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An Experimental Study of Hydrodynamic Tilting Thrust Bearings

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ABSTRACT

The performance of hydrodynamic tilting thrust bearing in terms of outlet film thickness, friction coefficient, and bearing temperature were measured experimentally. They are compared to other experimental and theoretical work. The bearing fluttering (vibration) at high speeds and light loads are also reported.

INTRODUCTION

Experimental investigation on the performance of hydrodynamic tilting thrust bearing have been reported in the past by Kettleborough (1955), Hemingway (1968), Tieu (1975), Gregory (1977,1979), Capitao et al (1976). The first three concerns with the hydrodynamic oil film gap and bearing friction. The remaining references dealt with oil flow, power loss from heat balance and bearing temperatures and did not attempt to correlate between theory and experiment. In this paper, bearing performance in terms of oil film gap, bearing friction and temperatures are correlated with the design calculations proposed by Engineering Science Data Unit ESDU (1975) and numerical simulation of three-dimensional oil film (Tieu,1975).

Most hydrodynamic thrust bearings operate with a convergent wedge consisting of two plane working faces. However the profile of the stationary bearing surface and the rotor surface are influenced by surface finish and waviness, machining accuracy during manufacture. Large size rotor surface can be manufactured and/or assembled with an amount of run-out comparable to the bearing film thickness. In these cases, the hydrodynamic load capacity of the bearings is contributed by the convergent pad tilt as well as the waviness of the rotor surface. Moreover, the rotor waviness contribute significant squeeze film effect to the hydrodynamic pressure at high speeds. Journal bearing surface waviness have been considered by Mokhtar et al (1984).

Boyd and Raimondi (1964) reported the 'spragging' phenomena of pivoted journal bearing, whereby the bearing vibrates or flutter under the conditions of high speeds, light loads and low viscosity. The spragg of thrust bearings was mentioned briefly by Gregory.

In this paper, the experimental investigation dealt with the performance of tilting thrust bearings which operate with a maximum rotor run out amplitude (single wave) of .22mm. The vibration and fluttering of the tilting pads are also reported here at high speeds and light loads.

TEST RIG

The diagram of the thrust bearing test rig is shown in Fig.1. The drive is powered by a 18.5 kW motor at maximum speed of 2700RPM. The variable speeds are provided by the TASC eddy current drive 180CD2/2. The tooth belt HTD drive has a step up ratio of 1.71:1. A rotor disc with the outside diameter of 0.6m is carried by a rotor shaft which is supported by roller bearings.

The thrust pads, made of phosphor bronze, has a width to length ratio of 1:1, with the width being 49mm, bearing area 2.17 x $10^{-3}m^2$. They are mounted on spherical pivots at the mean diameter of 0.5m, and therefore are free to tilt in circumferential and radial directions. The pivot is positioned at 60% of the pad mean circumferential length, from the leading edge.

The stator floats freely on spherical hydrostatic bearing, so that bearing friction can be measured. This was done by a stop of triangular plate fitted with strain gauges. The whole stator arrangement is loaded via by a double power screw. Loads are applied by dead weights through a mechanical lever system of ratio 92:1 to provide applied loads up to 10000N.

Rotor Runout

The rotor run out is plotted against the rotor circumference at the mean diameter of 0.5m in Fig. 2a. The results indicated that the rotor surface oscillates vertically through 0.22mm relative to a fixed point. This represents an amount of 0 to $26x10^{-6}m$ convergent or divergent wedge contribution to the oil film thickness over the pad length.

Lubrication

The lubricant used is Castrol Turbine Oil Perfecto T46 with Kinematic Viscosity of 46 cSt at 40^oC. The lubricant is supplied to each pad by direct spray method just before the pad leading edge. This reduces power loss due to churning loss normally experienced with fully flooded lubrication. The tapered roller bearing are lubricated by circulating oil of the same type. The oil temperature at the nozzle outlets and roller bearings were monitored by thermocouples.

Transducers

To measure the bearing film thickness both in the circumferential and radial directions, three Bently Nevada Proximity transducers 3000-190 are mounted on one pad, one positioned just before the leading edge and two at the trailing edge, as shown in Fig. 2b. Also shown are one copper-constantan thermocouple at the inlet edge and two at the trailing edge to monitor bearing temperature.

Test Procedure

With the same lubricant, rotor shaft speed and applied thrust load were varied. The oil sprays and hydrostatic bearings were turned on for at least $\frac{1}{2}$ to 1 hour to stabilise the temperature of the system. For any test the rotor shaft speed was set at the desired value, and the stator was raised closer to the rotor surface until the load lever started to move. This defines the 'no load' condition, which is required in the calculation of bearing friction. The load was then applied to mean bearing pressure of 55 to 1400 kPa. The oil temperature at the bearing leading edge was varied between 20 to 45 ^oC. A total of nine readings of the oil film thickness, friction torque, oil and bearing temperature were recorded on a Hewlett-Packard 3054DL data logger. The data logger scans continuously 0.025 second per channel.

RESULTS AND DISCUSSION

In this paper, rotor sliding speed, oil viscosity and pressure are grouped together into the duty parameter $\frac{\eta U}{P}$, since these variables affect the bearing thickness and friction. The viscosity η was evaluated at the pad mean temperature. In order to compare the results here with others for different size bearing, the film thickness values were scaled by the square root of the bearing width (ESDU,Jakobsen (1958))

The inlet and outlet film thickness are plotted against the parameter $\frac{\eta U}{P}$ in Fig. 3. Significant scattering of the inlet film thickness at light duty, i.e. large value of $\frac{\eta U}{P}$, due to bearing fluttering shall be discussed later. In Fig. 4 is the theoretical outlet film thickness obtained from the ESDU design standard for tilting thrust bearing and from Jakobsen (1958).The latter curve employs the experimental values of pad tilt ratio to derive the film thickness. These two film thickness curves almost coincide with each other, and therefore only one curve is presented here.

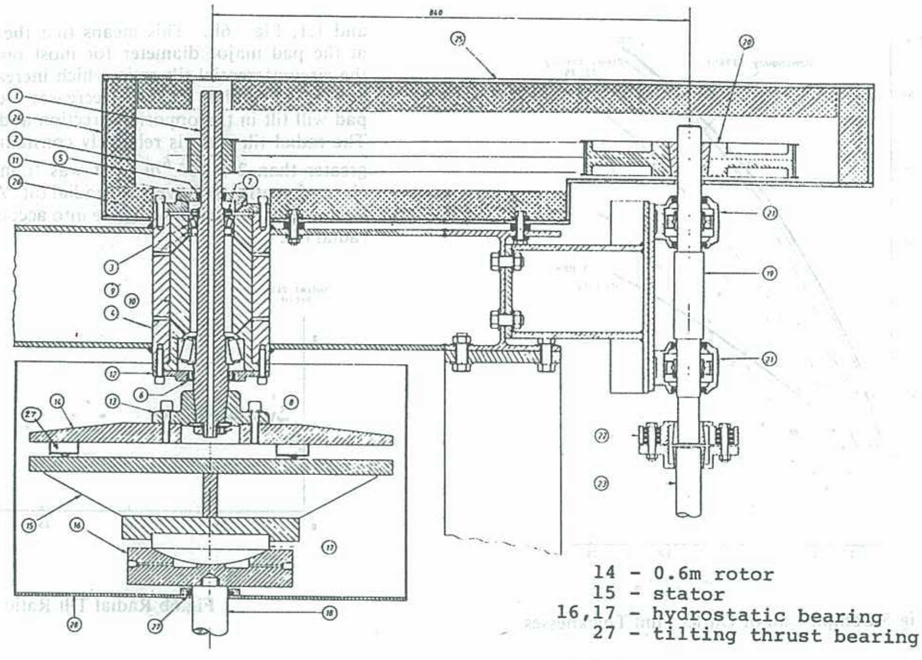

Fig.1 Diagrammatic View of Test Rig

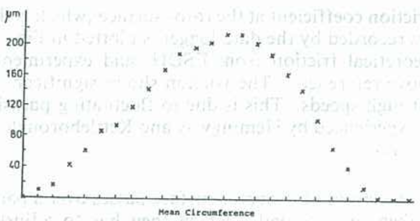

Fig.2(a) Rotor Runout Along The Mean Circumference

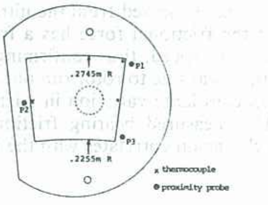

Fig.2(b) Location of Proximity Probes and Thermocouples

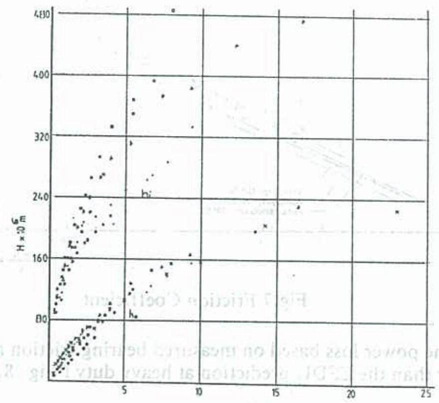

Fig.3 Inlet and Outlet Film Thickness Against $\frac{\eta U}{P}$

It can be seen that the measured film thickness fall below the theoretical curve which does not take into account the wave form of the rotor surface. This reduction in operating film thickness was also observed by Mokhtar (1984) for journal bearing with wavy surface.

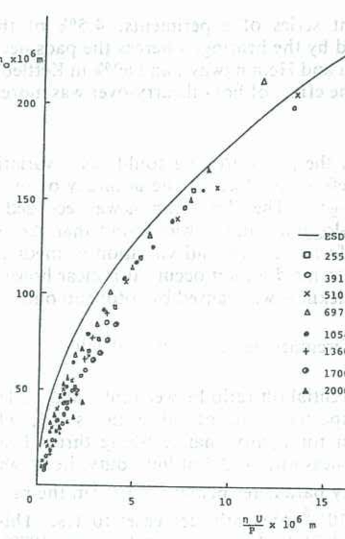

Fig.4 Outlet Film Thickness

The scatter of the experimental results is more evident on the expanded scale used in Fig. 5., and again they are well below the curves of Kettleborough ,Hemingway and Tieu . Kettleborough et al tested 6 pad bearings at 6.25 lower sliding speeds than the current experiment, and the viscosity was based on mean pad temperature. Experiments by Hemingway were carried out on three pad thrust bearing, at low speeds , and hence the results were not significantly affected by viscous thermal effects. Tieu allowed for thermal effects by using the effective viscosities within the three-dimensional oil film in the evaluation of the duty parameter. The comparable results from Gregory (1977) are also presented in Fig.5. In this case the viscosity η was based on the temperature in the middle of the pad. It can be seen that the results from Gregory agree with the ESDU curve at medium to heavy duty (3.44 and 2.06 MPa). At light duty (0.69 MPa), however, the film thickness fall sharply to those in this experiment. This could be due to the reported bearing vibration, namely 'spragging' effect , which was the result of oil starvation or excessive oil churning within the bearing casing.

There are many reasons that can be attributed to the discrepancy between the experiments. Firstly, individual pad in a set does not tilt the same as confirmed by Hemingway and Tieu. In these references, capacitance were mounted on the rotor and scan continuously over the three pads. The outlet film thicknesses varied from one pad to the other by as much as 8 microns, and they were based on the mean

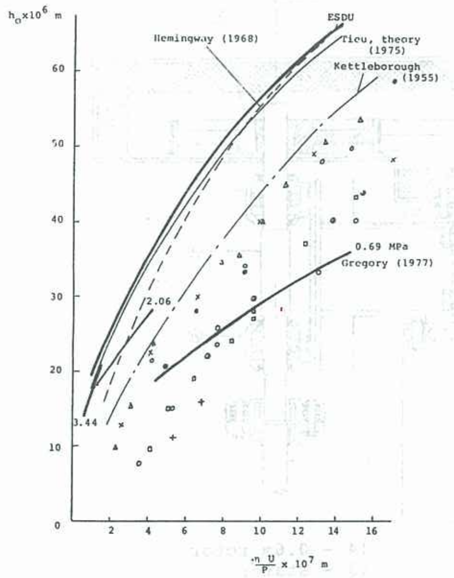

Fig.5 Comparison of Outlet Film Thicknesses

value of 9 readings, three from each pad. In this paper, all the measurements are confined to one pad.

In the current series of experiments, 4.5% of the working annulus was covered by the bearing, whereas the pads occupied 17% annulus area in Tieu and Hemingway , and 60% in Kettleborough. In the last reference, the effect of hot oil carry-over was more significant than in this paper.

At light duty, the pad fluttering could cause variation of film thickness, and therefore could affect the accuracy of the results obtained by the data-logger. The film thickness was recorded over a long test period of data-logging, and it was found that the results were not significantly affected by the pad vibration at medium to heavy duty, when pad fluttering does not occur. It is clear however that the reduction of film thickness was caused by rotor run out.

Circumferential and Radial Tilt

The circumferential tilt ratio between inlet and outlet is plotted in Fig.6a. The expected value of this ratio is 2.3, which is the theoretical optimum for hydrodynamic tilting thrust bearing. The measured film thickness ratio is 2.5 at light duty, i.e. high speed and light load . For duty parameter below 5 x 10^{-6}m the ratio increases rapidly. Above 8x10^{-6}, the ratio decreases to 1.8. This compares well with the values 1.74-1.77 measured by Gregory (1977).

The scatter of the results in this figure is quite large, especially when the duty parameter is greater than 5 x $10^{-6}m$. This indicates that the pad inlet edge continues to oscillate until the loading becomes heavy.

The radial tilt ratio is equal to the outlet film thickness on the pad minor diameter divided by that on the major diameter. It is smaller than the circumferential tilt ratio, and varies between 1.03

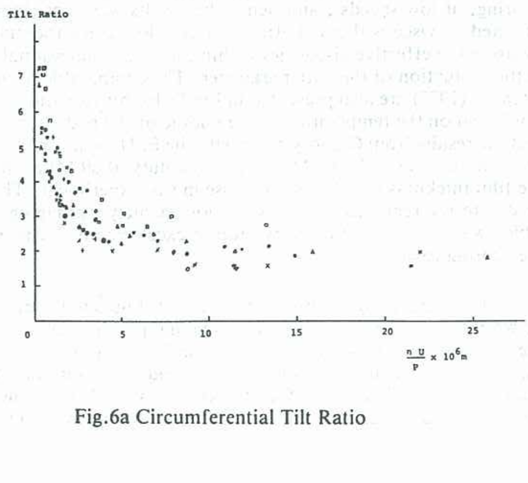

Fig.6a Circumferential Tilt Ratio

and 1.1, Fig. 6b. This means that the minimum film thickness is at the pad major diameter for most operating conditions. Unlike the circumferential tilt ratio which increases when heavy loads were applied, the radial tilt ratio decreases such that for heavy loads, the pad will tilt in the opposite direction and the tilt ratio is less than 1. The radial tilt ratio is relatively constant for duty parameter values greater than 2 x $10^{-6}m$. It was found that the tests with small circumferential tilt have large radial tilt. Any numerical simulation of the bearing oil film should take into account both circumferential and radial tilt.

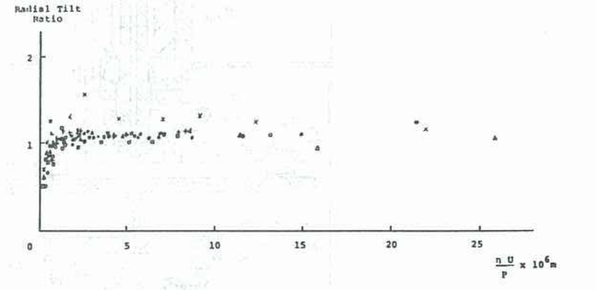

Fig.6b Radial Tilt Ratio

Friction

The friction coefficient at the rotor surface ,which is the average of 4 readings recorded by the data logger is plotted in Fig. 7 together with the theoretical friction from ESDU, and experimental results from the above references. The friction shows significant scattering ,especially at high speeds. This is due to fluctuating parasitic loss at 'no load' as experienced by Hemingway and Kettleborough, as well as to the rotor runout.

As the low point of the rotor surface passes over a pad, a higher loading is taken by that pad , which then has to adjust its angle of tilt, resulting in varying loads and frictional forces . Moreover because three pads experience different amount of waviness, and they themselves tilt differently, the resultant frictional forces from the three pads is varying. It was observed from the ultra-violet recordings that the waveform of the frictional force has a frequency which is three times that of the rotor speed, thus confirming that the oscillation of the frictional torque was due to rotor run out. The undulation of the rotor surface produces large variation in friction than those without any runout. The measured bearing friction is higher than other experimental work , again correlates with the finding of Mokhtar for journal bearing.

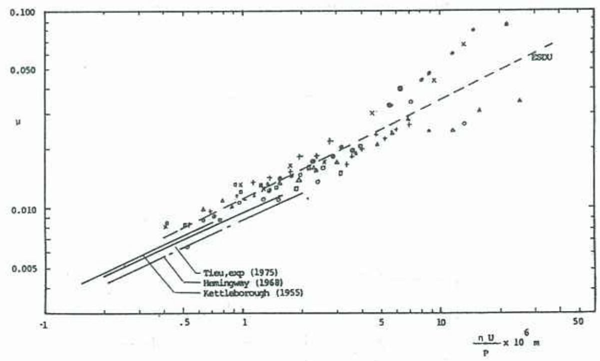

Fig.7 Friction Coefficient

The power loss based on measured bearing friction at 700 RPM is higher than the ESDU prediction at heavy duty (Fig. 8.).

Bearing Temperature Rise

The temperature rise above bearing inlet increases with loads and speeds is shown in Fig. 9.

Pad Flutter

Pad flutter can best be described as the rapid vibration of the thrust pads at high rotor speeds and relatively low loads. A low viscosity increases the probability that pad flutter will occur.

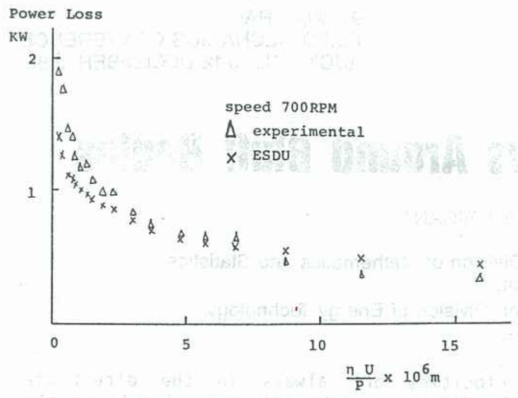

Fig.8 Power Loss

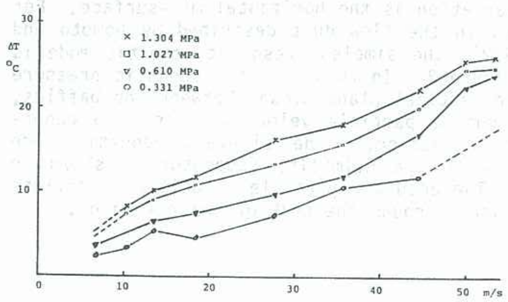

Fig.9 Temperature Rise Above Pad Inlet Temperature

At high speeds and low loads, knocking sound due to bearing pad fluttering against the rotor surface were quite audible , and they stopped if loading was increased. Results from the chart recorder indicated that pad flutter occurs when the parameter $\frac{\eta U}{P}$ is greater than $5 \times 10^{-6}m$. As the loading is increased, the amplitude of pad vibration decreases, particularly at the outlet edge. The scatter of the results decreases as the load increases or the speed is reduced.

The variation of film thickness and frictional torque are shown in Fig. 10. The two outlet film thicknesses oscillate together, indicating that the pad does not oscillate along a radial line. The inlet film thickness was not in phase with the outlet readings. The chart indicates that the pad is moving up and down through the stator movement on the hydrostatic oil film rather than rocking on its pivot.

The rotor run out causes the pad to oscillate to respond to the varying convergent wedge, especially from the 'run out' component. This causes the hydrodynamic load to vary, and the bearing pad in turn has to adjust its angle of tilt to accommodate the changing load. At the same time, the stator continually adjust its orientation on the hydrostatic oil film to accommodate different dynamic loads from the three bearing pads. Between 10 and $15 \times 10^{-6}m$, pad flutter is common in high speed tests.

To define the region in which pad flutter is most likely to occur, Fig. 11 is constructed. The thrust load is plotted on the horizontal axis, and the rotor speed along the vertical axis. The points plotted on the graph are defined by the load-speed combination at which pad flutter was found to occur.

The results is a rather wide region between the two curves in which flutter may occur. This region of uncertainty is due to the effects of variable viscosity. Above this region pad flutter will definitely occur. If a given combination of load-speed falls below the lower part of this region and the oil viscosity is high, pad flutter will not occur.

CONCLUSIONS

The experimental results presented in this paper indicate that theoretical design data are optimistic in their prediction of oil film thickness, especially with a rotor run-out that is comparable to the film thickness. The run out of the rotor reduces the bearing film thickness and increases significantly the bearing friction.

At high speeds and low loads, pad flutter occurs in a defined

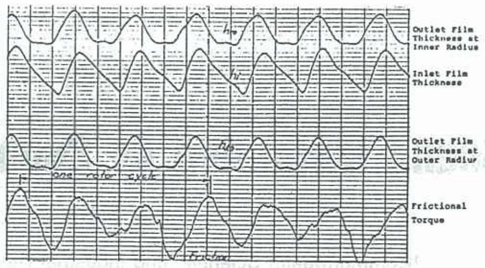

Fig.10 Fluctuation of Film Thickness and Frictional Torque

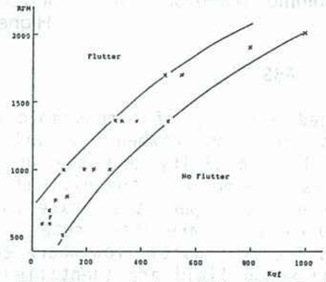

Fig.11 Fluttering Regions of Tilting Thrust Bearing In Terms of Loads and Speeds

combination of load and speed. This phenomena is similar to the spragging of tilting journal bearing. It is not a serious problem in normal application of pivoted pad bearing, but it is a phenomena which should be taken into account when dealing with high speeds, light loads and low viscosity lubricant.

ACKNOWLEDGEMENT

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