enhanced speeds and loads puts a premium on cooler-running bearing designs. For this reason, offset pivot pads which lead to significantly cooler bearings and increased hydrodynamic film thickness are frequently employed as a design option. Published experimental work with tilting-pad journal bearings has concentrated on bearings with center-pivoted pads and a recent paper (*1*) presented a new set of comprehensive data for such a five-pad, 200 mm diameter bearing.

The purpose of this new paper, which complements the one referred to above, is to present additional results particular to offset pads and compare these results with the already available center pivot findings. So far as the authors have been able to determine, with one exception, no similar experiments involving a journal bearing fitted with offset pivot pads have been previously reported. Hence, it is hoped that results of this paper will be of interest both to the academic community and to industry suppliers and users.

The exception mentioned above is an interesting paper by Brockwell et al. (*2*) mainly concerned with a rather different journal bearing, namely a tilting-pad, leading edge groove (LEG) bearing. For comparison purposes, Brockwell and his colleagues included some experiments with a 100 mm, conventional, i.e., non-LEG, offset pivot, journal pad bearing.

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EXPERIMENTAL BEARING

The experimental bearing consists of a steel housing supporting five tilting journal pads as Fig. 1 shows. The bearing used in this work program has been described previously and further information about its design is given in Ref. (*1*). In the case of the new work described in this paper, the pads are fitted with offset pivots situated such that the circumferential distance from the leading edge of each pad to the pivot is 0.6 of the overall circumferential length of the pad. Dimensional details for the bearing are given in Table 1.

The pads are preloaded, as is usual, by being bored to a diameter slightly greater than that justified by their radial distance from the axis of the bearing. Information about preload and the way preload ratio is defined is contained in Ref. (*1*).

The design of the bearing housing is such that it can be mounted as shown in Fig. 1 with the applied load acting vertically downward through the pivot line of one of the pads. Alternatively, the housing could be repositioned and rotated about the shaft axis so that the load line falls equidistant between the pivots of two adjacent pads. These two positions of the housing correspond to the "load-on-pad" and "load-between-pads" conditions defined in the earlier center pivot pad experiments. In practice, the transition between on-pad and between-pad configurations was brought about by rotating the bearing housing 180 degrees about its axis.

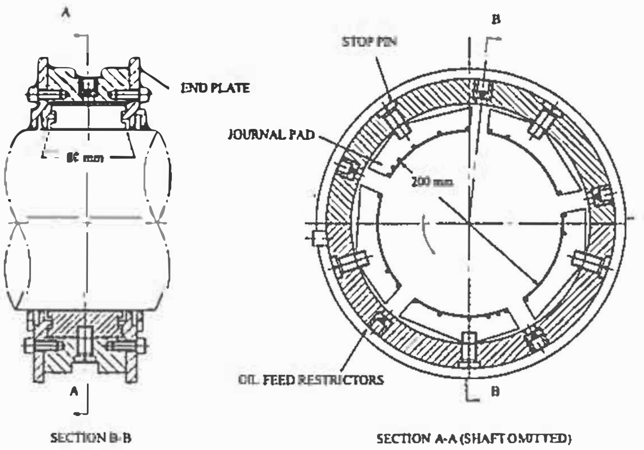

Fig. 1—Diagram of bearing used in experimental work.

TABLE 1—DIMENSIONAL DETAILS OF THE EXPERIMENTAL BEARING	
Bearing diameter	200 mm
Number of pads	Five
Offset pivots	
Pad axial length	80 mm
Pad subtended angle	60°
Nominal pad thickness	
at pivot	27.5 mm
average	24 mm
Diametric clearance	.22 mm
Preload ratio	.52

The bearing is lubricated by means of oil supplied under a modest positive pressure from an external source to a distribution annulus in the housing. The oil then passes into the bearing compartment through 5.8 mm diameter orifices situated between each pair of neighboring pads. Bronze endplates, bolted to the housing, provide axial location for the journal pads and are designed to maintain the bearing compartment flooded with oil. Oil drains from the bearing via a slot in each bottom half endplate. Any small amount of oil escaping along the shaft past the endplates is dealt with by a shaft flinger and double labyrinth seal arrangement which forms part of an outer casing for the bearing, which is not shown in Fig. 1.

A standard ISO VG 32 lubricating oil, delivered by the external system at 43°C +/− 1.5°C, was used throughout the experimental program. Oil supply pressure was measured at the inlet to the bearing's distribution annulus. Variation of this pressure in a range between 0.05 MPa and 0.15 MPa was used to control the amount of lubricant supplied to the bearing which, following the practice established in earlier work (*1*), was adjusted for each combination of speed and load to give a temperature rise between supply and drain of approximately 17°C.

APPARATUS AND INSTRUMENTATION

In the apparatus used for the work being described, the experimental bearing is mounted on a shaft between two slave bearings as described and illustrated in Ref. (*1*). Load

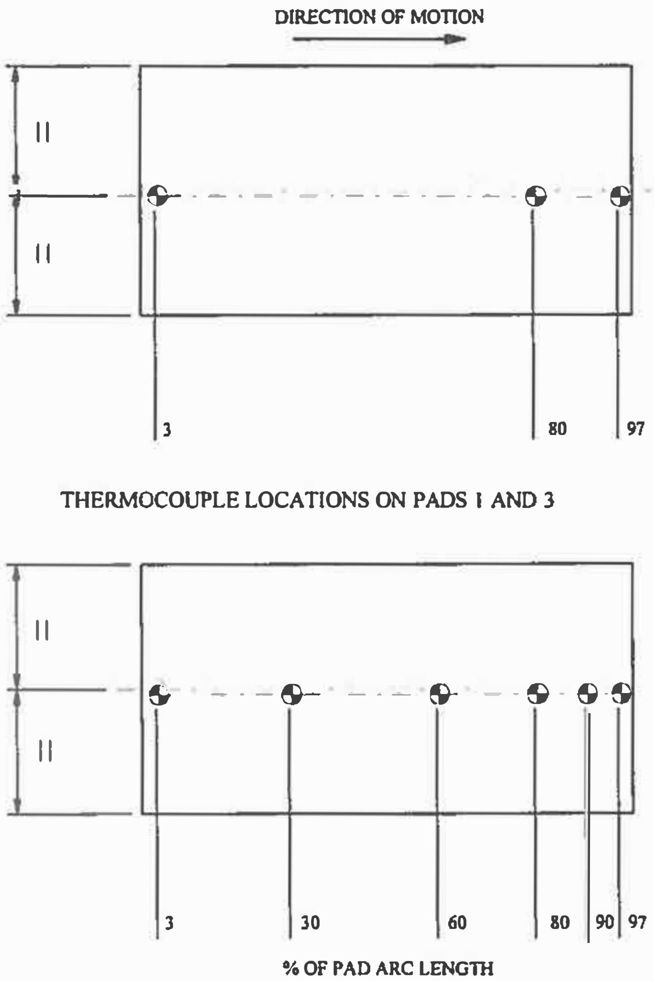

Fig. 2—Circumferential location of thermocouples relative to pad surfaces.

is applied to the casing of the experimental bearing by means of a hydraulic piston. The applied load is transmitted in turn to the shaft which reacts against the adjacent slave bearings. A variable speed drive is capable of providing rotational speeds up to 10,000 rev/min.

Thermocouples for measuring pad temperature were embedded beneath the whitemetal (babbit) surface of each offset pivot pad as shown in Fig. 2. Lubricant temperature was measured at the point of delivery to the bearing and at outlet in ports designed to ensure that the thermocouples were always immersed in hot oil exiting the system. Oil flow rate through the experimental bearing was measured using a variable area flow meter.

EXPERIMENTAL RESULTS

Two series of experiments were carried out to cover the load-on-pad and load-between-pads conditions referred to previously. The apparatus was operated with shaft speeds from 3000 rev/min to 10,000 rev/min. These operating speeds provide a sliding velocity range at the bearing's working surface of 31 m/s to 105 m/s. A series of loads was applied equating to bearing specific loads of 0, 1.38, 2.76 and 3.45 MPa. This set of operating speeds and loads, which cor-

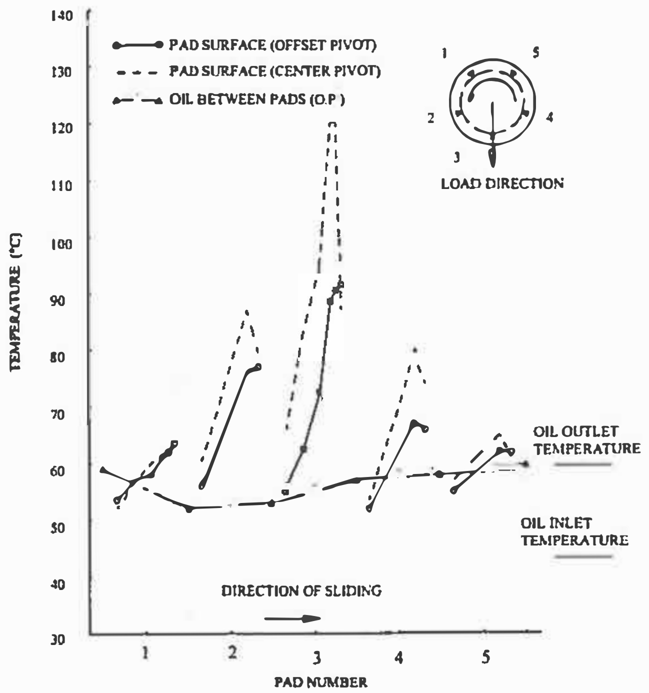

Fig. 3—Thermocouple readings obtained at 73 m/s and 2.76 MPa together with equivalent center pivot pad results from Ref. (1). Load-on-pad.

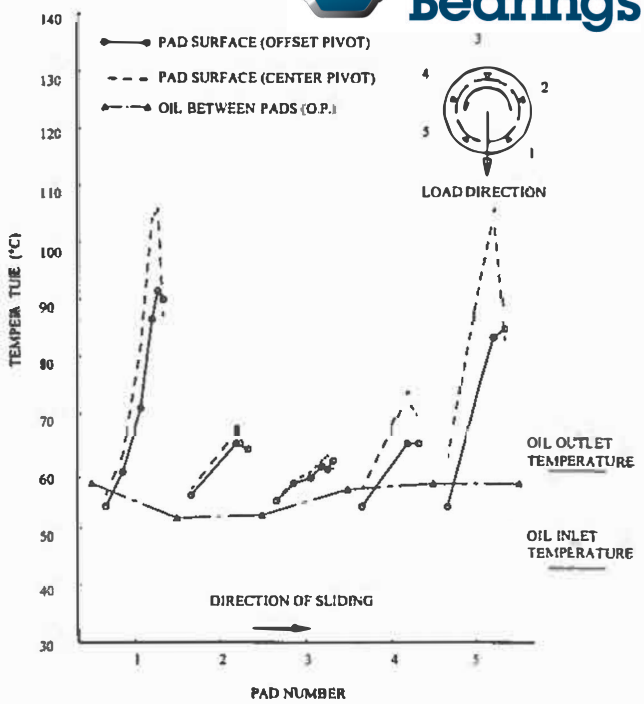

Fig. 4—Thermocouple readings obtained at 73 m/s and 2.76 MPa together with equivalent center pivot pad results from Ref. (1). Load-between-pads.

responds with that for the previous work involving center pivot pads (1), was chosen to cover typical duties expected of equipment powered by high-speed electric motors and tur-bine drives.

Figure 3 shows a typical set of results from the current experiments for the on-pad condition. Also shown in this fig-ure are the corresponding (same load, speed and orienta-tion) results from the center pivot pad experiments featured in Fig. 7 of Ref. (1). In the on-pad condition, load is sup-ported predominantly by the pad which sits directly beneath the shaft. As one would expect, this most highly loaded pad experiences the highest temperatures in both the offset and center pivot cases. Corresponding sets of results for the between-pads condition are shown by Fig. 4. Once again, the figure includes results from the current experiments and, for comparison purposes, the equivalent center pivot results drawn from Fig. 8 of Ref. (1). In this case, load is shared principally between the two heavily loaded pads in the lower half of the bearing casing.

The maximum temperatures of all the heavily loaded, off-set pivot pads given in Figs. 3 and 4 show a significant reduc-tion by comparison with the equivalent center pivot pad re-sults. There is, however, little difference between the maximum temperature of the most heavily loaded offset pivot pad in Fig. 3 and those for the two heavily loaded offset pivot pads in Fig. 4. This is in contrast to the center pivot experi-ments where the maximum temperature for the load-on-pad condition is significantly greater than for the load-between-pads configuration.

For the unloaded pads in the upper half of the casing, which are operating close to the oil outlet temperature, there is, by contrast with the heavily loaded pads, little temperature difference between the offset and center pivot results.