EXPERIMENTS ON THE BEHAVIOUR OF THRUST BEARING PIVOTED PADS

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SUMMARY
The present work regards an experimental investigation carried out on lubricated thrust tilting pads using a particularly versatile test rig. Simultaneous measurements of film thickness and friction are made. The tilt angle of the pads is measured by optical interferometry with monochromatic light with a glass disk simulating the collar in contact with the pad. The friction coefficient is evaluated by force measurements made possible by an aerostatic bearing supporting the pad structure. Visualisation of fluid flow and lubrication conditions is achieved by the use of a transparent glass disc and a video camera. All experimental data are recorded on a PC by means of a DAQ system. Experimental results are compared with numerical ones, obtained by a FEM code, generally showing a satisfactory agreement. Static pad characteristics have been calculated for different values of geometric parameters and operating conditions. The mutual influence between pads for an array of three pads has been also experimentally investigated.

Keywords: Hydrodynamics, Lubrication, Friction, Thrust bearings, Tilting pad.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Meaning</th>
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<tr>
<td>f</td>
<td>friction coefficient</td>
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<tr>
<td>F</td>
<td>friction force</td>
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<tr>
<td>h</td>
<td>film thickness at the pivot position</td>
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<td>u</td>
<td>tangential speed at the pivot position</td>
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<td>W</td>
<td>load</td>
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<td>α</td>
<td>tilt angle in the tangential direction</td>
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<tr>
<td>β</td>
<td>tilt angle in the radial direction</td>
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1 INTRODUCTION

The tilting pad thrust bearings are the most widely used hydrodynamic thrust bearings especially in the centred pivoted configuration and have been the object of extensive analytical, numerical and experimental research. Design charts have been provided for arbitrarily pivoted thrust bearing pads [1]. Thermal effects, pad deformation and turbulence have been investigated and complex analytical models devised using the ever more powerful available numerical tools [2-4]. On the other hand, in order to validate computer simulations refined experimental testing is required taking advantage of the presently available means of data acquisition and more efficient measuring instruments and techniques. Different experimental apparata have been set up. Testing to measure significant quantities such as friction for different load and speed combinations as well as pressure and thickness distribution has been performed on tilting pads with compliant surface in water [5]. The onset of turbulence has been investigated on a large-scale model of a single tilting pad working in air [6]. Dynamic effects have been studied in a high-speed machine [7].

The present work extends an experimental investigation started few years ago carried out on tilting pads using a particularly versatile test rig [8]. Simultaneous measurements of film thickness and friction are made and data recorded and processed with a PC. The tilt angle of the pads is measured by optical interferometry with monochromatic light simulating the collar in contact with the pads with a glass disk. In order to make visualization and image recording possible, tests are carried out at relatively low speeds and low loads so that the classical assumption of laminar flow and isoviscous lubricant holds.

The Reynolds equation is solved numerically by a code based on the finite element method (FEM), taking into account three-dimensional effects and results are compared with experimental ones.

2 EXPERIMENTAL APPARATUS

An experimental apparatus basically described in [9] and normally used for lubrication tests on nonconformal contacts [10, 11] has been employed. Little modifications have been introduced for carrying out tests on conformal surfaces such as pad and collar ones.

The lubricated contact occurs between the plane surface of a disc, simulating the collar supported by the thrust hydrodynamic bearing, and one or more pads. A schematic drawing of the rig is shown in Fig.1. Load is applied with weights and a lever mechanism having a radial gas bearing as fulcrum. The longitudinal motion of the support is constrained only by a load cell that measures the friction force on the pads.

The contact between the disc and the specimens can be regulated through a vertical positioning system. The disc is driven at the desired speed by an electric motor connected to the output of a digital acquisition board. The nominal speed range for the motor is between 0 and 3000 rpm. A helical planetary precision gear (1:20) connecting the shaft of the disc to its motor limits the disc speed to 150 rpm. A precise measurement of the speed is assured by the signal of an encoder in addition to that of the motor tachometric dynamo. With this system a precision of ±0.1 rpm is achieved.
An optical displacement sensor is used for film thickness evaluation. The laser sensor, also visible in Fig. 2, has a measuring range of ±0.25 mm.

The pads have an outer radius of 42 mm, an inner radius of 24 mm and a sector angle of 30° (the surface area is about 3 cm²). They are made of steel (AISI 430). Spheres with a diameter of 3.175 mm are used as pivots; they are placed at the mean radius of each pad at a distance of 0.8 mm from the pad centre (pivot offset), Fig. 3. The mean surface roughness is about \( R_a = 0.01 \, \mu \text{m} \).

The spheres are housed in conical seats in the pad support. Two chromel-alumel thermocouples are placed close to the pad inlet and outlet, Fig. 4.

Three different kinds of disc have been used: one made of steel and two of glass, Fig. 5. One glass disc is of crown glass and has a semireflecting chromium layer protected by a SiO₂ coating; the other one is an optical glass BK7 transparent disc. The diameter of the discs is about 160 mm. The surface roughness of the two glass discs is about \( R_a = 0.02 \, \mu \text{m} \) while that of the ground steel disc is about one order of magnitude greater.

The investigation on the flow is made possible by the transparent glass disc. The tilt angle can be evaluated in the case of the disc with the semireflecting layer by means of an optical interferometry system.

Signals are recorded from the load cell, the laser sensor, the encoder and the two thermocouples for contemporary measurements of the friction force, the mean film thickness, the speed and the lubricant temperature. Flow problems and tilt angle can also be investigated when the transparent glass disc and the one with the semireflecting layer are respectively used.

A phthalate diester, the bis(2-ethylhexyl)phthalate, has been used as lubricant. Its viscosity, measured with a rotational viscometer at the test temperature (20 °C), is \( \mu = 0.075 \, \text{Ns/m}^2 \).
The oil temperature is regulated by a bath/circulation thermostat. The inlet temperature was kept constant in the range of ±0.2 °C during tests.

3 EXPERIMENTAL DATA

Tests were carried out on both the configurations with one and three pads.

Three different loads per pad were tested: W = 10, 20 and 30 N (corresponding to W = 30, 60 and 90 N for the three pads). The range of W was chosen taking into account the characteristics of the experimental apparatus used (in particular the maximum load allowed by the glass discs).

For each load six different speeds were set: \( u = 0.05, 0.10, 0.15, 0.2, 0.3 \) and 0.4 m/s). The range of \( u \) was chosen in order to avoid cavitation and starvation phenomena as evidenced in previous testing [8]. Since the mean radius of the pad is equal to 51 mm, these velocities correspond to a range of rotational speed of the discs from 9.4 to 74.9 rpm.

Instantaneous measurements of the friction force, the pad displacement, the disc velocity and the temperatures near the pad leading and trailing edges were made. Data were recorded each second by a data acquisition system for a total time of one minute for each test condition at constant load and speed. For each load the experiments were carried out for all speeds in a completely automated way and the mean values of the different quantities were recorded after each minute test. Some variations were found during the one-minute tests due to some misalignment. However the investigation of time variant phenomena is beyond the scope of this work and only mean values are considered here.

The load cell and the laser sensor were calibrated at the beginning and end of each set of tests made with the same load, disc and pad(s). However, considering the transducer and electronic system errors and the fluctuations of the quantities, maximum errors have been valued at ±0.03 N for the friction force \( F \) and at ±0.5 \( \mu \)m for the measured displacement. In regard to the latter quantity some vibration phenomena caused a rather random offset of the zero position, in some cases up to about 10 \( \mu \)m.

The real film thickness, evaluated at the pad midpoint, is calculated from the measured displacement taking into account the distances of the measurement point and the pivot position from the fulcrum of the first order lever system used for the application of load.

Typical results of the friction force \( F \) and the pivot film thickness \( h \) obtained with different loads with the glass disc with the chromium semireflecting layer are reported in Fig.6 for the one pad (1P) configuration.

For the values of load and speed used, variations of temperature were not significant. A maximum increase of about 0.4 °C was detected from inlet to outlet. However the measurement of the oil temperature near the pads was useful for keeping a constant value regulating the thermostatic system.

The transparent disc with the semireflecting chromium layer was used for the evaluation of the pad tilt by optical interferometry with monochromatic light, Fig.8. Interference images were formed using a white light and an interference filter with a wavelength \( \lambda \) of 577 nm.
Due to the use of a spherical pivot, the pads can tilt in the tangential (motion) and radial directions. Therefore two angles characterize the inclination: $\alpha$ and $\beta$ shown in Fig.9. Each fringe corresponds to a zone with the same film thickness. A positive value of $\beta$ produces fringes like those shown in Fig.9b.

The tilt angles ($\alpha$ and $\beta$) are evaluated using the formulas:

$$\alpha = \arctan(\Delta h/\Delta x) = \arctan(\lambda/2n)/\Delta x$$

$$\beta = \arctan(\Delta h/\Delta y) = \arctan(\lambda/2n)/\Delta y$$

where $\Delta h$ is the difference of film thickness corresponding to the distance ($\Delta x$ or $\Delta y$) between two consecutive dark (or light) fringes, $\lambda$ the wavelength of the used light and $n$ the refractive index of the lubricant.

Examples of fringes due to pad tilt are reported in Fig.10. The vertical direction in the pictures corresponds to the pad radial one. The inclination of the fringes indicates a negative value of $\beta$. The interference images were stored by a video camera recorder and processed by a computer.

Both tilt angles $\alpha$ and $\beta$ increase with speed and decrease with load. Unfortunately for the higher speeds and lower loads vibration problems made it difficult to single out the fringes in the recorded images.

Since the planarity of the pads was not very good, the measured inclination of the pad surface due to the pad motion was affected by the pad crowning. An attempt to filter this unwanted effect has been made subtracting it as a systematic error, considering a rough linear dependence of the tilt angles on the Sommerfeld number described in the next paragraph. Some corrected results will be shown.

4 DATA PROCESSING

Raw data, in particular the friction values and the interferometric fringes, needed some processing.
Firstly, a correction was made to filter the friction effects due to the border of the support. The following formula was used:

\[
F = \frac{S_p \cdot (h + \Delta h)}{S_p \cdot (h + \Delta h) + S_b \cdot h} \cdot F_{\text{meas}} \tag{2}
\]

where \( \Delta h \) is the distance between the border of the support and the disc (\( \Delta h = 0.5 \text{ mm} \)), \( h \) is the mean film thickness, \( S_p \) and \( S_b \) are respectively the pad and border areas. The border effect is obviously greater in the 1P case (\( S_p = 310 \text{ mm}^2 \) for one pad, \( S_p = 930 \text{ mm}^2 \) for three pads, \( S_b = 545 \text{ mm}^2 \)) due to the ratio of the corresponding areas.

For a better comparison between the 1P and the 3P configurations, the differences of direction of the friction forces in the lateral pads were taken into account using the following formula for an equivalent mean force on a single pad in the 3P case:

\[
F = \frac{3}{1 + 2 \cdot \cos \gamma} \cdot F_{\text{meas}} \tag{3}
\]

where \( \gamma \) is the angle between pads.

Another difference between the 1P and the 3P configurations is that in the second case the two lateral pads present a slightly larger film thickness due to a different pivot displacement. By considering the lever arm, a difference of about 1% is found. Anyways, even including these corrections for the friction force, the friction coefficients, calculated dividing these forces by the normal load, show some differences in the tests made with one or three pads as shown in Fig.11.

In this figure, the friction coefficients calculated from the measured friction force using the glass disc with the semireflecting layer (“measured” in the legend) are plotted together with the values corrected by the use of equations (2) and (3). The friction coefficient for the 3P case is always smaller than that of the 1P case, but the effect of the corrections on the measured results is to draw results of the different cases closer. The remaining differences are then to be related to the mutual influence of the pads: the inlet effect present for the first pad is limited for the other ones and that reduces the drops of pressure and friction.

The results obtained with all three different discs are summarized in Fig.12. All quantities increase with \( u \). The increase of \( f \) with \( u \) along with the fact that \( f \) lowest values are obtained by increasing the load, clearly indicates full film lubrication conditions. The trends obtained with all discs are similar.
As expected, the different materials and surface roughness did not have a significant influence on the results because of the particular range of load and speed chosen, in which full lubrication conditions occur. However, some adhesion problems at the lowest speeds, more evident for the glass disc than for the metallic ones (including the steel one and the glass one with the chromium layer), indicate the importance of the materials when approaching mixed lubrication conditions. Experiments at low speeds in mixed and boundary lubrication conditions will be made in the future.

The differences in the friction coefficient for the 1P and 3P configurations seem to be rather independent on load and speed. At the same time, the differences in film thickness do not seem so significant considering the increasing uncertainties of this measure as load decreases.

5 COMPARISON BETWEEN EXPERIMENTAL AND NUMERICAL RESULTS

In order to calculate the pressure distribution in the pad film taking into account tilt in the radial direction, a FEM code [8] was used for the solution of the two-dimensional Reynolds equation.

Film geometry is completely defined by pivot film thickness and pad inclination. Once boundary conditions have also been defined (in this case equal to ambient) it is possible to determine film pressure.

Using first order triangular finite elements and linear interpolation functions the Reynolds equation can be expressed in terms of nodal pressure, yielding a system of linear equations, in matrix form

\[
KP = V
\]  

where \( K \) is a global “stiffness” matrix, \( P \) is the vector of unknown nodal pressures and \( V \) is a geometric vector.

Pad resultant forces and moments about the pivot can be determined summing up the element forces and moments. The moment tangential component may be also calculated and the tilt angle in the radial direction accounted for.

With the commonly accepted assumption of neglecting friction forces on the pad compared to the pressure resultant, for equilibrium the latter must have zero moment about the pad pivot. Such a condition, once the pivot film thickness is fixed, makes it possible to determine the pad tilt angles in both tangential and radial directions by means of an iterative method.

Numerical tests were carried out for different values of pivot film thickness with a mesh of 15×16 nodes. In fig.13 the numerical pressure field for the case of film thickness of 50 µm is presented. The results are reported as function of the Sommerfeld number

\[
S' = \sqrt{\frac{\mu \cdot u \cdot L}{W}}
\]

where \( \mu \) is the lubricant viscosity, \( u \) is the circumferential speed at the mean radius (pivot position), \( L \) is the pad length in the radial direction and \( W \) is the applied load.

Numerical and experimental results are compared in Fig.14. In this figure the friction coefficient \( (f) \), the pivot film thickness \( (h) \) and the tangential and radial tilt angles \((\alpha, \beta)\) are reported and compared also to the case of not allowed radial tilt. In the latter case it can be noted that all the calculated \((h, f, \alpha)\) quantities vary linearly with \( S' \). The effect of the radial tilt is a deviation of all curves from the linear behaviour to which they tend as \( S' \) decreases. Due to convergence problems, numerical results are extrapolated for very low values of \( S' \) (dotted lines).

The measured values of all quantities were elaborated as explained before. The agreement between numerical and experimental results seems quite good. The friction coefficient is lower for the 3P case (plain symbols) than for the 1P one for every load and disc. It tends to become greater than the calculated one for higher \( S' \) (that correspond to higher \( u \) and \( h \)). This could be explained by a greater influence of the support that inclines more as \( h \) increases, as confirmed by the preliminary tests with thicker pads described in the next paragraph.

Values of the film thickness and of the tilt angles deviate from the linear trends of the numerical solution not including the radial tilt possibility, in better agreement with the numerical results obtained considering this effect.

As regards the tilt angles, the results shown for the 3P case are the ones obtained for the central pad. Some differences were observed by similar measurements made for the last pad of the array. More in detail, lower values of the \( \alpha \) and \( \beta \) were found. This could be explained by the pad mutual influence, but also by a different curvature of the pad surface as well as by some small differences in the pad thickness.

Note that the radial tilt angle \( \beta \) shown in Fig.14 is negative, which means higher film thickness in the external zone.
6 EXPERIMENTS WITH DIFFERENT PADS

Some preliminary comparisons have been also made on a commercial pad (Glacier), Fig.15, with a support line contact, and a pivoted lab one having the same bearing area and eccentricity. The two pads are 0.9 mm thicker than the previously tested ones. Therefore the tests with these different pads were carried out also with the vertical positioning system lowered of 0.9 mm.

Some of the results obtained with the new pads are shown in Fig.16. Similar results obtained with the same disc (the glass one with the chromium layer) and one of the previous pads are also reported for comparison. It is interesting to note that the thicker pad (indicated with “high” in the legend of the figure) presents higher values of \( f \) compared to the thinner pad. This could be related to the influence of the support inclined in a different way, as proved by the lower values measured when the system is lowered of 0.9 mm (indicated with “-0.9” in the legend of the figure).

Anyway, the slopes of the friction curves are similar for the lab pad while the slopes of the ones obtained with the Glacier pad are different (in Fig.16 only the curves for 30 N are reported, but similar differences were found at different loads).

Figure 15: Tests with commercial pad; \( u = 0.3 \text{ m/s}, W = 10\text{ N} \)
It is also interesting to note that higher values of friction correspond to lower values of pivot film thickness. This is not strictly true for every condition, but it could be stated considering measurement uncertainties. The differences in behaviour can also be explained with the presence of the linear contact of the commercial pad.

7 CONCLUSIONS

An experimental investigation has been carried out on lubricated thrust tilting pads using a particularly versatile experimental machine on which it is possible to perform simultaneous measurements of several quantities. Film thickness, friction force, pad tilt angles and fluid flow conditions are investigated. In particular optical interferometry was used to measure contemporaneously radial and tangential tilt angles.

The case of one and three pads are investigated in similar conditions. Experimental results are compared with numerical ones generally showing a satisfactory agreement.

Lower values of friction have been found in the 3P configurations, showing a mutual apparently positive influence between pads.

Despite some experimental uncertainties, the adopted investigation methodology indicates fruitful possibilities of application and future developments.

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REFERENCES