Tilting Pad Journal Bearings—Measured and Predicted Stiffness Coefficients©

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This paper presents measured and calculated characteristics of a tilting pad journal bearing suitable for high speed machinery. Descriptions are given of the experimental techniques used with this variety of bearing and the theoretical model for predicting performance.

Measured values of pad temperature, eccentricity, attitude angle, and the four stiffness coefficients are given for a range of loads and rotational speeds. Data are given for both load on pad and between pad configurations, the two principal loading arrangements.

Comparisons are made between the measured and predicted bearing temperatures and stiffness coefficients over a wide range of values.

KEYWORDS

Bearings, Hydrodynamic, Tilting Pad, Load Carrying Capacity

INTRODUCTION

Tilting pad journal bearings have been used to support rotating shafts for a considerable number of years. They are more expensive and occupy a larger space than a corresponding conventional plain bearing. Nonetheless, they have been, and continue to be, selected by designers in applications where their superior stability, especially in the high speed/low load condition, is a paramount consideration.

Consequently, as with plain bearings, their widespread use has preceded knowledge of their performance characteristics and reliable theoretical models thereof. It is only in relatively recent times that the performance of tilting pad journal bearings has been studied. Theoretical studies (*1*)–(*3*) have been reported, and in particular Jones and Martin (*4*) have indicated the effect of changes in various geometrical parameters upon performance. Within the last few years, results of a number of experimental studies (*5*)–(*9*) have been published. Notwithstanding, there remains a scarcity of data.

This paper presents a contribution to this knowledge by comparing the measured performance of a typical tilting pad journal bearing with the predicted characteristics. Data are given for both the common operating configurations of load on and load between pads.

BEARING AND TEST DETAILS

The basic details of the tilting pad bearing selected for the test program are outlined in Table 1. The bearing was a standard 80 mm unit with a length to diameter ratio of 0.4 and a proprietary design as illustrated in Fig. 1. These units consisted of five tilting pads retained in a split-bearing housing, with end baffles to control the oil discharge. The

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NOMENCLATURE

a_{yy}, a_{xx}) = oil film stiffness coefficients
a_{yx}, a_{xy})
C_a = bearing assembled clearance, difference between journal radius and the distance from bearing center to pad surface, as installed
C_p = pad surface clearance, difference between pad face machined surface and journal radius
D = journal diameter
L = bearing length
N = rotational speed
P_h, P_v = forces applied to the bearing housing, horizontal and vertical
T = temperature
W = steady load
h, v = coordinates of journal center, horizontal and vertical
x, y = coordinates of journal center, perpendicular and parallel to the load direction
α = steady load direction (Fig. 3a)
ε = journal eccentricity
= attitude angle
= journal center displacements from equilibrium position due to the application of incremental loads ΔP_h and ΔP_v
η = viscosity of the lubricating fluid

Table 1—Bearing and Test Details	
Bearing geometry	
Journal diameter	80 mm (3.149 inch)
Journal length	32 mm (1.260 inch)
Clearances (radial)	
C_a (assembled)	0.047 mm (0.00185 inch)
C_p (pad surface)	0.104 mm (0.00409 inch)
Pad preload ratio	0.55 $(C_p - C_a)/C_p$
Number of pads	5
Circumferential arc	60 degrees
Lubrication	
Oil type	Shell Turbo 32
supply temperature	45–50°C
supply pressure	27–48 kPa (4.0–7.0 lbf/in^2)
Bearing duty	
Load	0.5 – 5.0 kN (112–1124 lbf)
Rotational speed	3000–8800 rpm

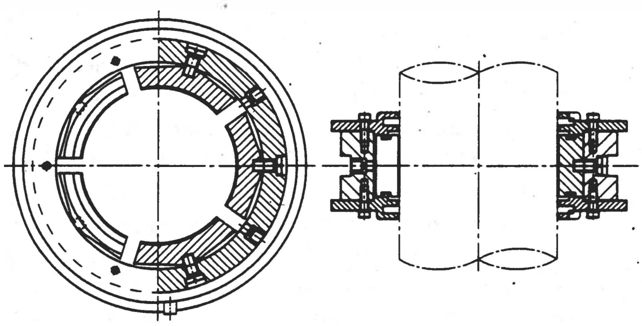

Fig. 1—Test bearing assembly.

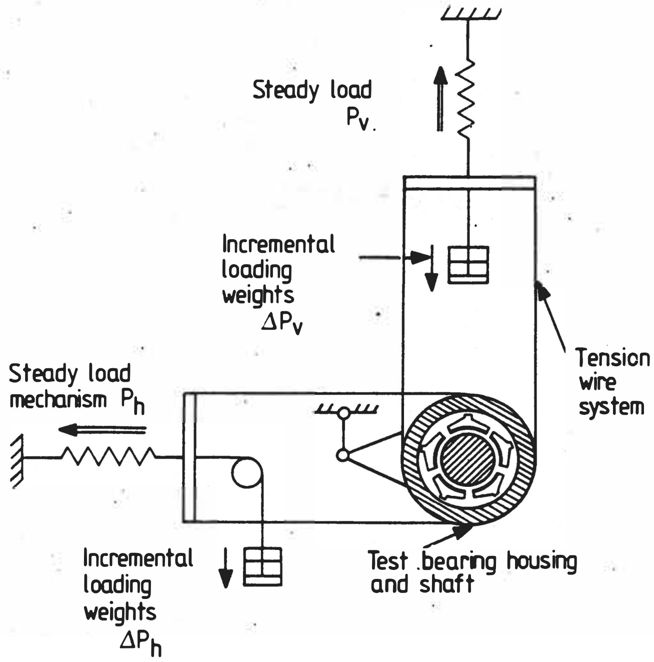

Fig. 2—Test equipment loading arrangement.

bearing normally operated with the housing filled with oil.

The tilting pads were held in the bearing housing so that they were preloaded towards the journal, and the five radial oil feed holes were located at the inter pad spaces. The pads themselves were of the central line pivot type, with a 0.5 mm thick white-metal (babbitt) face on a steel backing 12 mm thick. Apart from a break for the pad locating stop, the pivot extended the full width of the pad. This resulted in a pivot strength of more than 2.0E+9 Nm^{-1}, which is an order of magnitude greater than the expected lubricating film stiffnesses.

The lubricant used in this series of tests was an ISO 32 grade turbine oil (32cSt at 40°C and 21cSt at 50°C). Throughout all the tests reported herein, the temperature of the oil at inlet to the bearing assembly was maintained in the range 45° to 50°C. Typically the temperature variation during the course of a test sequence was less than 3°C.

Tests were performed at loads of 0.5, 1.0, 3.0, and 5.0 kN, which cover the typical working range of 0.2 to 2.0 MPa (30 to 300 psi) mean bearing pressure. The shaft speeds used for the test program were 3000, 5000, and 8800 rpm, this range being limited by the capabilities of the test stand.

Non-contacting displacement transducers mounted on the bearing housing measured the relative journal center position in the horizontal and vertical directions *(h,v)* at axial stations on either side of the bearing. The lowest pad was fitted with a chromel-alumel thermocouple adjacent to the pad pivot to give and indication of mean oil temperature in the lubricating films. Inlet and outlet bulk oil temperatures were also recorded.

EXPERIMENTAL TECHNIQUE

An outline arrangement of the test stand illustrating its use in the work reported herein is shown in Fig. 2. A full description of the experimental apparatus and the arrangements for loading the test bearing having been published previously *(10)*, *(11)*.

The test bearing was suspended on the shaft between two rolling element support bearings. Loads were applied to the test bearing through tensioned wires in both the horizontal and vertical directions *(h,v)*. By using an appropriate combination of forces P_h, P_v the steady load W could be applied at any angle α in a segment of 90° from vertical to horizontal ($0° \leq \alpha \leq 90°$). This provided a convenient means of directing the load either between pads 1 and 2 ($\alpha = 36°$), directly at pad 1 ($\alpha = 0°$), or directly at pad 2 ($\alpha = 72°$), see Fig. 3a, without re-establishing thermal equilibrium and/or changing speed.

Use of the test stand to measure the assembled bearing's clearance space in the cold, non-rotating state, has been described in *(10)*. The reference explains how a near complete clearance circle can be obtained for plain bushes by forcing the journal against the bearing surface at a large number of radial locations. However, the current test bearing's inherent pad tilting property meant that consistent readings could be obtained only when the journal was forced in the direction of an interpad gap. At all points in between, the journal slid along the tilted pad surface until it reached one of the interpad positions. Hence, only five consistent data points were reliably obtained.

Location of the effective bearing center in the hot rotating condition was accomplished using a new technique which

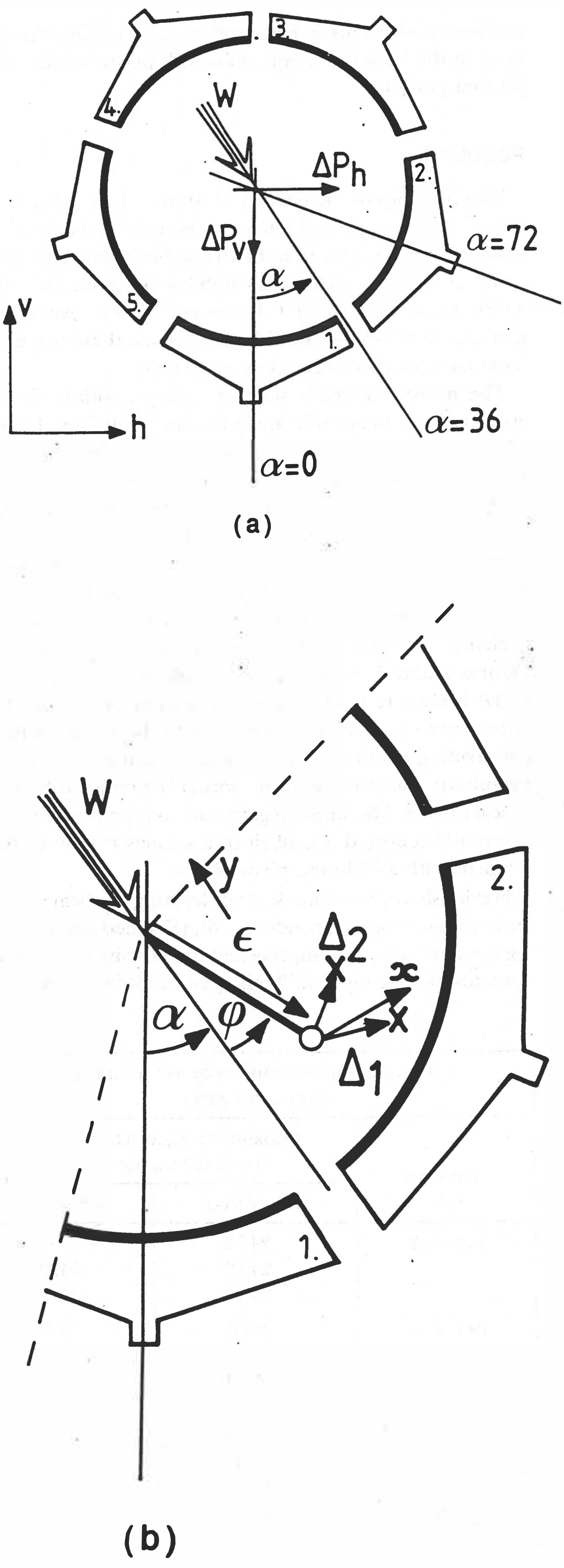

Fig. 3—Steady and incremental loading details.
(a) Static and incremental loading
(b) Bearing coordinates and displacements

was independent of the cold clearance space measurements. The new method utilized the ability of the test stand to direct the applied load to either of the pads at $\alpha = 0°$ and 72° (nos. 1 and 2), and the symmetry of these loading conditions. The procedure assumes that the attitude angle and eccentricity remain the same for operation with the load directed towards each of the pads 1 and 2. Thus, successive measurements of the journal position, having achieved thermal equilibrium, at these two load directions, enabled the bearing center to be located in the transducer coordinates for that duty.

If the coordinates of the journal center are given for the loading directed at pads 1 and 2, respectively, by h_1, v_1 and h_2, v_2 then:

$$\varepsilon = [(h_2 - h_1)^2 + (v_2 - v_1)^2]^{1/2} / 1.1756$$

Since the angle α could only be varied within the range $0 \leqslant \alpha \leqslant 90°$ when the test rig was in the hot rotating condition, then the hot bearing center location procedure used only pads 1 and 2 for the given direction of rotation.

Having established the bearing center, a simple manipulation of the journal center coordinate measurements produced the eccentricity and attitude angle for the load on pad condition. Assuming this bearing center is valid for the load between pad operation ($\alpha = 36°$), a further reading of the journal position gave the eccentricity and attitude angle for that case. This entire procedure formed the basis for determining the bearing's steady state operating condition for each and every load/speed combination.

The oil film stiffness coefficient a_{xx} - - - a_{yy} were measured using a previously applied modification to the traditional incremental loading technique (*10*), (*11*). If, as in the traditional technique, incremental forces are applied in directions parallel and perpendicular to the loading direction, then difficulties arise in the measurement of the consequential incremental displacements when operating at high eccentricities. Problems arise because under such conditions, the four coefficients have very different magnitudes. For example, a typical set of stiffness coefficients a_{xx}, a_{xy}, a_{yy}, a_{yx}, for results quoted in this paper for eccentricity ratio 0.75 are respectively 105, − 14, 105 and − 14 MNm^{-1} (0.6, −0.08, 4.0, −0.08 10^{+6} lbf $inch^{-1}$). Separate application of typical incremental forces, (294 Newtons) (66.2 lbf) in directions parallel and perpendicular to the steady load, would give corresponding incremental displacements of Δxa, Δxb, Δya, and Δyb of + 2.29, + 0.055, + 0.055, and 0.42 10^{-3} mm (110.0, 2.2, 2.2, 16.5 10^{-6} inch). These displacements have widely different magnitudes, in the extreme ratio of 50:1. Moreover, the values of 0.055 10^{-3} mm were judged to be less than the maximum possible error, not discrimination, of the measuring system when testing tilting pad bearings. Displacements between bearing and journal were measured in two planes on either side of the test bearing, from which values for the bearing center plane were computed. This meant that the maximum possible error for measurement in bearing center plane could take any value between zero and 0.127 10^{-3} mm (5.0 10^{-6} inch) depending on the sign of the error at each measuring station. A smaller value was obtained with plain bearings. From this data it follows that the error in the magnitude of the derived value of the indirect coefficients a_{xy}, a_{yx} would be in the range zero to ± 120 percent with a reversal of sign. However, if the same typical incremental loads are applied in two separate directions other than parallel and perpendic-ular to the steady load, at say an angle $\alpha = 45°$, then the corresponding displacements to be measured take nearly equal magnitudes and are well above the maximum possible error, i.e., 1.662, 1.187, 1.187, and 1.549 10^{-3} mm (65.45, 46.73, 46.73 and 61.0 10^{-6} inch). This reduces the range of possible error in derived cross coefficients to between zero and ± 30.7 percent, without a sign reversal. Previously reported applications of this approach (*10*), (*11*) describe testing of plain bearings with complete circumferential sym-metry and utilize angles of 30°, 40° and 50°.

However, the tilting pad bearings does not have such circumferential symmetry. Indeed, symmetry only arises with the five pad variety under test at 72° intervals. Hence, in the tests reported herein, use of $\alpha = 72°$ and 36° conve-niently provided load on and between conditions with non-zero angle α. However, at these values of α, the accuracy benefit for the cross coefficients a_{xy}, a_{yx} for $\alpha = 36°$ was less than that at 45°, namely, in the range zero to ± 45 percent instead of 30. 7 percent and for $\alpha = 72$ only a slight im-provement over the likely accuracy to be obtained at $\alpha = 0$. Accuracy of the determination of the direct coefficients a_{xx}, a_{yy} at $\alpha = 36°$, 72° was less markedly affected by choice of angle α, being within the ranges zero to ± 14.5 to ± 30.3 percent (a_{yy}) and 0.4 to 1.7 percent (a_{xx}). This ap-plication of the procedure produced a useful reduction in scatter for most coefficients under most, but not all, con- ditions, while retaining the benefit of simplicity.